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TÜRKİYE BİLİMSEL VE
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THE SCIENTIFIC AND TECHNICAL
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Mechanical Engineering, Material Sciences and
Manufacturing Systems Research Grant Committee

PANEL ISITMA SİSTEMLERİNİN MODELLEMESİ VE BİR STANDARD TASARIM ALGORİTMASININ GELİŞTİRİLMESİ

PROJE NO : MİSAG-12

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ANKARA

1. ÖNSÖZ

Bu araştırma 1.8.1991 tarihinde başlatılmış olup MİSAG-12 projesi ile TÜBİTAK tarafından iki yıl boyunca desteklenmiştir. Projenin ana amacına uygun olarak panel ısıtma ve soğutma konularında standart tasarım algoritmaları oluşturulmuştur. Bu çerçevede Türk Standartları Enstitüsü ve American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc. (ASHRAE) ile teknik, bilimsel ve idari ilişkiler tesis edilerek işbirliğinde bulunulmuştur. Projenin önemli ürünleri olarak iki adet Türk Standardı hazırlanmış, ASHRAE Handbook, Bölüm 6: "Panel Heating and Cooling" metni yeniden yazılmış, ayrıca endüstriyel uygulamalar için bilgisayar destekli tasarım programları geliştirilmiştir. Bu çalışmalarda ODTÜ/BİLTİR Merkezi ile HEATWAY firmasının teknik ve mali destekleri önemli ölçüde yarar sağlamıştır. Proje süresince ayrıca ulusal ve uluslararası düzeyde 26 adet yayın hazırlanmıştır (Ek-5).

Proje süresince 4 adet ve altı aylık faaliyetleri kapsayan ara raporlar TÜBİTAK'a sunulmuş, bu proje sonuç raporu ise üç bölümde hazırlanmıştır. Bunlar sırası ile idari rapor, mali rapor ve teknik rapor bölümleridir. Proje Sonuç Raporunun ana metnine ilaveten 10 ayrı ek şeklinde diğer gerekli bilgiler düzenlenmiştir.

2. İÇİNDEKİLER

3. TABLOLAR LİSTESİ

4. ŞEKİLLER LİSTESİ

5. ÖZ

Yapılarda panel ısıtma, yeni, yenilenebilir ve atık enerji kaynaklarının, daha yaygın, etkin ve verimli kullanılmasına büyük ölçüde imkan sağlayan bir sistemdir. Panel ısıtmaya özgü termo-fiziksel konfor koşulları sayesinde ısıtma yüklerinde de önemli ölçüde bir azalma ve enerji tasarrufu söz konusudur. Bu nedenlerle panel ısıtma, ilk yatırım maliyetlerinden işletme maliyetlerine ve enerji tüketimine uzanan ekonomik perspektif içersinde, ulusal ve evrensel enerji tasarrufu ile çevre sorunlarının hafifletilmesinde giderek önemli bir seçenek halini almaktadır. Bilinen bütün bu avantajlara ve uygulamalardaki olumlu gözlemlere karşın, uluslararası platform da dahil olmak üzere Ülkemizde panel ısıtmaya özgün tasarım yöntem ve yaklaşımları yeterince incelenmemiş ve sistem tasarımı ve analizi belirli bir sistematiğe kavuşturulmamış idi. Bu çalışmada uluslararası düzeyde ve bilimsel bir yaklaşımla, bir analitik model ve bilgisayar yardımı ile tasarıma yönelik bir algoritma hazırlanmıştır. Bu çalışmalar ASHRAE ve ISO kuruluşlarına da yansıtılmıştır. Ayrıca iki adet zorunlu TSE standardı hazırlanmış, bu kapsamda, panel ısıtmaya özgü ısı yükü hesap yöntemi de geliştirilmiştir. Ayrıca panel soğutma konusunda da çalışmalar yapılmıştır.

6. GİRİŞ

Ülkemizde döşemeden (yerden) ısıtma diye bilinen panel ısıtma uygulamaları özellikle son 10 yıldan beri yaygınlaşmaya başlamıştır. Daha çok konutlarda "konfor" ve "mimari estetik" sloganları ile pazarlanan döşemeden ısıtma sistemlerinin kendilerine özgü ve oldukça önemli avantajları vardır. Bunlar:

1. Isıtma ve soğutma hissedilir yüklerinde azalma,
2. Alternatif enerji kaynakları ile doğrudan çalışabilme,
3. Isı pompalarının performansını arttırma,
4. Çevre korumasına olumlu katkılar

şeklinde özetlenebilir.

Bu avantajların uygulamada gerçekleşmesi için detaylı ve hassas hesaplara dayanan bir tasarım yöntemi ile optimum çözümlerin aranması gereklidir. Bu düzeydeki bir teknik ve bilimsel bir altyapı ve araştırma birikiminin ülkemizde henüz oluştuğunu söylemek için vakit oldukça erkendir. Bu konuda faaliyet gösteren firmalar genellikle yurt dışındaki bazı yayınlarda yer alan ve çoğunlukla sadece bir mamul çeşidi ve uygulama tipi için geçerli abak veya basitleştirilmiş hesap yöntemleri ile iktifa etmektedirler. Bu konuda ilk tasarım yöntemi Makina Mühendisleri Odasında yayınlanmıştır (1989).

Kılkış (1988-a,b,c,d,e, 1989-a,b,c) döşemeden ısıtma konusundaki potansyel avantajları sistematik bir şekilde dile getirmiş ve olası bir tasarım algoritmasının anahatlarını çizmiştir. Mataracı (1988) elektriksel andırım yöntemi ile ODTÜ Makina Mühendisliği Bölümünde ısıtılan döşemeyi modellemiş ve tasarım nomogramları geliştirmiştir. M. Acar ise derlediği tasarım yaklaşımlarını makalesinde tanıtmıştır (1987).

Kılkış, piyasada kullanılan abakların yeterli olmadığını savunarak, oda boyu, yörenin denizden yüksekliği, döşeme kaplaması, iç ve dış sıcaklıklar gibi faktörlere göre düzeltilmesi gerektiğini öne sürmüş, tasarım abakları geliştirmiştir (Kılkış, 1990-b).

ODTÜ Döner Sermaye Projesi ile Mimarlık Bölümünde mevcut ODTÜ Güneş Evinin güneş enerjisi ile panel ısıtma ve soğutma projesi detaylandırılmış, fizibilite etüdleri yapılmış, ancak ödenek yetersizliğinden henüz uygulamaya geçmemiştir (Kılkış 1990-a, 1991-b).

Döşemeden ısıtma konusunda, bu projenin başlamasından önce hiç bir Türk Standardı veya ilgili bakanlık yönetmeliği mevcut değildi. Bu projenin doğrudan bir hizmeti ve ürünü olarak iki adet Türk Standardı hazırlanmıştır (1993-a, 1993-b). Özgün niteliği dolayısı ile İngilizce diline çevrilip ISO'ya teklif edilecektir.

Modern anlamdaki uygulamalar, Avrupa'da 1930'lu yıllarda daha çok tavandan ısıtma şeklinde (Crittall tavanları) başlamıştır. Şu anda Almanya'da yapılan yeni konutların % 50 'sinden fazlasında pael ısıtma ve/veya soğutma sistemi uygulanmaktadır. DIN Standartları bu konuda geniş bir çalışmaya girmiş ve tasarım açısından DIN 4725 'in tasarısını olgunlaştırmıştır (1990). Bu standart dört bölümden müteşekkil olup henüz kesinleşmemiştir. Fransız ve İngiliz standartları ise daha az bir kapsam ve teknik düzeydedir. ABD 'de ise konu İkinci Dünya savaşından sonra, Avrupa'dan dönen teknik subay ve askerlerce, bakır borulu sistem şeklinde yaygınlaştırılmış, fakat özellikle lehim malzemesi olarak uygun bir seçim yapılmaması, kontrol düzenlerinin yeterli olmayışı gibi başarısızlıklar nedeni ile 1950'li yılların ortasından itibaren unutulmaya başlamıştır. 1970'li yıllarda kendini gösteren enerji krizi, enerji tasarrufu ve çevre korumaya yönelik çalışmalar nedeni ile panel sistemler, gelişen metal olmayan boru teknolojisi ile de birlikte tekrar gündeme gelmiştir. ABD 'de tasarım için ASHRAE'nin El kitabında (1992) yer alan 6. bölüm kullanılmaktadır. Bu bölüm her ne kadar temel prensipleri içeriyor ise de optimum bir tasarım için gerekli kapsam ve hassasiyette değildir (Kılkış, 1992-d ve 1993-c). Bu projede geliştirilen algoritma, söz konusu el kitabının gelecek baskısında yer alacaktır. Yazılan Türk Standardı ve proje kapsamındaki bazı yazınlarda kaynak gösterilmiştir.

Aslen, ülkemizdeki uygulama ve genel kanının aksine, panel ısıtma yanısıra panel soğutma da yapılabilmektedir (Wilkins ve Kosonen, 1992). Panel olarak tanımlanan yüzey sıcaklığı kontrol edilen ve genellikle mevcut yapı bileşenlerinden (örneğin döşeme betonu gibi) faydalanılan ısıtma veya soğutma sistemleri tavanda, döşemede ve duvarda yapılabildiği gibi askı tavan, duvar panosu, ofis bölmesi şekillerinde de uygulanabilmektedir.

Panel sistemler her ne kadar tasarım ve uygulamada olası hata ve eksikliklere toleranslı sistemler olarak bilinse de, aslen panel ve oda ilişkisi, ısıl performans, ısıtma/soğutma yükleri ile panelin etkileşimi, konfor parametreleri ile etkileşimli ilişkiler, konuyu bilimsel yönden belki de en karmaşık bir problem haline getirmekte, olaya bir de enerji depolaması ve kontrol boyutlarını da eklemektedir. İnsanın ısıl konforundaki yeni şartlar da konunun ilgi alanını genişletmektedir.

Panel ısıtma/soğutma tatbik edilen bir iç mekanın ve panelin birbiri ve çevre ile olan ilişkilerinin hem ısıl hem de hidro-mekanik yönden tanımlanması gerekir. Zira panel ve iç mekanın performansı birbiri ile etkileşimlidir. Örneğin, ısıtılan bir döşeme yüzeyindeki tabii taşınımla ısı transferi; o mekandaki hava hareketine, iç hava sıcaklığına, hatta duvar yüzeylerinin ve pencerelerin ortalama iç yüzey ortalama sıcaklığına bağlıdır. Buna mukabil döşeme panelinin ısıl performansı da söz konusu mekanın iç sıcaklığına, hava hareketine, duvar sıcaklıklarına bağlıdır.

Panel yüzeyi referans alınarak aşağıda belirtilen sınırlardaki şartlar tanımlanmalıdır:

A- Panel yüzeyi/iç mekan,

B- İç mekan/dış çevre,

C- İç mekan/iç çevre,

D- Panel/iç ve/veya dış çevre.

Dünya literatüründe modern anlamdaki teknik çalışmalar da 1930 'lu yıllarda başlar. Çalışmalar genelde önceleri deneysel ve analitik olarak başlatılmış, son yıllarda sayısal modelleme çalışmaları ağırlık kazanmıştır.

Yukarıda sözü edilen sınırlarda hem ısıl (örneğin: yüzey ısı transfer katsayısı) hem de hidro-mekanik şartların (pencere ve kapılardan olan tabii infiltrasyon miktarı gibi) belirlenmesi gerekir. Bu sınır şartlarından birçoğu ise birbirleri ile etkileşim içindedirler (örneğin tabii infiltrasyon, mekan içindeki hava hareketine, sıcaklığa, dolayısı ile panel performansına bağlıdır). Ayrıca ısı depolama miktarı ve etkisi, ısıtma/soğutma yükleri, mekandaki basınç dağılımı gibi parametrelerin etkileşimli olarak tariflenmeleri gerekir. Örneğin tabii infiltrasyon, hacmin ısı yükünü, ısı yükü ise iç hava sıcaklığını, dolayısı ile panel performansını (ve panel ısı kayıp/kazançlarını) etkiler. İşte bu karmaşık problemin ideal bir biçimde çözülebilmesi için problemin bir bütün olarak ele alınması gerekir. Bundan önce yapılan çalışmalar istisnasız olarak problemin etkileşim özelliğini büyük ölçüde gözardı etmiş ve problemin belirli kısımlarını soyutlayarak tek bir kısmını çözmüşlerdir. Bu çalışmada ise, literatürde bu kapsam ve yaklaşımda yer almayan geniş bir tanımlama yapılmıştır.

Bu tanımlamadan hareketle iki tür modelleme çalışması gerçekleştirilmiştir:

a- Analitik modelleme,

b- Sayısal modelleme.

Problemin genel tarifinde yer alan etkileşim parametrelerini gözardı etmeden, fakat kolay uygulanabilir bir algoritmik seviyede, iç mekan ve panel analitik olarak ele alınmıştır.

Isı (soğuk) depolaması modeli, ısıtma/soğutma yüklerinin panel uygulamasına özgü hesap yönteminin geliştirme çalışmaları bitirilmiş, bulgular hem TSE'ye hem de ASHRAE'ye sunulan ortak proje raporunda yer almıştır (1993-c).

Sayısal modelleme çalışmalarında ise ANSYS Sonlu Elemanlar Analiz Programı ODTÜ/BİLTİR Merkezinde çalıştırılmıştır. Oda içi modellemesinin daha iyi bir şekilde yapılabilmesini teminen ANSYS programı ile FLOTRAN program pakedi birlikte kullanılmıştır. Bu yaklaşımla oda içi ısı transferinde taşınım/ışıma süperpozisyon modeli çözülmüş, analitik ısı yükü modelindeki tabii infiltrasyon düzeltme katsayıları mukayese edilmiştir. Örnek çözümler EK-8'de verilmiştir.

Proje amaçları içersinde iki bilgisayar programı geliştirilmiştir. Bunlar:

i - Döşemeden Isıtma Tasarım Programı,

ii - Bilgisayar Destekli Isıtma Yükü Programı'dır.

Bu programlara ilaveten Panel Soğutma ve Kar/Buz Eritme sistemlerinin tasarımı için iki ayrı bilgisayar programı daha hazırlanmıştır. Bu dört yazılım sırası ile Ek 6, 7, 9 ve 10'da tanıtılmıştır.

7. GENEL HATLARI İLE PROJE AMAÇLARI

1. Bu projenin ana amacı panel ısıtma ve soğutma konusunda geçerli olacak bir "uygulanabilir" tasarım algoritmasının ulusal ve uluslararası teknik işbirliği seviyesinde geliştirilerek tatbikata geçirilmesidir. Ülkemizde konu sadece döşemeden ısıtma alanında bilinmekte ve uygulanmaktadır. Hem ülkemizde yeni olan panel soğutma konusunun tanıtılması, hem de proje çalışmalarına uluslararası paralellik kazandırması amaçları ile proje sözleşmesinde yer almayan panel soğutma konusu, projenin iki yıllık tekamülü sırasında ister istemez ana amaçları arasına girmiştir. Bu çerçevede, hem panel ısıtma hem de panel soğutmada geçerli olan genel amaçlı bir algoritmanın geliştirilmesi mümkün olmuştur.

2. Araştırma projelerinin genel sıkıntısı, yeterli uygulama imkanı bulamamasıdır. TÜBİTAK'ın destekleme gayelerinden birisi de bu sıkıntıyı giderecek çözümlerin oluşturulmasıdır. Bu paralelde proje çalışmalarında aşağıdaki amaçlar hassasiyetle gözetilmiştir:

a) Geliştirilen algoritmanın belgelenmesi, yaptırım gücüne kavuşturulması: Bu amaçla iki yeni Türk Standardı hazırlanmış, kurum ve kuruluşların görüşleri de alınarak metinleri kesinlik kazanmış ve çalışmalar neticelendirilmiştir. TSE tarafından "Mecburi Standart" olarak yürürlüğe girmeleri ilgili Bakanlıklara teklif edilecektir.

b) Geliştirilen algoritmanın "uygulanabilir" olması: Çok karmaşık ve uzun hesaplamalar gerektiren, pahalı bilgisayar donanımına ihtiyacı olan bir algoritma, uygulamada ister istemez gerekli kabulü görmeyecek ve bilimsel bir rapor olarak kalacaktır. Bunun aksine, çok basitleştirilmiş bir algoritmada yeterli hassasiyeti ve doğruluğunu kaybedecek, bilimsel niteliğini yitirecektir. Bu projede söz konusu bu iki zıt faktörün "optimum" bir şekilde bağdaştırılarak, doğru ve hassas çözüm özelliğinden ödün vermeden, mümkün olan en basit ve uygulanabilir bir algoritmanın geliştirilmesine çalışılmıştır.

Bir algoritmanın uygulanabilir olması için kolay olması yeterli değildir. Bu algoritmanın yeterli bilgi desteğine, dökümantasyon ve belgelere de sahip olarak endüstriye bir paket olarak sunulması ve yeterli eğitimin sağlanması gerekir. Bu amaçla: Hem TSE standartlarındaki metinler eğitim ağırlıklı hazırlanmış, hem de kişisel bilgisayarlarda kolaylıkla kullanılabilir etkileşimli yazılımlar geliştirilerek kullanım kılavuzları hazırlanarak bu proje eklerinde verilmiştir.

c) Uygulayıcıların algoritmayı kabul edebilmeleri için, algoritmanın ve yazılımların geliştirilmesi sırasında firma ve kuruluşların görüşleri alınmış, istek ve ihtiyaçları gözönünde tutulmuştur.

3. Uluslararası nitelikte bir çalışma yapılması. Projenin sözleşmesinde de yer aldığı üzere ASHRAE ile proje özelinde ilişkiler tesis edilmiştir. Bu teknik işbirliği çerçevesinde ortak ODTÜ/TÜBİTAK/ASHRAE teknik raporu da düzenlenerek (Bakınız Ek-5, Yayın no: 19) aynı algoritmanın ASHRAE Handbook bölüm 6'da yer alması sağlanarak, belki de ilk kez böyle bir proje uluslararası alanda kabul görerek değerlenmiş olmaktadır. Türk Standartları bu el kitabında referans olarak gösterilmiş, yine bu projede hazırlanan yayınların bir kısmı muhtelif bölümlerle (ASHRAE Chapters: 48, 45, 6, 15, 12) kaynakça gösterilmiştir. Özgün niteliğinden dolayı ISO 'ya da teklif edilecektir.

4. Geliştirilen algoritmanın genel kapsamlı olması: Literatürde yer alan çalışmalar, istisnasız problemin sadece belirli bir kısmını ele almışlardır. Halbuki problem ısıtma soğutma yüklerinin hesabı da dahil olmak üzere çok yönlüdür. Dolayısı ile ilk kez bu projede ısıtma soğutma yüklerinin panel sistemler için hesabı sistematik bir algoritmaya kavuşturulmuştur. Bu algoritma TSE standardında yer almıştır. Projede:

a) Panel sistemler için ısıtma, soğutma yüklerinin hesap algoritmasının geliştirilmesi,

b) Panel sistemlerinde ısı transfer hesap yöntem ve algoritmasının geliştirilmesi,

c) Panel sistemlerinde özellik arzeden hidrolik devre basınç kayıp hesapları ve diğer kavramların da proje kapsamında incelenmesi

amaçlanmıştır.

Yukarıda özetlenen bütün amaçların gerçekleşmesi içinde analitik, sayısal benzetim ve deneysel yöntemler öngörülmüştür. Proje sözleşmesinde yer alan ve proje çalışmaları esnasında daha da genişleyen amaçlar çerçevesindeki konuların tümü analitik ve sayısal yöntemlerle çözülebildiği için, deneyler sadece çözümlerin doğrulanmadan ve mukayese amaçları ile sınırlı tutulmuştur.

5. Diğer çalışmalar:

i) Proje çalışmalarının tanıtımı: Yayınlar, seminer ve kurslar, yaz okulları düzenlenmesi

ii) Proje çalışmalarının devamı: Panel soğutma ve kar eritme konularında yeni proje teklifleri yapılması.

iii) Yeni uygulama alanlarının ülkemizde tanıtımı: Örneğin panel soğutma, kar eritme, toprak ısısı ve güneş enerjisi ile hem panel soğutma ve hem panel ısıtma konularında da yazılımlar hazırlanmış ve ilgili kuruluşlarla temaslar başlatılmıştır. Bu suretle proje kapsamı dışında da ülke ekonomisine yarar sağlanabilecektir.

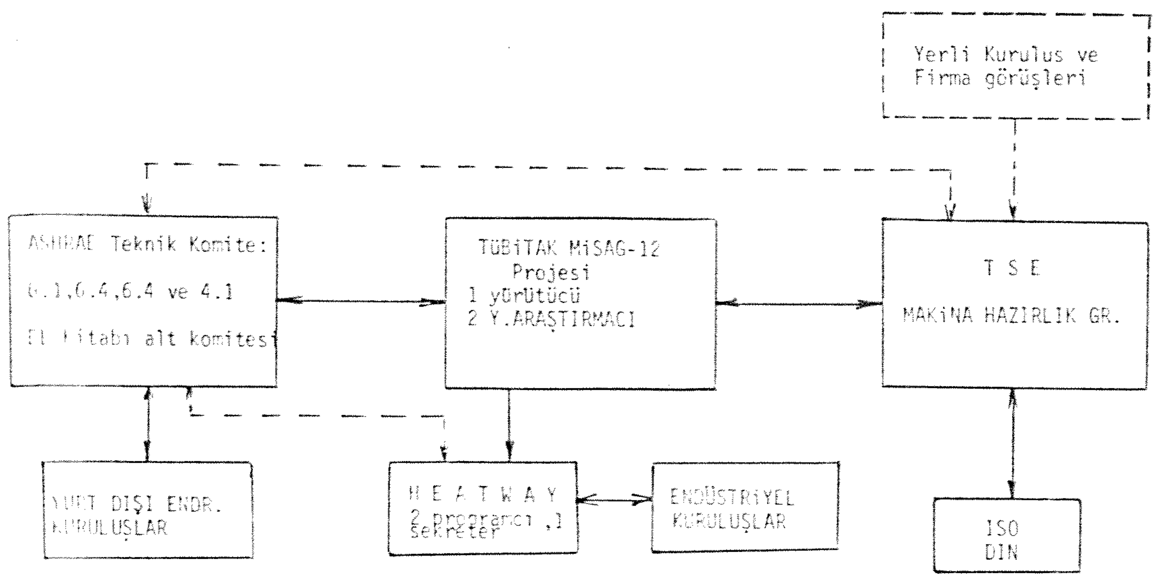
8. İDARİ RAPOR

8.1. Genel Bilgiler

Projenin çok kapsamlı amaçlarını gerçekleştirebilmek üzere TSE, ASHRAE ve ODTÜ'de hem bilimsel hem de idari faaliyetler belirli bir ko-ordinasyon içersinde yürütülmüş, HEATWAY Firmasının finans desteği ve teknik imkanları kullanılmıştır. Daha proje teklifi yapılırken TSE'de Makina Hazırlık Grubunda bir standart hazırlık grubu oluşturulmuştur. Proje kapsam ve amaçlarından bahisle ASHRAE'nin iki teknik komitesi ile doğrudan ilişki sağlanmıştır. Bu çerçevede, Proje Yöneticisi ASHRAE T.C.6.4. Convective Space Heating ve ASHRAE T.C.6.5. Panel Heating and Cooling Teknik komitelerine asli üye olarak seçilmiştir. Ayrıca T.C.4.1. Load Calculations ve T.C.6.1. Teknik komitelerinde sırası ile ısıtma, soğutma yüklerinin hesabı ve kar eritme sistemlerinin tasarımı konularında özel görev yapmış, tüm teknik komite toplantılarına katılmış, ASHRAE Handbook'un muhtelif ciltlerinde Chapter 6, Chapter 15, Chapter 48 ve Chapter 45'de yeni düzenlemeler yaparak Chapter 6'yı yeniden yazmıştır (Bakınız Ek-5, Yayın no: 19).

8.2. Proje Çalışmalarının Organizasyonu

Proje çalışmaları ile ilgili Genel Organizasyon şeması Şekil 1'de gösterilmiştir.



Şekil 1. Proje Çalışmalarında İdari Organizasyon.

MİSAG-12 projesinde sürekli olarak 2 adet yardımcı araştırmacı görev yapmıştır. Altı aylık dilimler şeklinde bu kişiler:

1. Y.Müh. Meriç Sapçı, Ar. Gör. Mahmut Uludağ, ODTÜ
2. Y.Müh. Meriç Sapçı, Ar. Gör. Mahmut Uludağ, ODTÜ
3. Selçuk Sağer, Mahmut Uludağ, Araştırma Görevlileri, ODTÜ
4. Selçuk Sağer, Mahmut Uludağ, Araştırma Görevlileri, ODTÜ

dir.

Projenin değişik safhalarında Meriç Sapçı ve Nuri Çarkoğlu'nun belirli çalışmaları hizmet alımı olarak değerlendirilmiştir. Proje çalışmalarında Sekreterya görevlerini ODTÜ Makina Müh. Bölümünden Gülseren Beyaz üstlenmiştir. ANSYS ve FLOTRAN paket programlarının kullanımında Figes Şirketinin teknik danışmanlığı, servis ve eğitim hizmetlerinden yararlanılmıştır.

Genel hizmetli olarak ODTÜ Mak. Müh. Bölümünden Semih Alten çalışmıştır. Isı transfer borularının ısıl deneyleri ODTÜ Makina Müh. Bölümünde gerçekleştirilmiştir. Ayrıca sayısal modelleme konusunda ODTÜ Ar. Gör. Aslı Aksel Doktora öğrencisi olarak çalışmalarda bulunmuş, Ar. Gör. Mahmut Uludağ'da Y.Lisans tezini, panel ısıtmada ısı yükünün bilgisayar destekli tesbiti konusunda yürüterek tez çalışmalarını proje sonunda bitirmiştir.

8.3. Diğer Kuruluşlarla İlişkiler ve İşbirliği

Diğer kuruluşlarla da hem geliştirilen algoritma hakkında görüşlerini almak, hem de proje ürünlerinin uygulamaya geçirilmesi amaçları ile ilişkiler sürdürülmüştür. Bunlardan en kayda değeri Yüksel Proje A.Ş. ile TÜBİTAK bünyesinde Türkiye Teknoloji Geliştirme Vakfına bir uygulamalı proje teklifinde bulunulmasıdır.

Bu proje teklifinin hazırlanmasında ve ön etüd çalışmalarında bilgisayar programları kullanılmaya başlanmıştır. Bu projede esas itiban ile 200 m^2 lik örnek konutda panel ısıtma ve soğutma sistemi kullanılacaktır. Panel sistem ise alternatif enerji kaynaklarından yararlanacaktır. Toprak ısısı kaynaklı bir ısı pompası proje direktörünün daha önceden hazırlamış olduğu TS 10055 standardı çerçevesinde değerlendirilmek üzere kullanılacaktır. İTÜ Makina Fakültesi ve Gazi Üniversitesi Makina Müh. Bölümü ile yapılması düşünülen ortak çalışmalar çerçevesinde ise güneş enerjisi tahrikli bir absorpsyonlu ısı pompası ile de panel soğutma araştırması gündemdedir. Bu projede geliştirilen panel soğutma tasarım programı ile ön etüdler bitirilmiştir. Absorpsyonlu ısı pompalı panel soğutma sistemi hakkında geliştirilen model iki ayrı yurtdışı makale olarak hazırlanmıştır (Ek-5 Yayın no.ları: 14, 16).

Ülkemizin tarihi yapılarında ve pre-fabrik yeni ahşap (MKEK imalatı) yapılarda tahta döşeme altından panel uygulamasına yönelik bir model ve bilgisayar yardımı ile tasarım programı geliştirilmiş ve ilgili kuruluş ve Bakanlıkla ilişkiye geçilmek üzere hazırlıklar bitirilmiştir. Bu özel model yurtdışı bir makale ile tanıtılacaktır (Ek-5, Yayın no: 25).

Panel ısıtma yükünün değişiklik arzetmesi nedeni ile TSE 'ye TS 2164 'de gerekli tadilat için başvurulmuş ve bu müracaat kabul edilmiştir. Aynı şekildeki bir başvuru ASHRAE 4.1 Teknik Komitesine de yapılmış olup, gelecek toplantıdaki bir özel seminerde bu konunun tartışmalı olarak ele alınması planlanmıştır.

ISO ve DIN kuruluşu ile de temaslar yapılmış, tasarı halindeki DIN 4725 standardı, ASHRAE Handbook Chapter 6 ve TS Standartları ile uyumun tesisi için ilk girişimler başlatılmıştır. İngilizce tercümelerinin bitirilmesini takiben hazırlanan iki Türk Standardı ISO'ya teklif edilecektir.

Terimler ve Tarifler Türk Standardı, ASHRAE 'nin Terminology in HVAC and R adlı terminoloji kitabı içinde panel ısıtma, soğutmaya ilişkin temel tariflerin proje yürütücüsü tarafından hazırlanmasında temel alınacaktır.

Kar eritme modelinin hazırlanması sırasında Devlet Meteoroloji İşleri Genel Müdürlüğünün kar rasat verilerinin bilgisayar ortamında değerlendirilmesi için ilişkiler başlatılmış olup, bu işbirliği sürmektedir.

Ayrıca, Merkezi Hollanda'da bulunan Int. Energy Agency, Heat Pump Center ile panel sistemlerin ısı pompaları ile kullanımındaki avantjalar konusunda işbirliği yapılmış, davetleri üzerine yayın yapılmıştır (Ek-5 Yayın no: 9 ve 22).

8.4. Diğer Faaliyetler

A- Yayın Faaliyetleri:

İki yıllık proje süresince toplam 26 adet bilimsel yayın hazırlanmıştır. Bunlardan ikisi Türk Standardı, ikisi Türkçe makale, birisi hem İngilizce hem Türkçe makale, diğerleri ise İngilizce dilinde makale, teknik rapor ve tebliğ niteliğindedir. Bu yayınlardan birisi ASHRAE Handbook: HVAC Systems and Equipment, Chapter 6 'dır. Bu yayın ASHRAE teknik kurullarında olumlu görüş almaktadır. Örnek bir yazı 8.7 bölümünde verilmiştir.

Bu yayınların tamamı ve listesi Ek-5 'de verilmiş olup, bu yayınlara yeri geldikçe Teknik Raporun değişik bölümlerinde atıf yapılacaktır. Bu suretle Sonuç Raporunda konuların tekrarı önlenecektir.

B- Teknik Toplantı, Simpozyum ve Uluslararası Konferanslara Katılım:

1) Yurt İçi:

- EE 2000 Enerji Tasarrufu ve Enerji Verimliliği Sempozyumu, 16-18 Kasım, 1992, Ankara (Tebliğ ile)
- İTÜ Makina Fakültesi, 7 Haziran 1993: "Panel Isıtma ve Soğutmanın Enerji Tasarrufundaki Yeri: Temel Teori ve Uygulama Esasları" (Davetli konuşmacı).
- Seminer: "Panel Isıtmada Isıl Konfor, Tasarım Esasları ve Kısıtları", 20 Haziran 1993, Ankara, YERPEX Firması.
- 22-24 Haziran 1993: ODTÜ/BİLTİR Merkezinde verilen ANSYS uygulayıcı eğitim seminerine iki yardımcı araştırmacı katılmış, ayrıca Ar. Gör. Selçuk Sağer 12-13 Temmuz 1993 tarihlerinde Bursa'ya Figes Şirketine eğitime gönderilmiştir.
- ASHRAE Winter Annual Meeting, 25-29 Ocak, 1992, Anaheim (Teknik Komite toplantıları).
- ASHRAE Annual Meeting, 27 Haziran - 2 Temmuz, 1992, Baltimore (Teknik Komiteler Üyesi).
- ASHRAE Annual Meeting.
- ASME Winter Annual Meeting, 8-13 Kasım, 1993, Anaheim (Tebliğ ile).
- Miami Üniversitesi, 9 Nisan 1993: "Radiant Panel Heating and Cooling: Fundamentals and Applications", Seminer (Davetli konuşmacı).
- Energy Seminar, 14 Nisan, 1993, Pocahontas (Davetli konuşmacı).
- ASHRAE Annual Meeting, Denver, 26-30 Haziran, 1993 (Tebliğ ile).

Ayrıca T.C.4.1, 4.5, 4.6 ve 6.1 Teknik Komite toplantılarının tümüne katılma.

Endüstriyel Temaslar:

İstanbul'da ISIYER, THY, DHMI, ALARKO, ESMAN firmaları, Ankara'da YÜKSEL Proje, TEKNİK, YERPEX, SEHA, CELAL OKUTAN firmaları ile 1993 Mayıs-Temmuz ayları arasında muhtelif görüşmeler yapılarak, proje ürünlerinin TÜBİTAK mevzuatı dahilinde ne şekilde uygulamaya geçirileceği araştırılmış, görüş ve teklifleri alınarak değerlendirilmiştir.

Bu temaslar sırasında aşağıdaki özel uygulamalar alanında da ürünlerin kullanılabileceği anlaşılmıştır (Rapor 4, Sayfa 9).

- Soğuk hava depolarında zemin betonu koruması,
- Hava meydanlarında kar ve buz eritmesi,
- Hangar ve depolarda (ASKERİ ve MKEK) emniyetli ısıtma ve soğutma (yangına ve patlamaya karşı)
- Müze ve tarihi yapılarda estetiği bozmadan ve emniyetli ısıtma soğutma.
- Toplu konutlarda doğal gaz tahrikli ısı pompaları ile ısıtma soğutma.

8.5. Proje Bitimini Takiben Yapılması Planlanan Faaliyetler

8.5.1. Genel

Projede uygulamaya dönük ve pazarlanabilir ürünler geliştirilmiştir. Bu ürünlerin, geliştirilen model ve Türk Standartlarının tanıtılması ve gerekli eğitimin verilmesi hem TÜBİTAK'a gelir getirici hem de konunun ülkemizde yaygınlaşmasına yardımcı olacaktır.

Pazarlanabilir ürünler:

1. Panel Isıtma Tasarımı Bilgisayar Programı: Bu program pakedi, kullanıcı klavuzu ile birlikte eğitimle beraber veya eğitimsiz olarak pazarlanabilir. Eğitim için 2 gün (1 gün teorik, 1 gün uygulama) gerekmektedir.

2. Panel Isıtmada Isı Yükü Hesabı Bilgisayar Programı: Eğitim için 2 gün önerilmektedir.

3. Panel Soğutma Tasarımı Bilgisayar Programı.

4. Kar ve Buz Eritme Tasarımı Bilgisayar Programı.
3 ve 4 sayılı ürünler içinde kullanıcı klavuzları.ve programlar ilgili Bölümlerde bu rapora eklidir. Eğitim için yine 2 gün önerilmektedir.

5. Panel Isıtma/Soğutma Teorisi ve Uygulama Esasları Eğitimi:
Bu projede üretilen tüm bilimsel ve teknik bilgilerin aktarımını içerecek bu eğitim için ek bir TÜBİTAK desteği ile seminer notlarının hazırlanması, hatta bir kitap şeklinde yayınlanarak satılması gereklidir. Seminer süresi 3 gün olarak önerilmektedir.

6. Panel Isıtma Soğutma Tasarım Nomogramı: Kullanım kılavuzu ile birlikte tek başına pazarlanabilir nitelikteki bu ürün 5 sayılı ürün ile birlikte de değerlendirilebilir.

7. Diğer bilgisayar programları:

i) Yükseltilmiş ahşap döşemede, çatıda, duvarda, bölmede ısıtma soğutma bilgisayar programları. Bu programlar için gerekli model ve temel yazılımlar mevcut olmakla birlikte uygulayıcıların özel istek ve ihtiyaçlarına göre geliştirilebilecektir.

ii) Soğuk hava depolarında temel ve zemin koruması tasarımı için bilgisayar programı.

8. Diğer gelir getirebilecek aktiviteler: Eğitim kursları ve seminerler, toplantılar ve tanıtım konuşmaları.

Bu alanda bir TÜBİTAK-TUSEP programı önerilecektir.

Planlanan diğer faaliyetler:

- NATO Yaz Okulu (ASI) düzenlenmesi,
- Lisansüstü ders içeriği hazırlanarak ilgili Üniversite, Bölüm ve kuruluşlara önerilmesi,
- Kamuoyu oluşturucu tanıtım broşürünün dizaynı ve metninin hazırlanması.
- İlgili bakanlıklarda bilgilendirme toplantısı ve seminerler düzenlenmesi.
- TSE'ye panel soğutma ve kar eritme konusunda standartlar hazırlanması için teklif yapılması.

8.5.2. Uygulama Özeti

Tahmini tarihler itibarı ile:

1994 :

1. TÜBİTAK Gebze tesislerinde TUSEP Semineri (Temmuz ayında).
2. Bilgisayar programlarının tanıtımı ve pazarlanmasını amaçlayan tanıtım broşürlerinin dizaynı.
3. Pilot kuruluşlarda bilgisayar programlarının kullanılması (Beta testing: değerlendirme).
4. Seminer notları ve kitap basımı için TÜBİTAK'tan destek talep edilmesi.
   Destek kabul edildiğinde seminer notu ve/veya kitabın yazımı ve basımı
   (bitiş tarihi: 1995).
5. TSE'ye yeni standart konularının teklifi.
6. ISO'ya TSE İngilizce tercümelerin gönderilmesi.
7. ASHRAE toplantılarına ve teknik komite oturumlarına katılım.
8. Kabul edildiği taktirde YÜKSEL Proje A.Ş. ile ortak projeye başlanılması.
9. Eğitim seminerlerinin planlanması.

1995 :

1. TIBTD, TÜBİTAK ve Endüstri desteğinde
   " 1. Panel Isıtma ve Soğutma Sempozyumunun İzmir'de düzenlenmesi".
2. NATO ASI Yaz Okulunun Çeşme'de düzenlenmesi.
3. Gelen talepler doğrultusunda eğitim seminerlerinin verilmesi.

8.6. Değerlendirme

Proje teklifi ve sözleşmesinde yer alan amaçların tamamı gerçekleştirilmiş, hatta bu amaçlar da aşılarak ek ürünler ve ek teknikler geliştirilmiştir.

Yurtdışı kuruluşlarla kurulması planlanan ilişki ve işbirliği tahmin edilenin üzerinde bir seviyede gerçekleşmiştir. Öngörülen iki standart zamanında yazılarak uygulamaya "mecburi standart" olarak sunulması sağlanmıştır. Hazırlanan analitik algoritma aynı etkinlikte hem panel ısıtma ve hem de panel soğutmada geçerlidir. Yine bu algoritma her tür panel (tavan, döşeme, duvar) pozisyonu ve konstrüksyonu (döşeme içi, askı metal tavan, yükseltilmiş döşeme altı, duvar içi gibi...) için geçerlidir. Bu sayede istek ve gereklere bağlı olarak özel uygulamalara yönelik ek bilgisayar programlarının yazımı mümkün olmuştur. Geliştirilen Algoritma uluslararası kabul görerek ASHRAE el kitabında yer alacaktır.

Panel ısıtma ve soğutmaya özgü yük hesaplarının belirli bir kaideye bağlanması Dünyada ilk kez gerçekleştirilmiştir. Yine Dünyada ilk kez hem ısıtma hem soğutma amaçlı ve her panel için geçerli bir "Üniversal Tasarım" nomogramı geliştirilmiştir. Bu ürün yine ASHRAE'nin el kitabında da yer alacaktır. ASHRAE inceleme komisyon üyeleri çok olumlu görüş beyan ederek TÜBİTAK'ın desteğine ve hazırlanan rapora katkıları için TSE ve ODTÜ'ye teşekkür etmişlerdir.

Bu çalışmalar sonunda bir yüksek lisans tezi bitirilmiş, bir doktora tezi ise devam etmiştir.

Sonuçta, kendi değerlendirmelerimiz, TSE ve ASHRAE beyanları çerçevesinde projenin teorik, bilimsel, teknik, ekonomik ve akademik katkıları hem ulusal hem de uluslararası düzeyde amaçlarını aşarak gerçekleşmiştir.

Proje çalışmaları mali açıdan da öngörülen miktar içersinde bitirilmiştir. Şu anda 13 Milyon TL dolayında bir meblağ bütçede bulunmaktadır.

Proje mali raporu 9. bölümde detaylı olarak verilmiştir.

8.7. Bazı Belgeler

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"Experts in Radiant Floor Heating Design and Equipment - Over 12 Years Experience"

October 20, 1993

Dr. Birol Kilkis

Dear Dr. Kilkis,

I have reviewed your final report on Chapter 6 of the ASHRAE Handbook: HVAC Systems and Equipment, 1996. It certainly is thorough. Although I don't feel confident in my ability to confirm all the algorithms contained in this work, it appears to be a work of merit.

I did not realize that you were working under such a significant grant. This explains why you were able to turn out so much information in such a short time. The vast majority of those on ASHRAE Technical Committees are volunteer or on a limited budget. We all stand to benefit from your work with the support of the Middle East Technical University.

The rearrangement of the chapter appears to be a positive improvement. For the most part I am in agreement with the revisions and modifications you have made and have few questions. The nomograph you have developed appears to address all of the issues which we have discussed. For ease of use, I feel it would be better to have the nomograph broken down into several graphs by surface and application. This would make the graph less intimidating for first time users. I also suggest that the range of AUST-t_a be extended to -8°F. This would allow the graph to be used for solariums and four season porches, etc. The mean water temperature should be stated in t_a-10, t_a-20, etc. to accommodate shops and warehouses. Although this makes the nomograph a little less "user friendly", it makes it more versatile.

Is there a particular reason why floor cooling has been omitted from the algorithm and nomograph? I realize that it is seldom possible to get 100% of the cooling load from a floor panel, but it can be a significant factor particularly in areas where the humidity is removed by other means or is not a factor. I have used floor cooling successfully, even in my own residence. I don't believe it should be overlooked.

Unfortunately, I do not have as much time available as I would like to study all the details of your report in depth. I have read through the report carefully and compared the results of examples with my own experience. The predictions are much closer to what I have seen in the field and therefore help to validate the procedure.

All in all, you are to be commended for your efforts on behalf of ASHRAE and the radiant panel industry. Should all of the algorithms be valid, as they appear to be, this work makes a big step in the direction of simplified yet more accurate procedures for designing radiant panel systems.

I look forward to further discussions with you and reviewing the responses of other committee members.

Sincerely yours,

Lawrence V. Drake
President

cc: Norman Buckley

Makina Mühendisliği Bölümü
ORTA DOĞU TEKNİK ÜNİVERSİTESİ
ANKARA

1956

15.6.1993

TSE
Standard Hazırlama Dairesi
İhtisas Kurulları Başkanlığı
Isı Özel Daimi Komitesi

Konu : - Döşemeden Isıtma Sistemleri - Terimler ve Tarifleri
- Döşemeden Isıtma Sistemleri - Projelendirme Kuralları

Adı geçen iki standard tasarısının birinci mütalaa yazısına verilen görüşler tarafımdan incelenmiş olup, değerlendirme raporum ekte arz olunmuştur. Görüşlerin redaksyonel mahiyette olmaları gözönünde tutularak, ikinci kez mütalaa istenmesine gerek görülmemiştir. Hazırlanan bu iki standard TÜBİTAK destekli MİSAG-12 projesinin özgün çalışmalarını yansıtmakta olup ISO standardları olarak ISO'ya teklif edilmesinin ülkemiz ve TSE adına çok yararlı olduğu görüşüme binaen konunun bu yönü ile de değerlendirilmesini gereği için saygılarımla arz ederim.

Prof. Dr. Birol Kılkış
ODTÜ Makina Müh. Bölümü

Ek : 1. Birinci mütalaa degerlendirme raporum
2. Konu ile ilgili uluslararası yayın

TÜRK STANDARDLARI ENSTİTÜSÜ
INSTITUT TURC DE NORMALISATION
TURKISH STANDARDS INSTITUTION

Necatibey Cad. 112, Bakanlıklar 06100 Ankara-Türkiye
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KALİTE YILI 1991

Dosya:B.02.2.TSE.0.11.03.07 16.04.92 10296

Sayın
Prof.Dr.Birol KILKIS
Orta Doğu Teknik Üniversitesi
Makina Mühendisliği Bölümü
ANKARA

İlgi: 9.3.1992 tarihli yazınız.

TS 2164 "Kalorifer Tesisatı Projelendirme Kuralları", adlı standard hakkındaki yazınız ve buna bağlı ek tetkik edilmiştir.

Kaynağın değişmesi üzerine adı geçen standarda revize edilmek üzere 1991-1992 iş programına alınmış olup halen üzerinde çalışılmaktadır.

Tasarı birinci mütalaa safhasına geldiğinde görüş için tarafınıza da gönderilecektir.

Bilgi edinilmesini rica ederim.

TÜRK STANDARDLARI ENSTİTÜSÜ
Mühendislik Hizmetleri
Hazırlık Grubu Başkanı

Polat OĞUN

12.05.92 12204

Dosya:B.02.2.TSE.0.11.03.05/4

Sayın Prof.Dr.Kemal GÜRÜZ
TÜBİTAK Başkanı
Atatürk Bulvarı No.221
06100 Kavaklıdere/ANKARA

Ülkemizde yapıların ısıtılmasında sarfedilen enerjinin yıllık enerji bilançosu içindeki payı yaklaşık %30 civarındadır. Dolayısı ile bu sektörde gerçekleştirilecek herhangi bir tasarrufun doğrudan milli enerji bilançomuzda ve çevre kirliliğinin azaltılması çalışmalarında müsbet tesirleri olacağı muhakkaktır. Yabancı ülkelerde oldukça yaygın bir şekilde tatbik edilmekte olan döşemeden ısıtma sistemi, uygun ve doğru bir projelendirme ile %40 dolayında enerji tasarrufu sağlayabilmektedir. Ülkemizde de yaygınlaşmaya başlayan bu ısıtma sistemine ait ilk Türk Standardlarının hazırlanması maksadı ile 13.8.1991 tarihinde Enstitümüz de bir Teknik Komite teşkil edilmiş bulunmaktadır.

Kurumunuzun desteklemekte olduğu "Panel Isıtma Sistemlerinin Modellemesi ve Bir Standard Tasarım Algoritmasının Geliştirilmesi" adlı MISAG-12 sayılı proje çerçevesinde çalışmalarını yürüten ve aynı zamanda teşekkül ettirdiğimiz Teknik Komitede konu raportörü olarak görev alan ODTÜ öğretim görevlisi Prof.Dr.Birol KILKIŞ bu konuda iki adet standard tasarısını hazırlayarak Enstitümüze teslim etmiş ve Teknik Komitede görüşülmeye başlanmıştır.

Bu standardlar sırası ile;

1-Döşemeden Isıtma Sistemleri-Terimler ve Tarifler,
2-Döşemeden Isıtma Sistemleri İçin Isıtma Yükünün Tayini ve Projelendirme Esasları'dır.

Bu standard hazırlık çalışmalarımızın bir TÜBİTAK Araştırma Projesi ile desteklenmesini ve bu araştırmaların özellikle ülkemiz için öneme haiz bir konuda Türk Standardlarına doğrudan yansımasını taktirle karşılamaktayız. Bu çalışmalar neticesinde ortaya çıkacak Türk Standardları, önemli bir konuda, büyük bir boşluğu dolduracaktır. Hazırlanmakta olan standardların, bu konuda sadece bir kaç yabancı ülkede mevcut olan standardlardan daha kapsamlı ve ileri seviyede olacağı izlenimi bizleri şimdiden ayrıca memnun etmektedir.

(EK-2 Devam)

TÜBITAK'ın desteklediği projelerin uygulamaya geçmesinde ilgili kurum ve kuruluşların işbirliğine örnek teşkil edecek bu çalışma nedeni ile teşekkür eder, bu ve buna benzer çalışmalarda işbirliğinin devamı temennisi ile saygılar sunarım.

TÜRK STANDARDLARI ENSTITÜSÜ
Başkanı V.

Prof.Dr.Tarık SOMER

Bilgi İçin: TÜBITAK, MISAG Yürütme
Komitesi Sekreterliği

23./4./1992 Rap.:A YENEN
28./04/1992 Müdür:A MUTLU
29./04/1992 Dai.Bşk.Y.:S DURGUT
../4./1992 Gn.Sek.Yrd.:İ ATIKLER
../../1992 Genel Sekreter:Ö PEKER

AY/FG 24.4.1992

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ASSET 15/3

bstracts of Selected Solar Energy Technology

olume 15, Number 3 1993

阳能 SOLAR ENERGY СОЛНЕЧНАЯ ЭНЕРГИЯ सौर ऊर्जा ENERGÍA SOLAR সৌর শক্তি
ONNENENERGIE ENERGIA SOLAR الطاقة الشمسية 太陽エネルギー ENERGI MATAHARI
NERGIE SOLAIRE सूर्य शक्ति ENERGIA SOLARE شمسی توانائی DAYANING SURYA 태양에너지
நாயிற்றியலான ஆற்றல் సూర్య శక్తి لمر دی طاقت دھوپ GÜNEŞ ENERJİSİ СОНЯЧНА ЕНЕРГІЯ
ĂNG LƯỢNG MẶT TRỜI ENERGIA SŁONECZNA સૂર્ય શક્તિ СОЛАРНА ЕНЕРГИЈА ಸೂರ್ಯ ಶಕ್ತಿ
انرژی خورشید സൂര്യോർജ്ജം KARFIN HASKEN RANA ସୂର୍ଯ୍ୟ ଶକ୍ତି NISHATI YA JUA နေစွမ်းအင်
พลังงานแสงอาทิตย์ ENERHIYA NG ARAW ENERGIE SOLARĂ SOLENERGI ZONNE ENERGIE
ਰਜ ਦੀ ਸ਼ਕਤੀ دلمر انرژی SLUNEČNÍ ENERGIE ÉNESI PANON POÉ قوت سورج NAPENERGIA
ENAGA SURIA AGBARA ÕRUN SEMBE DIGA NAANGE IKE ANYANWU ພະລັງງານແສງຕາເວັນ
MANDLA ELANGA සූර්ය ශක්ති බලය КУЁН ЭНЕРГИЯСИ انرژیی گئون HΛΙΑΚΗ ΕΝΕΡΓΕΙΑ
NERHIYA SA ADLAW سج جي قوت የፀሐይ ፡ ኃይል СЛЪНЧЕВА ЕНЕРГИЯ SUNČEVA ENERGIJA
NÈGI SOLÈY ሀሃይ ፡ ሓይሊ KOWADDA PANASSA AREH HERIN'NY MASOANDRO
NTI KALLPA INGUFU Z'IZUBA ཉི་མའི་ནུས་པ། КУН ЭНЕРГИЯСЫ OWIA AHOƆDEN DAI5YAŊ2 NƏŊ2
AURINKOENERGIA ENERHIA ITI INIT TIYA I NTANGU ENERGI DEL SOLELH SLNEČNÁ ENERGIA
ABARTA QORAXDA KUYAX ENERGIYISI ENERGJI DIELLORE ԱՐԵՎԱՅԻՆ ԷՆԵՐԳԻԱ
КҮНЭШ ЕНЕРКИЈАСЫ მზის ენერგია HXO BBU WE ENERHIYA SANG ADLAW NGUYA YA MOYI
TANAGO MATOARI ফিং চাওা জাক የፀሐይ ፡ ኃየሊ PAYU MATAN-AI روج تاب KILE BARKA
ENERHIA HALI SA ALDAW AWATANGENNA MATANNA ESSOE ODUDU UTIN ŊDO ZAZA
AMAANYI G'ENJUBA אנרגית השמש DONO KAUSU BE افتابک قوت هیزی هه تاوی
HINYA WA RIUA SAULES ENERGIJA BUKOLE BWA MUNYA UTAÑA UÃTUA TAIYANG NENG
NGUYA EA WANE WĨNTOOG TUULEM MPHAMVU YA DZUWA SIMBA RE ZUWA
MATLA A LETSATSI MOGOTE WA LETSATSI CUARAJHY MBARETÉ KATAN U JANTE BI

The United Nations University

teri

Tata Energy Research Institute

Bu tebliğ, EK-5,sıra no:14 de verilmi

the locally available materials and fabrication techniques. Fanger predicted mean vote—as a function of dry bulb temperature, wet bulb temperature, air-velocity, and mean radiant temperature—was calculated and recorded continuously by a data acquisition system. These values have been averaged to evaluate hourly comfort conditions in various zones of the test house. These were compared with the ambient conditions monitored simultaneously at the outdoor weather station to evaluate the potential of direct convective cooling as an energy saving technique. 6 refs.

0258/93 PANEL COOLING AND HEATING OF BUILDINGS USING SOLAR ENERGY. Kilkis, B. (Department of Mechanical Engineering, Middle East Technical University, Ankara, Turkey). (*Solar Energy in the 1990s: Presented at the Winter Annual Meeting of the American Society of Mechanical Engineers*, Dallas, Texas, USA, 25-30 November 1990). New York: The American Society of Mechanical Engineers. 7 pp.

This article reports on the basic theory and design principles for both the panel heating and panel cooling applications. A design study that has been carried out at Middle East Technical University for an existing solar house is presented. A single effect lithium-bromide water solar absorption unit rated at 3.5 kW to effectively cool the two storey solar house of 96.6 m^2 area was found to require 42.2 m^2 of flat plate collectors. For winter heating, floor heating panels, electrical water heating back-up and additional solar collectors were found to be necessary. 7 refs.

0259/93 PANEL COOLING OF BUILDINGS USING SOLAR ENERGY ABSORPTION SYSTEMS. Kilkis, B. (Middle East Technical University, Ankara, Turkey). (*Paper Presented at the Seminar on Solar Power Systems Alushta*, Yalta, USSR, 22-26 April 1991).

With respect to solar energy in buildings, it is abundant in summer months when there is no heating load, but instead, cooling is required. This paper presents panel cooling with solar absorption systems including the basic theory and a case-study carried out at Middle East Technical University. Advantages and technical details together with general economical features are also included and comparisons with conventional systems are presented. The case-study has revealed that solar system driven ceiling cooling is a serious alternative to conventional systems. The same piping system can be utilized in winter if the cooling circuit is by-passed. Comparing with chiller/fan-coil system and boiler/service water system, the system presented offered about 15% savings in initial investment cost, 25% energy load, and about 60% in fuel and electrical bills. 5 refs.

0260/93 RADIANT CEILING COOLING WITH SOLAR ENERGY: FUNDAMENTALS, MODELING AND A CASE DESIGN. Kilkis, B. (Department of Mechanical Engineering, Middle East Technical University, Ankara, Turkey). (*Proceedings of the International Symposium on International Efforts in Radiant Cooling*, ASHRAE Annual Meeting, Denver, Colorado, USA, 26-30 June 1993)

The paper provides the basic fundamentals and the design algorithm of radiant panel cooling systems with special emphasis given to solar energy utilization and to demonstrate its feasibility with a case design. A steady state heat transfer model was developed in order to determine the heat diffusion in a cooled ceiling slab and to predict its interactive performance in the conditioned space. More suitable tie-in features of the radiant ceiling cooling system with absorption type heat pumps and solar collectors were identified and explained in terms of the first and second laws of thermodynamics. This analysis indicated an increase in both heating and cooling coefficients of performance of the absorption heat pump and a substantial decrease in the total exergy loss in addition to an increase in the flat plate solar collector efficiency. The design analysis indicated that with radiant ceiling cooling the peak cooling load will be saved by about 25%. 27 refs.

0261/93 THE USE OF GRID-GENERATED TURBULENCE TO IMPROVE HEAT TRANSFER IN PASSIVE SOLAR SYSTEMS. Maad, RB; Belghith, A. (Laboratoire de Thermique, Faculte des Sciences de Tunis, Department de Physique, 1060 Tunis, Belvedere, Tunisia). *Renewable Energy—An International Journal*. 2(3), 1992, 333-6.

The paper investigates the natural convective flow between two heated vertical plates for designing a solar chimney of greatest efficiency for drying agricultural products. The equations of continuity, momentum and energy are solved by a finite difference method, taking into account the variation of the fluid properties with temperature and the pressure drop due to flow acceleration at the chemical entrance. Good agreement between the experimental results and the theoretical predictions was obtained. The average Nusselt number at the exit was calculated. The experimental results show that flow was no longer laminar. The experiments carried out showed that the use of grid generated turbulence mounted at the entrance effects the flow structure and intensifies the average heat transfer coefficient. 9 refs.

0262/93 THERMAL PERFORMANCE OF LIVED-IN PASSIVE SOLAR BUILDINGS: 2—SCHOOL RETROFIT AT

TÜ SEMİNER ÖZETİ

PANEL ISITMA VE SOĞUTMANIN ENERJİ TASARRUFUNDAKİ YERİ: TEMEL TEORİ ve UYGULAMA ESASLARI

Prof.Dr.Birol KILKIŞ
ODTÜ Mak.Müh.Bl.
ASHRAE Teknik Komite Üyesi

anel ısıtma ve soğutma,yeterli bir tasarım analizi ve eniyilemesi apılması kaydı ile uygulamada önemli ölçüde enerji tasarrufu ağlayabilmektedir. Ayrıca düşük yoğunluktaki alternatif ve atık nerji kaynakları ile çalışabilme özelliği ile bu kaynakların etkin e ekonomik kullanımına imkan sağlamaktadır. Isı pompası kullanıldığı amanlarda ise ısıtma ve soğutma tesir katsayıları artmaktadır. eminerde,önce ısıtma ve soğutma yüklerindeki azalmanın temellerine ğilinilmekte, panel tasarımı için geliştirilen bir analitik model anıtılmakta ve örnek çözümler sunulmaktadır. Havadan suya ısı ompalı bir ısıtma sistemi ile güneş enerjisi ile çalışan bir soğutma isteminin prensipleri çerçevesinde panel ısıtma ve soğutmanın vantajları tartışılmaktadır. Ayrıca konu ile ilgili TSE,TÜBİTAK ve SHRAE çalışmaları tanıtılmaktadır.

Buckley Associates

7700 Briarwood Drive
Crestwood, KY 40014
(502) 241-5925

NORMAN A. BUCKLEY, P.E.
Consulting Engineer

MIL BUCKLEY, CDP, CFPIM
Manufacturing Consultant

November 1, 1993

Dear Birol:

I received your letter of October 6 relating to Chapters 6 & 15. Although related, this does not address the need for coverage of both Infrared and Panel Systems in Chapter 51.

Your observation that "panel" is already used to identify Low Temperature Systems prompted a closer look. After much searching, it appears that this may be the solution to the problem.

Looking forward to your feedback on the enclosed proposed solution.

Sincerely

Norman A Buckley

Norman A. Buckley, PE

CKLEY ASSOCIATES
Briarwood Drive
wood, KY 40014
241-5925

MAN A. BUCKLEY, P.E.
ulting Engineer

BUCKLEY, CDP, CFPIM
ifacturing Consultant

March 10, 1993

TC 6.5 Handbook Subcommittee

ject: April 1

our meeting in Chicago April 1 was picked as the date to submit comments review materials for the Handbooks. You will be addressing Chapter 48 of Applications Handbook and Chapters 6 and 15 of the Systems and Equipment idbook.

ery important area, that we did not discuss, is improper placement of erials and duplication of materials in these Chapters. Due to reshuffling Application, Systems and Equipment materials in the Handbooks, several rs ago, some materials are now improperly located. April 1 is approaching is still the deadline. If you have a reason you can not meet this date ase contact me.

Chapters in the Handbooks are, in my opinion, ASHRAE's most powerful tool. ase get your contribution in the mail by April 1.

Sincerely,

Norm

Norman A. Buckley, PE

Bud -
Received your February 25 input.

Norm

BUCKLEY ASSOCIATES
7700 Briarwood Drive
Crestwood, KY 40014
(502) 241-5925

NORMAN A. BUCKLEY, P.E.
Consulting Engineer

MIL BUCKLEY, CDP, CFPIM
Manufacturing Consultant

February 1, 1993

Dear Dr. Kilkis:

Enclosed is a copy of the proposed revision of Chapter 48, for the '95 Applications Handbook, and Minutes of the Handbook Subcommittee Meeting in Chicago. Note that there are two pages of your suggested changes, that were attached at the end of CH 48, with comments by the subcommittee.

The minutes have a couple errors that should be noted:

Copies of CH 2 through 7 distributed include changes suggested by TC 6.5 Subcommittee Review members that were submitted to TC responsible for each chapter.

Chapters to be reviewed for '96 Systems and Equipment Handbook should be Chapters 6 and 15.

Looking forward to your input.

Sincerely

Norman A. Buckley

Norman A. Buckley PE

B

SHRAE American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
1 Tullie Circle, NE • Atlanta, Georgia 30329-2305 ☎ 404-636-8400 • Tlx 705343 ASHRAE • Fax 404-321-5478

es H. Norman
ager of Technical Services

M E M O R A N D U M

To: Mr Birol I Kilkis Date: April 24, 1992

From: Jim Norman

Subject: Appointment to Technical Committee 6.5, Radiant Space Heating and Cooling

We are pleased to confirm your appointment to TC 6.5, Radiant Space Heating and Cooling, as shown on the enclosed roster.

Your appointment begins at the close of the Annual (summer) Meeting, 1992, (approximately July 1) and will be effective through the Annual Meeting, 1993 (approximately June 30) and is subject to annual appointment (up to a maximum of four consecutive years for voting members). If you are new to the committee, you are invited to attend the June Meeting in Baltimore as a visitor.

Also enclosed is a copy of the scopes for the committees in Section 6. The acceptance of voting membership on a TC, TG or TRG implies a pledge of faithful attendence at committee meetings (usually a minimum of two per year in conjunction with the Winter and Annual Meetings) and meticulous attention to correspondence. The acceptance of corresponding membership implies participation in committee activities through correspondence or in-person involvement, although corresponding members may not vote. Corresponding members may be considered for future voting membership.

The committee roster includes the following information: position on the committee, member number, chapter number, name and address, starting year of service in the position indicated, and expected ending year of service. ASHRAE member code is shown between the member number and chapter number. "AM" "AS" "GS" and "PA" are ASHRAE member codes. Others indicate non-ASHRAE member. Because headquarters records are computerized, it is necessary to use the address and other information as recorded on each member's "master" record, which is used for all regular society correspondence (journals, handbooks, etc.). If you have not completed and returned a Membership Statistical Data form, please request one.

ASHRAE American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

1791 Tullie Circle, NE • Atlanta, Georgia 30329 ☎(404) 636-8400 • Fax (404) 321-5478

M E M O R A N D U M

To: Mr Birol I Kilkis   Date: April 29, 1993

From: Jim Norman

Subject: Appointment to Technical Committee, 6.4, In Space Convection Heating

We are pleased to confirm your appointment to TC 6.4, In Space Convection Heating, as shown on the enclosed roster. Your position on the committee as (voting) member, corresponding member or international member is shown on the line above your member number on the roster.

Your appointment begins at the close of the Annual (summer) Meeting, 1993, (approximately July 1) and will be effective through the Annual Meeting, 1994 (approximately June 30) and is subject to annual appointment (up to a maximum of four consecutive years for voting members). If you are new to the committee, you are invited to attend the June Meeting in Denver as a visitor.

Also enclosed is a copy of the scopes for the committees in Section 6. The acceptance of voting membership on a TC, TG or TRG implies a pledge of faithful attendence at committee meetings (usually a minimum of two per year in conjunction with the Winter and Annual Meetings) and meticulous attention to correspondence. The acceptance of corresponding membership implies participation in committee activities through correspondence or in-person involvement, although corresponding members may not vote. Corresponding members may be considered for future voting membership.

ASHRAE American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

1791 Tullie Circle, NE • Atlanta, Georgia 30329 ☎(404) 636-8400 • Fax (404) 321-5478

MEMORANDUM

To: Mr Birol I Kilkis Date: April 27, 1993

From: Jim Norman

Subject: Appointment to Technical Committee 4.1, Load Calculation Data and Procedures

We are pleased to confirm your appointment to TC 4.1, Load Calculation Data and Procedures, as shown on the enclosed roster. Your position on the committee as (voting) member, corresponding member or international member is shown on the line above your member number on the roster.

Your appointment begins at the close of the Annual (summer) Meeting, 1993, (approximately July 1) and will be effective through the Annual Meeting, 1994 (approximately June 30) and is subject to annual appointment (up to a maximum of four consecutive years for voting members). If you are new to the committee, you are invited to attend the June Meeting in Denver as a visitor.

Also enclosed is a copy of the scopes for the committees in Section 4. The acceptance of voting membership on a TC, TG or TRG implies a pledge of faithful attendence at committee meetings (usually a minimum of two per year in conjunction with the Winter and Annual Meetings) and meticulous attention to correspondence. The acceptance of corresponding membership implies participation in committee activities through correspondence or in-person involvement, although corresponding members may not vote. Corresponding members may be considered for future voting membership.

ASHRAE American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
1791 Tullie Circle, NE • Atlanta, Georgia 30329 ☎(404) 636-8400 • Fax (404) 321-5478

M E M O R A N D U M

To: Mr Birol I Kilkis Date: April 29, 1993

From: Jim Norman

Subject: Appointment to Technical Committee, 6.1, Hydronic and Steam Equipment and Systems

We are pleased to confirm your appointment to TC 6.1, Hydronic and Steam Equipment and Systems, as shown on the enclosed roster. Your position on the committee as (voting) member, corresponding member or international member is shown on the line above your member number on the roster.

Your appointment begins at the close of the Annual (summer) Meeting, 1993, (approximately July 1) and will be effective through the Annual Meeting, 1994 (approximately June 30) and is subject to annual appointment (up to a maximum of four consecutive years for voting members). If you are new to the committee, you are invited to attend the June Meeting in Denver as a visitor.

Also enclosed is a copy of the scopes for the committees in Section 6. The acceptance of voting membership on a TC, TG or TRG implies a pledge of faithful attendence at committee meetings (usually a minimum of two per year in conjunction with the Winter and Annual Meetings) and meticulous attention to correspondence. The acceptance of corresponding membership implies participation in committee activities through correspondence or in-person involvement, although corresponding members may not vote. Corresponding members may be considered for future voting membership.

Mechanical Engineering Department
MIDDLE EAST TECHNICAL UNIVERSITY
ANKARA, TURKEY

1956

TÜBITAK

MISAG Yürütme Komitesi Sekreterliği

Ankara 28/5/1993

ayı: MİSAG-12-93/7

onu: Tebliğ ile Konferansa Katılma

7 -29 Haziran Tarihleri arasında Denver de Yapılacak olan Annual ASHRAE Meeting de ntl.Conf. on Panel Cooling Simpozyumunda sunulmak üzere kabul olunan tebliğimi sunmak ve gen rojemiz çerçevesinde ASHRAE Teknik komitelerine katılmak üzere anılan toplantıya atılmak istemekteyim. En ucuz hava yolu bağlantısısın THY ile yapılabildiğini de tesbit tmiş bulunmaktayım. Projemizin ilgili kaleminden gidişdönüş biletimin ve 4 günlük harcırahımın karşılanabilmesi ususunu saygılarımla onayınıza arz ederim. Tebliğ sahibi olduğum için kaydiye ücreti öz konusu olmayıp,diğer giderlerimi de başka kaynaklardan temin edeceğim.

İlgilerinize arz ederim.

aygılarımla

rof.Dr.Birol KILKIŞ

roje Yöneticisi

:1

OHIO
COMPUTER
AIDED
ENGINEERING
1612 Georgetown Road
Hudson, Ohio 44236
(AKR) 216/650-2057
(CLE) 216/656-1029
(FAX) 216/656-4024

July 22, 1993

Dr. Birol Kilkis
Middle East Technical University
Mechanical Engineering Department
06531 Ankara,
TURKEY

Dear Dr. Kilkis:

I am pleased to receive your abstract entitled "Simulation of Radiant Floor Heated Room" for the 1994 ANSYS Conference. Actual acceptance will be based on the placement of an abstract in a session and the successful review of a draft of the complete paper.

First drafts are due September 30, 1993. If you have any questions or need more information about our Conference, please contact me.

Once again, thank you for your interest in the 1994 ANSYS Conference.

Sincerely,
OHIO COMPUTER AIDED
ENGINEERING, INC.

James A. Knopp

James A. Knopp
Session Developer

JAK/kgg

Erwin G. Hansen
Le Danae
80, avenue du Bois de Cythère
06000 Nice, France

17 November 1992

Prof. Dr. Birol Kilkis

Dear Professor Kilkis,

Returning from an extended trip I find your letter of 10/1/92 forwarded to me by Syska & Hennessy in New York.

I appreciate your comments concerning my book. You are of course entirely right in your observation that the square sign is missing from the otherwise unnecessary brackets in the denominator of the formula on page 53. I thank you for bringing this typo to my attention. It had slipped me by.

Unfortunately the book abounds with typos, one even worse than this one. I shall add this one to the lengthy list of corrections that I prepared and sent to McGraw-Hill some three years ago.

At that time I could not budge my editor to issue an errata list, and he was certainly not contemplating a revision. But since sales have now passed the 4500 mark and the book seems to be selling steadily, even if moderately, at about 400 copies a year, he may more more receptive to the thought of a revised copy now.

I would welcome a revision. The book was written in 1980, before the advent of the widespread use of personal computers in engineer's offices. Thus the section written for programmable calculators, so dear to McGraw-Hill at the time, is completely out of date. It should be updated.

If McGraw-Hill change their mind I would gladly add a section on the flow in radiant panels and would take you up on your offer to supply information with the appropriate credit given. However, as I am sure you realize, my book treats only the performance on the water side. It would not get into the radiant heat emission aspect.

I thank you for the material on radiant panels which accompanied your letter. I shall read them with interest and comment on them in due course.

To my enormous surprise I have just been elevated to the grade of Fellow by ASHRAE. Although I have not been to one in 20 years, I will now have to go to the Winter Meeting in Chicago in January '93 to receive the honor. If you are attending we may then discuss our mutual interest viva voce. Otherwise you may reach me at my Winter quarters in Florida which I expect to take up in mid-December. My coordinates there are

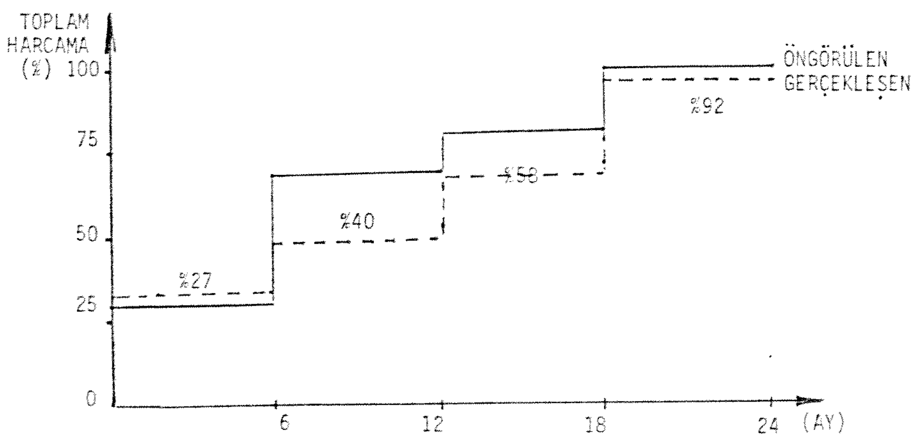
250 Carolina Avenue, Apt 405, Winter Park, FL 32789
Tel.: 407-629-1908

Cordially,

9. MALİ RAPOR

9.1. Genel Bilgiler ve Demirbaşlar

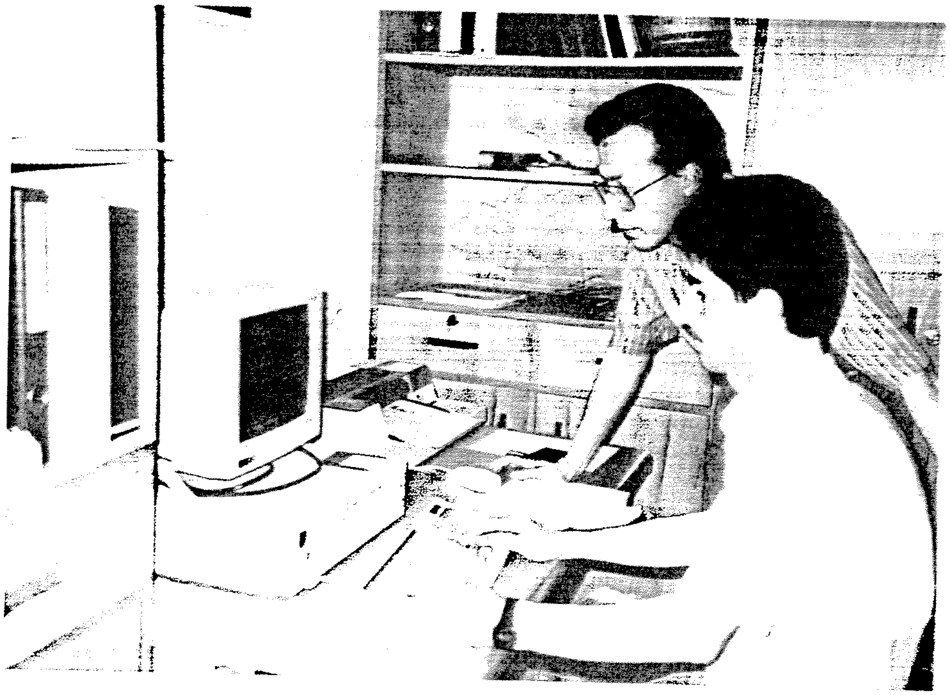
TÜBİTAK proje harcamaları için öngörülen ve gerçekleşen akış mukayesesi Şekil 2 'de gösterilmiştir. Bu şekilden de görüleceği üzere harcamalar genel plana uygun seyretmiştir. Bu arada kalemler arası bazı transferler TÜBİTAK onayları ile yapılmıştır. Şu anda son telif ödemeleri de çıktıktan sonra 13 Milyon 016 869 TL bulunmaktadır. Buna karşılık Proje Yürütücüsünün enerji 2000 ve ASHRAE 1993 toplantılarına tebliğle katılma harcamalarına ilişkin uçak bileti talepleri henüz sonuçlandırılmamıştır. Bu ödemeler de yapıldığı taktirde, bütçe tam denk olarak kapatılmış olacaktır.



Şekil 2. TÜBİTAK Bütçesi Harcama Akışı.

Yukarıdaki şekildeki gerçekleşen değerlere, o dönemin telif ödemeleri de dahil edilmiştir. Tahsil edilemeyen uçak biletleri hariç olmak üzere tüm harcamalar toplamı 24 ay sonunda % 92 olarak gerçekleşmiştir.

TÜBİTAK bütçesinden alınan bilgisayar donanımı ODTÜ-BİLTİR Merkezinde G-11 numaralı odada iki yıl süre ile kullanılmıştır. Proje odası olarak kullanılan odadan ve çalışmalardan bir görüntü Şekil 3'de gösterilmiştir. Ayrıca ODTÜ-BİLTİR Merkezindeki genel kullanım alanlarından ve bilgisayarlarından da yararlanılmıştır.



Şekil 3. Proje Çalışmalarından Bir Görünüş.

Demirbaşa alınan aşağıdaki donanım halen ODTÜ-BİLTİR Merkezindedir.

Demirbaş Sıra No	Donanım	Adet
1	40 MB HD-PC 286 Bilgisayar	1
2	Mono VGA Ekran	1
3	Mouse	1
4	Disk sürücü, 1.2 K.	1
5	HP A4/A3 Çizici	1

Bu donanımın ODTÜ-BİLTİR Merkezinde kalması için 22.6.1993 tarih ve MİSAG-12-93/8 sayılı yazı ile TÜBİTAK'a başvuruda bulunulmuştur.

9.2. Bütçe Harcamalarının Seyri ve Ödemeler

Çizelge 1'de ise kalemlere ve dönemlere göre harcama akışı verilmiştir.

Çizelge 1. Harcama Akışı

Dönem	Telif	Personel	Teçhizat ve Sabit Y.	Sarf Malzemesi	Seyahat ve Nakliye	Hizmet Alımı	Bilgi İşlem Harcamaları	Kırtasiye ve Basım	Diğer	Toplam
1	13000	2484	14411.4	2101.5	11778.3	-	-	-	-	41484.2
2	18000	2484	-	-	-	-	-	-	-	20484
3	19200	1242	-	2000	-	3000	-	-	-	25442
4	25200	-	-	-	913	185143	931.8	1688.7	435	47682.8
Sonuç Raporu	-	-	-	-	300	-	3500	2700	-	6500
									TOPLAM	142093.-

Destekleyen diğer kuruluşun katkılarının yaklaşık dökümü ise şöyledir:

1 adet 4 ay adam (Araştırmacı)	:	4 500 $	
2 adet 24 ay adam (Bilgisayar programcısı	:	20 000 $	(yarı zaman)
Araştırma giderleri için diğer personel ödemesi	:	30 000 $	
Bilgisayar donanımı	:	5 000 $	
Sekreterya	:	2 500 $	
Literatür tarama ve yayın satınalma	:	2 500 $	
Deney düzeneği için harcamalar	:	10 000 $	
Haberleşme, telefon giderleri	:	2 000 $	
Kırtasiye giderleri	:	1 000 $	
Hazır yazılım satınalma	:	1 000 $	
Kongre ve benzeri seyahatler	:	10 000 $	
TOPLAM	:	88 500 $	

Bunun TL karşılığı yaklaşık 485 Milyon TL etmektedir.

9.3. Değerlendirme

Proje harcamaları öngörülen düzeyde ve akışta gerçekleşmiş, kongreye katılım, toplantılara ve seminerlere katılım, tebliğlerin yazılması gibi ve benzeri değer çalışmalara ilişkin giderlerin büyük bölümü destekleyen kuruluş tarafından karşılanmış, bütçe faslında yeterli ödenek olmasına ve zamanında talep yapılmasına karşın, Ankara-Şikago ve Ankara-Denver-Ankara uçak giderleri için projeden herhangi bir ödeme yapılmamıştır.

10. TEKNİK RAPOR

10.1. Giriş

Bu bölümde, MİSAG-12 projesi kapsamında yapılan çalışmalar izah edilerek, problemin tanıtımı ve temel bilgiler verilmekte, gerektiğinde Ek-5'deki yayınlara da atıflar yapılarak çözümler literatürdeki çalışmalara da değinilerek tanıtılmaktadır. Panel ısıtma ve soğutma konusunda şimdiye kadar yapılan çalışma ve değerlendirmeleri iki sınıfta toplamak mümkündür. Bunlar:

a) Deneysel çalışmalar: Bu çalışmalarda ağırlık panel ve oda performansı ve ısıl konfor konularında olmak üzere, 1930'lu yıllardan başlayarak incelenmiş, performansa etken temel parametrelerin etkileri açıklanmaya çalışılmıştır. Sartain ve diğerleri (1953) ilk kez sistematik olarak döşeme panel performansını incelemiştir.

b) Teorik çalışmalar: Isıtılan veya soğutulan paneldeki ısı transferi matematiksel olarak ilk kez Faxen (1937) tarafından etüd edilmiş, Rydberg ve Huber (1955) panel içersindeki ısı transferine ilişkin Laplace diferansyel denklemini hem homojen bir panel bloğu hem de çok tabakalı bir ısıtılan levhadaki sıcaklık alanını tavandan ısıtma özelinde Fourier serisi açılımı ile çözmüşlerdir.

Panelin kanatçık modeli ile incelenmesinin esasları Kolemar ve Liese (1957) tarafından öne sürülmüş ve tek boyutlu kanatçık modeli ile probleme yaklaşılmıştır.

1970'li yıllarda ise önce sonlu farklar ve daha sonraları da sonlu elemanlar yöntemi ile hem panel içersindeki ısı transferi hem de ısıtılan veya soğutulan bir odanın termofiziği incelenmiştir. Zhang ve Pate (1986) sonlu farklar yöntemi ile ısıtılan bir tavan içersindeki ısı transferini çözmüşlerdir. Yine Zhang ve Pate (1987) ısıtılan paneldeki ısı transferi için yarı-analitik bir yöntem geliştirmiştir.

Isıtılan ve soğutulan bir odadaki termofiziksel koşullar sayısal ve analitik yaklaşımlarla bir çok araştırmacı tarafından panel performansından bağımsız olarak incelenmesine rağmen ısıtma ve soğutma yüklerinin klasik sistemlere göre farklı olacağına Zmeureaunu ve diğerleri (1988) dışında şimdiye kadar hiç bir araştırmacı işaret etmemiştir.

Teorik çalışmalarda ise panel, ısıtılan/soğutulan oda, çevre sentezi yapılmamış, sadece bu elemanlardan birisi örneğin panel incelenmiştir.

Bu çalışmada ise panel, ısıtılan/soğutulan oda, çevre sentezi gözetilerek panel, oda termofiziği üzerine analitik ve sayısal çalışmalar yapılmış, ilk kez ısı yükü hesabı için özel bir algoritma geliştirilmiş, elde edilen modeller bilgisayar ortamına aktarılarak tasarım ve analiz programları geliştirilmiştir.

10.2. Problemin Tanımı

Probleme döşemeden ısıtma özelinde bakıldığında aşağıdaki tanımların yapılması gerekir:

a) Panel/oda ısı transferi,

b) Oda/çevre ısı transferi,

c) Panel içi ısı transferi,

d) Oda içi termofiziği,

e) Panelden çevreye olan ısı kayıpları/kazançları.

Şekil 4'de problemin genel modeli görülmektedir. Bu modelde q_Y, birim alanda panel yüzeyindeki ısı transferini simgelemektedir. Bu ısı transferinde doğal taşınım ve ışıma ısı transferi hakimdir:

$$q_Y = q_{yr} + q_{yc} \tag{1}$$

$$= A \cdot (T_p - AUST) + B \cdot (T_p - T_a)^c \tag{2}$$

A, linearize edilmiş ışıma ısı transfer katsayısıdır. B ve C ise doğal taşınım ısı transferi katsayıları olup, literatürde deneysel çalışmalar ışığı altında oldukça yerleşmiş ve kabul görmüş değerler mevcuttur. T_p ise boru aralığına ve diğer panel ve oda özelliklerine ve ısı transfer miktarına bağlı olarak yüzeyde belirli bir salınıma sahip olup aslen sabit değildir. AUST ise alan ağırlıklı olmak üzere ısıtılmayan/soğutulmayan yüzeylerin ortalama iç yüzey sıcaklığıdır. T_a ise oda sıcaklığı olup aslen bu da sabit değildir. Oda içersindeki basınç (P), hava hızı (V) ve hava sıcaklığı (T), panel yüzeyindeki B ve c katsayılarını, duvar yüzeylerindeki film katsayılarını (α_i) ve doğal infiltrasyon miktarını etkiler. Oda ısı kayıpları/kazançları ise dış ve çevredeki diğer iç ortam koşullarına, duvar, kapı pencere ısıl ve hidrolik özelliklerine bağlıdır. Denklem 2 'den görüleceği üzere panel yüzeyindeki ısı transferi (q_Y) oda termofiziğine bağlıdır. Buna mukabil ısı kayıp ve kazançlarını etkileyen çevre koşullarına bağlı olarak T, P ve V 'deki değişimler (oda termofiziği) de q_Y 'ye ve T_p 'ye bağlıdır. Örneğin belirli bir ortalama su sıcaklığına (T_s) haiz döşemeden ısıtılan bir odada, dış koşulların iyileşmesi sonucu önce T_a ve .... sıcaklıkları artar. Denklem 2 'ye göre bunun sonucu olarak q_Y azalır. Diğer bir deyişle panel belirli bir düzeyde ısı transferini oda koşullarına bağlı olarak ayarlamış olur.

Görüldüğü üzere oda termofiziği ve panel performansı birlikte incelenmelidir.

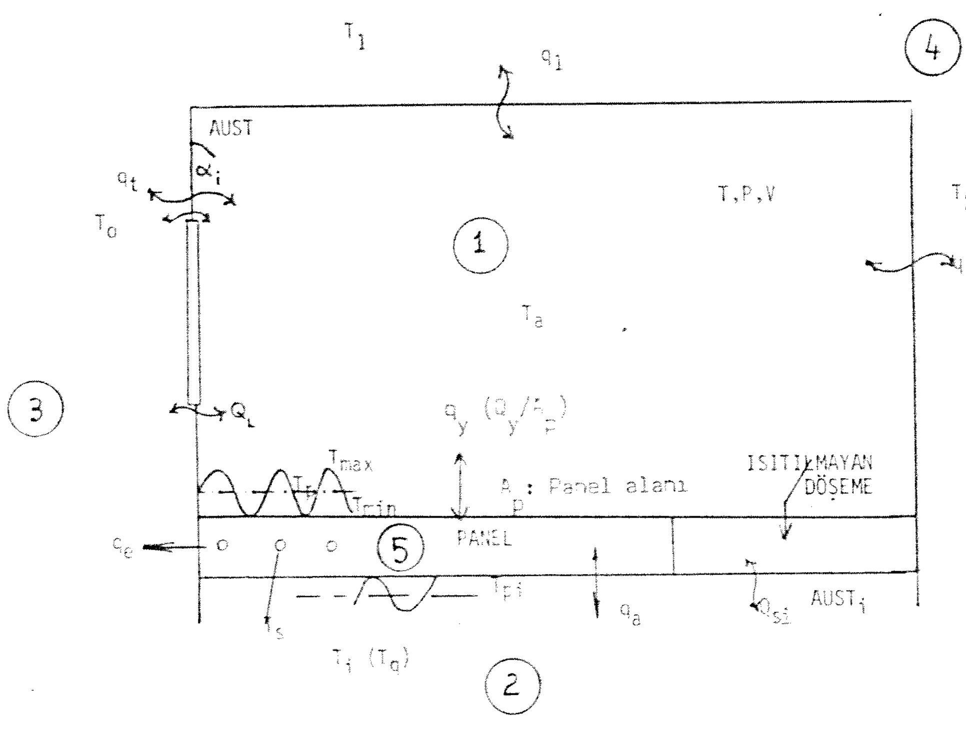
Panel içindeki ısı transferi ise ısı iletimi niteliğinde olup q_Y 'yi etkileyen sınır şartları ile birlikte çözülmelidir. Ayrıca yüzeydeki sıcaklık dağılımı da gözönünde tutulmalıdır. Panelden olan ısı kayıpları q_a ve q_e (veya soğutmada kazançlar) çevre koşullarına bağlıdır. Şekil 4 'de, eğer panelin altında başka bir oda varsa, aynı panel bu komşu oda nazarında bir ısıtılan tavan paneli niteliğindedir. O taktirde denklem 2 'ye benzer bir şekilde:

$$q_a = A' . (T_{pi} - AUST_i) + B' . (T_{pi} - T_i)^{c'} \quad (3)$$

Toprak temaslı bir döşemede ise q_a , T_s 'ye ve yalıtıma bağlı olan bir ısı iletimi niteliğindedir. Isı transferinde toprak sıcaklığı T_g , T_i 'nin yerini alır. q_e ise genelde ısı iletimi ve/veya ısı taşınımını içerir.

Şekil 4 'den görüldüğü üzere problemde gözönünde tutulmas gereken 5 ortam vardır:

1. Oda,
2. Panel arkasındaki ortam,
3. Dış ortam (var ise),
4. Diğer iç çevre,
5. Panel içi.



Şekil 4. Problemin Genel Modeli.

Sistemin analizi için 2, 3, 4 ve 5 yüzeylerindeki sınır şartları belirlenmeli ve 5, 1, 5/1 ilişkisi, 1/3, 1/4 ve 5/2 ilişkileri çözülmelidir.

Tam bir çözüm için bu şekilde bir sentez gereklidir. Ancak burada cevaplanması gereken soru, bu kapsamdaki bir sentezin mühendislik hesaplarında gerekli olup olmadığıdır. Örneğin 5/2 ilişkisi içersinde panel performansının T_i 'ye ve q_a 'ya ne kadar bağımlı olduğu tartışılabilir. Bu noktadan hareketle sistemin basitleştirilmesi - istenirse - mümkündür görüşü ile probleme genel sentez açısından yaklaşılmış, geliştirilen analitik hesap algoritmasında belirli basitleştirmeler yapılmıştır. Sayısal analizlerde ise genel sentez prensibine sadık kalınmıştır.

10.3. Teori

10.3.1. Simgeler*

a	Sızdırganlık katsayısı, $m^3/h\ m$
A	Panel yüzeyinden-odaya ışıma ısı transfer katsayısı, W/m^2K
A'	Panel arkasından-arkadaki hacme ışıma ısı transfer katsayısı W/m^2K
A_p	Panel yüzey alanı, m^2
A_i	Normal ölçülerdeki bir insanın yüzey alanı, m^2
AUST	Isıtılan (soğutulan) mahalde paneli gören ve ısıtılmayan (soğutulmayan) yüzeylerin alan ağırlıklı ortalama sıcaklığı, K (°C)
AUST*	Isıtılan (soğutulan) mahallin tüm iç yüzeylerinin (panel yüzeyleri dahil) alan ağırlıklı ortalama sıcaklığı, K (°C)
$AUST_i$	Panel arkasındaki mekanda bulunan tüm yüzeylerin alan ağırlıklı ortalama sıcaklığı, K (°C)
B	Panel yüzeyinden-odaya taşınım ısı transfer katsayısı W/m^2K
B'	Panel arkasından-arkadaki hacme taşınım ısı transfer katsayısı W/m^2K
c	Panel yüzeyinden-odaya taşınım ısı transfer üssü

* SI birimleri temel alınmakla birlikte, uygulamadaki tercihler nedeni ile raporun muhtelif bölümlerinde IP ve Metrik birimlere de yer verilmiştir.

c'	Panel arkasından-arkadaki hacme taşınım ısı transfer üssü ayrıca: Panel yüzeyindeki ışıma ve taşınım gradyanları katsayısı (Denklem 36), K (°C)
C	Panel ısıtmada pik yük traşlama katsayısı
C_p	Özgül ısı, kj/kg K
C_s	Panel soğutmada pik yük traşlama katsayısı
CLO	Giysi yalıtım indeksi
d	Oda pozisyon kodu, m
D_e	Odanın eşdeğer çapı, m
D_i	Boru iç çapı, m
Do	Boru dış çapı, m
ERF	Etkin ışıma ile ısıtma (soğutma) akısı (Effective Radiant Flux), W/m²
e	Işıma yayınım katsayısı
F_r	Işıma katsayısı
g	Yer çekimi ivmesi, m/s²
h	Denizden olan yükseklik, m
h_a	Panel arkasindaki yüzeyde ısı taşınım katsayisi, W/m² K
H	Yapı özelliği, kcal/m³°C
k	Panel malzemesinin ısı iletim katsayısı, W/m K
k_e	Kanatçığın eşdeğer ısı iletim katsayısı (panel yüzeyine paralel yönde), W/m K
k_h	Boru malzemesinin ısı iletim katsayısı, W/m K
k_i	Panel yüzeyindeki her bir kaplamanın (örtünün) ısı iletim katsayısı, W/m K i:1..n_k
k_s	Sıvının ısı iletim katsayısı, W/m K
L	Panelin kanatçık içersinde kalan kısmının kalınlığı, m
m	Kanatçık katsayısı, m^{-1}
M	Boru merkezleri arası mesafe, m
MET	Aktivite düzeyi
MRT	Ortalama ışıma sıcaklığı (Mean Radiant Temperature), K
n_k	Panel yüzeyindeki kaplama (örtü) adedi
P	Hava basıncı, Pa
Pr	Prandtl sayısı

q_a Panel arkasından olan birim ısı kaybi (kazanci), W/m² (Kcal/m²)

q_d Tavan panelinde toplam birim ısı transferi, W/m² (Kcal/h m²)

q_e Panel kenarlarından olan birim ısı kaybı (kazancı), W/m² (Kcal/h m²)

q_t Mahal ile dış ortam arasinda birim yüzeydeki iletim ısı transferi, W/m² (Kcal/h m²)

q_y (veya q_u)

Döşeme panelinin birim yüzeyinde toplam ısı transferi (q_{yr}+q_{yc}), W/m² (Kcal/h m²)

q_{yc} Döşeme panelinde taşınım ile gerçeklesen birim ısı transfer W/m² (Kcal/h m²)

q_{yr} Döşeme panelinde ışıma ile gerçeklesen birim isi transferi, W/m² (Kcal/h m²)

Q_i İnsan vücudundan bir saatde yayılan hissedilir isi miktarı, (Kcal/h)

Q_L Tabii infiltrasyon ısı kaybı (kazancı), W (Kcal/h)

Q_y Panel ısıtma yükü (-soğutma yükü), W (Kcal/h)

Q_{si} Panelin bulunduğu yapi elemanının (ör:döşeme) isıtılmayan (soğutulmayan) kismındaki ısi kaybi(kazanci), W (Kcal/h)

q_j j:1,2,3.. sayılı ısıtılmayan (sogutulmayan) yüzeylerdeki bi ısı transferi, W/m² (Kcal/h m²)

r Işıma ile ısı transferi ifadesi doğrusallastırma katsayısı,

R Oda özelliği

r_a Panel arkasındaki toplam ısı iletim direnci (boru merkezind itibaren), m²K/W

r_j j:1,2,3.. sayılı ve iç içe geçmis boruların yarı çapları (Şekil 11), m

r_y Panel üzerindeki toplam ısı iletim direnci (boru merkezinde

r_p Panel gövdesinin ısı iletim direnci, m²K/W

r_s (veya R)

Boru dış yüzeyi ile panel malzemesi arasındaki temasın ısıl direnci, m²K/W

r_t Boru et kalınlığının ısı iletim direnci, m^2K/W

r_u Panelin toplam ısı iletim direnci ($r_s+r_t+r_p+r_c$), m^2K/W

r_i Boru iç yarı çapı, m

s_ϕ Isı kazancı (kaybı) genel ifadesi

t Panel kalınlığı (Denklem 19), m

T Sıcaklık, K (°C)

T_a Ortalama iç hava sıcaklığı, K (°C)

T_b Dış hesap sıcaklığı, K (°C)

T_d Boru dış yüzey sıcaklığı, K (°C)

T_g Toprak hesap sıcaklığı, K (°C)

T_i Alltaki mahallin ortalama hava sıcaklığı, K (°C)

T_j j:1,2,3.. sayılı ısıtılmayan (soğutulmayan) iç yüzeylerin sıcaklıkları, K (°C)

T_{max} Panel yüzeyindeki en yüksek sıcaklık, K (°C)

T_{min} Panel yüzeyindeki en düşük sıcaklık, K (°C)

T_o Dış hava sıcaklığı, K (°C)

T_{op} Oda operatif sıcaklığı, K (°C)

T_p Etkin panel yüzey sıcaklığı, K (°C)

$T_p(x)$ (x) noktasinda panel yüzey sicaklığı, K (°C)

T_{pi} Alttaki mahallin ortalama tavan yüzey sıcaklığı, K (°C)

T_s Ortalama sıvı sıcaklığı, K (°C)

T_{si} Cilt (giysi) sıcaklığı, K (°C)

T_{total} Toplam kanatçık kalınlığı, m

T_u Isıtılmayan (soğutulmayan) her hangi bir oda iç yüzeyinin sıcaklığı, K (°C)

U Isı transfer katsayısı, W/m^2K

U_c Taşınım ısı transfer katsayısı, W/m^2K

U_r Işıma ısı transfer katsayısı, W/m^2K

v_s Sıvı hızı (boru içersinde), m/s

V Hava hızı, m/s

W Boru açıklığı yarı mesafesi $(M-D_o)/2$, m

X	Panelin ısıtma (soğutma) verimi	
x,y,z	Kartezyen ko ordinat eksenleri	
	veya:	
	z	Dış sıcaklık faktörü, K (°C)
σ	Stefan Boltzmann sabiti, $5.67 \cdot 10^{-8}$ W/m^2K^4	
η	Kanatçık verimi	
α_B	Sıvı-boru ışı taşınım katsayısı, W/m^2K	
α_i	Duvarın mahalle bakan yüzeyinde ısı taşınım katsayısi, W/m^2K	
α_y	Panel yüzeyinde ısi taşınım katsayısi, W/m^2K	
φ	Oda içinde termo hidrolik bağımlı değişkenlerin genel ifadesi	
Γ_ϕ	Bağımlı değişken katsayısi	
β	Isıl genleşme katsayısı, K^{-1}	
μ	Viskosite	
ρ	Yoğunluk, Kg/m^3	

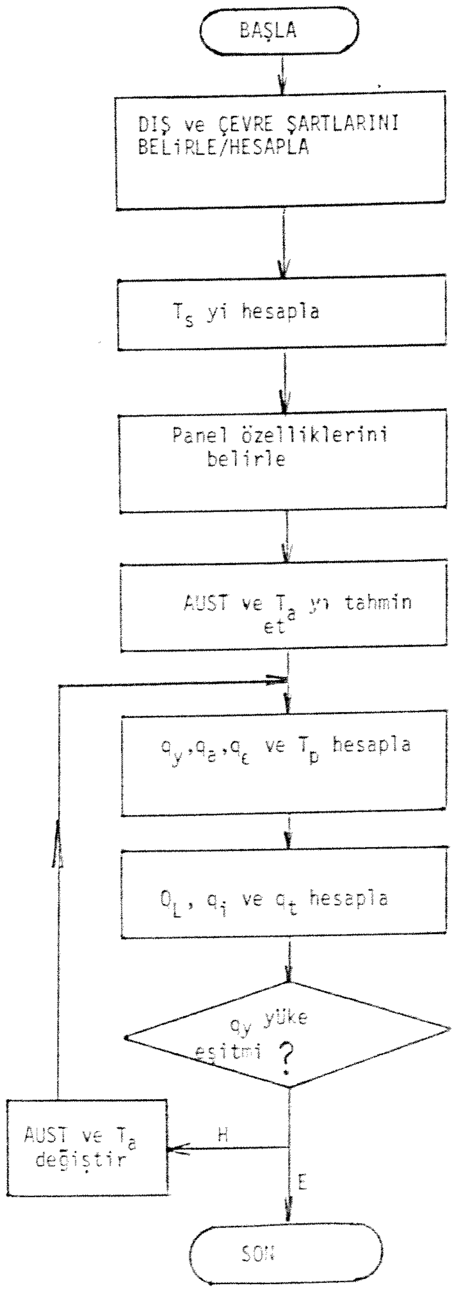
10.3.2. Genel Yaklaşım

Yukarıda sözü edilen sentez, sayısal bir yaklaşım için teknik olarak mümkündür. Ancak proje amaçlarından olan analitik bir tasarım algoritmasında bu boyutta bir sentezin getireceği hesap yükü çok fazladır. Bu noktada iki çözüm düşünülebilinir: Birincisi belirli bir iterasyon düzeni içersinde senteze sadık kalmak (Şekil 5) veya hesapların doğruluğu ve hassasiyetinden fazla bir ödün vermeden basit ve daha kısa bir algoritma geliştirmek. İki çözümün ne kadar birbiri ile tutarlığı olduğunu gerektiğinde sınamak için iki çalışma yürütülmüştür. Bunlar:

a) Sayısal çözüm,

b) Analitik yaklaşım'dır.

Sayısal çözümde oda ve panel incelenmiş, analitik yaklaşımla da panel ısı transferi ve oda ısıtma/soğutma yüküne yaklaşık çözümler getirilmiştir.



Şekil 5. İterasyon ile Çözüm.

10.3.3. Panelin Modellenmesi

10.3.3.1. Panel Yüzeyinde Isı Transferi

<u>Doğal Taşınım</u> :

Panel yüzeyindeki taşınım ısı transferi, panelin konumuna bağlıdır. Çeşitli araştırmacılar panel konumuna bağlı olarak, doğal taşınım ısı transferi için Denklem 1'deki B ve c katsayılarını vermişlerdir.

Çizelge 2. Panel Yüzeyinde Doğal Taşınım Katsayıları.

Pozisyon	B (W/m² °C)	C	Yazarı
Döşeme	2.67	1.25	Griffiths ve Davis (1931)
	3.26	1.25	Nusselt ve Hencky (1921)
	2.12	1.31	Min ve diğerleri (1956)
	2.67	0.12	Wilkes ve Peterson (1938)
Tavan	0.64	1.25	Dulong ve Petit (1922)
	0.87	1.25	Kollmar ve Liese (1957)
	0.4	1.25	Zhang ve Pate (1986)
	1.28	1.25	Griffiths ve Davis (1931)
	0.134	1.25	Min ve Diğerleri (1956)
	0.15 ~ 0.6	1.25	Krause (1959)
Duvar	2.56	1.25	Nusselt, W. (1909)
	1.98	1.25	Griffiths ve Davis (1931)
	1.78	1.32	Min ve Diğerleri

Aslında yüzeydeki doğal taşınım ısı transfer katsayısı odanın büyüklüğüne, deniz seviyesinden olan yüksekliğe, yüzeyi süpüren havanın hızına, tavandan ısıtmada ise ısıtılan yüzeyler arasında "soğuk" alan olup olmamasına bağlıdır. Wilkes ve Peterson (1938) bu katsayıların (T_p - T_a) farkına da bağlı olduğunu öne sürmüşlerdir.

Yüzeyi süpüren hava hızına bağlı olarak zorlamalı taşınım da söz konusu olabilir. Özellikle soğuk pencere yüzeylerinden aşağıya doğru olan hava akımları bu bölgeye yakın panel yüzeylerinde taşınımı arttırır (Sartain ve Harris, 1959) (Drake, 1993).

<u>Işıma</u> :

Işıma ısı transferi linearize edildiğinde, A katsayısı:

$$A = r \cdot F_r \cdot \sigma \qquad (4)$$

r, linearizasyon katsayısı olup, F_r yüzey soğurma katsayısı e 'ye eşit alınabilir.

$$r = 4 \cdot \left(\frac{(T_p + 273)}{2} + \frac{(AUST + 273)}{2}\right)^2 \qquad (5)$$

olup Kılkış (1993) işletme koşullarında geçerli olan:

$$r \cong \left[0.0105 \cdot \left(\frac{T_p + AUST}{2}\right) + 0.7955\right] \cdot 10^8 \qquad (6)$$

$$\left(15\,°C \leq \frac{T_p + AUST}{2} \leq 30\,°C \text{ için}\right)$$

denklemini vermiştir.

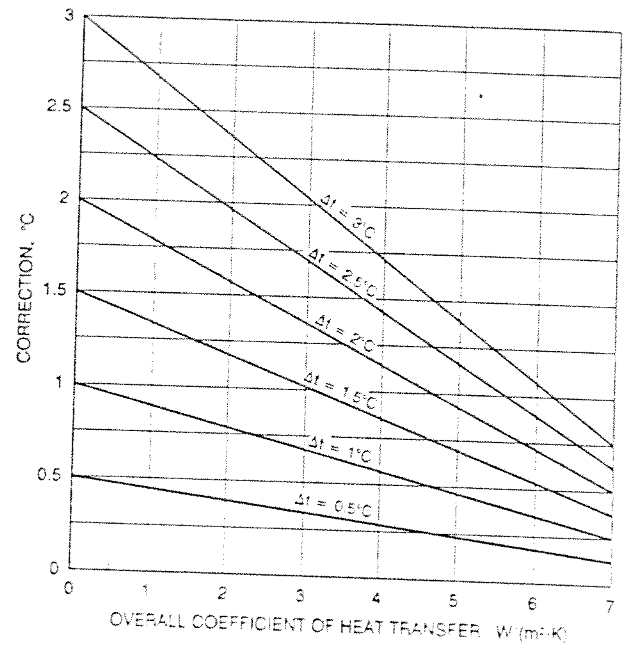
Kılkış (1993) ısı taşınımı düzeltmesi için:

$$\left(\frac{4.96}{D_e}\right)^{0.08} \cdot \left(1 - 2.22 \cdot 10^{-5} \cdot h\right)^{2.627} \qquad (7)$$

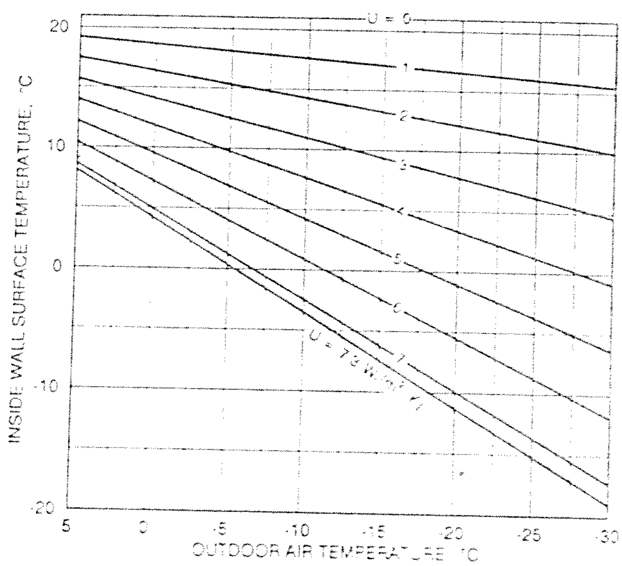
ifadesini kullanmıştır.

Burada D_e odanın eşdeğer çapı, h ise denizden olan yüksekliktir.

Işıma ısı transferinin hesabında AUST'un bulunması problemin en zor kısımlarından birisidir. ASHRAE (1992) el kitabı bölüm 6 'da iç duvar yüzeylerinde sıcaklığın iç hava sıcaklığına eşit alınabileceği görüşü ileri sürülmektedir. Zmeurenau ve diğerleri (1987) bu sıcaklığın oda sıcaklığının üzerinde olabileceğine sayısal modellerinde işaret etmişlerdir. Dış duvarlarda ise bu sıcaklık duvarın ısıl direncine ve dış sıcaklığa ve rüzgar hızına iç ve dış yüzeylerindeki ısıma ısı transferine ve bir miktarda iç yüzeydeki film katsayısına (α_i) bağlıdır. ASHRAE (1992) bu ilişkiyi basitleştirerek T_u değerini dış sıcaklık ve ısı geçirim katsayısına bağlı olarak Şekil 6-a 'da görüldüğü üzere vermiştir. T_a : 21 °C için geçerli olan bu ilişki için düzeltme miktarları da Şekil 6-b 'de verilmiştir.

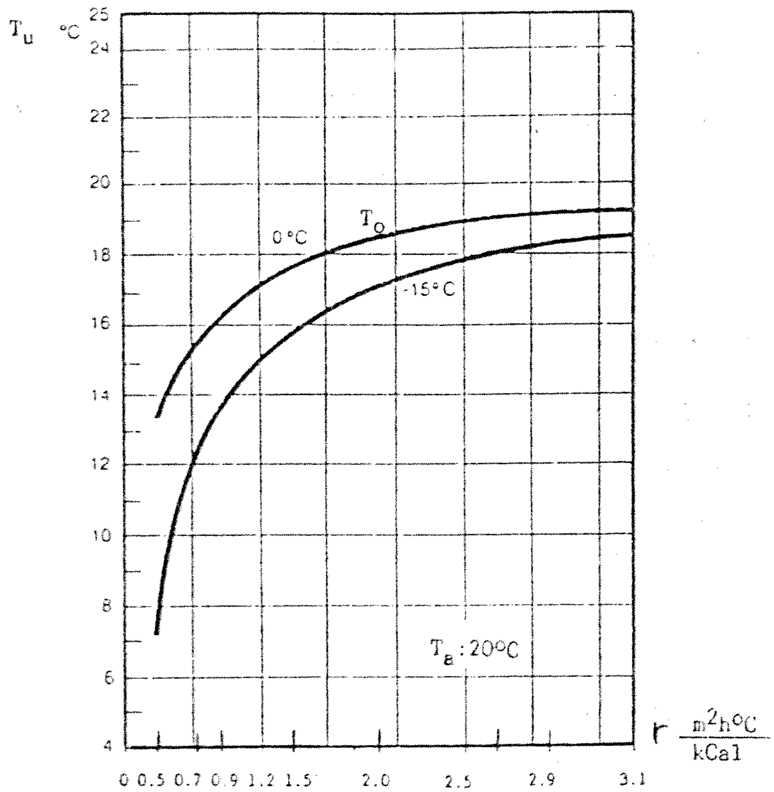


Şekil 6-a. Dış Duvarların İç Yüzey Sıcaklığı.



Şekil 6-b. Dış Duvarların İç Yüzey Sıcaklıkları için Düzeltme ($T_a \neq 21$ °C).

Şekil 6-a 'ya uyum gösterir nitelikteki bir ilişki ise Kömatherm Firması kataloğunda (1988) verilmiştir (Şekil 7).



Şekil 7. Isıtılmayan Dış Duvar İç Sıcaklığı.

Steinman ve diğerleri (1989) bu ilişkilerin cam yüzeyi fazla olan ve veya az yalıtımlı köşe odalarında geçerli olmayacağını bildirmişlerdir.

Shutrum ve diğerleri (1953a ve 1953b) nin deneyleri ve Kalisperis (1985) in bilgisayar modeli AUST'un T_a 'ya yakın olduğunu bildirmişlerdir. Drake (1993) ise oda konumu ve dışa bakan duvar, pencere, kapı alanına bağlı olmak kaydı ile AUST'un T_a 'dan 7 °C daha az bir sıcaklığa inebileceğini göstermiştir. Chapman ve Jones (1993) ise geliştirdikleri bilgisayar programında kullandıkları Monte Carlo yöntemi ile AUST T_a ilişkisinin büyük ölçüde oda konumu, yalıtımı, panel pozisyonu, dış duvar, pencere, kapı alan yüzdesine, döşeme ve tavan özelliklerine bağlı olduğunu göstermiştir.

Sonuç olarak, AUST değerinin tesbiti, panelin performansını önemli ölçüde etkilediğinden, hesaplarda ağırlık kazanmaktadır. Kılkış ve Sağer (1993) sayısal modelleme çalışmalarında elde ettikleri örnek özümlerde AUST'un T_a 'dan az olacağını göstermişlerdir. Elde edilen sonuçlarla Kılkış'ın (1993) geliştirdiği basit analitik algoritma uyum göstermişlerdir.

Belirli bir ölçüde panel ısıtma için çözümlenen AUST değerinin tesbiti problemi, panel soğutma açısından hemen hiç ele alınmamıştır. Kılkış (1993) bu konuda da basitleştirilmiş bir analitik yöntem geliştirmiştir. Panel ısıtma ve soğutma için AUST değerleri bu yönteme göre:

$$\text{AUST} : T_a - d \,.\, z \tag{8}$$

Burada d oda pozisyon kodudur:

d : 0.5 ; iç oda

d : 1 ; tek kenarı dış temaslı oda (pencere yüzey oranı ≤ % 5)

d : 2 ; tek kenarı dış temaslı oda (pencere yüzey oranı > % 5)

d : 3 ; iki veya daha çok kenarı dış temaslı oda.

z ise dış sıcaklık faktörüdür:

$$z \approx \frac{15}{25 + T_b} \qquad \{ T_b \geq -20\,°C \} \;;\; \text{ısıtmada} \tag{9-a}$$

$$z \approx \frac{7}{T_b - 45} \qquad \{ 26 \leq T_b \leq 36\,°C \} \;;\; \text{soğutmada} \tag{9-b}$$

Külpmann (1993) tavandan soğutmada AUST'un T_a genellikle çok yakın olacağını deneysel olarak göstermiştir. Bu farkın soğutma yükü ile doğrusal orantılı olduğu da izlenmiştir.

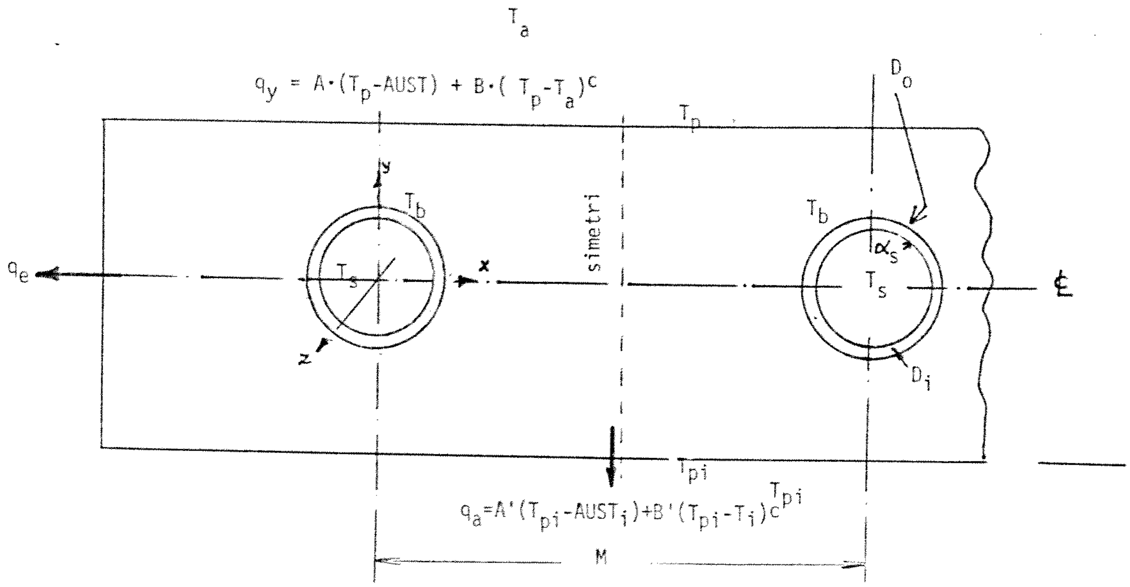
10.3.3.2. Panelde Isı Transferi

Panel yüzeyindeki ısı transferi sınır şartı olarak ele alındığında ve panel arkasından olan ısı kayıpları (kazançları) içinde benzer bir ifade yazıldığında panel içindeki ısı transferi iki boyutlu olarak ele alınabilir. Burada yapılan varsayım, ısı transfer sıvısında ve borusunda z yönünde bir sıcaklık farkı olmadığı şeklindedir.

Boru kesitinde sıcaklığın sabit olduğu varsayılabilir. Paneli besleyen hidrolik devre $(q_Y + q_e + q_a)$ ısı miktarını sağlar. Bu miktarın q_Y bölümü ısıtılan mekana arz olunur. O halde ısıtma verimi X :

$$X = \frac{q_Y}{q_Y + q_e + q_a} \quad \text{'dır.} \tag{10}$$

Boru içinde ortalama sıcaklığı T_s olan ve v_s hızı ile hareket eden bir sıvının türbülanslı akımda oluşturduğu zorlamalı ısı transferi (Mc.Adams, 1954):



Şekil 8. Panelde Isı Transferi.

$$\alpha_s = \frac{k_s}{D_i} \, [\, 0.023 \, . \, R_e . \, Pr^{0.4} \,] \tag{11}$$

Burada k_s sıvının ısı iletim katsayısıdır. Türbülanslı akım için Reynolds sayısı 6000 'den, tercihan 10000 'den fazla olması gerekir. Kreider ve Kreith (1981), güneş kollektörlerindeki borular içinde dolaşan sıcak su için α_s 'yi W/m^2.K cinsinden:

$$\alpha_s : \; 1056 \, . \, [\, 0.02\,(T_s + 273\,) - 4.06\,] \, . \, v_s^{0.8} / D_i^{0.2} \tag{12}$$

şeklinde vermiştir.

Dairesel bir boru için:

$$T_d = T_s - \frac{q_Y \, . \, M}{X} \; . \; [\, \frac{1}{\alpha_s \, . \, D_i} + \frac{1}{2 \, . \, k_h} \; . \; \ln\,(D_o / D_i)\,] \tag{13}$$

Burada T_d boru dış yüzey sıcaklığıdır. k_h ise boru malzemesinin ısı iletim katsayısıdır. Panel içersindeki ısı transferi M/2 noktasındaki dikey çizgiye göre simetriktir. Boru merkezlerinden geçen yatay çizgi ise bir çok araştırmacı ve yayın tarafından (Krinninger, 1989, DIN 4725, 1990) yukarı ve aşağı ısı transferinde ayırım çizgisi olarak tanımlamışlardır. Bu varsayımın geçerliği olduğu Kılkış ve Sağer'in (1993) sayısal çözümlerindeki normal uygulama şartlarında görülmüştür (Ek-8 'deki çözümlere bakınız).

Sonuç olarak, T_s sıcaklığındaki ortamdan hareketle ısının q_Y bölümü yukarıdaki yüzeye ulaşacaktır. Yüzeyin etkin sıcaklığı T_p olarak literatürde geçmekte ise bu sıcaklık h[illegible]li bir dağılım gösterir. Gelecek bölümde görüleceği üzere bu dağılımın sinüs eğrisi olduğunu ileri süren araştırmacıların bu varsayımı analitik ve sayısal çözümler (Kılkış, Sağer, 1993) ve Kollmar ve Liese (1957) tarafından doğrulanmamıştır.

Isının q_a bölümü boru merkezlerinden geçen çizgiden başlayarak aşağıya transfer edilir. Kenardaki borudan ise q_e miktarında yatay ısı transferi söz konusu olup araştırmacılar bu miktarı panel içi ısı transferinde ihmal etmişlerdir. Bu ısı transferinin panel içi sıcaklık dağılımına etkisi bu yüzeydeki sınır şartları yazılarak sayısal yöntemlerle hassas bir biçimde bulunabilir. Analitik yöntemlerde ise bu ısı kaybı (kazancı) yaklaşık olarak hesaplanabilirse de panel içi sıcaklık dağılımına etkisinin analitik çözümü pratik anlamda imkansızdır.

Panel içindeki ısı transferinin deneysel olarak tesbiti oldukça zordur ve aslında uygulamada pek bir anlamı da yoktur. Amaç panel performansı ve ısı kayıpları/kazançlarının tesbiti ile sınırlandığında, literatürde panel performansını inceleyen deneysel çalışmaların bulunduğu görülür. Bu deneysel çalışmaların istisnasız tamamı panel yüzey sıcaklığındaki dağılımı gözönünde tutmamışlar, T_p tarifini yeterli bulmuşlardır. Sartain ve Harris (1956) döşemeden ısıtılan deneysel odalarında döşeme panelinin performansını, değişik döşeme kaplamaları ve iç oda sıcaklıklarında yüzey ısı transferi şeklinde değerlendirmişlerdir. (İç sıcaklık - dış sıcaklık) ile döşeme kaplaması ısıl direncinin ortalama su sıcaklığına etkilerinin lineer olduğunu deneylerinde göstermişlerdir. Ayrıca panele sevkedilen ısının ne kadarının yukarı ulaştığını ne kadarının ise aşağıya ve yana kaçtığını ölçmüşlerdir. Beklendiği üzere ısı kayıplarının döşeme kaplaması ısıl direncine orantılı olduğunu ve bu miktarın (iç sıcaklık - dış sıcaklık) farkına da bağlı olduğunu göstermişlerdir.

Leigh ise (1991) beton döşeme panellerinde döşemeden ısıtma uygulamasına yönelik kontrol stratejilerinin geliştirilmesi amacı ile panelin süreksiz eğimdeki ısıl davranışını oda ısı yükü ve çevre koşullarına göre incelemiştir.

10.3.3.2.1. Analitik Yaklaşım

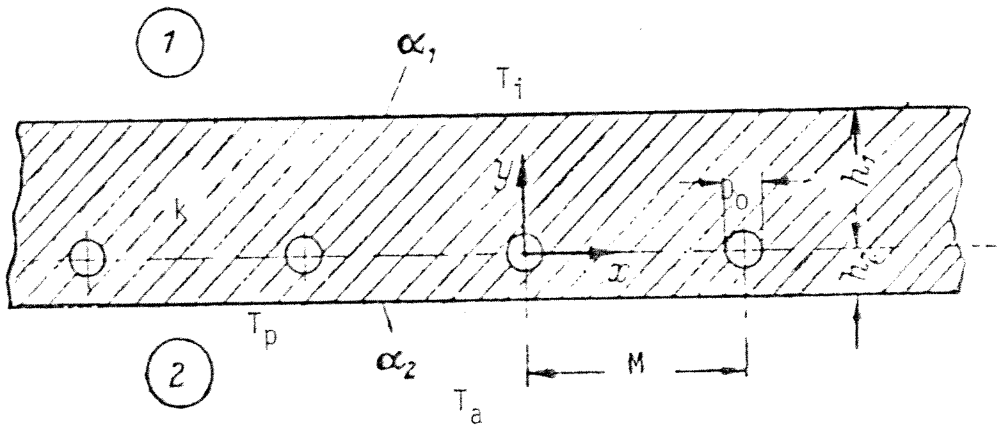
Rydberg ve Huber (1955), Faxen'in (1937) bir etüdüne dayanarak içinde boru bulunan, paralel yüzeyli, homojen veya çok tabakalı bir tavan ısıtıcı panelini incelemişlerdir. Şekil 9'da homojen tavan panelinin modeli görülmektedir.

Sıcaklık alanı T (x, y):

$$T(x,y)\,.\,\frac{M}{\pi\,.\,A} = -G_1 y - |y| - G_2 + \frac{M}{\pi}\sum_{s=1}^{\infty}\frac{1}{s}\left[e^{-\frac{2\pi s|y|}{M}} + g_1 e^{-\frac{2\pi sy}{M}} + g_2 e^{-\frac{2\pi sy}{M}}\right.\qquad(14)$$

$$\cos e^{-\frac{2\pi sx}{M}}$$

olarak ifade edilmiştir.



Şekil 9. Rdydberg ve Huber Modeli.

Şekilden görüleceği üzere, boru merkezlerinden geçen düzlem ve iki komşu boru arası orta nokta, analiz için temel seçilmiştir.

Burada;

D_o	boru çapı,
M	borular arasındaki açıklık,
h_1, h_2	boru eksenleriyle levha yüzeyleri arasındaki uzaklık,
T	levhanın herhangi bir yeriyle ortam arasındaki pozitif sıcaklık farkı,
α_1, α_2	levha yüzeyleri üzerindeki ısı iletim katsayıları, levhanın ısı geçergenlik katsayısı,
x_1, x_2	boru eksen düzleminin üst ve alt tarafına doğru oluşan kısmı ısı iletimi ile ilgili direnç katsayıları,
k_p	panelin ısı iletim katsayısıdır.

Bu modelde kenar kayıpları gözönünde tutulmamış, yüzeylerdeki ısı transfer katsayısı tarifinde ışıma ve doğal konveksyon ayırımı yapılmamıştır. Dolayısı ile α_1 ve α_2 $(T_p - T_a)$ sıcaklık farkında toplam ısı transferini tarifleyen katsayılardır. Bunun anlamı AUST = T_a olmaktadır.

A sabiti sınır şartlarının katılması için bir katsayı olup;

$$A = \{ \ln \frac{M}{\pi . D_o} - \frac{\pi}{M} G_2 + \sum_{s=1}^{\infty} \frac{g_1+g_2}{s} \} \frac{1}{\Delta T_o} \tag{15}$$

ΔT_o = Boru ile ortam arasındaki sıcaklık farkıdır.

$$G_1 = \frac{x_1 - x_2}{x_1 + x_2} , \tag{16-a}$$

$$G_2 = -2 k_p \frac{1}{x_1 + x_2} \tag{16-b}$$

dir. g_1 ve g_2, $s = 1, 2, 3, ... \infty$ için, aşağıdaki iki denklem sistemi aracılığı ile tayin edilir:

$$(\frac{\alpha_1}{k_p} - \frac{2\pi s}{M}) (1 + g_1) e^{-\frac{4\pi s h_1}{M}} + (\frac{\alpha_1}{k_p} + \frac{2\pi s}{M}) g_2 = 0 , \tag{16-c}$$

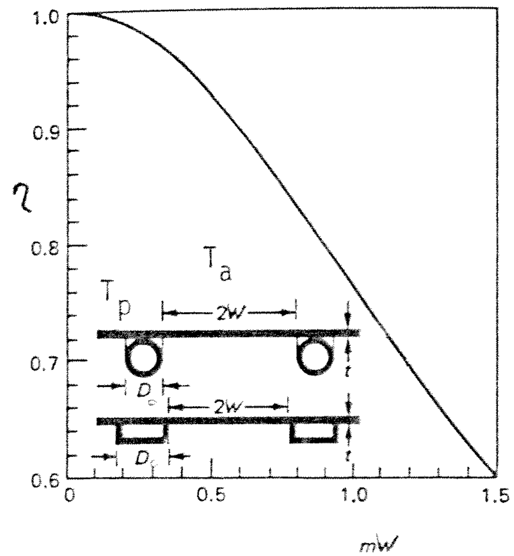
$$(\frac{\alpha_2}{k_p} - \frac{2\pi s}{M}) (1 + g_2) e^{-\frac{4\pi s h_2}{M}} + (\frac{\alpha_2}{k_p} + \frac{2\pi s}{M}) g_1 = 0 . \tag{16-d}$$

Yüzeyde ek katmanlar var ise çözüm daha da karmaşık bir durum aldığından çözümleri abaklar şeklinde vermişlerdir. Bu paralel yüzeyli ve çok tabakalı levhadaki sıcaklık dağılımı çözüm yöntemi ise ancak panelin çeşitli tabakalarına ait kalınlık ve ısı iletim katsayıları aynı mertebede olduğu taktirde geçerlidir. Bu çözüm beton gibi kalın ve ısı iletim katsayısı fazla olan panel üzerinde ince fakat ısıl direnci çok olan bir katmanın (örneğin lastik tabanlı halı veya tahta parke) bulunduğu durumlarda geçerli değildir.

Uygulamadaki zorlukları nedeni ile bu yöntem önerildiği zamanda itibar görmemiş, bilgisayarların geliştiği çağlarda ise sonlu eleman ve sonlu farklar yöntemleri ile daha hassas çözümler elde edilebildiğinden uygulamaya geçememiştir. Aslen Krittal tavanlar ile ısıtma için öngörülen bu yöntem α_1 ve α_2 değerleri değiştirilerek döşemeden ısıtmada veya (+) işaretleri değiştirilerek panel soğutmada da kullanılabilir.

Bu yöntemin diğer bir avantajı ise yüzeydeki sıcaklık dağılımını ($y = -h_2$ ve $T(x, y) : T_p(x, y)$) verebilmesidir.

Kreider ve Kreith (1981) ise panel uygulamasına benzerlik arzeden düzlemsel güneş kollektörü plakasında ısı transferini tek boyutlu kanatçık modelini kullanarak incelemişlerdir. Şekil 10 'da bu model görülmektedir.



Şekil 10. Güneş Kollektör Plakasında Kanatçık Modeli.

Bu model, yükseltilmiş döşemelerde, döşeme altına tesbit edilen borular ile tamamen benzerlik arzetmekle birlikte, beton içi uygulamalarında problem aslen iki boyutludur. Kreider ve Kreith'in modeli kullanıldığında:

$$\text{Yüzeydeki maksimum sıcaklık, } T_{max} : T_a + \frac{(T_p - T_a) . M}{[2 . W . \eta + D_o]} \qquad (17)$$

olmaktadır. Burada

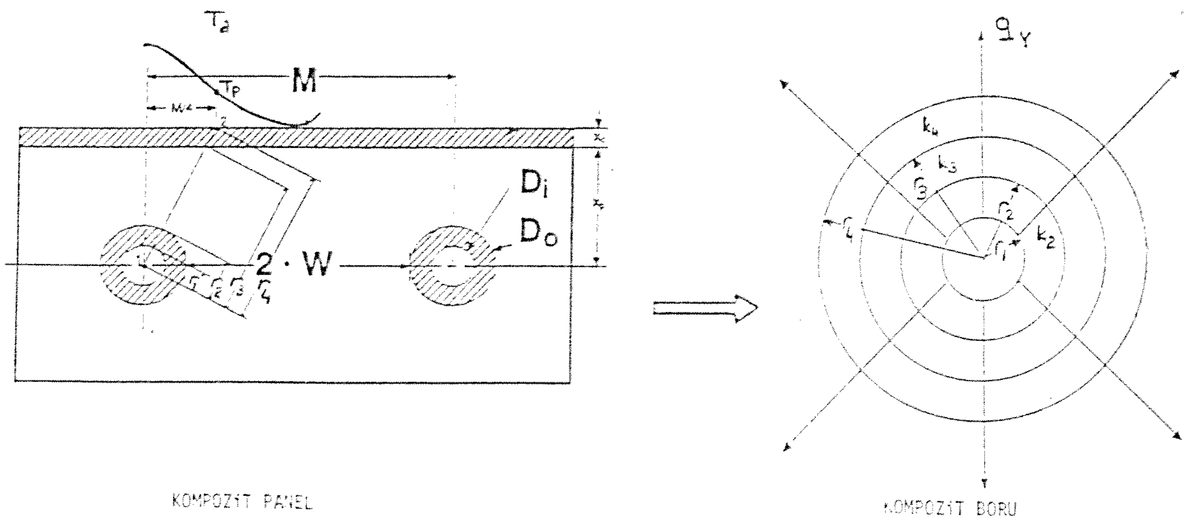
$$\eta = \frac{\tan h\,(m . W)}{m . W} \qquad (18)$$

$$\text{olup} \quad m = \sqrt{\frac{q_Y / (T_p - T_a)}{k_p . t}} \qquad (19)$$

Bu basit yöntemde çok katmanlı panelin modeli için eşdeğer bir k_p kullanılması gerektiği gibi, panel yüzeyindeki ısı transferinde yüzeyden ışıma ve doğal konveksiyon ile T_a AUST ayırımı yapılamamaktadır.

Kollmar ve Liese [1957] bu modeli aynen döşeme içi ısıtmaya uygulamışlar, ancak yüzey kaplamalarının k_p 'ye olan etkisini, dolayısı ile kanatçık verimine etkisini tam olarak izah edememişler, problemi tek boyutlu şekli ile bırakmışlardır. Ülkemizde de Makina Mühendisleri Odası, Kalorifer Tesisatı Proje Hazırlama Teknik Esasları (1989) adlı yayınında bu modeli önermiş ve bu model ile hazırlanan tasarım abakları vermiştir. Bu tasarım yönteminde panel ısı kayıpları, panel yüzeyindeki ışıma, doğal taşınım ayırımı gözetilmediği gibi, döşeme kaplamalarının etkisi panel ısı yüküne zam şeklinde öngörülmüştür ki bu uygulama Kılkış tarafından (1990-b) kritik edilmiştir.

Krinninger (1989) ise döşeme içine gömülü boruların oluşturduğu paneli, yüzey kaplamaları da dahil olmak üzere iç içe geçmiş borular şeklinde modellemiştir. Boruların yarıçapları, panel yüzeyinde, M/4 noktasından geçecek şekilde tayin edilmektedir. Krinninger'e göre bu noktada yüzey sıcaklığı T_p 'ye eşittir. Yüzeydeki sıcaklık profili ise sinüsoidaldir. Şekil 11'de döşeme modellemesi gösterilmiştir.



Şekil 11. Döşeme Panelinin Kompozit Boru Şeklinde Modellemesi (Krinninger, 1989).

Buradaki ısı transferi ifadesinden:

$$T_d = \frac{q_Y \cdot M}{X \cdot \pi \cdot 2} \cdot \left[\frac{\ln(r_2/r_1)}{k_2} + \frac{\ln(r_3/r_2)}{k_3} + \dots \right] + T_p \qquad (20)$$

Bu yaklaşımın iki ana sakıncası vardır:

1. Şekil 11 'den görüldüğü üzere, 2π 'lik tüm boru çevresini süpüren yüzeyde ısı transferinin homojen olduğu varsayılmaktadır. Halbuki boru çevresindeki ısı transferi her noktada eşit değildir (Kılkış, Sager, 1993). Sonlu elemanlar yöntemi ile yapılan çözümler de buna işaret etmektedir (Bkz. Bölüm 10.3.3.2.2, Şekil 18 ve Ek-8).

2. Sadece üst yüzeyde serili halı, bu modelde borunun tüm etrafını sarmaktadır. Tabii bu fiziki gerçeklere uymamaktadır. Bu nedenle çıplak beton panellerde oldukça iyi netice veren bu yaklaşım, ısıl direnci yüksek kaplamalara haiz döşemelerde hassasiyetini yitirmektedir.

Ayrıca bu modelde su/boru iç yüzeyi film katsayısı ihmal edilmiştir.

Yüzeydeki sıcaklık dağılımının sinüsoidal olmadığı ise bu projede yapılan sayısal çözümlerle kanıtlanmış, ayrıca Kollmar ve Liese (1957) ve diğer araştırmacılar da aynı noktaya işaret etmişlerdir. Bu durum, bölüm 10.3.3.2.3 'de verilen örnek çözümde ve Ek-8 'deki sonlu elemanlar çözümü ile elde edilen hassas yüzey sıcaklık dağılımı grafiklerinde açıkça görülmektedir. Aslen (T_p, M/4) noktasından sonsuz sayıda sinüs eğrisi geçtiğinde yüzeydeki sıcaklık dağılımı, dolayısı ile T_{maks} ve T_{min} bulunamamaktadır.

Zhang ve Pate (1987) ise döşeme içi panel için yarı analitik bir çözüm geliştirmişlerdir. Kartezyen ko-ordinatlarda bir çözüm elde etmek için silindirik boruları prizmatik bir geometri ile idealize etmişlerdir. Bu sakıncayı gidermek için çözümlerine bir düzeltme faktörü dahil etmişlerdir. Bu düzeltme faktörünün niteliği ve niceliğini ise sonlu elemanlar yöntemi ile elde edilen çözümlerle yapılan mukayeselerde tesbit edilmişlerdir.

Panel ısıtma/soğutma problemine yakın ideal analitik modeller de mevcuttur. Kutateladze ve Borishanskii (1966) daimi rejimde, yarı sonsuz bir homojen malzeme içinde yan yana dizili borular ile üst yüzeye bakan ortam arasındaki ısı transferini aşağıdaki şekilde vermişlerdir:

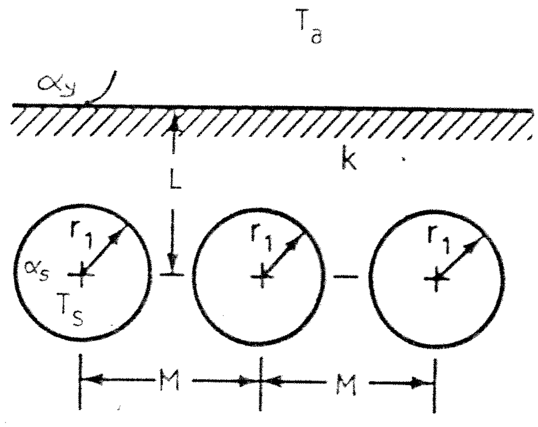
$$q_Y = \frac{2 . \pi . k . (T_s - T_a)}{1 / Bi_1 + \ln \{ (L / \pi . D . r_1) \sinh [\pi . 2 . (D + D / Bi_2)] \}} \qquad (21)$$

Burada,

$$Bi_1 = \alpha_s r_1 / k \quad ; \qquad Bi_2 = \frac{\alpha_Y . L}{k} \qquad \text{ve} \qquad D = L / M \qquad (22)$$

α_s üst yüzeydeki toplam ısı transferi katsayısıdır.

Bu ifade ile boru ısıl özellikleri, panel ısı kayıpları ve yüzeydeki AUST, T_a ve ışıma taşınım ısı transferi ayırımı yapılamadığı gibi homojen olmayan yani panel yüzeyinde kaplama(lar) bulunan paneller için geçerli değildir. Bu model Şekil 12 'de gösterilmiştir.



Şekil 12. Yarı Sonsuz Malzemeye Gömülü Borular Modeli.

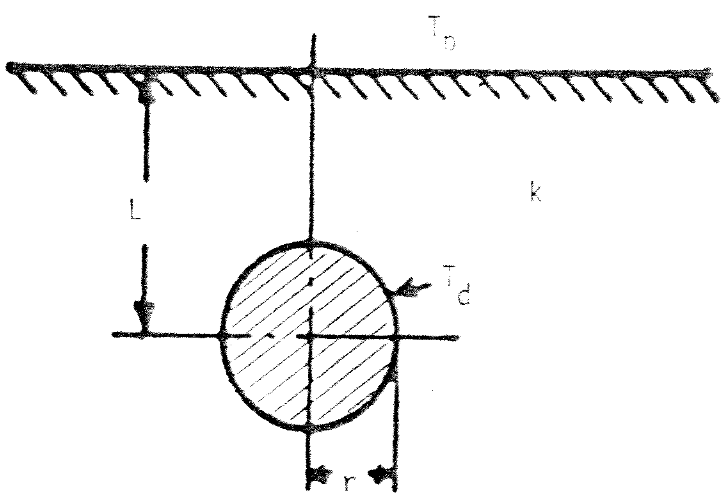
Yukarıdaki ifade T_s ve T_a 'ya bağlı olarak hem ısıtma hem soğutma için geçerlidir. Buna benzer bir ifade ise ASHRAE (1993) tarafından tek bir silindir için verilmiştir. Bu nedenle boru aralığı az olan sistemlerde geçerli değildir. Bu model için yine yukarıdaki mahzurlar söz konusudur:

$$q_Y = \frac{(T_d - T_p) . 2 . \pi . k}{\ln[(L + \sqrt{L^2 - r^2})/r]} \qquad (23)$$

ve $L >> r$ koşulu, ki panel sistemler için geçerlidir;

$$q_Y = \frac{(T_d - T_p) . 2 . \pi . k}{\cosh^{-1}(L/r)} \qquad (24)$$

Bu yöntemin uygulanabilirliği, özellikle boru aralığının gözönünde tutulmaması nedeni ve yukarıdaki mahzurlar nedeni ile çok kısıtlıdır. Şekil 13 'de bu model gösterilmiştir.

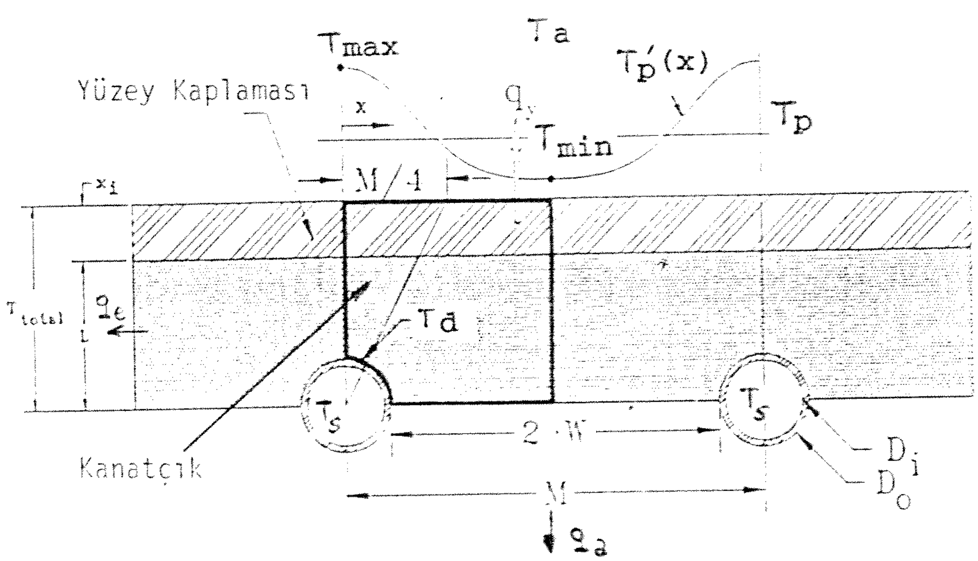


Şekil 13. Toprağa Gömülü Tek Silindir Modeli.

Bu proje çalışmalarında ise homojen olmayan (komposit kanatçık) modeli geliştirilmiştir. Yüzey ısı transferinde $T_a \neq$ AUST şartına ve taşınım-ışıma ayırımına itibar edildiği gibi, yüzey kaplamalarının etkisi çok gerçekçi bir biçimde çözüme yansıtılmıştır. Bu yöntem sonlu elemanlar yöntemi ile mukayese edilerek çok yakın neticeler elde edilmiştir (Ek-5; Yayın no: 11). Bu yöntemin hem soğutma hem ısıtma için ve diğer panel pozisyonları için de geçerli olması ve hassas neticelerin alınması nedenleri ile ASHRAE Handbook, Bölüm 6 'da yer alacaktır (Ek-5, Yayın no: 19).

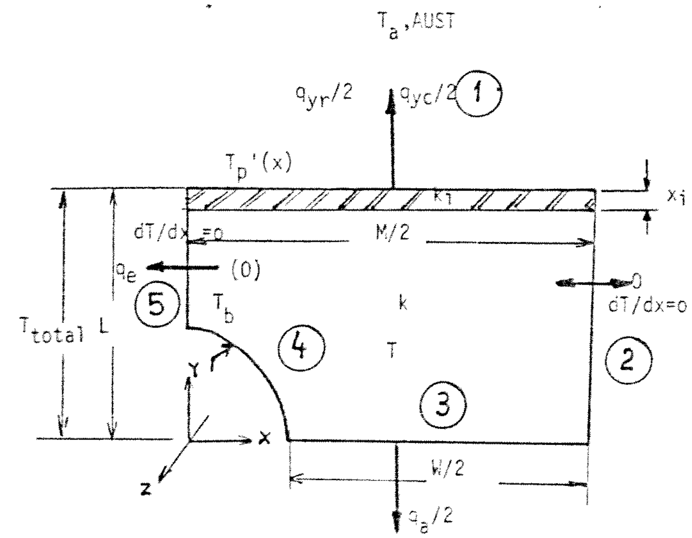
Kullanımın basit ve kısa olması bakımından uygulayıcı ve mühendisler tarafından kolaylıkla benimsenmesi beklenmektedir. Bu yöntemle yüzeydeki sıcaklık dağılımı her noktada çözülebilmekte, T_{maks} ve T_{min} değerlerinin önemli olduğu uygulamalarda (örneğin kar ve buz eritme sistemlerinde) bu çözüm avantaj sağlamaktadır (Ek-5, Yayın no: 2 ve 13).

Şekil 14 'de çözüme esas teşkil eden kanatçığın panel içersindeki genel konumu görülmektedir. Simetriden dolayı merkezleri arasındaki mesafe M olan iki komşu boru arasındaki bölümün yarısı incelenmektedir. Boru merkezlerinden geçen yatay düzlem ise yukarı ve aşağı yöndeki ısı transferini sınırlamaktadır.



Şekil 14. Panel İçersinde Kanatçığın Konumu (Kılkış, Sağer, 1993).

Şekil 15 'de ise problemin idealizasyonu görülmektedir.



Şekil 15. Kompozit Kanatçık Modeli (Kılkış, Sağer, 1993).

z yönündeki ölçü 1 'dir.

Yüzeylerdeki genel şartlar:

1. yüzey :

$T_p'(0) = T_{max}$; $\quad T_p'(M/2) = T_{min}$

$$T_p = \frac{1}{M/2} \cdot \int_0^{M/2} T_p'(x) . dx \qquad \text{(Etkin panel yüzey sıcaklığı)}$$

Isı kaybı (kazancı) $= q_Y . M/2 = M/2 . (q_{yr} + q_{yc})$

2. yüzey :

$q = 0$

$dT / dx = 0$

3. yüzey :

Isı kaybı (kazancı) $= q_a / 2$

4. yüzey :

$$T = T_b$$

Boru yüzeyindeki sıcaklık dağılımı sayısal yöntemle incelenmiş ve yukarıdaki varsayımın pratik tasarım ölçüleri içersinde geçerli olacağı görülmüştür.

5. yüzey :

Isı kaybı (kazancı) $= q_e$ (kenar boruda)
$= 0$ (panelin diğer bölümlerinde)

Bir numaralı yüzeyde, x noktasında ışıma ve ısı iletim transferi:

$$q_{yr} = U_r \,.\, (T_p'(x) - AUST) \tag{25-a}$$

$$q_{yc} = U_c \,.\, (T_p'(x) - T_a) \tag{25-b}$$

$$U_r = F_r \,.\, \sigma \,.\, r \tag{26}$$

$$F_r \approx e \tag{27}$$

$$U_c = (1 - 2.22 \,.\, 10^{-5} \,.\, h)^{2.627} \,.\, (4.96 / D_e)^{0.08} \,.\, 2.67 \,.\, (T_p - T_a)^{0.25} \tag{28}$$

Yukarıdaki denklemde U_c , T_p 'ye bağımlı tutularak, panel yüzeyinde sabit alınmıştır.

x yönündeki ısı transferinden hareketle;

$$\frac{U_r}{k_e \,.\, T_{TOTAL}} \,.\, [T_p'(x) - AUST] + \frac{U_c}{k_e \,.\, T_{TOTAL}} \,.\, [T_p'(x) - T_a] = \frac{d^2 T}{d x^2} \tag{29}$$

$$\theta_1 = T_p'(x) - AUST \tag{30}$$

ve

$$\theta_2 = T_p'(x) - T_a \tag{31}$$

Tarifi ile;

$$\frac{U_r}{k_e \cdot T_{TOTAL}} \cdot [\theta_2 + (T_a - AUST)] + \frac{U_c}{k_e \cdot T_{TOTAL}} \cdot \theta_2 \cong \frac{d^2\theta_2}{d x^2} \cong \frac{d^2\theta_2}{d x^2} \tag{32}$$

$$\frac{d^2\theta_2}{d x^2} - \theta_2 \left[\frac{U_r + U_c}{k_e \cdot T_{TOTAL}}\right] - \frac{U_r}{k_e \cdot T_{TOTAL}} \cdot (T_a - AUST) \cong 0 \tag{33}$$

$$U = U_r + U_c \tag{34}$$

ve

$$m^2 = \frac{U}{k_e \cdot T_{TOTAL}} \tag{35}$$

tanımından ve

$$c' = \frac{U_r}{k_e \cdot T_{TOTAL}} \cdot (T_a - AUST) \tag{36}$$

basitlemesi ile çözüm:

$$= C_1 e^{-mx} + C_2 e^{mx} - C / m^2 \tag{37}$$

Sınır şartı olarak:

$$\theta(x) = T_{max} - T_a \tag{38}$$

ve

$$\frac{d\,\theta(w)}{dx} = 0 \tag{39}$$

kullanılarak, yüzeyde:

$$T_p'(x) = \frac{\cosh[m(M/2 - x)]}{\cosh m \cdot W} \cdot (T_{max} - T_a + c'/m^2) + T_a - c'/m^2 \tag{40}$$

35 ve 37 numaralı denklemlerden:

$$c'/m^2 = c \text{ tarifiile } = \frac{U_r}{U} (T_a - AUST) \tag{41}$$

Son denklemde T_a = AUST şartında c sıfır olmaktadır.

40 numaralı denklemden;

$$T_{min} = T(M/2) = (\cosh[m.W])^{-1}.(T_{max} - T_{a+c}) + T_{a-c} \tag{42}$$

Kanatçık yüzeyindeki toplam ısı transferi:

$$[W.\eta + D_o/2].U_r.(T_{max} - AUST) + [W.\eta + D_o/2].U_c.(T_{max} - T_a) = q_Y.M/2$$

$$A = [2.W.\eta + D_o].U_r \tag{43}$$

$$B = [2.W.\eta + D_o].U_c \tag{44}$$

tarifleri ile

$$T_{max} = \frac{q_Y.M + A.AUST + B.T_a}{A + B} \tag{45}$$

Burada q_Y, birim panel yüzeyindeki ısıtma (soğutma) yüküdür:

$$q_Y = \frac{Q_Y}{A_p} \times 1000 \tag{46}$$

Burada A_p panel yüzey alanıdır. η ise Denklem 18 'de verilmiştir.

45 numaralı denkleme AUST = T_a şartı uygulandığında 17 numaralı denkleme dönüşebilmesi için U_r 'nin U_c 'ye eşit olması gerekir ki, bu şartın geçersiz olduğu 26 ve 28 numaralı denklemlerden görülür. Dolayısı ile Kreider ve Kreith'in modelinin panel ısıtma veya soğutma için matematiksel olarak geçerli olmadığı söylenebilir. Ancak, yukarıda verilen algoritmanın günlük uygulamada daha basit ve kullanışlı olmasını teminen 17 numaralı denklem kullanılabilir. Bunun ne kadar geçerli olduğunu sınamak için bölüm 10.3.3.2.3 'de verilen örnek ele alındığında:

$U_r = 4.99 \ W/m^2 K$

$U_c = 4.77 \ W/m^2 K$

$T_{max} = 30.1 \ °C$ (Denklem 45 'den)

$T_{max} = 31.8 \ °C$ (Denklem 17 'den)

Sonlu elemanlar, çözümü 29.6 olarak verdiğinden 45 numaralı denklemin daha hassas olduğu görülmektedir.

Yukarıdaki farkın nedeni; AUST = 17.3 °C (T_a = 18 °C) ve U_r = 4.99 (U_c = 4.77) olmasından kaynaklanmaktadır.

Önceki denklemlerde kullanılan k_e , yatay yöndeki ısı transferinde geçerli olan eşdeğer ısı iletim katsayısıdır. Aynı zamanda problemin 2 boyutlu adaptasyonunu sağlamaktadır. Kompozit bir kanatçık içersinde, n_k tane yüzey kaplaması var ise:

$$k_e = 2 \ . \ \frac{\left[\sum_{i=1}^{n_k} (k_i . x_i) + k . L \right]}{T_{TOTAL}} \tag{47}$$

Belirli bir yükün (q_Y) karşılanabilmesi için yüzeydeki T_{max} değeri çözüldükten sonra, gerekli T_d değeri:

$$T_d = T_{max} + q_Y . [R_{co} + (L - D_o/2) k + R] \tag{48}$$

Burada R_{co} döşeme kaplamalarının dikey yöne itibarla, toplam ısıl direncidir:

$$R_{co} = \sum_{i=1}^{n_k} (x_i / k_i) \tag{49}$$

Boru ile panel malzemesi arasındaki ısıl direnç, R ise genellikle ihmal edilebilir.

Gerekli akışkan sıcaklığı ise 13 numaralı denklem ile bulunabilir.

Yukarıdaki denklemlerin tamamı soğutma için de geçerlidir.

Isıtmada	AUST < T_p	$T_a < T_p$,
Soğutmada ise	AUST > T_p	$T_a > T_p$

şartları geçerli olduğundan q_Y ısıtmada (+), soğutmada ise (-) olarak tariflendiği sürece denklemler hiç bir değişikliğe gerek duymayacaklardır (Ek-5, Yayın no: 19). Şekil 16 'da tavanda soğutma paneli ve kompozit kanatçık görülmektedir. Burada T_{max} ve T_{min} yer değiştirmektedir.

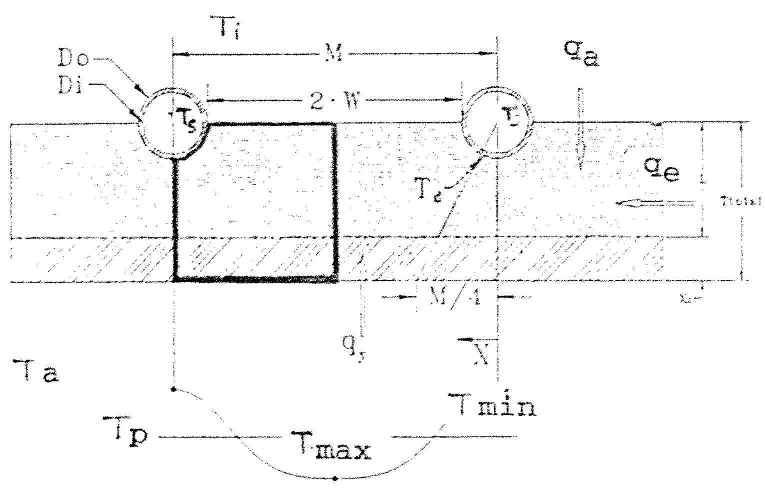
Panel arkasındaki ısı transferi için geliştirilen basitleştirilmiş yöntemle:

$$q_a \cong \frac{1}{r_a + h_a} \cdot (T_d - T_i) \tag{50}$$

burada r_a , betonun kanatçık altında kalan kısmı da dahil olmak üzere bütün katmanların toplam ısıl direncidir. Eğer alttaki ortam yine iç bir mekan ise, Min ve diğerlerinin (1956) verilerine dayanarak:

$$h_a \cong \sigma . e . F_r + 0.138 . (T_{pi} - T_i)^{0.25} \tag{51}$$

En kenardaki boruya ilişkin kanatçıktan olan kenar ısı transferinde de denklem 46 'ya benzer bir yaklaşım geçerlidir. Panelin diğer kısımlarında $q_e = 0$ şartı tatbik edilir.



Şekil 16.Tavandan Soğutma Paneli Modeli.

Panel soğutma ile ilgili teori, model ve uygulamalar Ek-9 'da verilmiştir.

10.3.3.2.2. Sayısal Yaklaşım

Panel içi ısı transferinin prensip itibarı ile sayısal olarak çözülmesinde bugünkü yazılımlar ve yöntemler çerçevesinde teknik bir zorluk olmamasına karşın, şimdiye kadar yapılan çalışmalarda kenar ve aşağıya olan kayıplar ihmal edilerek, problem sadece 1, 2 ve 4 yüzeylerindeki şartlara göre çözülmüş (Şekil 15), yüzeydeki ısı transferi için ise toplam ısı transfer katsayısı ve AUST = T_a şartı kullanılmıştır.

Zhang ve Pate (1986) sonlu farklar yöntemi ile tavandan ısıtma sistemini incelemişler ve yüzey sıcaklığının zamana göre değişimini de vermişlerdir.

Bu çalışmada ise ANSYS sonlu elemanlar pakedi kullanılarak daha hassas çözümler elde edilmiş ve analitik yöntemin değerlendirilmesinde kullanılmıştır. Şekil 17 'de verilen problem ve buna benzer problemlerin çözümü ile diğer modellerin de geçerliği incelenmiştir.

Bu çözüm, odaya ilişkin çözümle doğrudan doğruya birlikte veya, AUST ve T_a değerlerinin iterasyonu ile bağımsız bir şekilde yürütülebilinir.

Bu çalışmaya ilişkin detaylar ve örnek çözümlerin bütün çıktıları Ek-8 'de verilmiştir.

10.3.3.2.3. Örnek Çözüm

Şekil 17 'de, ele alınan örnek panel gösterilmiştir. Bu örnekte, şap içersine 20 cm aralıklı, 2.1 cm dış çapı olan lastik borular döşenmiştir. Boruların hemen altında 2 cm strapor yalıtım, daha altta 10 cm kat betonu bulunmaktadır. Isıtılan oda sıcaklığı 18 °C, alttaki mahal sıcaklığı ise 15 °C alınmıştır. Yüzeyde bir adet kaplama (halı) vardır. Borular içersinden 55 °C da su geçirilmektedir. α_s değeri 2910 W/m^2 K alınmıştır.

İstanbul'daki (T_b = - 3 °C) bir yapının dış cepheli (pencere oranı > % 5) bir odasında ısı yükü Q_Y 3.1 kW 'dır. Panel alanı ise 30 m^2 'dir. Oda boyutları 5x8 m 'dir. (D_e = 6,15 m).

Bu verilerden hareketle ve deniz seviyesi için:

$$U_c = 2.66 \cdot (T_p - 18)^{0.25}$$

$$U_r = 4.98 \ (e = 0.85 \text{ için})$$

Çizelge 3. Örnek Çözüm Mukayesesi.

Parametre	Geliştirilen Algoritma	Sonlu Elemanlar	DIN (1990) ve Krinninger (1989)
q_Y [W/m²]	103.3	103.8	119.7
T_p [°C]	28.4	28.6	28.6
T_{max} [°C]	30.1	29.6	-
T_{min} [°C]	26.8	27.6	-
q_a [W/m²]	39.0	37.5	-
X	0.72	0.73	-
T_s [°C]	52.0	55.0	59.9

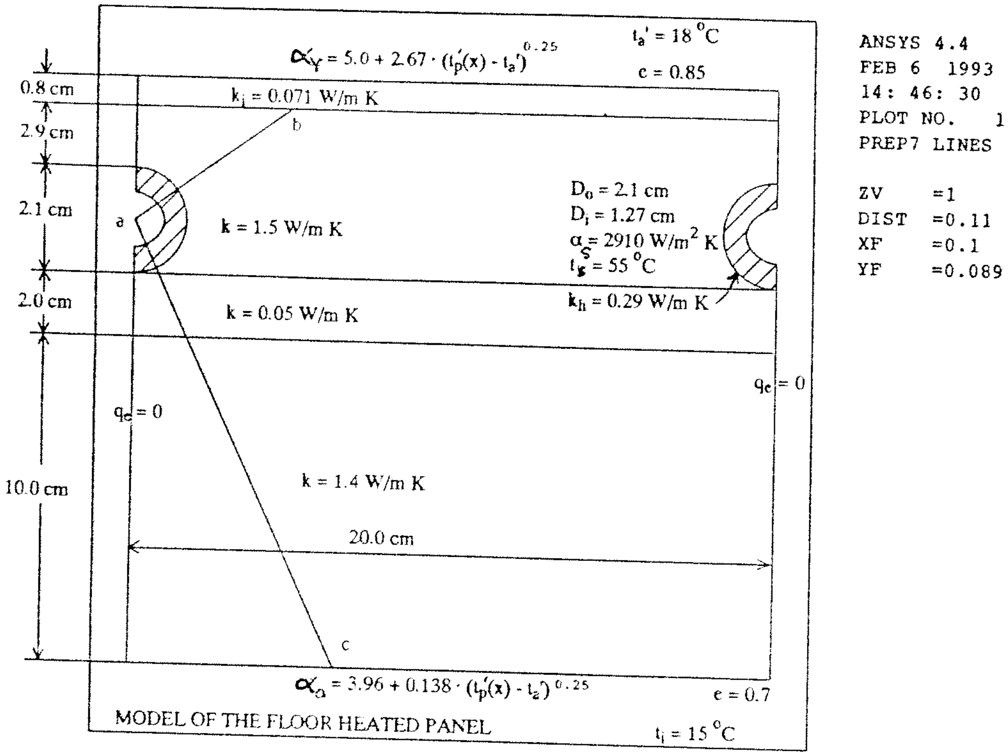
Bu mukayeseden geliştirilen algoritmanın sonlu elemanlar sonuçları ile en iyi uyumu soğladığı anlaşılmaktadır. Örneğin 52 °C su sıcaklığı hesaplanmış, buna mukabil DIN standardına göre bu su sıcaklığı 59.9 °C olarak bulunmaktadır. Sonlu elemanlar yönteminde ise bu değer, aynı ısı çıktısı için 55 °C 'dir.

Aynı problem değişik boru aralıkları için çözülmüştür. Sonuçlar Çizelge 4 'de verilmiştir.

q_Y = 3100 / 30 = 103.3 W/m²
z = 0.68 °C
AUST = 17.3 °C (d = 2 şartı için)

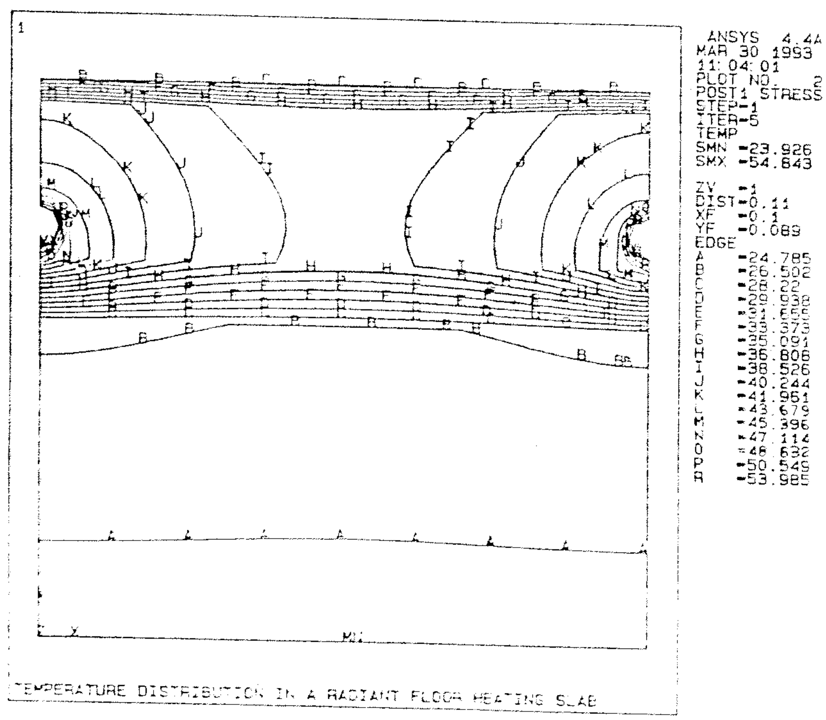
Gerekli birim ısıtma yükünü karşılamak için gerekli
T_p : 28.4 °C (Denklem 21-a, 21-b, 22, 23 ve 24 'den)
k_e = 1.26
η = 0.83 (18 numaralı denklemden)

Yukarıda izah edilen algoritma kullanılarak:
T_{max} = 30.1 °C
T_{min} = 26.8 °C
T_d = 43.7 °C
q_a = 39 W/m²
X = 0.72
T_s = 52 °C



Şekil 17. Örnek Problem.

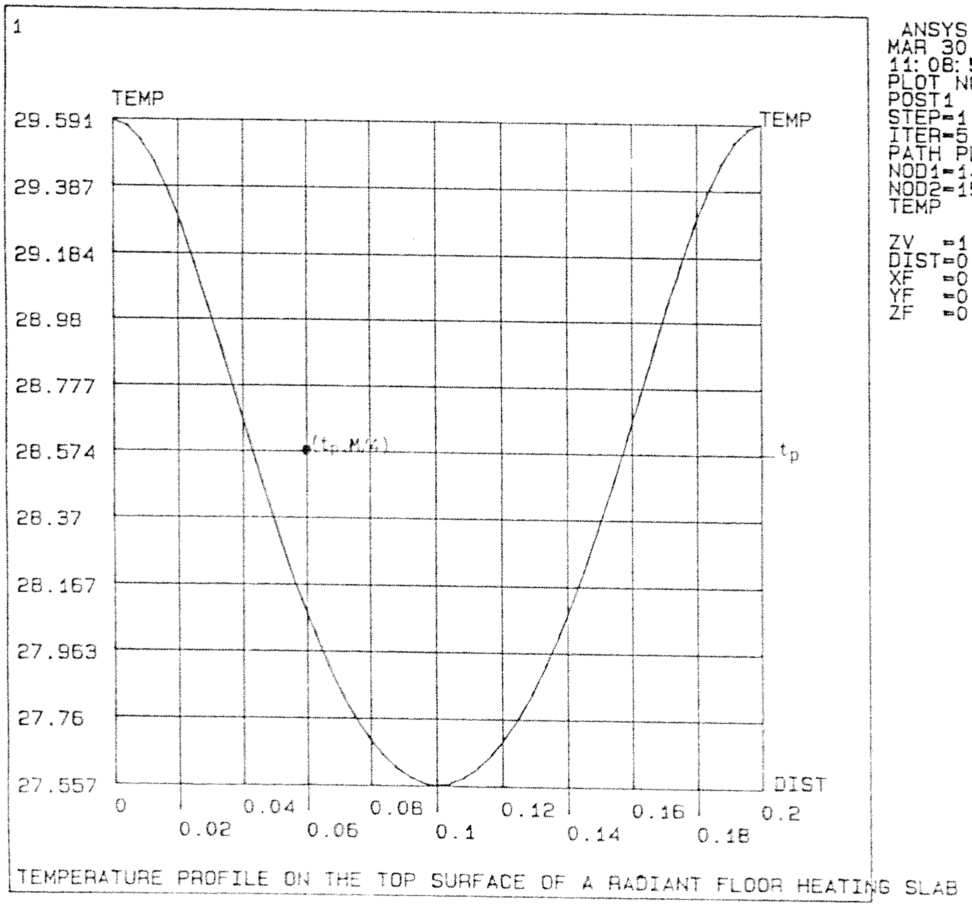
Şekil 18 'de ise aynı problemin ANSYS sonlu elemanlar paketi ile çözümü ile elde edilen sıcaklık dağılımı görülmektedir.



Şekil 18. Sonlu Elemanlar Çözümü (Kılkış, Sağer, 1993).

Bu şekilden görüleceği üzere boru çevresinde ısı akısı homojen ve simetrik değildir. Bu durum, Krinninger'in (1989) simetrik ısı akısı varsayımını doğrulamamaktadır. Buna mukabil Ek-8 'de verilen daha detaylı çizimlerden görüleceği üzere boru yüzeyindeki sıcaklık pratik ölçüler içersinde oldukça homojendir.

Şekil 19 'da ise yüzeydeki sıcaklık dağılımına ilişkin çizici çıktısı görülmektedir.



Şekil 19. Yüzeyde Sıcaklık Dağılımı.

Bu şekilden görüleceği üzere yüzeydeki sıcaklık 29.6 °C ile 27.6 °C arasında değişmektedir. 55 °C su sıcaklığı sınır şartından yüzeydeki toplam ısı transferi, integrasyondan sonra 103.85 W/m^2 olarak bulunmuştur. Aşağıya olan ısı kaybı ise 37.55 W/m^2 dır. $q_c = 0$ şartında, ısıl verim X 0.73 olmaktadır. Yine aynı şekilde görüleceği üzere profil sinüs eğrisi olmayıp, çözüm (T_p, M/4) noktasından da geçmemektedir.

Çizelge 3 'de ise geliştirilen algoritma ile, sonlu elemanlar sonuçları ve Krinninger ve DIN Standardının öngördüğü yöntemin sonuçları mukayese edilmiştir.

Steinman ve diğerleri (1989) bu ilişkilerin cam yüzeyi fazla olan ve veya az yalıtımlı köşe odalarında geçerli olmayacağını bildirmişlerdir.

Shutrum ve diğerleri (1953a ve 1953b) nin deneyleri ve Kalisperis (1985) in bilgisayar modeli AUST'un T_a 'ya yakın olduğunu bildirmişlerdir. Drake (1993) ise oda konumu ve dışa bakan duvar, pencere, kapı alanına bağlı olmak kaydı ile AUST'un T_a 'dan 7 °C daha az bir sıcaklığa inebileceğini göstermiştir. Jones ve diğerleri (1990) ise geliştirdikleri bilgisayar programında kullandıkları Monte Carlo yöntemi ile AUST ile T_a ilişkisinin büyük ölçüde oda konumu, yalıtımı, panel pozisyonu, dış duvar, pencere kapı alan yüzdesine, döşeme ve tavan özelliklerine bağlı olduğunu göstermiştir.

Sonuç olarak, AUST değerinin tesbiti, panelin performansını önemli ölçüde etkilediğinden, hesaplarda ağırlık kazanmaktadır. Kılkış ve Sağer (1993) sayısal modelleme çalışmalarında elde ettikleri örnek çözümlerde AUST'un T_a 'dan az olacağını göstermişlerdir. Elde edilen sonuçlarla Kılkış'ın (1993) geliştirdiği basit analitik algoritma uyum göstermişlerdir.

Çizelge 4. Boru Aralığının Panel Performansına Etkisi.

	q_Y [W/m^2] üst oda	q_a [W/m^2] alt oda	q_Y [W/m^2] üst oda	q_a [W/m^2] alt oda	q_Y [W/m^2] üst oda	q_a [W/m^2] alt oda
Oda Sıcaklığı, T_a / Boru Aralığı, M	18 °C	15 °C	20 °C	15 °C	24 °C	15 °C
10 cm	135.86	47.38	-	-	-	-
20 cm	103.85	37.55	98.15	38.21	-	-
25 cm	92.68	33.5	-	-	-	-
30 cm	83.12	30.1	-	-	65.43	32.9
35 cm	75.5	27.23	-	-	-	-

M = 20 cm değeri için $X : \frac{135.86}{135.86 + 47.38} \cong 0.73$ 'dür. Geliştirilen TSE standardında (1994-b) ısı verimi, X için ayrı bir yaklaşık yöntem verilmiştir (Ek-5, Yayın no: 2). Buna göre:

$$X \cong \frac{1}{1 + \frac{r_y}{r_a} \cdot \frac{(T_d - T_i)}{(T_d - T_a)}} \tag{52}$$

Söz konusu problemde r_y : 0.132 ve r_a : 0.47 olduğuna göre ve hesaplanan T_d değeri (43.7 °C) kullanılarak X : 0.76 olarak bulunur.

Bu sonuç ta TSE 'de verilen yaklaşık yöntemin dahi oldukça gerçekçi olduğunu göstermektedir.

10.3.4. Isıtılan Mekanın Modellemesi

Hem panel performansına etki eden T_a , AUST ve hava dolaşımının tesbiti hem de ısıtma (soğutma) yükünün panel sistemlere özgü bir biçimde hesaplanması için alışılagelen yöntemler yeterli değildir (Bakınız Ek 1 ve 2). Panel sistemlerde ve diğer ısıtma soğutma sistemlerinde mekan içersindeki havaya ilişkin temel termo-hidrolik denklemler aynı olmakla birlikte sınır şartları değişiktir. Mekan içersindeki geçerli denklem:

$$\frac{\partial}{\partial t}(\rho \phi) + \frac{\partial}{\partial x_i}(\rho V_i \phi) = \frac{\partial}{\partial x_i}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_i}\right) + S_\phi \tag{53}$$

$i = 1, 2, 3 \qquad x_1 = x, \qquad x_2 = y, \qquad x_3 = z$

Burada ϕ genel bağımlı değişken, Γ_ϕ , ϕ 'nin katsayısı,. S_ϕ ise uygun kazanç (source) veya kayıp (sink) ifadesidir.

Süreklilik için:

$\phi = 1, \qquad \Gamma_\phi = 0, \qquad S_\phi = 0;$

Momentum için:

$\phi = V_i, \qquad \Gamma_\phi = \mu, \qquad S_\phi = -\frac{\partial P}{\partial x_i} - \rho g_i (1 - \beta \Delta T);$

Burada $\Delta T = T - T_o$ olup T_o referans sıcaklığıdır.

Enerji denklemi için:

$$\phi = T\ , \qquad \Gamma_\phi = \frac{k}{c_p}\ \text{(ısıl diffüsivite)}, \qquad S_\phi = \text{Isı kazanç/kayıp ifadesidir.}$$

3 boyutlu bir çözüm için i : 1, 2, 3 'tür.

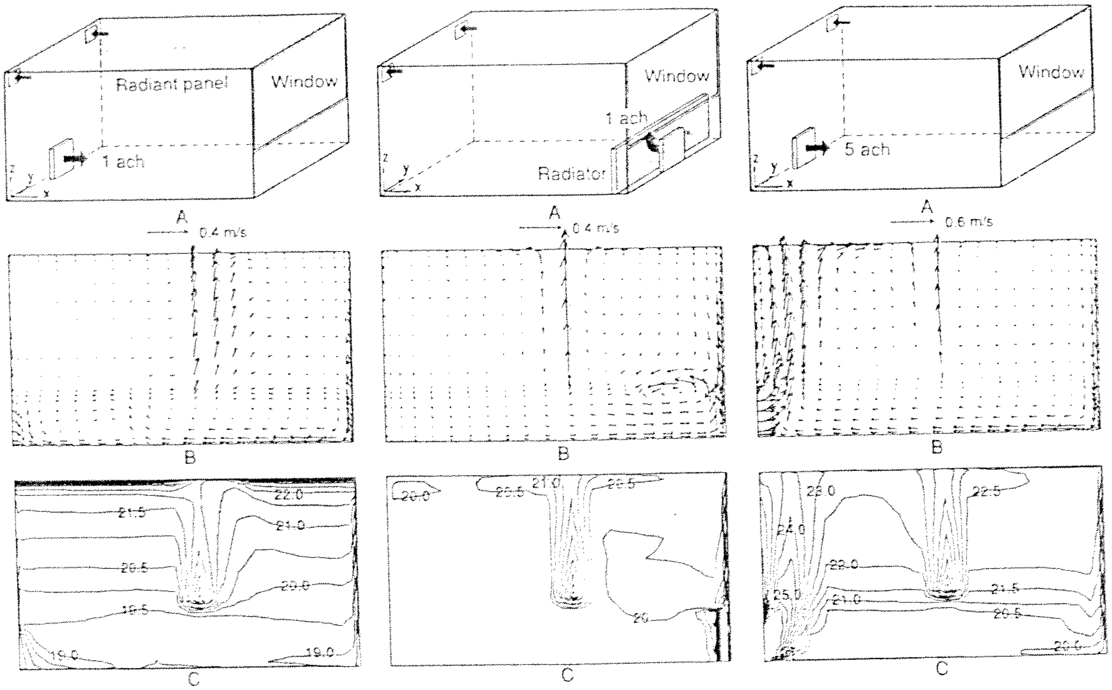
Nielsen (1974), sayısal yöntemleri oda içi hava hareketi ve ısı transferine tatbik eden ilk araştırmacılardan biridir. Problemi iki boyutlu olarak çözmüş, türbülansı ise (k - ε) modeli ile uygulamıştır. Hava hareketi kaldırma (bouyancy) terimi ihmal edilerek hesaplanmış ve dikey yöndeki sıcaklık değişimi oda yüksekliği boyunca incelenmiştir. Quingyang ve Jiang (1992) sınır şartları uygun tatbik edildiği taktirde (k - ε) modelinin hala en geçerli türbülans modeli olduğunu belirtmiştir. Bununla birlikte, özellikle mekanik hava değişiminin olmadığı ve panel ısıtma/soğutma tatbik edilen bir ortamda, duvara yakın bölgelerde k - ε modelinin adapte edilmesi gerekir (Patel ve diğerleri, 1985). Bu amaca en uygun modelin Launder ve Sharma (1974) 'nın modeli olduğu belirtilmiştir. Duvara yakın bölgelerdeki bu tanım, panel yüzeylerine yakın bölgeleri de içerir, sayısal çözümler için çok küçük elemanlar alınması gerekli olur.

Şu ana kadar ki çalışmalar genellikle 2 boyutlu olmuştur. Yukarıdaki genel modelin 3 boyutlu olarak çözümü sonlu hacımlar yöntemi ile de mümkündür (Patankar, 1980). Bu alandaki genel çalışmaları Whittle (1986) detaylı bir şekilde özetlemiştir.

Yukarıda da belirtildiği gibi genel çözümün aynı olmasına rağmen, panel ısıtma ve soğutma sınır şartları ile çözüm literatürde çok az dikkat çekmiştir.

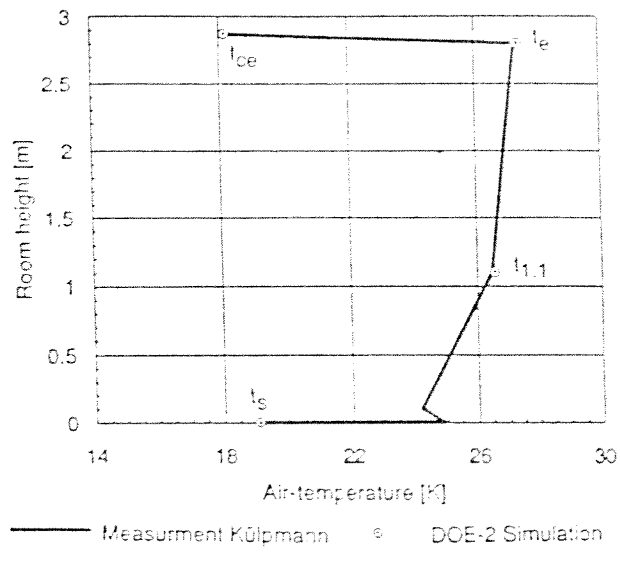
Zumeurenau ve diğerleri (1987) panel ısıtma sistemi ile sıcak hava sistemini mukayese amacı ile bir bilgisayar programı geliştirmişlerdir. Elde ettikleri sonuçlarla ısı yükünün tavandan panel ısıtmada daha az olacağını göstermişlerdir. Program, matematiksel bir model ile çalışmakta, hissedilir ısıtma yükünün saatlik değişimlerine müsaade etmektedir. Hesaplarında ısıtma yükünün tüm öğeleri ile kısa dalga güneş ışıması ısı transferini de gözetmişlerdir. Bu yaklaşımla problem global olarak çözülmüş, hava hareketlerinin yerel değişimi detayına girilmemiştir. 6x6x3.6 metre boyutlarında ve bir kenarı dışarı bakan bir örnek odada ısıtma yükünün, sıcak havalı sisteme göre % 38 daha az olacağını hesaplamışlardır.

Quingyan (1990) PHOENICS-84 (1987) program pakedi ile üç ayrı sistemi mukayese etmişlerdir. Bunlar tavandan ısıtma, radyatör ve sıcak havadır. Tavandan ısıtma ve radyatörlü sistemde saatte 1 hava değişimi, sıcak havalı sistemde saatte 5 hava değişimi alınmıştır. Hava hızına ve sıcaklığına ilişkin örnek çözümler Şekil 20 'de gösterilmiştir.



Şekil 20. PHOENICS-84 Programı ile Örnek Çözüm (Quingyan, 1990).

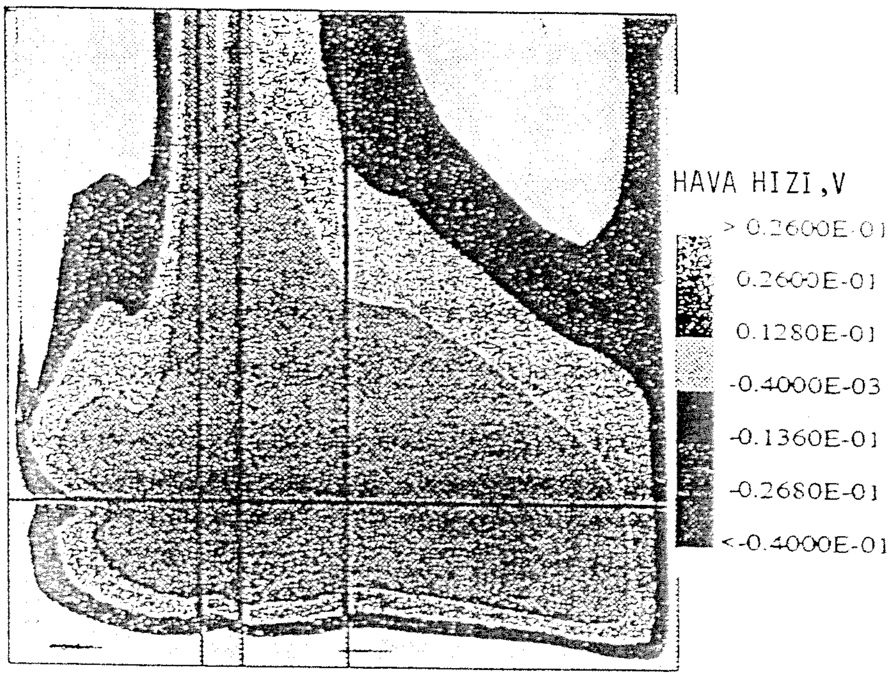
Quingyan aynı zamanda ısı yüklerindeki farklılıkları araştırmış, tavandan ısıtmada ısı yükünün radyatörlü sisteme oranla % 17 daha fazla olacağını hesaplamıştır. Ancak bunun nedenini tavan panelinin arkasının yalıtılmamış varsayıldığı şeklinde açıklamıştır. Qingyan ve Kooi (1988) kişisel bilgisayarlarda kullanılabilecek ikinci bir bilgisayar programı daha geliştirmişlerdir. Bu program soğutma sistemleri için kullanılmaktadır. Zweifel (1993), DOE-2 programını 2 bölgeli model şekline dönüştürerek hava hareketlerini ve sıcaklık dağılımını hesaplamıştır. Şekil 21 'de bulgular deneysel ölçümlerle mukayeseli olarak görülmektedir.



Şekil 21. Panel ile Soğutulan Odada Sıcaklık Dağılımı (Zweifel, 1993).

Udagawa (1993) ise havanın ısı ve nem dengesi denklemlerini kullanarak, sonlu farklar yöntemi ile panel ile soğutulan oda için bir model geliştirmiştir.

Ochocinski (1992), Harwell Laboratuvarında (İngiltere) FLOW 3D adlı sonlu hacımlar programı ile tavandan askı panel ile ısıtma problemini büyük bir oda için incelemiştir. Örnek bir hız dağılımı çözümü Şekil 22 'de verilmiştir.



Şekil 22. Askı Tavan Paneli ile Isıtma için Sayısal Çözüm (Ochocinski, 1992).

Jiang, Z. ve diğerleri (1992) geliştirdikleri sayısal yöntemle, tavandan soğutmada daha homojen bir hava sıcaklığı oluştuğunu, buna mukabil hava hızının arttığını görmüşlerdir.

Geliştirilen sayısal çözüm tekniklerinin sınanmasında kullanılmak üzere 7.2 m x 5.4 m x 2.4 m boyutlarındaki bir odada hız, türbülans kinetik enerjisi ve hızlar tam 205 noktada ölçülerek bir veri bankası oluşturulmuşsa da, bu odada panel sistemler incelenmediği için, panel sistemlerde kullanılacak özel paket programlar için bu düzeyde deneysel verilere hala gerek vardır. Chapman ve Jones ise (1992) Monte Carlo yöntemi ile bir kapalı hacmin panel ısıtma özelinde ortalama ışıma sıcaklığını (MRT) hesaplayacak bir bilgisayar programı geliştirmişlerdir.

Yukarıdaki bölümde izleneceği üzere özellikle son yıllarda etkin ve güçlü bilgisayar modelleri ve paket programları geliştirildiği halde, döşemeden ısıtma konusunda sistematik bir araştırma yapılmamıştır. Isıtma yükünün azalmasına etken parametrelerin ise tümü incelenmemiş veya yeterli bilgi birikimi oluşmamıştır. Bu noktada yine sorun, bu kuvvetli bilgisayar programlarının proje yapan herkezce kolaylıkla satın alınıp kullanılabilecek düzeyde olmayışıdır. Bu nedenle projenin amacı, sayısal yöntemlerle elde edilecek bilgiler ışığı altında daha kolay bir algoritmanın oluşmasına katkıda bulunmaktır.

Bu amaçla ODTÜ Biltir Merkezinde ANSYS programı ve FLOTRAN yazılım paketi kullanılmıştır. Bu yöntemle döşemeden ısıtma, radyatörlü sistem mukayesesi yapılacak şekilde uyarlamalar yapılmıştır.

10.3.4.1. Sayısal Yaklaşım

Bu yaklaşımın detayları Ek-8 'de örnek çözümlerle birlikte verilmiştir. ANSYS ve FLOTRAN paketi çözümleri ışıma ve taşınım için ayrı ayrı çözülmüş ve süperpozisyon yöntemi ile bitiştirilerek çözümler anlaşana kadar sürecek iterasyon döngüleri öngörülmüştür.

Panel ışı çıktısının iki ögesi olan ışıma, ısıtılmayan duvarların ortalama sıcaklığına (AUSI), taşınım ise iç hava sıcaklığına (T_a) bağıntılıdır.

Literatürde bir çok araştırmacı tarafından varsayıldığı gibi hava ile panel arasında ışıma ısı transferi ihmal edildiğinde, bu iki ısı transfer mekanizması pratik olarak birbirinden bağımsız mütalaa edilebilir. Ancak, iç mekandaki sınır koşulları bu iki mekanizmaya da bağlı kaldığından ayrı yapılacak çözümler arasında uyum ve anlaşma sağlanmalıdır.

Denklem 1 ve 2 'de görüldüğü üzere zaten bu iki ısı transfer mekanizması birbirinden ayrı mütalaa edilmiştir (q_{yr} ve q_{yc}). Sayısal olarak yapılacak iki ayrı çözüm neticesinde elde edilecek sınır çözümleri birbirlerini etkilemektedir. Örneğin ışıma çözümü ile elde edilen duvar sıcaklıkları, doğal taşınım çözümü ile elde edilen hava dolaşımı ve yere hava sıcaklığı ile değişime uğramaktadır. Bu yaklaşıma ilişkin genel çözüm mantığı Şekil 23 'de verilmiştir. Bu akış içersinde iki temel döngü vardır. Bunlardan birincisi, AUST değerinin odaklanmasına ilişkindir. İkinci döngü ise panel ısı çıktısı ile ısı yükünün dengelenmesine kadar sürer. Üçüncü bir döngü ise panel yüzeyindeki ışıma ve doğal taşınım katsayılarının düzeltilmesi için gerekebilir.

Jiang ve diğerleri de (1992) kullandıkları sayısal yöntemde benzer bir yaklaşımla, önce duvar sıcaklığını tahmin ederek elde ettikleri çözümü geri besleyerek, duvar sıcaklığını düzelterek döngüye devam etmişlerdir. Ancak bu döngü içersinde panel sıcaklığını sabit kabul etmişlerdir.

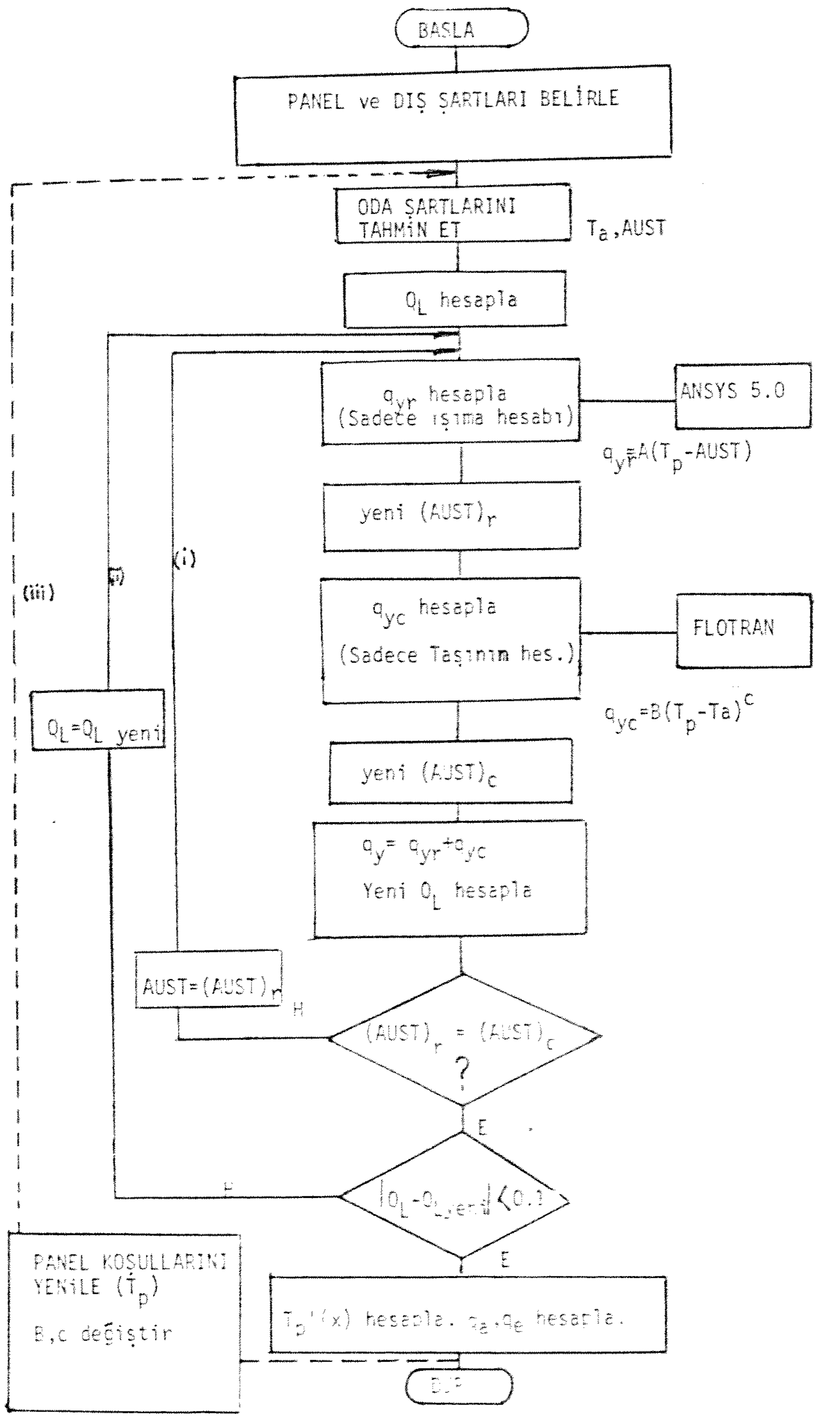
Şekil 24 'de döşemeden ısıtılan ve tek tarafı dış cepheli olan bir odada elde edilen örnek bir hava hızı dağılımı çözümü görülmektedir.

Şekil 25 'de ise duvar iç sıcaklıkları da dahil olmak üzere sıcaklık dağılımı verilmiştir.

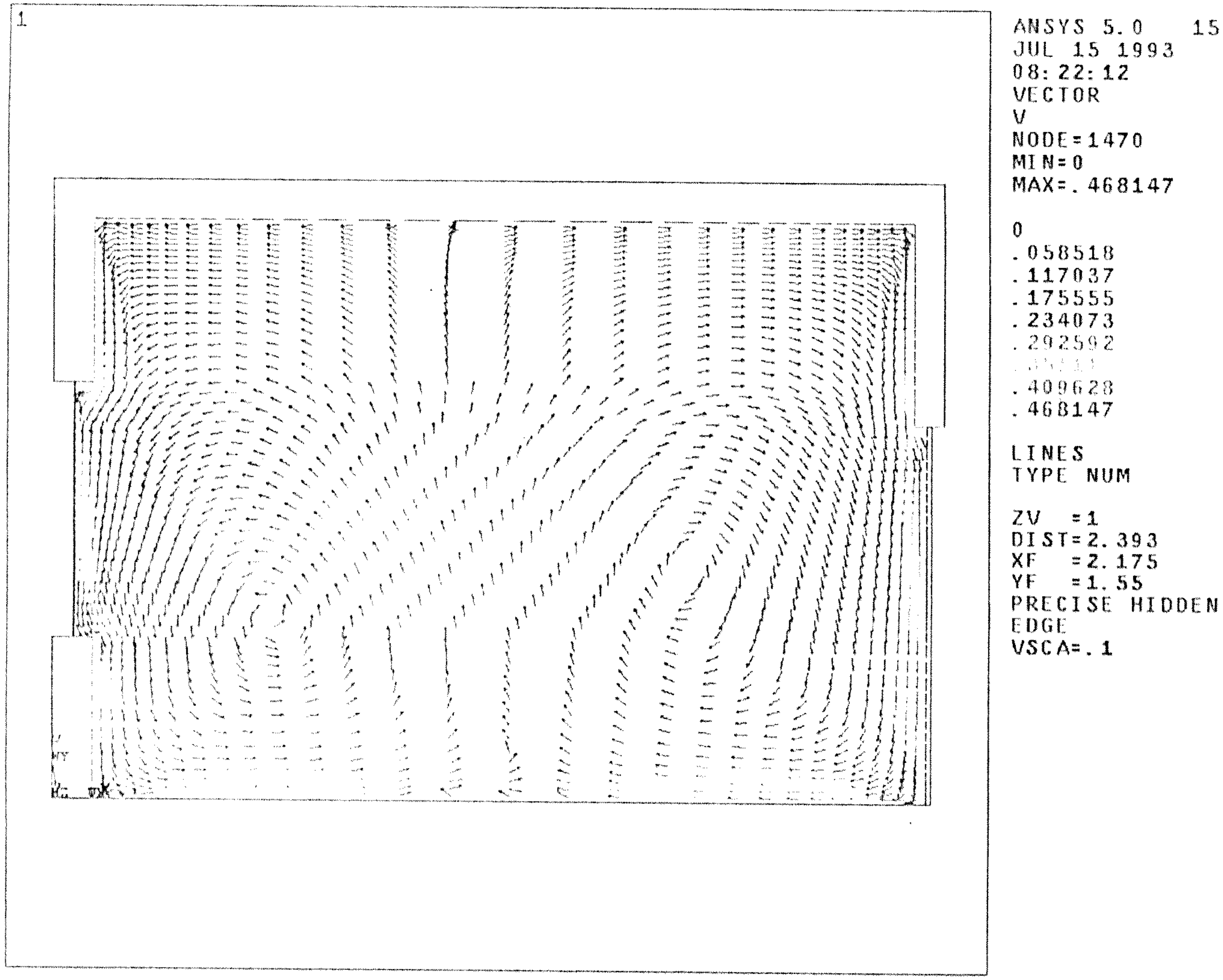
Şekil 26 'da ise basınç dağılımı görülmektedir.

Bu modelde, önceki bölümde açıklanan panel içi ısı transferine ilişkin ANSYS çözümünün entegrasyonu mümkündür. Bu şekli ile problem ilk kez hem döşeme içi, hem de ısıtılan mekan entegrasyonu içersinde çözülecek bir algoritmaya kavuşturulmuş olmaktadır.

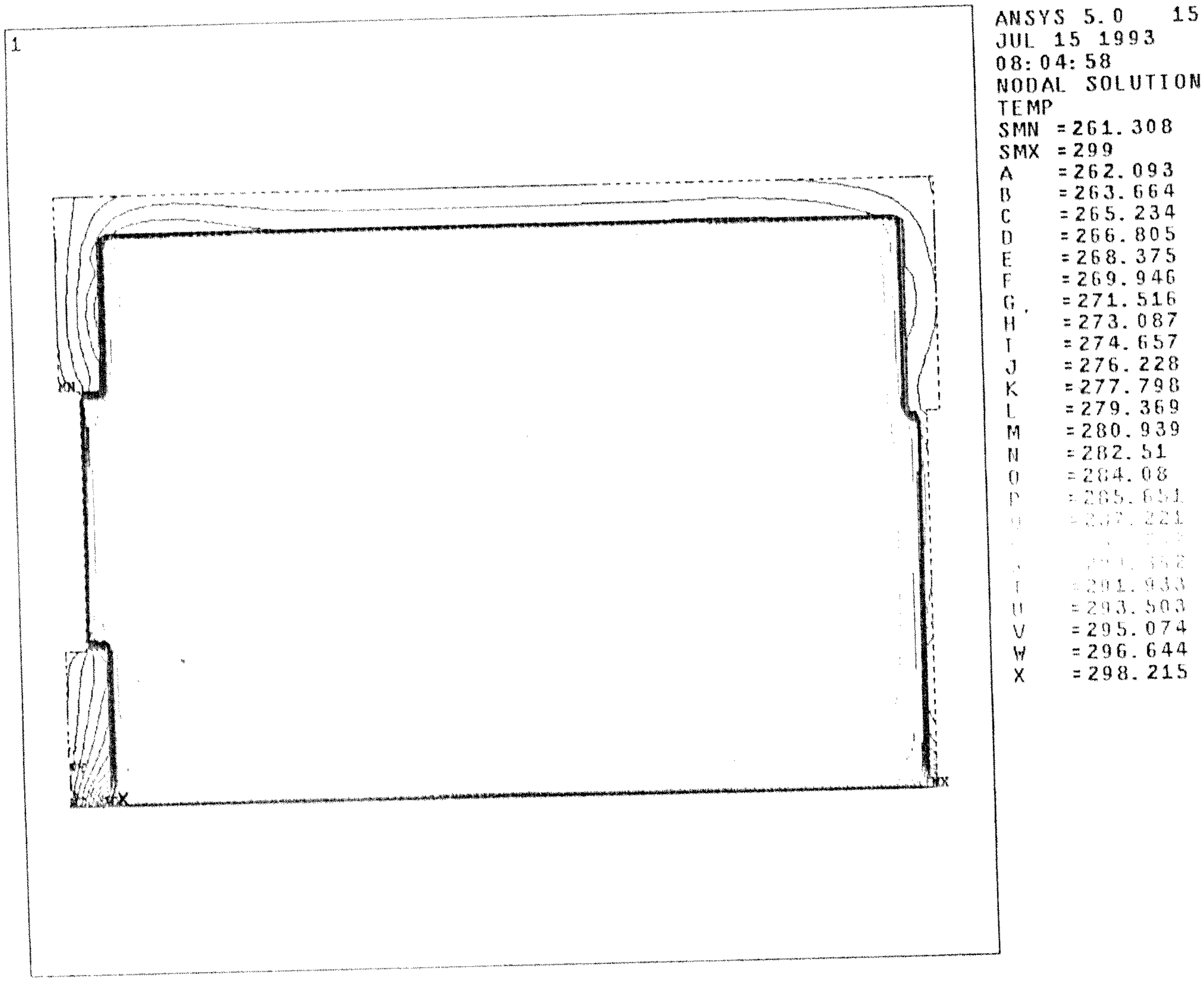
Diğer bir ilginç konu ise, aynı zamanda ısıtma yükünün panel performansı ile ilişkilendirilebilir olmasıdır.



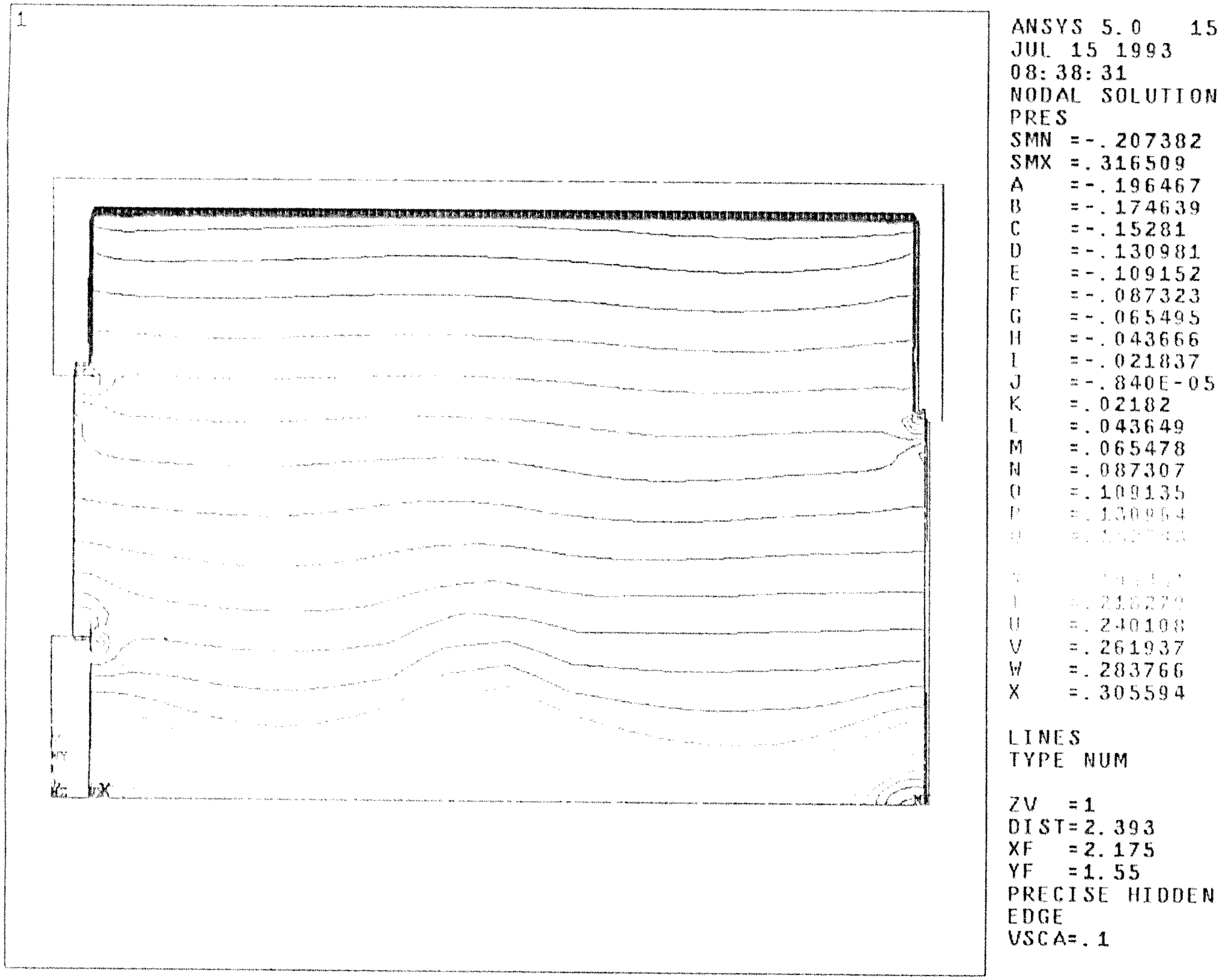
Şekil 23. Sayısal Çözümde İterasyon (Kılkış, Sağer, Uludağ, 1994).



Şekil 24. Örnek Hava Hızı Çözümü (Kılkış, Sağer, Uludağ, 1994).



Şekil 25. Örnek Sıcaklık Dağılımı Çözümü (Kılkış, Sağer, Uludağ, 1994).



Şekil 26. Örnek Basınç Dağılımı Çözümü (Kılkış, Sağer, Uludağ, 1994).

10.3.4.2. Yaklaşık Çözüm

Önceki bölümde tartışıldığı üzere hergünkü kullanıcıya karmaşık, zor ve uzun gelen sayısal çözümlerin anlamlı bir şekilde irdelenmesi ve analitik bir yapıya kavuşturulması gerekir. Kılkış'ın önceki yaklaşımları da gözönünde tutularak, TSE standardında (1993-b) verilen algoritma hazırlanmıştır. Bu algoritma iki amaca hizmet etmektedir:

a) Isıtma yükü hesabı,

b) AUST ve eşdeğer konfor için standart iç hava sıcaklıkları.

Isıtma yükünde döşemeden ısıtmaya özgü ısıl konfor önem taşımaktadır. Ek-1 'de verilen yaklaşımla, eşdeğer ısıl konfor için standart iç hava sıcaklıkları hesaplanmıştır (Bakınız, Ek-5, Yayın no: 2, Çizelge 2).

Isıtma yükünde önemli bir yeri olan doğal infiltrasyon sızdırganlık katsayıları için gerekli düzeltmeler incelenmiş, duvar iç yüzeylerindeki film katsayısındaki olası değişimler sayısal çözümler ışığı altında incelenmiştir. Yapı bileşenlerinde ısı depolaması için hazırlanan "Pik Yük Traşlama" katsayıları ve ısıtma yükü hesaplama algoritması Ek-2 'de sunulmuştur. Bu yönteme ve hazırlanan TSE 'ye uygun bir bilgisayar programı ise Ek-7 'de tanıtılmıştır.

Tavandan soğutma için soğutma yükü hesabı için geliştirilen hesap yöntemi ise özetle:

$$\text{AUST} = \frac{7}{T_b - 45} \qquad \text{(Kılkış,1993-b)} \qquad (54)$$

Burada:

z oda pozisyon faktörü olup Denklem (8) için verilen değerler geçerlidir.

Sızdırganlık katsayıları : 0.7 * standart sızdırganlık katsayısı,

İç konfor sıcaklığı : Standart sıcaklık değeri + 2 °C,

Panel soğutmada yük : C_s * Hesaplanan Yük.

C_s katsayısı Çizelge 5 'de verilmiştir.

Çizelge 5. Soğutma Hissedilir Yükü Düzeltme Katsayısı, C_s.

İklim Koşulu	C_s Yapı Türü		
	Hafif	Normal	Çok Yoğun
1. Isı İklim Bölgesi	0.90	0.85	0.80
2. Isı İklim Bölgesi	0.85	0.75	0.70
3. Isı İklim Bölgesi	0.80	0.70	0.65

m_y ise binanın ağırlık yoğunluğu (kg/m^2) olup ilgili Türk Standardında (1993-b) Çizelge 1 dip notu olarak verilmiştir.

10.4. Proje Gerçekleşen Ürün ve Hizmetler

Proje başlangıcından, sonuç raporunun yazımına kadar geçen süre içersinde üretilen ürün ve hizmetler şu şekilde özetlenebilir:

a - Panel Isıtma ve Soğutma Tasarım Algoritması

i - Panel içi ısı transferi :

Panel içi ısı transferinin hesabında kullanılacak olan bu algoritma ASHRAE Handbook Chapter 6 'da yer alacağı gibi, hazırlanan Türk Standardının (1993-b) ikinci kısmında verilmiştir (ayrıca bakınız Ek-5, Yayın no.ları : 7, 10 ve 15 (Basitleştirilmiş yöntem), 11 (Detaylı yöntem)). Soğutma yükünün (-) işaret ile gösterilmesi ile aynı algoritma panel soğutmada geçerli olmaktadır. Tek değişiklik T_{max} ve T_{min} 'in yer değiştirmesidir. Dolayısı ile prensip itibarı ile geliştirilecek olan herhangi bir bilgisayar programı, nomogram veya hesap cetveli (örnek taslak için Şekil 30 'a bakınız) de hem ısıtma hem de soğutma için geçerli olmaktadır. Dolayısı ile bu kapsamdaki bir algoritma literatürde ilk kez geliştirilmiş olmaktadır. Ayrıca panel soğutma için, yeterli bir hesap yöntemi ilk kez uygulamaya çıkacaktır. Bu konudaki iki diğer çalışmadan birincisi Dr. Feustel tarafından çalışmaları sürdürülen (Lawrence Berkeley Lab. de) panel soğutma bilgisayar programı, ikincisi ise DIN tarafından yakında ilk tasarısı görüşe sunulacak olan panel soğutma Alman Standardıdır. Literatürdeki diğer münferit deneysel ve sayısal çalışmalar ise uygulamaya yaygın ve sistematik bir şekilde henüz yansımamıştır. Bu nedenle hazırlanan "SOĞUTMA" programı da ilk tasarıma uygulanabilir bilgisayar programı niteliğini taşımaktadır (Ek-9).

Geliştirilen algoritma tasarıma ilişkin bütün değişkenler arasındaki etkileşimi bir arada gözönünde tutmaktadır. Bu nedenle seçilecek bir amaç fonksiyonu çerçevesinde sayısal, analitik ve/veya iteratif bir optimizasyona imkan sağlar. Panel içi ısı transferi algoritması, geliştirilen panel ısıtma/soğutma yükü tasarım algoritması ve hidrolik hesaplarında iyileştirmeler ile birlikte kullanıldığında bütün halinde tasarıma imkan tanımaktadır. Yine bu durumu ile genel optimizasyon mümkün olmaktadır. Örneğin bir tasarımcı minimum işletme ve yatırım masrafı amacı ile bina optimum yalıtımı ile bu döşemeye serilecek boru optimum aralığı çözülebilir. Bu tür uygulama yöntemlerinin geliştirilmesi ise algoritmayı kullanacak uygulayıcı ve araştırmacılara bırakılmıştır. Elektrikli panel ısıtmada ise aynı algoritma geçerlidir. Burada, boru dış sıcaklığı elektrik kablosu sıcaklığına dönüşmekte, T_s değişkeni kalkmaktadır (TSE 1993-b). Çizelge 6 'da geliştirilen algoritmanın ana hatları özetlenmiştir.

Çizelge 6. Panel İçi Isı Transferi Algoritması

Ana Hesap Girdisi*	İkincil Hesap Değişkeni	Açıklama
Odanın ısıtma/soğutma yükü, Q_Y panel sayısı (n_p)	Panel alanı, A_p	Panel birim yükü, q_Y hesaplanır (soğutmada - q_Y)
Oda boyutları	D_e (Eşdeğer çap)	Oda büyüklüğü düzeltme katsayısı hesaplanır.
Denizden olan yükseklik, h		Yükseklik düzeltmesi hesaplanır.
Panel yüzey yayınım katsayısı, e Oda iç sıcaklığı, T_a, panel arkası komşu sıcaklık, T_i		U_r hesaplanır.
Oda konumu, kat tipi, dış sıcaklık, T_o	d, z	AUST ve U_c hesaplanır.
q_Y, AUST, T_a, U_r, U_c		Gerekli T_p hesaplanır.
Yüzey kaplama kalınlığı (x_i), ısı iletim katsayısı (k_i), panel ısı iletim katsayısı (k), ısıtıcı/soğutucu elemanın pozisyonu (L), dış çapı (D_o) ve aralığı (M).	Eşdeğer ısı iletim katsayısı (panel boyunca), k_e Panel ısıl direnci, r_p	Kanatçık verimi h hesaplanır.

* TSE 1993-b 'de değişik kullanılan simgeler için bu standard da EK-A'ya bakınız.

Çizelge 6. (Devamı)

q_Y, M, AUST, T_a, U_r, U_c		T_{max} (soğutmada T_{min}) ve T_{min} (soğutmada T_{max}) hesaplanır.
Panel ısıl direnci, r_y	T_d	Boru (elektrik kablosu) dış sıcaklığı
Panel arkası ısıl direnci (r_a)	X (yaklaşık olarak)	Panel ısıl verimi kazan yükü hesaplanır.
Kenar (q_e) ve arka (q_a) ısıl kayıpları (kazançları)	X (kesin)	Panel ısıl verimi kazan yükü hesaplanır.
Hidrolik sistemde:		
D_i, k_h akışkan özellikleri	Akışkan debisi, α_s hesaplanır.	T_s hesaplanır.
ΔT, ΔP	ΔP (basınç kaybı) hesaplanır.	$T_{gidiş}$ / $T_{dönüş}$ hesaplanır.
Elektrik sistemde: Tel genleşme ve direnç özellikleri	T_b sıcaklığına göre düzeltilir (TSE 1993-b)	Gerekli akım ve elektrik gücü hesaplanır.

NOT : Terimler ve Tarifler için TSE (1993-a) 'ya bakınız.

Bu algoritma ile panelin performansı ısı alışverişinde bulunduğu oda ve komşu ortam ile ilişkilendirilmiş olmaktadır. Bu algoritmada yer alan analitik ifadeler günlük tasarımda kolaylıkla uygulanabilecek düzeydedir (Kılkış, 1993-c. Bu yayın Ek-5, 19. sırada verilmiştir). Panel ısıl direncinin hesabında geliştirilen yöntem ise Ek-3 'de özetlenmiştir (Kılkış, 1993-c). Bu suretle genelde ihmal edilen veya yeterli hassasiyette hesaplanmayan boru direnci hesabına açıklık getirilmiştir (Kılkış, 1993-c).

ii - Panel Isıtma (Soğutma) Yükü :

Panel ısıtma yükünün hesabında üç ana düzeltme faktörü geliştirilmiştir. Bunlar, pik yük azalması (Peak Load Shaving), tabii infiltrasyon katsayılarında azalma, iç standart hesap sıcaklıklarında azalma. Ayrıca iç yüzeylerde film katsayılarında azalma için yaklaşık bir katsayı önerilmiştir. Bu algoritma kullanılarak etkileşimli bir bilgisayar programı EXPERT system anlayışı ile geliştirilmiştir (Ek-7). Bu program standart ısı yükü hesabı da yaparak sonuçları mukayese edebilmektedir. Soğutma yükünün hesabında ise aynı yöntem, değişik katsayılar geçerlidir (Çizelge 5).

b - Geliştirilen Bilgisayar Programları

Proje çalışmaları neticesinde uygulamaya dönük, etkileşimli 4 adet bilgisayar programı geliştirilmiştir. Bu programlar takibeden sayfada disket olarak verildiği gibi, rapor eklerinde detaylı olarak açıklanmış, örnek çözümler sunulmuştur. Kısa kullanıcı klavuzları da hazırlanmıştır.

Çizelge 7. Bilgisayar Programları

ADI	NİTELİĞİ	Detaylı Bilgi için:
ISITMA 2	Döşemeden ısıtma sisteminin tasarım ve analizini etkileşimli olarak yapar.	EK - 6
SOĞUTMA	Tavan içinden soğutma sisteminin tasarım ve analizini etkileşimli olarak yapar.	EK - 9
ISI 2	Tez çalışması olarak geliştirilen bu program hazırlanan Türk Standardına uygun panel ısıtma yükünü hesaplar.	EK - 7
KAR	Panel içi ısı transferi modeli kullanılarak dış ortamlarda kar ve buz eritme sistemlerinin tasarım ve analizini etkileşimli olarak yapar.	EK - 10

Tüm programlar DOS ortamında ve IBM uyumlu PC bilgisayarlarda çalışır tarzda hazırlanmıştır. Gerekli grafik ve matematiksel işlem yazılımları da yüklü olduğundan, normal şartlarda .EXE dosyaların işletimi ile programların çalışması beklenir. Herhangi bir kullanım ve işletim problemi ile karşılaşıldığında, Ar. Gör. Mahmut Uludağ (Ankara, ODTÜ Biltir Merkezi : 312 210 12 76) ile temas kurulmalıdır. Bu dört ana programa ek olarak, endüstrinin gerek duyacağı özel uygulama programları da hazırlanabilecektir. Bu programlar ekli poşet içersinde 3 1/2" disketlerde verilmiştir.

c - Tasarım Nomogramı

ASHRAE 'ye de teklifi ve kabulü yapılan bir Tasarım Nomogramı geliştirilmiş olup detaylı bilgi aşağıda verilmiştir.

d - Yayınlar

Proje süresince hazırlanan 26 adet yayın kopyeleri tarih dizini ile EK - 5 'de verilmiştir.

e - Diğer Hizmetler

İki adet Türk Standardı hazırlanmış olup, kamu oyunun panel ısıtma ve soğutma konusunda eğitilmesi ve bilgilendirilmesi için girişimler devam etmekte olup, önerilen çalışmalar 8. bölümde verilmiştir. Adı geçen standartlar da Ek - 5 'de verilmiştir.

Projeye tamamen uluslararası nitelik kazandırılmış, ASHRAE ile teknik ve idari konularda en üst düzeyde ve etkin ilişkiler tesis edilmiştir. TÜBİTAK / ODTÜ / ASHRAE ortak kesin raporu (Ek-5, Yayın no: 19) ise proje çalışmalarını özetleyen ve ASHRAE Handbook Chapter 6 özelinde irdeleyen geniş kapsamlı ve özgür bir rapordur.

Şekil 27 ve 28 'de, hazırlanan Üniversal Panel Isıtma ve Soğutma Abakları SI ve Metrik Birimlerde verilmiştir. Bu tasarım abağının özellikleri:

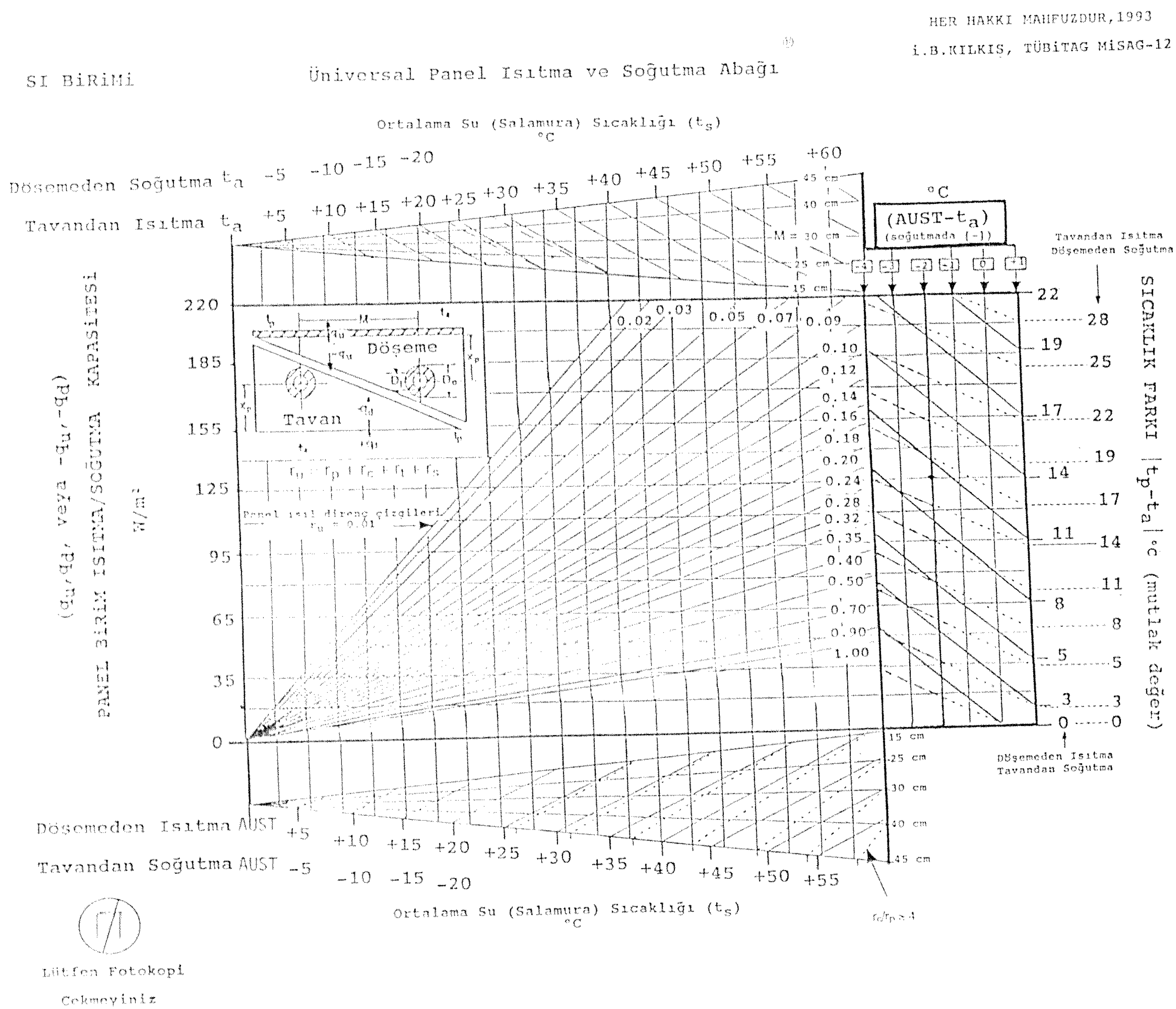
i - İlk kez panel soğutma ve ısıtma tasarımı tek bir abakta yapılabilmektedir.

ii - İlk kez değişik AUST değerleri, oda konumuna göre girilebilmektedir.

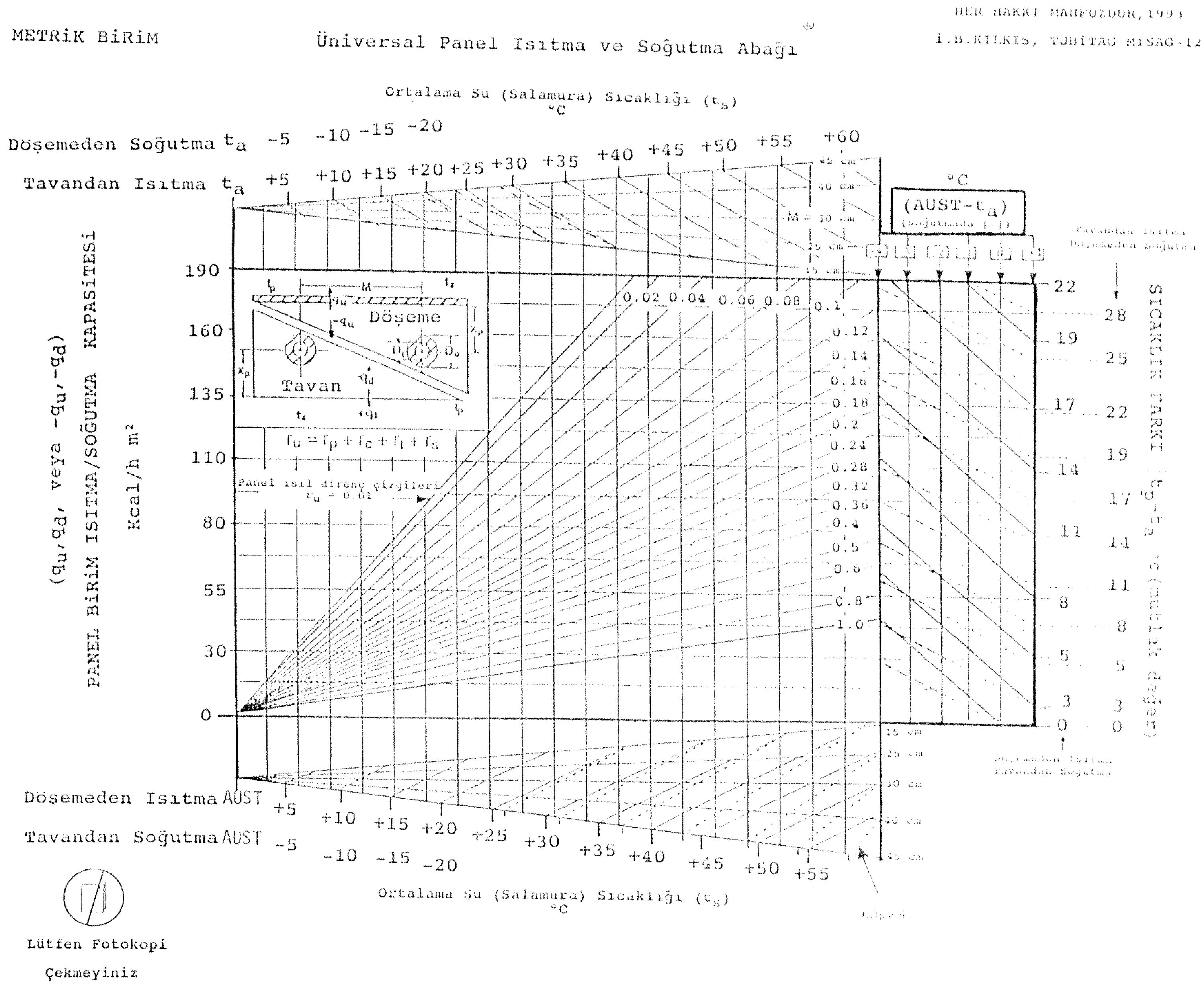
iii - İlk kez geliştirilen panel ısıl direnç tarifi ve "karakteristik kanatçık kalınlığı" tarifleri ile türlü panel pozisyonu (tavan ve döşeme) ve pratik olarak her türlü panel tipi için kullanılabilmektedir.

iv - Değişik oda sıcaklıklarına göre düzeltmeyi kendi kendine yapabilmektedir.

Bu nomogramın SI ve IP birimlerinde olan versiyonları ASHRAE Handbook, Chapter 6 'da da yer alacaktır (Kılkış, 1996). Nomogramların kullanımı ve panel dirençlerinin hesabı, Ek-5, Yayın no: 19 'da verilmiştir. Ayrıca Ek-3 'de özetlenmiştir.



Şekil 27. Üniversal Tasarım Nomogramı, SI Birimi (Kılkış, 1993-c).



Şekil 28. Üniversal Tasarım Nomogramı, Metrik Birim (Kılkış, 1993-c).

Örnek 1 : Döşemeden Isıtma

Panel birim ısıtma yükü : 125 W/m^2 (q_u)
T_a = 18 °C, r_u = 0.09
AUST = 16 °C
Boru aralığı = 25 cm

Çözüm :

Soldaki dik eksenden, 125 W/m^2 girilir (q_u)
AUST - T_a = 16 - 18 : - 2 °C olduğundan, ilgili sağ eksendeki - 2 çizgisinden gerekli etkin döşeme yüzey sıcaklığı, T_p okunur :
T_p - T_a ≅ 10 °C olduğundan, T_p = 18 + 10 = 28 °C.

İkinci hareket olarak, 125 çizgisi ile r_u = 0.09 çizgisinin kesiştiği noktadan, aşağıya, boru aralığı M = 25 cm eğik çizgisi ile buluşana dek inilir. Bu noktada, T_s ≅ AUST + 23 °C okunur. Dolayısı ile ortalama su sıcaklığı olarak
T_s ≅ 16 + 23 = 39 °C gerekmektedir.
Aynı nomogram, belli bir su sıcaklığı, yüzey sıcaklığı da girilerek diğer yönde, örneğin panel birim ısıtma/soğutma kapasitesinin tahmininde de kullanılabilir.

Örnek 2 : Tavandan Soğutma

Panel birim soğutma yükü : 95 W/m^2 (- 9 d)
Boru aralığı = 15 cm
r_u = 0.03
T_a = 24 °C
AUST = 25 °C

Çözüm :

Yine soldaki dik eksen ile aşağıdaki ortalama su sıcaklığı ekseni kullanılır.
AUST - T_a ≅ (25 - 24) = - 1 °C olduğuna göre | T_p - T_a] ≅ 8 ° C
T_p = T_a - 8 °C = 24 - 8 = 16 °C. Bu sıcaklığın yoğuşmaya neden olup olmayacağı psikiyometrik abaklarda kontrol edilmelidir.
(Soğutmada mutlak değerler okunur).
r_u = 0.03 çizgisi ve M = 15 çizgileri aracılığı ile,
T_s = 25 - 13 °C = 12 °C.
Eğer boru aralığı 30 cm olsa idi,
T_s = 25 - 15 °C = 10 °C.

Örnek 3 : **Tavandan Isıtma**

Panel birim ısıtma yükü : 155 W/m^2 (q_d)

Boru aralığı : 25 cm

r_u = 0.03

T_a = 20 °C

AUST = 20 °C ; AUST - T_a = 0 °C

Bu sefer yukarıdaki eksen ve sağdaki eksenin kırık çizgileri kullanılır.

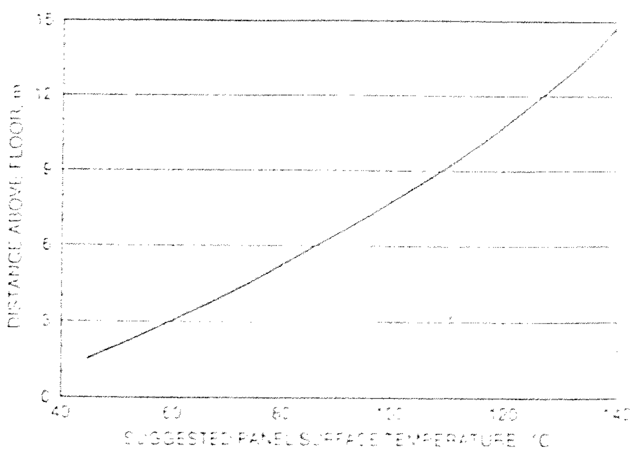
Sağ eksenden, AUST - T_a = 0 için, ve kırık çizgiler kullanılarak,

T_p - T_a = 21 °C

Dolayısı ile,

T_p = 20 + 21 = 41 °C

Bu tavan sıcaklığının insan konforuna göre uygun olup olmadığı kontrol edilmelidir. Bunun için ASHRAE El Kitabı, Bölüm 6 'dan (SI version), Şekil 16 aracılığı ile oda yüksekliğine göre müsaade edilebilir sıcaklık (2.1 m oda tavanı yüksekliği için) 60 °C okunur. Dolayısı ile tasarım uygundur. İlgili şekil, Şekil 29 'da verilmiştir.



Şekil 29. Tavan Panel Sıcaklığı Müsaade Edilebilir Sıcaklığı (ASHRAE, 1993).

Aynı bölüm, döşeme için bu sınırı 29 °C olarak vermektedir. Hesaplara devamla,

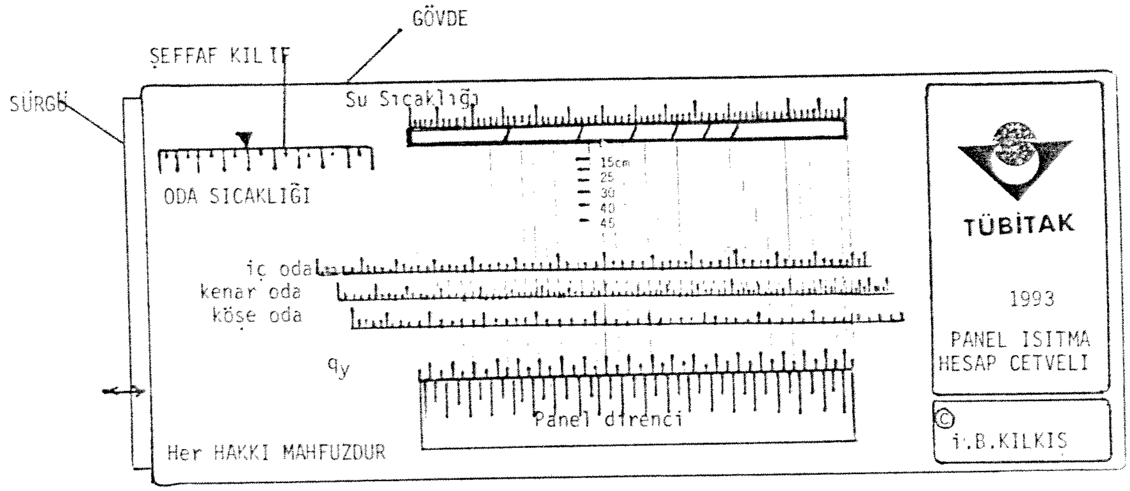
Üst eksenden,

T_s = T_a + 28 °C = 20 + 28 °C = 48 °C olarak bulunur.

Eğer boru aralığı 45 cm olsa idi, T_s yaklaşık olarak 20 + 35 , (55 °C) olacaktı.

Ortalama su sıcaklığını tayin eksenindeki kırık çizgiler ise panel malzemesi ile kaplama malzemesi arasında büyük ısıl direnç farkı olduğu zaman ($r_c / r_p \geq 4$) kullanılır. Bu durum örneğin lastik tabanlı kalın bir makine halı ve beton döşeme için geçerlidir.

Bu nomogramdan hareketle, bir tasarım cetvelinin bütün esasları hazırlanmıştır. Buna ilişkin dizayn Şekil 30 'da gösterilmiştir.



Şekil 30. Tasarım Cetveli Taslağı.

Bu tasarım cetveli üç parçadan müteşekkildir:

1 - Gövde

2 - Sürgü

3 - Şeffaf (naylon) kılıf

Gövde üzerinde, panel ısıl direnci r_y 'nin ayarlanmasına ve ortalama akışkan sıcaklığı T_s 'nin okunmasına yarayan iki göz vardır. Gövde içersindeki sürgü sağa sola hareket ettirildiğinde alttaki gözden r_y değeri, üstteki gözden ise her biri sırası ile 15, 25, 30, 40 ve 45 cm.lik boru aralığı (M) 'ye tekabül eden ve renk kodlu çizgiler gözükmektedir. Bu çizgilerin tamamı sürgü üzerine hesaplanarak çizilmiştir. Sürgü sağa sola hareket ettirildikçe bu çizgiler de sağa sola kaymaktadır. Bu çizgilere karşılık gelen ortalama akışkan sıcaklığı skalası şeffaf kılıf üzerine basılmıştır. T_s ve T_p , oda sıcaklığına doğrudan ve lineer olarak bağımlıdır. q_Y ise T_p 'ye bağlı olduğu için, oda sıcaklığına bağlıdır. İşte bu bağımlılıklar şeffaf kılıfın gövde üzerinde sağa sola kaydırılması ile sağlanır. Bu ayar ise gövde üzerindeki oda sıcaklığı T_a skalası ile şeffaf kılıf üzerindeki ($\rightarrow$) okunun istenen T_a 'da çakıştırılması ile sağlanır. Aynı şeffaf kılıfta alttaki 3 adet T_p sakalasında basılıdır. Bu skalalar ayrı oda konumlarının AUST üzerine etkisini simgeler. Şeffaf kılıf aynı zamanda, mukavva gövdeyi ve skalaları dış etkilerden korumaktadır. Panel soğutma için ise aynı şeffaf skalanın arkası öne getirilerek kullanılır.

Bu şeffaf kılıf, istenen oda sıcaklığına ayarlandıktan sonra gövdenin alt kısmındaki panel birim ısı çıktısı ($\geq$ panel birim ısı yükü) skalası üzerinde istenen r_y değeri sürgü vasıtası ile çakıştırıldıktan sonra, en üstteki skaladan T_s , istenen (M) değeri için okunur. Burada interpolasyon mümkündür. Ayarı bozmadan kılıf üzerindeki skaladan oda tipine göre (iç oda, kenar oda, köşe oda) gerekli döşeme sıcaklığı (T_p) bulunur. (T_p) ve (T_s) skalaları ile q_Y skalası arasındaki göz ilişkisini kolaylaştırmak üzere dikey-milimetrik çizgiler gövde üzerinde çizilmiştir.

Bu cetvele değişik noktalardan girmek ve çıkmak mümkündür: nomogram ve cetvel, $x = 0.90$ değeri için hazırlanmıştır.

Örnek 1 :

Su sıcaklığı (T_s), oda sıcaklığı (T_a) ve q_Y bellidir. r_y de bilinmektedir. Bu durumda önce kılıf T_a 'ya göre ayarlanır. Sürgü ise q_Y ve r_y çakışana dek oynatılır. Sonra, üst skaladan T_s 'ye karşılık gelen M ve orta skaladan, oda tipine göre T_p okunur.

Örnek 2 :

T_a , T_p , M ve r_y bellidir. T_a ayarını takiben oda tipine uygun T_p skalasından q_Y okunur. Bu q_Y değerine çakışan r_y değeri ile M vasıtası ile üst skaladan gerekli T_s okunur.

Örnek 3 :

r_y , q_Y , M ve T_s bellidir. T_a ve T_p 'yi bulunuz.

Bu durumda iterasyon gereklidir.

T_a tahmin edilir. Kılıf bu tahmini değere göre ayarlanır. Orta skaladan, oda tipine göre T_p okunur. Bu T_p değerinden aşağıya inerek q_Y ve r_y 'nin aynı hizada çakışıp çakışmadığına bakılır. Çakışmıyor ise T_a ayarı ile oynanarak işlem tekrar edilir.

Tasarım değişkenleri, T_a , oda pozisyonu, r_y , q_Y , T_p , T_s ve M 'dir.

Görüldüğü üzere tek bir sonuç elde etmek için 7 değişkenden 6 'sının bilinmesi gerekir. Daha çok örneğin iki bilinmeyen olduğu taktirde yukarıdaki örnekte görüldüğü üzere iterasyon gerekir. Daha çok bilinmeyen olduğu taktirde birden fazla çözüm noktası olur. Cetvelin arka yüzeyinde ise tasarımın diğer safhalarına ilişkin hesaplar örneğin boru hesabı vs. konabilir. Bu hesaplar daha basit olduğu için bu konu, uygulayıcıların arzusuna ve çalışmasına bırakılmıştır. Nomogram ve hesap cetvelinde 7 değişkenin bir arada mütalaası ve değiştirilmesi ilk kez gerçekleşmektedir.

Yukarıda verilen iki nomogram ve hesap cetveli içerik ve hazırlandıkları yöntem bakımından Dünyada ilk kez geliştirilmiş örnek niteliğindedir. Tamamen Proje Yöneticisi Prof. Dr. İbrahim Birol Kılkış'ın şahsi çalışmalarının ürünüdür. TÜBİTAK sözleşmesi çerçevesinde bu iki ürünün "GİZLİ" tutularak önce patentlerinin alınması, bilahare gerekli albeni tasarımı ve imalat aşamalarını takiben de piyasaya arzını öneriyorum.

Patent için gerekli metalip yazılımının TÜBİTAK hukuk danışmanlığı ile ortak çalışarak hazırlanmasında gerekli teknik done ve bilgiler tarafımdan verilecektir. Çalışmanın ortak yapımında yarar görmekteyim. Ürünün piyasa standartlarında imalatı için TÜBİTAK gerekli finans kaynaklarını ayırmalıdır. Gerekli teknik destek, GEBZE ve TÜBİTAK matbaasından istenebilir. Bu konuda TÜBİTAK Başkanlığına ayrı bir yazı yazılmıştır.

Hazırlanan bilgisayar programları da özgün çalışmalar olup, bu ürünler üzerindeki telif haklarının proje çalışanları kısmının dökümü ise aşağıda verilmiştir. Bilindiği kadarı ile bilgisayar programlarında artık fikir ve uygulama orijinaliteleri bakımından patent konusu olabilmektedir.

Çizelge 8. Ürünlerin ve Fikirlerin Telif ve Patent Haklarının Çalışanlara Dağılımı (TÜBİTAK payı çıktıktan sonra)

Ürün	Çalışanların katkı oranı (% olarak) B. Kılkış	S. Sağer	M. Uludağ	M. Sapçı
Tasarım Nomogramları (Metrik ve SI birimde)	100	-	-	-
Tasarım Cetveli	100	-	-	-
Isıtmada Panel İçi Isıl Analizi	100	-	-	-
Kar Eritme Yükü Hesap Yöntemi	100	-	-	-
Soğutmada Panel İçi Isıl Analizi	100	-	-	-
Panel Isıtma ve Soğutma Kitabı*	100	-	-	-
KAR Programı	100	-	-	-
SOĞUTMA Programı	100	-	-	-
ISITMA 2 Programı	70	10	10	10

* Kitap yazımı için TÜBİTAK'tan ek destek talep edilecektir.

f - Diğer Hizmetler

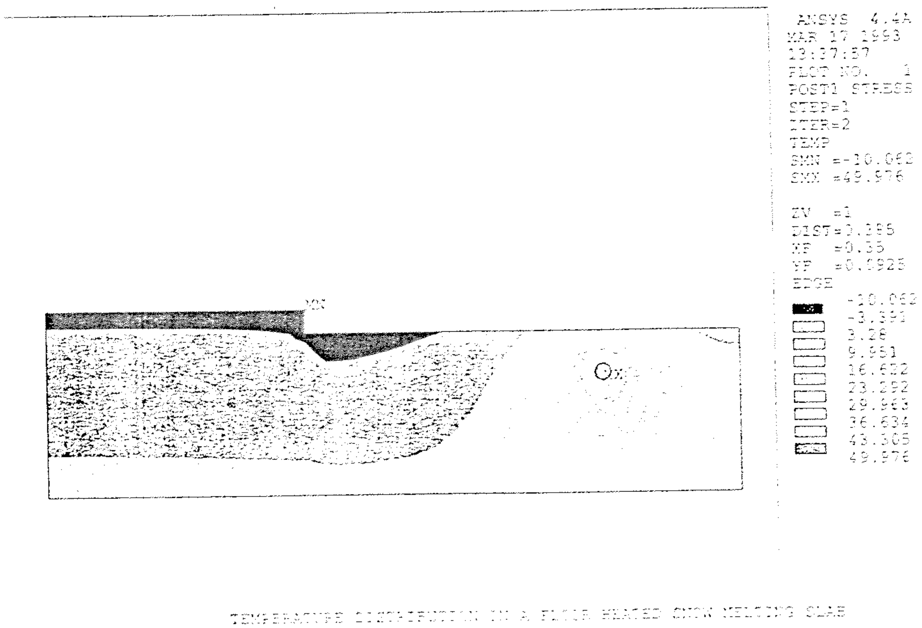
Panel sistemlerde basınç kayıplarının hesabında, alışılagelen yöntem, en uzun parkur ve üzerindeki yüke göre pompa seçimidir. Bu yöntemin, aynı kollektörlere bağlı ve birbirinden farklı uzunluklara haiz paralel hidrolik devreler için gerekli olmadığını Kılkış göstermiştir (1992-c). Bu konu ilk kez Hansen (1985) tarafından dile getirilmiş, fakat uygulamaya geçmemiştir. Kılkış Hansen'in kitabındaki matematiksel hatayı da düzelterek, yeni bir algoritma geliştirmiştir. Bu algoritma Ek-5, Yayın no: 15, 3.4. bölümünde verilmiştir.

Ayrıca, metal olmayan borulardaki oksijen geçiriminin korozyona katkısı da incelenmiştir (Kılkış, 1992-c).

Kılkış tarafından yurt içi ve dışı toplantılara tebliğle katılınılmış, seminerler verilmiştir. Bu faaliyetler, raporun 8.4. bölümünde verilmiştir.

Ayrıca kar eritme konusunda yeni kar eritme yükü hesap usulü ve tasarım algoritmaları geliştirilmiştir (Ek-5, Yayın No: 12, 13 ve Ek-10).

Ayrıca ANSYS programı kar eritme panelinin analizinde ve geliştirilen modelin sınanmasında kullanılmıştır. Örnek bir sayısal çözüm Şekil 31 'de verilmiştir.



Şekil 31. Kar Eritme Paneli Örnek Sayısal Çözümü (Kılkış, 1993).

Bu uygulama Ülkemizde hiç bilinmemekle birlikte milli ekonomimize büyük katkıları olacağı kesindir. Bu bakımdan gerekli kuruluşlarla temas edilmiştir.

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12. EKLER

Bu bölümde 10 adet Ek halinde yapılan çalışmaların detayları verilmektedir. Ek 1 ve Ek 2 'de panel ısı yükünün hesabı için gerekli konfor, infiltrasyon, ısı kayıpları ve ısı depolaması üzerinde durulmuştur. Ek 3 'de Kılkış'ın (1993-c) geliştirdiği yeni panel ısıl direnci hesap algoritması açıklanmaktadır. Ek-4 'de ise panel ısıtma ve soğutma deney düzeni bilgileri verilmektedir. Mukayese amaçlı bazı deneyler yapılmış ve sonuçları verilmiştir. Projeyi müteakiben gerek duyulan sistematik deneyler, endüstriden de talep geldiğinde yapılabilecektir. Ek 5 'de proje süresince yapılan 26 adet yayının kopyeleri verilmiştir. Diğer eklerde ise sırası ile ISITMA 2, ISI 2, SOĞUTMA VE KAR programları örnekleri ile tanıtılmaktadır. Ek-8 'de ise Sayısal Modelleme ve Analiz çalışmaları anlatılmaktadır.

EK-1. Döşemeden Isıtmada Isıl Konfor

Isıtılan veya soğutulan bir iç mekandaki ısıl konfor esasları ANSI/ASHRAE 55-1992 standardında genel çerçevede belirlenmiştir.

Panel sistemlere özgü ayrı bir konfor standardı olmamakla birlikte genel konfor esasları geçerlidir. Ancak aşağıdaki farklar gözönünde tutulmalıdır:

i - Panel sistemler sadece hissedilir yüklere cevap verebilir.

ii - İnsan-ısıtılan/soğutulan mekan ilişkisinde ışıma ısı transferi ağırlıklıdır. Bu nedenle insan giysileri ve çıplak yüzeyleri ortalama sıcaklığı standartlarda öngörülen değerlerden çok az farklıdır.

iii - Hava hızı daha az olduğundan insan yüzeyi ısı iletim katsayısı farklıdır.

iv - Işıma asimetrisi yüksek sıcaklıktaki ısıtıcı yüzeyleri veya enfrarüj ısıtıcılardaki kadar yüksek olmamakla birlikte çok sıcak tavanlarda önemli olabilir (Fanger ve diğerleri, 1982).

v - Hava hızı az olduğu için insan bedenindeki sübjektif konfor hisleri daha olumludur.

vi - Panel ısıtmada veya soğutmada iç hava sıcaklığı ve hava hızı daha homojen olduğu için yine sübjektif konfor hisleri daha olumludur.

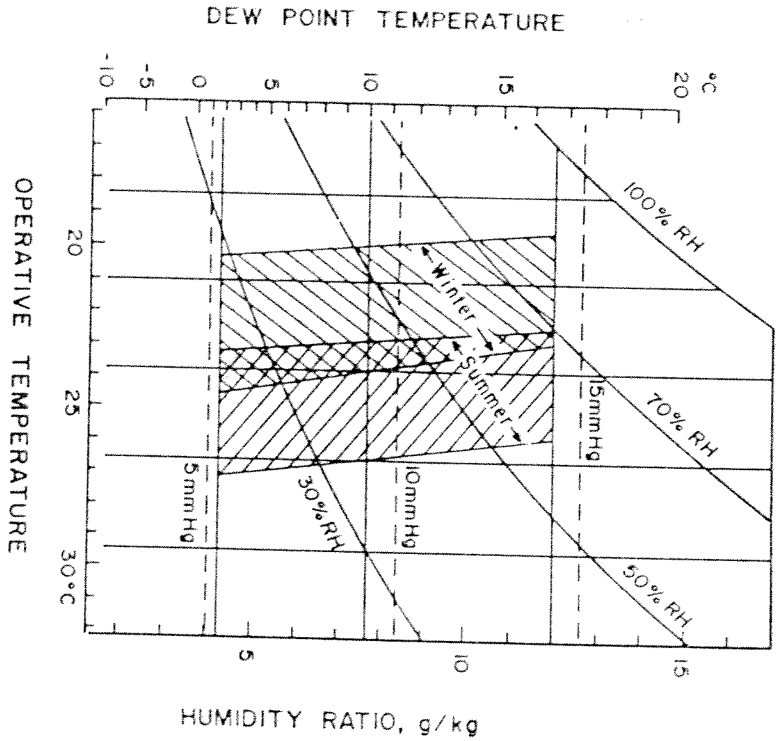
vii - Isıtma ortalama ışıma sıcaklığı (MRT) daha yüksek, soğutmada daha düşüktür.

viii - Geçici rejimde ve ani yük değişimlerinde insan konforu ihtiyaçları daha farklı olup daha kritiktir (Wang ve Peterson, 1993). Panel sistemlerdeki ısıl atalet ve panel performansının adaptif nitelikli olması nedeni ile geçici rejim konfor şartları daha kolay sağlanır.

Panel sistemlerinde konfor 5 parametre ile ilintilidir. Bunlar ortalama ışıma sıcaklığı (MRT), T_a, Operatif Sıcaklık (T_{op}), Düzeltilmiş kuru termometre sıcaklığı, etkin ışıma yoğunluğu (ERF) 'dir.

Ayrıca insan yüzeyindeki ışıma ve iletim katsayıları h_r ve h_c sabit kabul edilebilmekle birlikte h_c hava hızına, insan hareketine, h_c ise insan giysisine, pozisyonuna bağlıdır.

Psikiyometrik Diyagramda konfor bölgesi Panel ısıtma için ve Panel soğutma için düzeltilmiştir (Şekil 1-1). Buradaki yatay eksen operatif sıcaklık T_{op} 'tur. T_{op} ise giysi yalıtım indeksi CLO ve aktivite düzeyine bağlıdır (ASHRAE, 1991, Bölüm 48).



Şekil 1-1. Panel Sistemler için Düzeltilmiş Konfor Bölgesi (ASHRAE, 1991, Bölüm 48).

Döşemeden ısıtmada geçerli olmak üzere gerekli iç hava sıcaklığı standart değerli bu çalışmada yeniden hesaplanmıştır. Bu çalışmada oda fonksiyonuna bağlı olarak tipik insan aktivite seviyesi, giysi türü, pozisyonu ve yaş ortalaması (ve boyutları) gözönünde tutularak operatif sıcaklıktan hareketle Şekil 1 'den en uygun iç sıcaklıkları hesap edilmiştir. Bu amaçla, Modera'nın (1993) insan deri sıcaklığı değerleri, metabilzmik ısı yayınımları (ASHRAE 1992, Bölüm 3) ve oda fonksiyonuna bağlı tipik panel konumları kullanılmıştır. Sonuçlar hazırlanan Türk Standardı (1993-b), Çizelge 2 'de verilmiştir.

Görüldüğü üzere T_a genelde 2 °C kadar daha düşüktür. Müsaade edilebilir döşeme sıcaklıkları tesbitinde ise ANSI/ASHRAE 55-1992 ile DIN 4725 (1990) standardı ile Nevin's ve diğerlerinin çalışmaları (1958, 1964) gözönünde tutulmuştur. Soğutma için aynı genel yöntemler kullanılır. Ancak, tavandan soğutma için müsaade edilebilir panel sıcaklığında konfordan önce yüzey yoğuşması önem taşımaktadır. Bu nedenle T_{min} değerinin yerel çiğ noktası sıcaklığından az olmama koşulu tasarım sırasında gözönünde tutulmalıdır. Ayrıca Külpman'ın (1993) "soğuk tavan" konforsuzluğu üzerine yaptığı deneysel çalışmalar da gözönünde tutulmuştur. Bu çalışmaya temel alınan sonuçlara göre $(T_a - T_p)$ değeri genelde 13 °C 'ı geçmemelidir.

Ayrıca Kılkış (1993-e, Ek-5 : Yayın no: 15) daha çabuk bir hesap yöntemi geliştirmiştir. Buna göre:

$$q_r : 4.6 \ (T_{si} - AUST^*) \qquad [1\text{-}1]$$

$$q_c : 3 \ (T_{si} - T_a) \qquad [1\text{-}2]$$

ve insan bedeninden yayılan hissedilir ısı Q_i ve ortalama yüzey alanı A_i ise:

$$Q_i / A_i = 4.6 \ (T_{si} - AUST^*) + 3 \ (T_{si} - T_a) \qquad [1\text{-}3]$$

Burada AUST* panel yüzeyini de kapsar.

Q_i ise insan aktivitesine bağlı olup, literatürde verilen çeşitli değerler kullanılabilir:

Örnek :

$Q_i = 80$ W
$A_i = 1.6$ m^2
$T_{si} = 26.6$ °C
$AUST^* = T_a + 3$ °C
1-3 denkleminden T_a : 18.2 °C bulunur.

EK-2. Döşemeden Isıtma Yükü

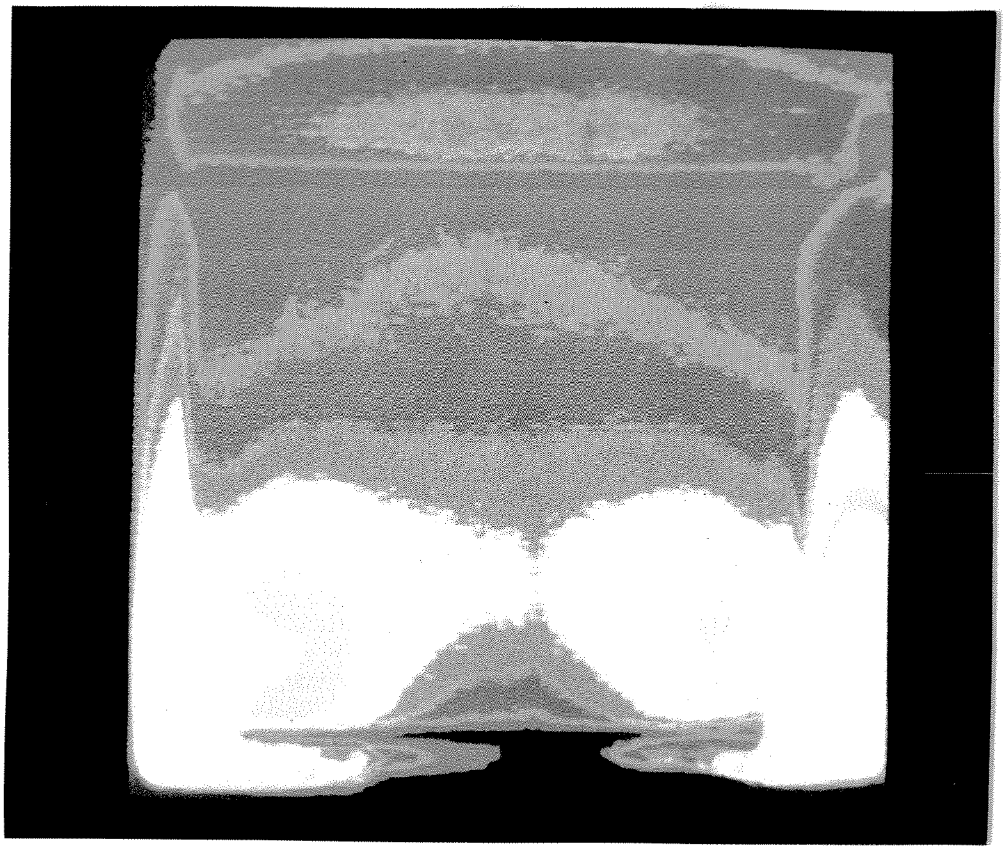
Döşemeden ısıtma yükü hesabında aşağıdaki düzeltmeler gereklidir:

i - İç sıcaklık değeri düzeltilmelidir.
ii - Doğal infiltrasyon (mekanik havalandırma yoksa) düzeltilmelidir.
iii - Isı depolaması nedeni ile pik yük düzeltilmelidir (Kılkış, 1992, Andersen, 1992).
iv - İç yüzeylerin film katsayısı düzeltilmelidir.
v - Panelin bulunduğu duvarda ısıtılmayan/soğutulmayan bölüm var ise bu bölümdeki ısı kayıpları/kazançları sadece kazan (soğutucu) yüküne eklenmelidir.

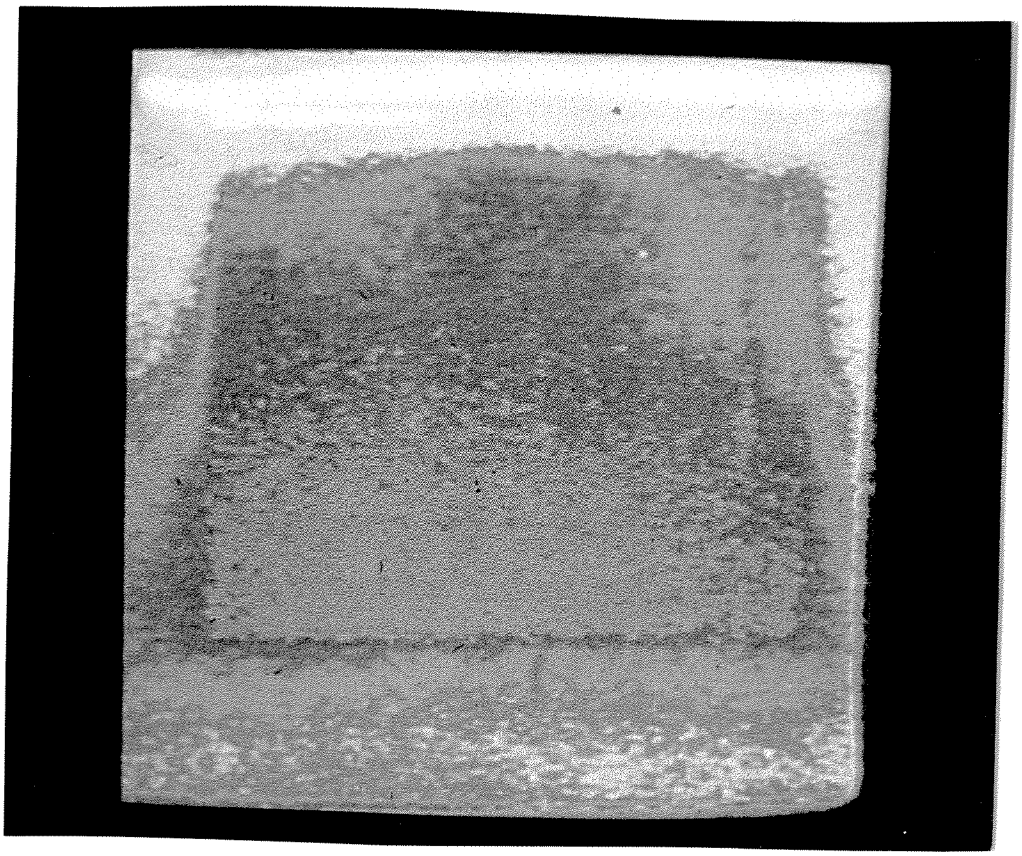
İç sıcaklık düzeltilmiş değerleri Ek-1 'de izah edilmiştir. Doğal infiltrasyonda ise düzeltme iki nedenle gereklidir. Birincisi iç sıcaklıktaki düzeltmedir (Buckley, 1989). İkincisi ise odadaki hava termo mekaniğinin özellik arzetmesidir. Kılkış, Sağer ve Uludağ (1994-f) dış koşullar aynı kalmak şartı ile ve aynı insan konforu sağlamak üzere radyatörlü sistemle ısıtılan bir odayı panel ile ısıtılan bir oda ile karşılaştırmışlar ve FLOTRAN pakedi ile hava basınç, sıcaklık ve hızını çözerken, tipik sızdırma noktalarından (örneğin pencere gibi) geçen hava miktarını da gözlemişlerdir. Mukayeseler % 50 dolayında sızdırma miktarında azalma tesbit etmişlerdir. Bu nedenle de TS 2467 'deki kaloriferli sistem için geçerli sızdırganlık katsayısı, oda özelliği (R) ve yapı özelliği (H) katsayıları baştan düzenlenmiştir.

Literatürde ise ısıtma yükünün yeni bir yöntemle hesabının gerektiğini vurgulayan bazı araştırmacıların dışında böyle bir algoritmayı geliştirmemişlerdir (Lafontaine, 1990, Buckley, 1989). Bu arada yapılan deneysel çalışmalarda ise bu özellik genel olarak tesbit olunmakla birlikte Dole ve Ackerman (1990) aksini savunmuşlardır. ASHRAE teknik komitesi TC. 6.5 ise bu tebliğde yer alan sonuçları incelemiş ve bulgunun genelleştirilemiyeceği sonucuna varmıştır.

Şekil 2-1 'de ise, literatürde yapılan termografik bir çalışma örneği görülmektedir. Bu örneğe göre oda içi hava sıcaklık dağılımı çok daha homojendir.



a - Radyatör



b - Panel

Şekil 2-1. Oda İçi Sıcaklık Dağılımı (Wirsbo, 1990).

Geliştirilen algoritma TSE (1993-b), Bölüm 1.1 'de verilmiştir.

‹NİK KÖŞE

HAVADAN/SUYA TİP, ELEKTRİK MOTORU TAHRİKLİ ISI POMPASI İLE YERDEN ISITMA SİSTEMİ İÇİN KULLANICI YÖNÜNDEN ENERJİ MALİYETİ FİZİBİLİTESİ

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paları çevre, alternatif, jeotermal ve atık n etkin kullanımında büyük avantaj sağ- ıdırlar. Bu suretle düşük sıcaklıklardaki ›ğun enerjinin daha yüksek sıcaklıklarda ›ir biçimde, örneğin mekan ısıtmasında ması prensip itibari ile mümkün olmakta- :ak klasik ısıtma sistemlerinde mevsimin uk gününde 90/70°C su gidiş ve dönüş jı gerekli olmaktadır. Isı pompalarının su caklığı ise 55°C, en çok 60°C ile sınırlıdır. r, başlıbaşına ısı pompalarının radyatörlü ma sisteminde kullanılmasını,-ılık günler nemen hemen imkansız kılmakta ve alışı- ış bir destek kazanına çoğunlukla gerek mektedir. Öte yandan havadan enerji ’an ısı pompalarının ısıtma tesir katsayıla- = Isı pompasından elde edilen enerji/Isı asının tahrikinde kullanılan enerji) dış ha- :aklığı ile orantılıdır. Bu nedenle soğuk rde ısı pompasının ısıtma tesir katsayıları ekte, kısaca,birim ısıtma için daha çok tüketme zorunluluğu ortaya çıkmaktadır. soğudukça ısı pompasından elde edilen n ısı miktarı da düşmektedir. Bu ise bina- ›ğuk hava ile artan ısıtma ihtiyacı ile tam zat teşkil etmektedir. Bu iki zıt faktör nede- belirli bir dış hava sıcaklığından daha so- jünlerde ısı pompası ısıtmaya yeterli olma- adır. Ayrıca klasik kazanlarda 90°-70°C = olan gidiş dönüş su sıcaklığının 20°C ol- na mukabil ısı pompalarında en çok 10°C müsaade edilmektedir. Bu durum Şekil 1' österilmiştir. Görüldüğü üzere +7 °C dola- ı klasik kazan devreye girmeye başlamak- · 5°C dolayında da gerekli gidiş-dönüş su :lık farkı 10°C'ı aştığından ısı pompası dev- şı kalmaktadır.

Özetle:

i- Isı pompası su çıkışının 55°C ile sınırlı olması,

ii- Isıtma tesir katsayısının soğuk havada azalması (örneğin 2.75'den, -7°C dış sıcaklıkta 1.90'a düşmesi, (Şekil 2)),

iii- Hava soğudukça artan ısı yüküne zıt olarak ısı pompası ısıtma kapasitesinin düşmesi (Şekil 1),

iv- Gidiş dönüş su sıcaklık farkının 10°C ile sınırlı olması,

nedenleri ile bir ısı pompasının radyatörlü ısıtma sistemine entegrasyonu, çok ılıman yerler dışında, teknik olarak mümkün olmamaktadır. Bunda en önemli etkenler yukarıda sıralanan (i) ve (iv) numaralı faktörlerdir. Bu nedenle ısı pompalarının ısıtmada kullanılamıyacağı kanaati dahi toplumda oluşmuştur. Bu yanlış izlenimin giderilmesi gerekli olduğu gibi aslında yukarıda sıralanan sorunlara tam anlamı ile çözüm getiren bir ısıtma sistemi mevcuttur. Bu sistem döşemeden ısıtma sistemi olup, daha ilk ağızda (i) ve (iv) numaralı sorunları gidermektedir. Zira,

(i) Döşemeden ısıtma sistemi zaten 55°C ve daha düşük su sıcaklıklarında çalışmaktadır.

(iv) Döşemeden ısıtma sistemi zaten $\triangle T =$ 10°C'a göre tasarımlanmaktadır. (ii) ve (iii) numaralı sorunların tersine döndürülmesi imkansız olduğuna göre yerden ısıtma bu konuda nasıl yardımcı olabilmektedir? Herşeyden önce yerden ısıtma sisteminde ısı yükleri azalmaktadır. Bunun çeşitli nedenleri vardır:

ENERJİ BÜLTENİ
EİE ÜÇ AYLIK ENERJİ TASARRUFU BÜLTENİ OCAK 1992 No 9

ENERJİ 1992

10.DOĞAL GAZ, KÖMÜR, PETROL, SU, GÜNEŞ, RÜZGAR, HİDROJEN, BİOGAZ, YERALTI ISISI, NÜKLEER ENERJİ VE DİĞER ALTERNATİF ENERJİ KAYNAKLARI VE ENERJİ TASARRUFU FUARI
11-15 OCAK 1992
tarihleri arasında İstanbul Tepebaşında
TÜYAP SERGİ SARAYINDA
yapılmaktadır.

T.C.Enerji ve Tabii Kaynakları Bakanlığı Enerji Tasarrufu Koordinasyon kurulu Başkanlığı ve TÜYAP işbirliği ile hazırlanan, Türkiye Enerji Tasarrufu Haftası içinde yer alan bu fuarda;

Enerji Kaynakları,yeni ve yenilenebilir enerji kaynakları konularının yanısıra ısıtma,havalandırma,aydınlatma ile ilgili ürünler ve enerji nakline ilişkin araç ve gereçler tanıtılacak bunların tasarruf sağlayan özellikleri gerek ilgililere gerekse kamuoyuna topluca sunulacaktır.

Ayrıca Fuar döneminde 14-15 Ocak 1992 günleri düzenlenecek seminerde fuara katılan yüksek eğitim kurumları ve kuruluşlar çalışmalarına ilişkin tebliğler sunacaklardır. Konut,sanayi, doğal gaz, çevre başlıkları altında yaklaşık 20 tebliğ sunulacaktır. Fuar hergün $11.^{00}$ - $20.^{00}$ saatleri arasında ziyarete açık tutulacaktır.

Çizelge 13. Elektrikli Isıtıcı Eleman Merkezleri Arası Mesafe (M) Değerleri

M (mm)				
40	50	60	90	120

Çizelge 12. Bronz ve Bakır Rezistans Teli Özellikleri

Rezistans Tel Çapı (mm)	Dış Çap, Do (Yalıtım dahil) (mm)	Birim Ağırlık (kg/m)	Uygulanabilir Maksimum Gerilim (V)		RL Ohm/m (25 oC'da)
Tek Rezistanslı Isıtıcı Eleman (E)					
0.40	3.05	22	300	E	2.001
0.45	3.25	30			1.190
0.57	3.68	51			0.984
0.72	3.81	57			0.656
1.01	4.19	67			0.328
1.44	4.44	83	300		0.164
1.83	5.08	107			0.098
2.31	5.46	127			0.066
3.25	6.25	179			0.033
0.72	4.65	79	600	E	0.656
1.01	5.21	103			0.344
1.29	5.33	110			0.197
1.29	5.46	127			0.115
2.05	6.10	149			0.082
2.31	6.43	167			0.066
2.89	7.04	213			0.043
4.11	6.10	301			0.020
1.01	5.06	97	600	E	0.021
1.29	5.46	109			0.013
1.62	5.84	131			0.008
2.05	6.25	156			0.005
2.54	7.04	201			0.003
Çift Rezistanslı Isıtıcı Eleman (E veya B)*					
0.45	4.19	75	300	E	2.625
0.72	4.65	100			1.312
1.01	6.25	143			0.410
1.29	8.64	271	600	E	0.230
1.62	9.42	328		E	0.144
2.05	10.21	387		E	0.092
1.29	8.64	271		B	0.027
1.62	9.42	328		B	0.017

* E = Bronz
B = Bakır

Çizelge 11. Çeşitli Döşemeden Isıtma Boru Malzemesi Sıcaklık Dayanım Sınırı ve Bükülebilir En Küçük Çap (D_{min}).

Boru Malzemesi	T_s (°C) Sürekli rejimde	T_s (°C) Kısa süre için	D_{min}
A- Termoplastik Borular:			
Poli Büten*	50	55	$10.D_o$**
Poli Propilen***	50	60	$12.D_o$
Polietilen-Cross link****(ağdalı)	55	65	$10.D_o$
B- Lastik esaslı borular	70	80	$(6\sim7).D_o$
C- Metal borular	Döşeme malzemelerinin ısıl dayanımı ile sınırlıdır.		——

* Polibüten (PB) boru malzemesi yapılarda döşemeden ısıtmada kullanılmaz. Polietilen, PVC ve benzeri düşük yoğunluktaki ve homopolimer efsaflı başka malzemeler döşemeden ısıtmada hiç bir şekilde kullanılamaz.

** D_o = Boru ortalama dış çapı [m].

*** Polipropilen (PP) malzemelerden tercihan Polipropilen Random Copolimer kullanılmalıdır. Bu malzeme mevcut değilse daha düşük evsaflı Polipropilen Co-polimer malzeme kullanılabilirse de, polipropilen Homopolimer boru hiç bir surette döşemeden ısıtmada kullanılamaz.

**** Ağdalama işlemleri şu metodlarla uygulanabilir (Giderek azalan termomekanik kalite sıralaması ile):
a- Peroksit ağdalaması (Engel metodu)
b- Hidrosilikon ağdalaması
c- Elektron huzmesi ile ağdalama
d- Azo metodu
Mamuller üzerinde, kullanılan yöntem a,b,c,d harflerinden birisi ile belirtilir. Ağdalamada esas, mümkün mertebe bütün boru et kalınlığına ağdalamanın tatbikidir.

a-metodu (Engel) tatbik olunduğunda T_s sürekli rejimde 70°C, (madde 1.2.3.1.7 'ye bakınız) kısa süre için 80°C alınır.

NOT 1. Lastik esaslı borularda, yanmaya, basınca ve ezilmeye karşı özel katmanlar ve istenirse oksijen geçirimsiz zarf bulunmalıdır.

NOT 2. Herhangi bir borunun "oksijen geçirimsiz (bariyerli)" olarak pazarlanabilmesi için oksijen geçirim katsayısı 0.1 g/m^3.gün 'den fazla olmamalıdır.

Çizelge 10'un devamı: Döşemeden ısıtma sistemlerinde dikişli çelik boru tesisatı yaklaşık boru hesabı çizelgesi
Basınç kaybı, R = 28,5-100 mm.s.s/m

Anma ölçüsü	1/2"	4/3"	1 "	11/4"	11/2"	2"	21/2"	3"	4"	5"	6"	8 5/8"	10 3/4"
İç Çap mm	16,0	21,6	27,2	35,9	41,8	53,0	68,8	80,8	105,3	130,0	155,4	210,1	263,0
R mm.s.s/m	$\Delta T_s = 10\ ^oC$ ve $T_s \cong 55\ ^oC$ için												
28,5	Q_s: 3525 V_s: 0,50	7900 0,65	14950 0,75	31600 0,90	46100 1,00	87500 1,10	180500 1,40	276500 1,50	545000 1,80	945000 2,00	1500000 2,20	3130000 2,60	5650000 3,00
31	3670 0,55	8200 0,65	15550 0,80	32900 0,95	48000 1,00	91500 1,20	187500 1,40	287000 1,60	565000 1,90	985000 2,20	1570000 2,40	3245000 2,60	5850000 3,00
33	3810 0,55	8500 0,70	16150 0,80	34100 0,95	49700 1,10	94500 1,20	194000 1,50	298000 1,70	585000 1,90	1020000 2,20	1625000 2,40	3365000 2,80	—
36	4025 0,60	8950 0,70	16950 0,85	35850 1,00	52500 1,10	99500 1,30	204000 1,60	312000 1,70	615000 2,00	1060000 2,20	1710000 2,60	3545000 3,00	—
40	4215 0,60	9350 0,75	17750 0,90	37500 1,10	55000 1,20	104000 1,30	213000 1,60	325500 1,80	640000 2,00	1160000 2,40	1770000 2,60	—	—
44	4465 0,65	9900 0,80	18700 0,95	39600 1,10	58000 1,20	110000 1,40	225000 1,70	344000 1,90	675000 2,20	1180000 2,60	1880000 2,80	—	—
50,5	4750 0,70	10550 0,85	19900 1,00	42100 1,20	61500 1,30	117500 1,50	240000 2,00	364500 2,00	725000 2,40	1250000 2,60	1995000 3,00	—	—
55	5000 0,75	11150 0,90	21000 1,00	44500 1,30	64500 1,40	124000 1,60	253500 2,20	385500 2,20	755000 2,40	1320000 2,80	2100000 3,00	—	—
60,5	5250 0,75	11700 0,95	22050 1,10	46800 1,30	68000 1,50	130500 1,70	266500 2,20	404500 2,20	795000 2,60	1390000 3,00	—	—	—
66	5500 0,80	12300 1,00	23100 1,20	48950 1,40	71000 1,50	137000 1,80	278500 2,40	424000 2,40	830000 2,80	1450000 3,00	—	—	—
71	5750 0,85	12800 1,00	24050 1,20	51000 1,50	74500 1,60	142000 1,80	290000 2,40	442000 2,40	865000 2,80	—	—	—	—
77	5950 0,90	13350 1,10	25050 1,30	53000 1,50	77000 1,60	148000 1,90	300500 2,60	459000 2,60	900000 3,00	—	—	—	—
82	6200 0,90	13800 1,10	25900 1,30	55000 1,60	80000 1,70	153500 2,0	311000 2,60	475500 2,60	930000 3,00	—	—	—	—
88	6400 0,95	14300 1,20	26850 1,30	55500 1,60	82500 1,80	158500 2,0	321500 2,80	494000 2,80	—	—	—	—	—
100	6800 1,00	15200 1,20	28550 1,40	56000 1,70	88000 1,90	169000 2,20	342000 3,00	520000 3,00	—	—	—	—	—

Çizelge 10'un devamı: Döşemeden ısıtma sistemlerinde dikişli çelik boru tesisatı yaklaşık boru hesabı çizelgesi
Basınç kaybı, R = 6,6-26,5 mm.s.s/m

Anma ölçüsü	1/2"	4/3"	1 "	11/4"	11/2"	2"	21/2"	3"	4"	5"	6"	8 5/8"	10 3/4"
İç Çap mm	16,0	21,6	27,2	35,9	41,8	53,0	68,8	80,8	105,3	130,0	155,4	210,1	263,0
R mm.s.s/m	$\Delta T_s = 10\ ^oC$ ve $T_s \cong 55\ ^oC$ için												
6,6	Q_s: 1590 V_s: 0,24	3600 0,30	6800 0,34	14450 0,42	21400 0,46	40950 0,55	84000 0,65	129000 0,70	253000 0,85	439000 0,95	710000 1,10	1465000 1,20	2665000 1,40
7,1	1670 0,24	3760 0,30	7050 0,36	15100 0,44	22350 0,48	42550 0,55	87500 0,65	134500 0,75	263000 0,85	460000 1,00	735000 1,10	1530000 1,30	2770000 1,50
7,7	1730 0,26	3910 0,32	7400 0,36	15750 0,44	23250 0,50	44200 0,55	91500 0,70	140000 0,80	273500 0,90	477000 1,00	770000 1,20	1585000 1,30	2875000 1,50
8,2	1800 0,26	4060 0,32	7650 0,38	16300 0,46	24100 0,50	45950 0,60	94500 0,75	145500 0,80	284000 0,95	496000 1,10	795000 1,20	1645000 1,40	2980000 1,50
8,8	1830 0,28	4195 0,34	8000 0,40	16850 0,48	24900 0,55	47450 0,60	98000 0,75	150500 0,85	295000 0,95	510000 1,10	820000 1,20	1700000 1,40	3085000 1,60
10.0	1990 0,30	4470 0,36	8500 0,42	17950 0,50	26500 0,55	50500 0,65	104000 0,80	159500 0,90	312000 1,00	545000 1,20	870000 1,20	1805000 1,50	3270000 1,70
11.0	2115 0,32	4740 0,38	9000 0,44	18950 0,55	28000 0,60	53500 0,70	109500 0,85	168500 0,95	331000 1,10	575000 1,20	900000 1,40	1910000 1,60	3450000 1,80
12.0	2225 0,32	5000 0,40	9450 0,48	20000 0,55	29400 0,65	56500 0,75	115000 0,90	177000 1,00	347000 1,10	605000 1,30	975000 1,50	1995000 1,60	3630000 1,90
13.0	2325 0,34	5250 0,42	9900 0,50	20900 0,60	30800 0,65	59000 0,75	120500 0,95	185000 1,00	363500 1,20	635000 1,40	1020000 1,50	2050000 1,70	3795000 2,00
14.0	2425 0,36	5500 0,44	10350 0,50	21800 0,60	32100 0,70	61500 0,80	125500 0,95	193000 1,10	378000 1,20	655000 1,40	1060000 1,60	2180000 1,80	3945000 2,00
15.5	2525 0,38	5700 0,46	10750 0,55	22650 0,65	33400 0,70	64000 0,85	131000 1,00	205000 1,10	393500 1,30	685000 1,50	1090000 1,60	2270000 1,90	4110000 2,20
16,5	2620 0,38	5900 0,48	11100 0,55	23500 0,65	39600 0,75	66000 0,85	135500 1,00	207000 1,20	407000 1,30	710000 1,50	1140000 1,70	2350000 1,9	4250000 2,20
17,5	2715 0,40	6100 0,50	11500 0,60	24350 0,70	35800 0,75	68500 0,90	140000 1,10	214500 1,20	422000 1,40	735000 1,60	1180000 1,80	2430000 2,00	4410000 2,40
19	2800 0,42	6300 0,50	11850 0,60	25150 0,70	36950 0,80	70500 0,90	145000 1,10	221000 1,20	435000 1,40	760000 1,60	1215000 1,80	2505000 2,0	4545000 2,40
20	2890 0,42	6500 0,50	12200 0,60	25950 0,75	38050 0,80	73000 0,95	149000 1,10	228000 1,30	450000 1,50	780000 1,70	1255000 1,90	2590000 2,20	4675000 2,40
21	2985 0,44	6700 0,55	12600 0,65	26700 0,75	39200 0,85	74500 0,95	153500 1,20	234500 1,30	461000 1,50	800000 1,70	1265000 1,90	2650000 2,20	4805000 2,60
22	3070 0,46	6850 0,55	12950 0,65	27450 0,80	40200 0,85	76500 1,00	157500 1,20	240500 1,30	475000 1,60	825000 1,80	1325000 2,00	2725000 2,20	4930000 2,60
24	3225 0,48	7250 0,60	13650 0,70	28900 0,80	42300 0,90	80500 1,00	165500 1,30	253500 1,40	497500 1,60	865000 1,90	1385000 2,00	2870000 2,40	5200000 2,80
26,5	3370 0,50	7550 0,60	14250 0,70	30300 0,85	44250 0,95	84500 1,10	173500 1,30	265000 1,50	520000 1,70	900000 1,90	1445000 2,20	2995000 2,40	5400000 2,80

Çizelge 10'un devamı: Döşemeden ısıtma sistemlerinde dikişli çelik boru tesisatı yaklaşık boru hesabı çizelgesi
Basınç kaybı, R = 1,45-6,0 mm.s.s/m

Anma ölçüsü	1/2"	4/3"	1 "	11/4"	11/2"	2"	21/2"	3"	4"	5"	6"	8 5/8"	10 3/4"
İç Çap mm	16,0	21,6	27,2	35,9	41,8	53,0	68,8	80,8	105,3	130,0	155,4	210,1	263,0
R mm.s.s/m	$\Delta T_s = 10\ ^oC$ ve $T_s \cong 55\ ^oC$ için												
1,45	Q_s: 680 V_s: 0,10	1540 0,12	2940 0,15	6300 0,18	9300 0,20	18000 0,24	36950 0,28	57000 0,32	112000 0,36	197000 0,42	319500 0,48	665000 0,55	1235000 0,65
1,5	710 0,10	1605 0,13	3070 0,15	6550 0,19	9750 0,20	18650 0,24	38650 0,30	59000 0,34	116500 0,38	205000 0,44	333500 0,50	700000 0,60	1290000 0,70
1,6	735 0,11	1665 0,13	3190 0,16	6800 0,19	10100 0,22	19400 0,26	40100 0,30	61500 0,34	121000 0,40	212500 0,46	345000 0,50	720000 0,60	1330000 0,70
1,7	765 0,11	1725 0,14	3310 0,17	7100 0,20	10450 0,22	20050 0,26	41500 0,32	64000 0,36	125500 0,42	220000 0,48	358500 0,55	745000 0,60	1380000 0,75
1,85	790 0,12	1785 0,14	3425 0,17	7350 0,20	10750 0,24	20900 0,28	42950 0,34	66000 0,36	129000 0,42	226500 0,48	369500 0,55	770000 0,65	1425000 0,75
2,0	815 0,12	1840 0,15	3535 0,18	7550 0,22	11150 0,24	21450 0,28	44250 0,34	68000 0,38	183500 0,44	235000 0,50	382500 0,60	790000 0,65	1475000 0,80
2,1	840 0,12	1895 0,15	3660 0,18	7800 0,22	11500 0,24	22000 0,28	45550 0,36	70000 0,40	138000 0,46	241000 0,50	393000 0,60	820000 0,70	1510000 0,80
2,2	860 0,13	1950 0,16	3735 0,19	8000 0,22	11850 0,26	22700 0,30	46700 0,36	72000 0,40	141500 0,46	248500 0,55	403000 0,60	835000 0,70	1540000 0,80
2,4	905 0,13	2005 0,16	3925 0,20	8450 0,24	12450 0,26	23850 0,30	49250 0,38	75500 0,42	149000 0,46	260000 0,55	420500 0,65	880000 0,75	1590000 0,85
2,6	950 0,14	2160 0,17	4125 0,20	8800 0,26	13050 0,28	25000 0,32	51500 0,40	79000 0,44	156000 0,48	273000 0,60	439000 0,65	920000 0,75	1650000 0,85
2,85	995 0,15	2260 0,18	4330 0,22	9200 0,26	13600 0,30	26200 0,34	53500 0,42	80250 0,46	163000 0,50	284000 0,60	458000 0,70	960000 0,80	1730000 0,90
3,10	1040 0,15	2350 0,19	4510 0,22	9600 0,28	14150 0,30	27250 0,36	56000 0,44	86000 0,48	170000 0,55	297500 0,65	475000 0,70	990000 0,80	1800000 0,95
3,30	1085 0,16	2445 0,20	4690 0,24	10000 0,28	14650 0,32	28200 0,36	58000 0,44	89000 0,50	175500 0,55	307000 0,65	493500 0,75	1030000 0,85	1870000 1,00
3,6	1145 0,17	2605 0,22	4950 0,24	10500 0,30	15450 0,34	29600 0,38	61000 0,46	93500 0,50	184500 0,60	323000 0,70	520000 0,80	1065000 0,90	1955000 1,00
3,95	1200 0,18	2730 0,22	5200 0,26	11000 0,32	16250 0,34	31000 0,40	64000 0,50	98000 0,55	193000 0,60	338500 0,75	540000 0,80	1130000 0,95	2055000 1,10
4,4	1270 0,19	2890 0,24	5450 0,28	11650 0,34	17100 0,36	32750 0,42	65250 0,50	104000 0,60	203500 0,65	357500 0,80	570000 0,85	1195000 1,00	2160000 1,10
4,95	1355 0,20	3090 0,24	5850 0,30	12400 0,36	18250 0,40	34900 0,46	72000 0,55	111000 0,60	216500 0,65	379000 0,80	610000 0,90	1265000 1,00	2300000 1,20
5,5	1440 0,22	3150 0,26	6150 0,30	13150 0,38	19300 0,42	36950 0,48	76000 0,60	117000 0,65	229000 0,70	400000 0,85	640000 0,95	1335000 1,10	2425000 1,30
6,0	1515 0,22	3440 0,28	6450 0,32	13850 0,40	20450 0,44	38900 0,50	80000 0,60	123000 0,70	241500 0,80	421500 0,90	675000 1,00	1410000 1,20	2540000 1,30

Çizelge 10'un devamı: Döşemeden ısıtma sistemlerinde dikişli çelik boru tesisatı yaklaşık boru hesabı çizelgesi
Basınç kaybı, R = 0,026-1,4 mm.s.s/m

Anma ölçüsü	1/2"	4/3"	1 "	11/4"	11/2"	2"	21/2"	3"	4"	5"	6"	8 5/8"	10 3/4"
Iç Çap mm	16,0	21,6	27,2	35,9	41,8	53,0	68,8	80,8	105,3	130,0	155,4	210,1	263,0
R mm.s.s/m	$\Delta T_s = 10\ ^oC$ ve $T_s \cong 55\ ^oC$ için												
0,26	Q_s: 257 V_s: 0,04	595 0,05	1135 0,06	2440 0,07	3600 0,08	7000 0,09	14650 0,11	22600 0,13	44750 0,15	79500 0,17	128500 0,19	275500 0,22	505000 0,26
0,28	270 0,04	620 0,05	1190 0,06	2555 0,07	3270 0,08	7400 0,10	15350 0,12	23600 0,13	46550 0,15	83000 0,18	134500 0,20	288000 0,24	530000 0,28
0,31	282 0,04	650 0,05	1240 0,06	2665 0,08	3935 0,08	7650 0,10	16000 0,12	24600 0,14	48650 0,16	87000 0,19	140500 0,22	298500 0,24	555000 0,30
0,33	294,5 0,045	675 0,05	1290 0,06	2775 0,08	4090 0,09	7950 0,10	16650 0,13	25550 0,14	50500 0,17	90000 0,19	145500 0,22	310000 0,26	570000 0,30
0.36	311.5 0,045	710 0,06	1360 0,07	2925 0,08	4305 0,09	8400 0,11	17550 0,13	27000 0,15	53000 0,17	94500 0,20	152000 0,22	325000 0,26	605000 0,32
0,40	328 0,05	735 0,06	1425 0,07	3070 0,09	4530 0,10	8800 0,11	18350 0,14	28300 0,16	55500 0,18	100000 0,22	160000 0,24	340500 0,28	635000 0,34
0,44	349 0,05	790 0,06	1515 0,08	3260 0,09	4815 0,10	9350 0,12	19350 0,15	29900 0,17	59000 0,19	105000 0,22	170000 0,26	360000 0,30	670000 0,36
0,50	373.5 0,05	845 0,07	1615 0,08	3490 0,10	5150 0,11	10000 0,13	20700 0,16	31950 0,18	62500 0,20	112500 0,24	179500 0,26	382500 0,32	720000 0,38
0,55	397 0,06	895 0.07	1720 0,09	3695 0,11	5450 0,12	10650 0,14	21850 0,17	33850 0,19	66500 0,22	119000 0,26	190500 0,28	405000 0,34	750000 0,40
0,60	420 0,06	945 0.08	1815 0,09	3900 0,11	5750 0,12	11100 0,14	23100 0,18	35650 0,20	70000 0,22	125000 0,26	201000 0,30	426000 0,36	790000 0,42
0,65	439.5 0,06	995 0,08	1905 0,10	4115 0,12	6050 0,13	11700 0,15	24200 0,19	37450 0,20	73500 0,24	131000 0,28	211000 0,32	444000 0,36	825000 0,44
0,71	460 0,07	1040 0,08	1995 0,10	4295 0,12	6350 0,14	12250 0,16	25250 0,19	38150 0,22	77000 0,26	137500 0,30	221000 0,34	463500 0,38	860000 0,46
0,77	478.5 0,07	1085 0,09	2080 0,10	4470 0,13	6600 0,14	12700 0,16	26300 0,20	40750 0,22	79500 0,26	142000 0,30	229000 0,34	475000 0,40	900000 0,48
0,82	498.5 0,07	1130 0,09	2155 0,11	4650 0,13	6850 0,15	13250 0,17	27450 0,22	42300 0,24	83000 0,28	148000 0,32	238500 0,36	500000 0,42	930000 0,50
0,88	515 0,08	1170 0,09	2235 0,11	4815 0,14	7100 0,15	13800 0,18	28500 0,22	43800 0,24	85500 0,28	152500 0,32	247500 0,38	515000 0,42	960000 0,50
1,00	550 0,08	1250 0,10	2395 0,12	5150 0,15	7600 0,16	14650 0,19	30450 0,22	46650 0,26	91000 0,30	162000 0,34	263000 0,40	550000 0,46	1020000 0,55
1,1	585 0,09	1330 0,11	2540 0,13	5450 0,16	8050 0,17	15550 0,20	32100 0,24	49400 0,26	97000 0,32	171500 0,36	278000 0,42	580000 0,48	1075000 0,55
1,2	615 0,09	1405 0,11	2675 0,13	5750 0,16	8450 0,18	16450 0,22	33800 0,24	52000 0,30	102000 0,34	18000 0,38	292000 0,44	610000 0,55	1135000 0,60
1,4	650 0,10	1470 0,12	2810 0,14	6000 0,17	8900 0,19	17150 0,22	35500 0,26	54500 0,30	106500 0,34	189000 0,40	306000 0,46	645000 0,55	1190000 0,65

Çizelge 10: Döşemeden ısıtma sistemlerinde dikişli çelik boru tesisatı yaklaşık boru hesabı çizelgesi
Basınç kaybı, R = 0,060-0,24 mm.s.s/m

Anma ölçüsü	1/2"	4/3"	1 "	11/4"	11/2"	2"	21/2"	3"	4"	5"	6"	8 5/8"	10 3/4"
İç Çap mm	16,0	21,6	27,2	35,9	41,8	53,0	68,8	80,8	105,3	130,0	155,4	210,1	263,0
R mm.s.s/m	$\Delta T_s = 10\ ^oC$ ve $T_s \cong 55\ ^oC$ için												
0,060	Q_s: V_s:	282 0,02	481 0,025	1045 0,03	1570 0,035	3075 0,04	6350 0,05	9900 0,06	19450 0,06	35200 0,08	56000 0,08	124000 0,10	231500 0,12
0,065	-	254,5 0,02	510 0,025	1100 0,03	1640 0,035	3215 0,04	6650 0,05	10400 0,06	20900 0,07	36750 0,08	59000 0,09	131000 0,11	243000 0,13
0,072	-	268 0,02	530 0,025	1145 0,035	1710 0,035	3385 0,045	7000 0,05	10900 0,06	22050 0,07	38250 0,08	61500 0,09	136000 0,11	252500 0,13
0,077	-	280,5 0,025	555 0,03	1200 0,035	1780 0,04	3520 0,045	7300 0,06	11350 0,06	22550 0,07	40100 0,09	64500 0,10	142500 0,12	264000 0,14
0,082	-	294,5 0,025	575 0,03	1245 0,035	1855 0,04	3710 0,05	7600 0,06	11800 0,07	23600 0,08	41500 0,09	66500 0,10	147500 0,12	272500 0,14
0,088	-	306,5 0,025	595 0,03	1295 0,035	1920 0,04	3830 0,05	7850 0,06	12300 0,07	24300 0,08	42850 0,09	69000 0,10	153500 0,13	283000 0,15
0,10	-	330,5 0,025	635 0,03	1690 0,04	2060 0,045	4065 0,05	8400 0,06	13050 0,07	25800 0,08	45900 0,10	73500 0,11	164000 0,14	302000 0,16
0,11	150 0,02	354 0,03	675 0,035	1480 0,04	2185 0,045	4355 0,06	8900 0,07	13900 0,08	27450 0,09	48400 0,10	78500 0,12	172500 0,14	320000 0,17
0,12	159 0,025	375 0,03	720 0,035	1555 0,045	2310 0,05	4565 0,06	9400 0,07	14600 0,08	28800 0,09	51000 0,11	82000 0,12	182500 0,15	338000 0,18
0,13	167 0,025	397,5 0,03	755 0,04	1645 0,045	2430 0,05	4770 0,06	9900 0,08	15350 0,09	30400 0,10	54000 0,12	86500 0,13	192000 0,16	354500 0,19
0,14	176 0,025	416 0,035	795 0,04	1725 0,05	2545 0,05	4965 0,06	10350 0,08	16050 0,09	31900 0,10	56000 0,12	91000 0,14	200500 0,17	369000 0,19
0,15	184 0,025	434,5 0,035	830 0,04	1800 0,05	2645 0,06	5250 0,07	10750 0,08	16700 0,09	33150 0,11	59000 0,13	94500 0,14	205000 0,17	385000 0,20
0,16	192,5 0,03	452 0,035	865 0,045	1870 0,05	2760 0,06	5400 0,07	11200 0,09	17350 0,10	34300 0,11	61000 0,13	99000 0,15	216500 0,18	398500 0,20
0,17	200,5 0,03	467 0,04	900 0,045	1940 0,06	2860 0,06	5550 0,07	11650 0,09	18000 0,10	35700 0,12	63500 0,14	102000 0,15	228000 0,19	412500 0,22
0,18	208,5 0,03	485,5 0,04	935 0,045	2005 0,06	2965 0,06	5750 0,07	12000 0,09	18600 0,10	36800 0,12	65500 0,14	106000 0,16	231000 0,19	425500 0,22
0,20	215,5 0,03	500 0,04	965 0,05	2075 0,06	3065 0,07	6000 0,08	12400 0,10	19250 0,11	38200 0,13	68000 0,15	109000 0,16	238000 0,20	439500 0,24
0,21	223 0,035	515 0,04	895 0,05	2140 0,06	3155 0,07	6150 0,08	12800 0,10	19800 0,11	39250 0,13	70000 0,15	113000 0,17	244500 0,20	451000 0,24
0,23	230,5 0,035	535 0,045	1025 0,05	2205 0,06	3250 0,07	6300 0,08	13200 0,10	20400 0,11	40250 0,13	72500 0,16	115500 0,17	252000 0,20	462500 0,24
0,24	244 0,035	565 0,045	1080 0,05	2320 0,07	3435 0,07	6700 0,09	13950 0,11	21500 0,12	42500 0,14	75000 0,16	122000 0,18	264000 0,22	487000 0,26

Çizelge 9 'un devamı

Malzeme	k değeri (W/m K)
PARKE	
Meşe (% 12 nem)	0.18
Kayın (% 12 nem)	0.17
Çam	
(% 12 nem)	0.14
(% 2 nem)	0.17
Sedir (% 2 nem)	0.14
Çınar (% 2 nem)	0.17
Köknar (% 12 nem)	0.11
PORSELEN	1.55
BRANDA	0.17
LASTİK, sert	0.15
LASTİK, yumuşak vulkanize	0.14
LASTİK, taban malzemesi	0.06
LASTİK, stiren-butadien	0.11
LASTİK, köpük	0.04
KAROLAR	
Seramik	0.7
Kil	0.84
Çimento	1.1
Mantar tabanlı	0.07
Mermer	2.0
PVC asbest	0.84

Çizelge 9. Döşeme Kaplama (örtü) Malzemelerinin Isı İletim Katsayısı

Malzeme	k değeri (W/m K)
TUĞLA	0.93
HALILAR	
Akrilik	
cm^2 de 12 bukle, 0.5 cm kalınlık	0.03
Köpük tabanlı, 0.6 cm kalınlık	0.03
cm^2 de 7 bukle, 1.2 cm kalınlık	0.05
cm^2 de 9 bukle, 1.75 cm kalınlık	0.05
Naylon	
cm^2 de 14 bukle, 0.3 cm kalınlık	0.03
cm^2 de 7 bukle, 0.3 cm kalınlık	0.02
cm^2 de 10 bukle, 0.5 cm kalınlık	0.03
cm^2 de 12 bukle, 0.3 cm kalınlık	0.02
cm^2 de 12 bukle, 2 cm kalınlık	0.07
İnce doku, 0.6 cm	0.03
Yün	
cm^2 de 7 bukle, 1.2 cm kalınlık	0.02
Polyester	
cm^2 de 8 bukle	0.02
Uzun bukleli polypropilen	0.03
MUKAVVA	0.43
MUKAVVA, BİTÜMEN	0.18
SIKIŞTIRILMIŞ YONGA LEVHA	0.14
JÜT, 0.6 cm kalınlık	0.02
ŞAP	1.40
MANTAR ELYAFI	0.16
CİPSUM	0.17
FİBER LEVHA	0.07
GRANİT	2.50
SUNTA	0.12
SERT TAHTA	
1 cm kalınlık	0.10
0.6 cm kalınlık	0.11
Döşeme kaplaması, 2 cm kalınlık	0.15
LİNOLYUM	0.28
DERİ	0.15
MİKA	0.43

Çizelge 7: Döşeme Kaplama Malzemesi İçin Yayınım Katsayıları (ε)

Malzemenin Cinsi	Yayınım Katsayısı (ε)
Parke (parlak, cam cila)	0.88
Parke (mat)	0.90
Marley (açık renk)	0.90
Marley (koyu renk: kırmızı-kahverengi)	0.92
Mermer (tabii renkler)	0.89
Mermer (koyu renkler)	0.91
Siyah döşemelik plastik	0.90-0.95
Beton	0.88
Halı (uzun tüylü veya açık renk)	0.90
Halı (kısa tüylü makine halısı veya koyu renkli)	0.95
Tahta	0.92
Sunta	0.90
Siyah lastik muşamba	0.95
Linolyum (açık renkler)	0.90
Linolyum (koyu renkler)	0.92
Seramik karo	0.87

Çizelge 6. Tabii Hava Sızıntısı Yükseklik Zammı, ε_y

	Mahalin Zeminden Yüksekliği (m)										
	0	5	10	15	20	25	30	35	40	45	50
ε_y	1.0	1.0	1.0	1.2	1.4	1.5	1.6	1.7	1.9	2.0	2.0
	55	60	65	70	75	80	85	90	95	100	
ε_y	2.1	2.2	2.3	2.4	2.4	2.5	2.6	2.7	2.7	2.8	

TS 2164 Çizelge 5 (aynen) : Yön Zammı Z_H

Yön	G	GB	B	KB	K	KD	D	GD
Z_H	-0.05	-0.05	0	+0.05	+0.05	+0.05	0	-0.05

Çizelge 1: Sürekli Rejimde Isıtma Yükü Düzeltme Faktörü, C

	C		
	Birim Yapı Kütlesi m_Y (kg/m^2)		
	$m_Y < 600\ kg/m^2$	$600 \leq m_Y \leq 1400\ kg/m^2$	$m_Y > 1400\ kg/m^2$
	Hafif Yapılar	Normal Yapılar	Çok Yoğun Yapılar
I. ISI İKLİM BÖLGESİ	1.0	0.88	0.80
II. ISI İKLİM BÖLGESİ	1.0	0.85	0.78
III. ISI İKLİM BÖLGESİ	1.0	0.80	0.75

Çizelgede m_Y değerleri aşağıdaki formül yardımıyla bulunur:

$$m_Y = \frac{\overline{m}}{\Sigma A_{dış}} \qquad (kg/m^2)$$

burada:

$$\overline{m} = \sum_{Dış} (0.5 \text{ x çelik yapı elemanları toplam kütlesi} + 2.5 \text{ x tahta yapı elemanları toplam kütlesi} + \text{diğer yapı elemanları toplam kütlesi})$$

$$+ 0.5 \sum_{iç} (0.5 \text{ x çelik yapı elemanları toplam kütlesi} + 2.5 \text{ x tahta yapı elemanları toplam kütlesi} + \text{diğer yapı elemanları toplam kütlesi}) \quad (kg)$$

$\Sigma_{Dış}$: dış temaslı elemanlara ait toplamı,

$\Sigma_{iç}$: iç elemanlara ilişkin toplamı belirtir.

$\Sigma A_{dış}$: Binadaki dış duvar ve pencere toplam alanı, (m^2)

dır.

EK-A. (Devamı)

α_c	:	elektrik direnci sıcaklık katsayısı	$°C^{-1}$
	:	0.0039 $°C^{-1}$, bakır tel için	
	:	0.0040 $°C^{-1}$, bronz alaşımlı tel için	
α_o	:	ısıl genleşme katsayısı	$°C^{-1}$
	:	$16.7x10^{-6}$ $°C^{-1}$, bakır tel için	
	:	$13x10^{-6}$ $°C^{-1}$, bronz alaşım tel için	
C_D	:	ışıma katsayısı, (5.67 . ε W/m^2 K^4)	
c	:	hacmin yapı içersinde bulunduğu konum katsayısı (Çizelge 8)	
n_k	:	Döşeme kaplaması (örtü) sayısı,	
t_j	:	(j) sayılı kaplamanın (örtü) kalınlığı	m
k_j	:	(j) sayılı kaplamanın (örtü) ısı iletim katsayısı	W/mK
L_T	:	Bir panel devresindeki ısıtıcı eleman toplam uzunluğu	m
M	:	Isıtıcı eleman merkezleri arası mesafe (modulasyon)	mm

EK-A. SEMBOLLER ÇİZELGESİ

Sembol		Sembolün Anlamı	Birimi
Q_T	:	zamlı toplam ısı kaybı	kW
Q_L	:	hava sızıntısı (infilitrasyon) ve yenilenmesi toplam ısı kaybı	kW
C	:	ısıtma yükü düzeltme katsayısı	
Z_H	:	TS 2164'den alınan yön zammı	
n_y	:	ısı kaybı olan yüzeylerin sayısı	
k_i	:	ısı kaybeden yüzeyin ısı geçirme katsayısı	kW/m °C
A_i	:	ısı kaybeden yüzeyin net alanı	m^2
ΔT	:	sıcaklık farkı	°C
a	:	sızma katsayısı (Çizelge 3)	m^3/m
l	:	sızma uzunluğu (Fuga boyu)	m
R	:	oda özellik katsayısı (Çizelge 4)	
H	:	yapı ısı özelliği katsayısı (Çizelge 5)	$kcal/m^3$ °C
Z_E	:	köşe katsayısı (TS 2164)	
ε_y	:	bina yükseklik katsayısı (Çizelge 6)	
T_a	:	eşdeğer konfor iç hesap sıcaklığı (Çizelge 2)	°C
T_d	:	dış hesap sıcaklığı (TS 2164)	°C
V_L	:	değişen taze hava miktarı (yenilenen hava)	m^3/h
C_p	:	havanın özgül ısısı	kJ/m^3°C
Q_{Top}	:	toplam ısıtma kapasitesi	kW
Q_D	:	destek ısıtıcı ısıtma kapasitesi	kW
Q_h	:	döşemeden ısıtma yükü	kW
Q_K	:	panel devresinin ısı kaybı	kW
Q_{Yi}	:	(i) sayılı panelin ısıtma kapasitesi	kW
n_p	:	ısıtılan mahaldeki panel sayısı	
A_{pi}	:	(i) sayılı panelin döşeme yüzeyinde etkili olduğu alan	m^2
q_{Yi}	:	(i) sayılı panel birim ısıtma kapasitesi	W/m^2
A_{pr}	:	panel kullanımına müsait toplam yüzey alanı	m^2
q_{ri}	:	panelin ışıma ile birim ısıtma kapasitesi	W/m^2
q_{ci}	:	panelin taşınım ile birim ısıtma kapasitesi	W/m^2
h	:	yörenin denizden olan yüksekliği (rakım)	m
$D_{eş}$	:	hacim döşemesinin eşdeğer çapı	m
M	:	boru merkezleri arası mesafe	mm
k_h	:	boru malzemesi ısı iletim katsayısı	W/mK
k_p	:	şap malzemesi ısı iletim katsayısı	W/mK
R_D	:	döşeme kaplama malzemesi toplam ısıl direnci	m^2 K/W
γ	:	ısıtıcı akışkanın işletme sıcaklığındaki özgül ağırlığı	kg/m^3
ΔT_s	:	ısıtıcı akışkanın panele giriş ve çıkış sıcaklık farkı, (Genellikle, 10 °C)	°C
T_n	:	alt kattaki komşu mahal iç tasarım sıcaklığı	°C
R_y	:	yukarı yönde toplam ısıl direnci	m^2 K/W
λ	:	boru sürtünme katsayısı	
D_i	:	boru iç çapı	m
D_o	:	Boru dış çapı	m
v_s	:	boru içersindeki akışkan hızı	m/s
γ	:	tasarım ortalama su sıcaklığında akışkan özgül ağırlığı	kg/m^3
g	:	yer çekimi ivmesi	9.81 m/sn^2
Q_Y	:	panelin gerekli ısıtma kapasitesi	kW
X	:	panel ısıtma verimi	
E	:	mevcut sistemdeki elektrik potansyeli (voltaj)	volt
R_L'	:	T_b tel dış sıcaklığına göre düzeltilmiş birim elektrik direnci	ohm/m
R_L	:	25 °C sıcaklıkta birim elektrik direnci (Çizelge 12)	ohm/m
Re	:	Panel devresininin elektrik direnci	ohm

ATIF YAPILAN STANDARTLAR

TS …… Döşemeden Isıtma Sistemleri - Terimler ve Tarifler
TS 2164
TS 388
TS 825

} yayın tarihleri, İngilizce ve Türkçe isimleri kapak arkasına yazılacak.

Boruların dilatasyon aralığından geçirilmemesine özen gösterilir. Bir panel hiç bir zaman dilatasyon ile bölünmemelidir. Sadece gidiş ve dönüş boruları dilatasyon aralığından geçirilebilir. Bu geçişlerde borular iç çapı $2 \cdot D_0$ olan ve en az 30 cm uzunluğundaki elastik kılıflar içerisinden geçirilir.

1.2.3.3.2. Elektrikli Döşemeden Isıtma Sistemi

Elektrikli ısıtma sistemleri, sıcak sulu döşemeden ısıtma sistemi bulunduğu döşemelerde, banyo küveti altında, döşemeye oturan sabit eşya altında, açık havada, iç veya dış mahal merdivenlerinde ve havuz kenarlarında uygulanmamalıdır.

Duvardan ısıtma yapılması durumunda panel, döşeme yüzeyinden en az 10 cm yukarıda başlamalı ve panelin üst sınırı döşemeden en çok 1.5 m yükseklikte olmalıdır. Her panel yüzeyinde kolayca görülecek ve silinmeyecek bir uyarı levhası bulunmalı, bu uyarı sistemin projesinde belirtilmelidir.

Isıtıcı elemanlar döşeme kaplamasından en çok 2 cm aşağıda bulunmalı, ısıtıcı elemanlar altındaki şap veya beton kalınlığı en az 5 cm olmalıdır. Isıtıcı elemanların hem elektrik hem elektromanyetik yalıtımı sağlanmalıdır. Her panelin kendisine ait topraklama ve devre kesici emniyet tertibatı bulunmalıdır. Elektrikli döşemeden ısıtma sistemlerinde sadece Açık-Kapalı türde (on-off) kontrol uygulanmalı, akım şiddetini değiştirmek suretiyle kontrol yapılmamalıdır.

Canlıların elektromanyetik dalgalardan etkilenmemesini teminen her komşu ısıtıcı eleman diğerine göre zıt yönde akım taşıyacak şekilde serilmelidir.

- Her mahalde bir pano, ana şalter, gerekli akım kesici emniyet tertibatı bulunmalıdır.
- Pano kolay görülebilen bir yere yerleştirilmeli ve örtülmemelidir.
- Her mahalde, elektrik tesisatı ile ilgili ikaz yazısı açıkça görülür ve okunur şekilde bulundurulmalıdır.
- Elektrikli ısıtma sisteminin panel adedi, anma boyutları, konumları, rezistans tipi, çektiği akım ve ısıtma kapasitesi proje üzerinde belirtilmelidir.
- Elektrikle ısıtma panelleri döşemede hiç bir zaman dilatasyon ile bölünmemelidir.

Tasarım sonunda özgül ısıtma kapasitesi, (q_E) ayrıca kontrol edilir:

$$q_E = W_{TOP} \cdot 1000 / L \qquad (W/m) \qquad (52)$$

q_E değeri 5 W/m 'den az, 10 W/m 'den çok olmamalıdır.

1.2.3.3. Genel Esaslar

Bu standardda yer alan tasarım algoritmasına ve sınırlamalara tam uyulmak kaydı ile bilgisayar programları hazırlanıp kullanılabilir. Bu gibi programlarla hazırlanacak ve tasarımı yapılan hacme münhasır abaklar da tasarım raporuna örnek olarak eklenebilir. Bu abaklarda şu bilgiler bulunmalıdır:

- Eşdeğer konfor iç sıcaklığı
- Odanın boyutları
- Dış hesap sıcaklığı
- Kullanılan ısıtma elemanı tipi, malzemesi ve anma boyutları, varsa standardı
- Mahalin kullanım maksadı
- Döşeme konstrüksyonu
- Döşeme kaplamaları detayı

Bu bilgilere haiz bir abakta birim ısı çıktısı (q_y), ortalama su sıcaklığı (T_s) ve etkin köşeme sıcaklığı (T_p) eksenlerine göre değişik boru aralıkları için eğriler çizilir.

Bu gibi abaklar genel tasarım maksadı ile kullanılamaz.

1.2.3.3.1. Sıcak Sulu Döşemeden Isıtma Sistemleri

Döşemeden ısıtma sistemlerinde pompalı, atmosfere kapalı hidrolik devre sistemi tatbik olunur. Bu amaçla atmosfere kapalı genleşme deposu, hava pürjörü ve kompensatör kullanılır. Dolaşım pompası, su ısıtıcı eleman (kazan) çıkışına ve güvenlik borusu, su besleme cihazı, pürjor ve atmosfere kapalı genleşme deposunu takiben konur. Sistemde eşanjör var ise, döşemeden ısıtma sistemi devresine ayrı pompa, pürjör, atmosfere kapalı genleşme deposu konulur. Sistemde 3 veya 4 yollu vana var ise bu vana(lar) dolaşım pompasından önce yerleştirilir.

Kollektörler, uygun bir yerde, sabit kollektör dolabı içersine alt ve üstlü olarak konulur. Alttaki kollektörün gidiş kollektörü olarak kullanılması tavsiye edilir. Genleşme deposu, genleşme düzenleyici sistem(ler), güvenlik düzenleri TS 2164 'de verilen esaslar dahilinde projelendirilir.

Kullanılan borularda oksijen geçişine karşı bariyer bulunmuyor ise sıcak su üretici (kazan) devresi ile döşemeden ısıtma sistemi devresi birbirlerinden korozyona dayanıklı bir ısı değiştirici (eşanjör) ile ayrılmalı, ve/veya hidrolik devrenin tümünde, dolaşan sıvıya uygun miktar ve kalitede korozyon önleyici sıvı eklenmelidir. Bu sıvının termoplastik veya lastik boru malzemesi ile uyumu kontrol edilmelidir. Korozyon önleyici sıvının miktarı ve tipi projede belirtilir. Bu sıvı, fonksyonunu geniş bir miktar yüzdesi toleransı içinde korumalıdır.

Boruların hangi demet tarzında mahal döşemesine serileceği her mahal için ayrı ayrı proje çiziminde gösterilir. Sadece boru uzunluğu ve aralığı belirtilmesi ile yetinilmez.

Kollektör ve kollektör ağızları numaralandırılır her ağzın sağladığı ısı miktarı, bağlı olduğu borunun toplam uzunluğu çizim üzerinde gösterilir.

Çizimlerde, her mahal için ayrıca aşağıdaki bilgiler bulunmalıdır:

- Eşdeğer konfor iç sıcaklığı,
- Döşeme kaplamaları kalınlık ve türleri,
- Işıma ile ısıtma oranı,
- Döşeme yüzey sıcaklığı,
- Aynı mahalde farklı M değerleri var ise, her (M) değerine göre boru demet çizimleri ayrı ayrı verilmelidir.
- Destek ısıtıcı var ise konumu, tipi ve kapasitesi, bağlantıları.

Borular, bükülebilir en küçük çaptan (D_{min}) daha küçük bir kavisle bükülmemelidir (Çizelge 11).

Çok soğuk bölgelerde sistem donmaya karşı korunmalıdır.

1.2.3.2. Elektrikli Döşemeden Isıtma Sistemi Projelendirmesi
1.2.3.2.1. Mahal Isı Yükü Hesabı

Elektrikli Döşemeden Isıtma Sistemi ile ısıtılacak bir mahalin ısıtma yükünün hesabı Madde 1.1 de belirtildiği gibi yapılır. Sistemin sadece duvar veya tavandan uygulanması halinde mahalin ısı yükü TS 2164'e göre hesaplanır.

1.2.3.2.1. Elektrikli Isıtıcı Elemanlar

Isıtılan hacme konacak panel sayısı n_p 'nin bulunması için her panel'in yaklaşık 1 kW 'lık ısıtma kapasitesi ile sınırlı tutulması gözönünde tutulur :

$$n_p = Q_h \ / (1 \text{ kW}) \qquad (45)$$

Her bir panel için gerekli toplam elektrik gücü (W_{TOP}), aşağıdaki formül ile hesaplanır.

$$W_{TOP} = Q_Y / X = Q_h / (n_p \,.\, X) \qquad (\text{kW}) \qquad (46)$$

Panel ısıtma verimi X, normalde 0.9, taban ve çevre yalıtımı olmayan döşemeler için 0.8 olarak alınır. Panel içersinde yerleştirilmesi gerekli elektrik teli miktarı (L), gerekli panel elektrik direnci (R_e) ve düzeltilmiş birim elektrik direnci (R_L') 'ye bağlı olarak aşağıdaki formülle bulunur:

$$L = R_e / R_L' \qquad (\text{m}) \qquad (47)$$

R_e değeri aşağıdaki formülle hesaplanır:

$$R_e = E^2 / (W_{TOP} \,.\, 1000) \qquad (\text{ohm})$$

Döşemeden ısıtmada yaygın olarak kullanılan elektrikli ısıtıcı elemanların özellikleri Çizelge 12'de verilmiştir. Bu çizelgede elektrikli ısıtıcı elemanların birim elektrik direnci (R_L), 25 °C oda sıcaklığında verildiğinden, bu değerler tasarım şartlarındaki ısıtıcı elemanın dış yüzey sıcaklığı T_b 'ye göre aşağıdaki formülle düzeltilmelidir:

$$R_L' \cong R_L \,.\, \frac{(1 + \alpha_e T_b)}{(1 + \alpha_o T_b)} \qquad (\text{ohm/m}) \qquad (48)$$

Formüldeki ısıtıcı eleman dış yüzey sıcaklığı (T_b) Madde 1.2.3.1.1'de belirtildiği gibi hesaplanır. Eğer elektrikli döşemeden ısıtma sistemi döşemede değil ise, 10 numaralı eşitlikteki 2.67 sabiti duvar için 0.75 tavan için 0.50 katsayısı ile çarpılır. Panelin kısmi trafo yükünün (Q_e) hesabında panel terminali ile, trafo bağlantısını sağlayan elektrik kablosunun uzunluğu, (ΔL), gözönünde tutulur. Bu bağlantı kablosunun uzunluğu en az 2 m, en çok 3 m olmalıdır.

Panelin kısmi trafo yükü (Q_e);

$$Q_e = W_{TOP} + E^2 / (\Delta L \,.\, R_L \,.\, X) \qquad (\text{kW}) \qquad (49)$$

formülü ile hesaplanır. Hesaplanan bu değer 1.20 kW'ı geçmemelidir. Toplam elektrik akımı (I) :

$$I = E / (R_e + \Delta L \,.\, R_L) \qquad (\text{A}) \qquad (50)$$

Elektrikli ısıtıcı eleman merkezleri arasındaki mesafe (M) değerleri Çizelge 13'de verilmiştir.

Panel içerisindeki ısıtıcı eleman uzunluğu (L) bilindiğine göre, öngörülen panel yüzey alanı kullanılarak M değeri aşağıdaki formülle hesaplanır:

$$M = \frac{A_p \,.\, 1000}{L} \qquad (\text{mm}) \qquad (51)$$

Bulunan M değeri Çizelge 13'de verilen en yakın değer olarak seçilir ve yukarıdaki formülde yerine konularak panel yüzey alanı yeniden hesaplanır. Bu alan 1 m^2 'den az, 10 m^2 'den çok olmamalıdır.

1.2.3.1.6. Boru Hesapları

1.2.3.1.6.1. Kolon Borularının Çap Tayini

Kolon borularının çap tayini TS 2164 standardında verilen usulle yapılır. Bu amaçla TS 2164, Föy 9 kullanılır. Ancak döşemeden ısıtma sistemlerinde; ΔT_S 'nin genellikle 20 °C yerine 10 °C olarak kullanılması ve ısıtıcı akışkan ortalama sıcaklığının daha düşük olması sebepleri ile gerekli hız (V) ve birim uzunluktaki basınç kaybı (R) değerleri TS 2164 yerine ekte verilen Çizelge 10'dan alınır.

1.2.3.1.6.2. Isıtıcı Borularda Basınç Kayıpları

Döşemeden ısıtma sistemlerinde çapları önceden tesbit edilen borulardaki basınç kayıpları aşağıdaki formüller yardımıyla hesaplanır:

$$R = (\lambda / D_i) \cdot (v_S^2/2) \qquad \text{(Pa/m)} \tag{37}$$

veya

$$R = (\lambda / D_i) \cdot (v_S^2/2.g) \qquad \text{(mm.ss/m)} \tag{38}$$

Boru sürtünme katsayısı, λ aşağıdaki formül aracılığı ile hesaplanır:

$$\lambda = 0.0055\ (1 + (20.\varepsilon/D_i) + \upsilon / (v_S.D_i) \cdot 10^6)^{1/3}) \tag{39}$$

Thermoplastik ve lastik borular için ε değeri 0.1 mm olarak alınabilir.

Isıtıcı akışkanın ortalama tasarım sıcaklığındaki (T_S) kinematik viskositesi (υ), su için aşağıdaki formülle bulunur:

$$\upsilon_{su} = \{1.37 - 0.024\ T_S + 1.53\mathrm{x}10^{-4}\ T_S^2\} \times 10^{-6} \quad (\mathrm{m^2/s}) \tag{40}$$

Isıtıcı akışkan özgül ağırlığı, ısıtıcı akışkan ortalama tasarım sıcaklığı (T_S)'ye göre, su için aşağıdaki gibi hesaplanır:

$$\gamma_{su} = 1000.7 - 0.0585\ T_S - 0.0039\ T_S^2 \qquad (\mathrm{kg/m^3}) \tag{41}$$

Boru dönüş ve bükümlerinde oluşan özel basınç kayıpları (37) veya (38) numaralı formüllerden elde edilen R değerinin yaklaşık % 10 'u olarak hesap edilebilir.

Panel devresindeki diğer özel direnç basınç kayıpları, (Z), TS 2164'de belirtilen hesap usulleri ile hesaplanır.

Bu işlemlerden sonra, toplam basınç kaybı (H) aşağıdaki formülle bulunur:

$$H = \Sigma\ R.\ L_t + \Sigma\ Z \qquad \text{(Pa)} \tag{42}$$

L_t, panel devresi içerisinde (L), ayrıca kollektörden panele gidiş, panelden kollektöre dönüş toplam parkur uzunluğu (ΔL) de dahil olmak üzere toplam boru uzunluğudur (m).

$$L_T = L + \Delta L \tag{43}$$

Panel devresi içersindeki boru demeti uzunluğu, L :

$$L = A_p/M \cdot 1100 \quad \text{(m)} \tag{44}$$

Burada 1100 sayısı boru kavislerini de yaklaşık olarak hesaba katmaktadır. MM 'nin birimi mm.dir.

1.2.3.1.7. Döşemeden Isıtmada Kullanılacak En Yüksek Isıtıcı Akışkan Sıcaklığı

Boru malzemesinin ısıya dayanımına bağlı olarak döşemeden ısıtma sistemlerindeki ısıtıcı akışkan sıcaklıkları Çizelge 11'de verilen değerleri geçmemelidir ancak konutlardaki döşemeden ısıtma sistemlerinde ise malzeme uygun olsa bile 70 °C sınırı aşılmamalıdır.

$$R_y = \sum_{j:1}^{n_k} t_j / k_j + \{ \frac{1}{\alpha_y} . 0.4 \} \quad \text{(Bakınız Şekil 1)} \tag{31}$$

$$\underbrace{\qquad\qquad}_{V}$$

R_D = Döşeme kaplaması toplam ısıl direnci

R_D hesabında, döşeme kaplamalarında her türlü muhtemel değişiklikler gözönünde tutulması tasarım emniyeti açısından uygun olacaktır.

$(1/\alpha_y)$ döşemeden mahale olan ısı taşınım katsayısı aşağıdaki gibi hesaplanır:

$$1/\alpha_y = \frac{1}{2.67\,(T_p - T_a)^{0.25}} \qquad (m^2.K/W) \tag{32}$$

R_a ise boru merkezlerinden geçen düzlem itibarı ile aşağıda kalan bütün malzemelerin toplam ısıl direnci olup aşağıdaki formülle hesaplanır:

$$R_a = \sum_{j=1}^{n_a} t_j / k_j + (\frac{1}{\alpha_a}) \tag{33}$$

Alt kattaki mahal ile tavanı arasındaki ısı taşınım direnci de katılır. Bu taşınım direnci $(1/\alpha_a)$ ısıtılan bir komşu mahal için , (4.3 m^2.K/W)'dır. Isıtılmayan bir komşu mahal için ise (1.07 m^2.K/W)'dır.

1.2.3.1.3. Kazan (Trafo) Isı Yükünün Hesabı

Panelin kazan (trafo) ısı yükü (Q_B) aşağıdaki formüllerden biri ile bulunur:

$$Q_B = \frac{Q_Y}{X} \quad \text{veya} \quad Q_B = Q_Y + Q_A \qquad (kW) \tag{34}$$

Kazan (trafo) ısı yükü ise yapıdaki bütün panellerin kısmi kazan (trafo) ısı yükleri toplanarak bulunur.

1.2.3.1.4. Destek Isıtma Kapasitesi

Çeşitli sınırlamalar sebebi ile destek ısıtmaya ihtiyaç var ise, bu kapasite aşağıdaki formülle bulunur:

$$Q_D \geq (Q_h - Q_{Top}) \qquad kW \tag{35}$$

1.2.3.1.5. Işıma ile Isıtma Oranının Hesabı

Bir mahalde ısıtmanın, en az % 60 oranında ışıma ile gerçekleşmesi gerekir. Bu oran her mahalde kontrol edilmelidir:

$$I = \frac{\sum_{i:1}^{n_p} \frac{q_{ri} \cdot A_{pi}}{1000} + Q_D \cdot 0.2}{Q_{Top} + Q_D \cdot 0.8} \tag{36}$$

NOT : % 60 'dan az oranlara haiz mahallerde ısı yükü TS 2164'e tamamen uygun olarak hesaplanır.

Yüzey sıcaklığı fark katsayısı, (t_W) aşağıdaki formülle hesaplanır:

$$t_W = (T_{max} - T_{min}) / T_p \tag{23}$$

Yaşanan hacimlerde (t_W) 0.25 değerini aşmamalıdır.

Isıtıcı eleman dış sıcaklığı aşağıdaki formülden bulunur:

$$T_b = T_{max} + q_Y . (R_D + (T - D_o/2)/k_p) \tag{24}$$

1.2.3.1.2. Ortalama Su Sıcaklığının (T_s) Hesabı

Ortalama su sıcaklığı (T_S), aşağıdaki formülle hesaplanır:

$$T_s: \frac{q_Y . (M/1000)}{X . \pi} . \left(\frac{1}{\alpha_s . D_i} + \frac{1}{2 k_h} \ln\left(\frac{D_o}{D_i}\right)\right) + T_b \tag{25}$$

(α_S) Su/boru iç cidarı ısı taşınım katsayısı aşağıdaki şekilde hesaplanır:

$$\alpha_S = 1056\ [0.02\ (T_S + 273) - 4.06]\ v_S^{0.8} / D_i^{0.2},\ (W/m^2K) \quad \text{(su için)} \tag{26}$$

Bu eşitlikte (T_S) değeri bulunduğu için 25 numaralı formül doğrudan (T_S) için kullanılamaz. Elle yapılan hesaplamalarda basitleştirme sağlamak için,

$$\alpha_S = 3700\ W/m^2K$$

değeri kullanılabilir.

(v_S) Boru içerisindeki ortalama ısıtıcı akışkan hızı, aşağıdaki formülle hesaplanır:

$$v_s = \frac{4 . V}{\pi . D_i^2 . 3600} \quad (m/s) \tag{27}$$

$$(V)\ \text{Su debisi} = \frac{Q_Y / X}{\gamma . \Delta T_s} \quad (m^3/h) \tag{28}$$

olmalıdır. Döşemeden Isıtma Sistemleri için genelde giriş-çıkış su sıcaklık farkı (ΔT_S) 10 ^oC olarak alınır.

(X) Panel ısıtma verimi,

$$X = \frac{Q_Y}{Q_Y + Q_A} \tag{29}$$

dir.

(X) değeri yaklaşık olarak,

$$X \cong \frac{1}{1 + \frac{R_y}{R_a} . \{ \frac{(T_b - T_a)}{(T_b - T_n)} \}} \tag{30}$$

alınabilir. Mecburi haller dışında X değeri 0.8 'den az olmamalıdır.

1.2.3. Döşemeden Isıtma Sistemlerinin Projelendirilmesi
1.2.3.1. Sıcak Sulu Döşemeden Isıtma Sistemi Projelendirilmesi
1.2.3.1.1. Isıtıcı Eleman Dış Yüzey Sıcaklığının (T_b) ve Döşeme Yüzeyi Maximum (T_{max}) ve Minimum (T_{min}) Sıcaklık Değerlerinin Hesabı

Hesaplanan birim ısıtma kapasitesinin sağlanabilmesi için ısıtıcı eleman, dış yüzeyi belirli bir sıcaklıkta (T_b) bulunmalıdır. Bu sıcaklık değerinden hareketle boru içerisinden geçirilen suyun ortalama sıcaklığı (T_s) hesap edilir. Elektrikli döşemeden ısıtma sistemlerinde ise elektrikli ısıtıcı eleman dış yüzey sıcaklığı (T_b) 'nin hesabı ile yetinilir.

Döşeme yüzey sıcaklık dağılımı ve döşeme içi ısı transferi hesapları için kullanılan model Şekil 1'de gösterilmiştir.

Çeşitli malzemelerin k değerleri Çizelge 9 'da verilmiştir.

Sıcak sulu döşemeden ısıtma sistemlerinde, tavsiye edilen M değerleri sırası ile 100, 150, 200, 250, 300, 350 ve özel hallerde 400 mm.dir. Koridor ve benzeri dar yerlerden geçen gidiş/dönüş boruları için bu değerler dışına çıkılabilirse de, bu mesafe 100 mm.den az, 750 mm.den çok olmamalıdır.

Şekil 1'deki model esas alınarak, (T_{max}) aşağıdaki formülle belirlenir:

$$T_{max} = T_a + \frac{(T_p - T_a) \cdot M / 1000}{[2 \cdot W \cdot \eta + D_o]} \quad (^{\circ}C) \qquad (17)$$

Döşeme yüzeyi maksimum sıcaklığı (T_{max}) yaşanan hacimlerde 40 °C değerini geçmemelidir.

NOT : (T_{max}) ve (T_{min}) değerleri, döşeme malzemesinde meydana getireceği ısıl gerilmeler ve insan fizyolojisine etkileri açısından dikkate alınmalıdır.

(η) kanatcık verimi aşağıdaki formül yardımıyla bulunur.

$$\eta : \tanh (m.W) / (m.W) \qquad (18)$$

$$\tanh (m.W) = \frac{e^{m.W} - e^{-m.W}}{e^{m.W} + e^{-m.W}} \qquad (19)$$

Burada $W = \left(\frac{M}{1000} - D_o\right) / 2$.

(m) değeri aşağıdaki formülle hesaplanır:

$$m = \sqrt{\frac{q_{ü} / (T_p - T_k) + q_{a} / (T_p - T_a)}{k_{eş} \cdot T_{TOP} \cdot 1{,}25}} \qquad (20)$$

Eşdeğer ısı iletim katsayısı ($k_{eş}$) aşağıdaki formül yardımıyla hesaplanır:

$$k_{eş} = \frac{\sum_{j:1}^{n_k} t_j \cdot k_j + T \cdot k_p}{T_{TOP}} \qquad (21)$$

Burada n_k üstüste konulmuş döşeme kaplamalarının sayısıdır.

Döşemeden ısıtma sistemlerinde şap malzemesi içersindeki agrega anma boyu 16 mm'yi geçmemeli, (k_p) değeri de 0.86 W/mK değerinden az olmamalıdır.

$$T_{min} = T_a + \frac{2 \cdot (T_{max} - T_a)}{e^{m.W} + e^{-m.W}} \quad (^{\circ}C) \qquad (22)$$

Burada n_p, mahaldeki panel sayısıdır. Ancak, dış duvar ve pencereler yakınında, ayrıca üzerinde çok gezilmeyen yüzeylerdeki panellerde q_{Yi} değeri daha yüksek seçilebilir. Bunlara ek olarak, aşağıdaki şart gözetilmelidir:

$$\sum_{i:1}^{n_p} A_{pi} \leq A_{pı} \tag{9}$$

1.2.2. Döşeme Yüzey Sıcaklığının (T_p) Hesabı

Herhangi bir panelin birim ısıtma kapasitesi hesaplandıktan sonra, bu kapasiteyi sağlayacak döşeme yüzey sıcaklığı, aşağıdaki formülle ve deneme yanılma metodu ile hesaplanır:

$$q_{Yi} = \underbrace{\sigma \,.\, C_D \,.\, (T_p - T_k)}_{q_{ri}} + \underbrace{2.67\,(T_p - T_a)^{1.25} F_1 \,.\, F_2}_{q_{ci}} \quad \text{W/m}^2 \tag{10}$$

(σ) ışıma ısı transferi lineer ifade katsayısı ($°C^3$) aşağıdaki eşitlik yardımıyla hesaplanır:

$$\sigma = 0.0105 \,.\, (\frac{T_p + T_k)}{2}) + 0.7955 \quad \{15\ °C \leq (\frac{T_p + T_k)}{2}) \leq 30\ °C\ \text{için}\} \tag{11}$$

(T_k) : Isıtılmayan iç yüzeylerin ortalama sıcaklığı (°C) aşağıdaki formülle hesaplanır:

$$T_k \equiv T_a - c \,.\, W_d \qquad (14\ °C \leq T_k \leq 19\ °C\ \text{için}) \tag{12}$$

Burada;

c : Oda pozisyon faktörü (Çizelge 8)

(W_d) : Dış sıcaklık faktörü aşağıda belirtildiği gibi hesaplanır:

$$W_d = \frac{15}{25 + T_d}, \quad (°C) \quad (T_d \geq -20\ °C\ \text{için}) \tag{13}$$

(F_1) : Deniz kotuna göre düzeltme katsayısı aşağıdaki formülle bulunur:

$$F_1 = (1 - 2.257x10^{-5} \,.\, h)^{2.6275} \tag{14}$$

(F_2) : Mahal büyüklüğü düzeltme katsayısı aşağıdaki formülle bulunur:

$$F_2 = (4.96/D_{eş})^{0.08} \tag{15}$$

Bütün değerler hesaplandıktan sonra 10 numaralı formül aracılığı ile panel yüzeyinin sahip olduğu varsayılan homojen yüzey sıcaklık değeri (T_p) deneme yanılma metodu ile hesaplanır. Bu hesaplamaya, q_{yi} değerine % 2 yaklaşım sağlanana kadar devam edilir.

Bu sıcaklık (T_p) genelde 29 °C'ı aşmamalıdır. Ancak gerekli olduğu durumlarda ve mahalin ayni kişilerce kullanım süresi az olan bölümlerinde sürekli olarak 31 °C'a, kısa aralıklarla da 35 °C'a çıkılmasına müsaade edilebilir. İnsanların fizyolojik konforu gözönünde tutularak tespit edilen ve sürekli olarak aynı insanların kullanacağı döşeme yüzeyi sıcaklık değerleri Çizelge 2'de verilmiştir.

NOT : Deneme yanılma işlemine iyi bir tahminle başlamak için aşağıdaki formül kullanılabilir:

$$q_{Yi} \cong 8.92\,(T_p - T_a)^{1.1} \qquad (\text{W/m}^2) \tag{16}$$

NOT : Her mahalde yer alan panel veya panellerin hesap edilen T_p değerleri proje üzerinde her panel için açıkça yazılarak belirtilir. T_p değerinin Çizelge 2'de belirtilen sınırı aştığı durumda, sebepleri (ör: değişik şahısların geçtiği koridor .... gibi) ve varsa alınacak tedbirleri dip not olarak projeye yazılır.

(ΔT) sıcaklık farkı, hesap yapılan mahallin eşdeğer konfor sıcaklığı ile dış hava veya komşu hacımlar sıcaklıkları arasındaki farktır. Isıtılmayan komşu hacimlerin sıcaklıkları TS 2164'den alınır veya TS 2164'e göre hesaplanır. Döşemeden ısıtılan mahallere ait eşdeğer konfor iç hesap sıcaklıkları (T_a) Çizelge 2 'de verilmiştir.

1.1.2. Hava Sızıntısı ve Yenilenmesi ile Meydana Gelen Isı Kayıplarının Hesabı

(Q_L) aşağıdaki formüller yardımı ile hesaplanır:

- Hava sızıntısı ile meydana gelen ısı kaybı:

$$Q_L = \sum_A (a.l) \,.\, R \,.\, H \,.\, Z_E \,.\, \varepsilon_y \,.\, (T_a - T_d) / 860 \quad \text{kW} \tag{4}$$

- Hava yenilenmesi ile meydana gelen ısı kaybı:

$$Q_L = V_L \,.\, C_p \,(T_a - T_d) / 3600 \quad \text{kW} \tag{5}$$

NOT: (C_p)'nin değeri normal şartlarda 0.074 kJ/m^3 ^oC alınır.
NOT : Hava sızıntısı (Denklem 4) ve hava yenilenmesi (Denklem 5) ile meydana gelen ısı kayıpları toplanır.

1.1.3. Toprak Temaslı Döşemelerden Meydana Gelen Isı Kaybının Hesabı

Toprak temaslı döşemelerden ısıtma yapılmıyorsa ısı kaybı (Q_n) TS 2164'e göre yapılır. Toprak temaslı döşemelerden ısıtma yapılıyorsa, ısı kaybı hesabında (Q_A) TS 2164'deki (t_i) değeri T_s değerine eşit alınır.

1.2. Döşemeden Isıtma Sistemlerinin Projelendirilmesi

Döşemeden ısıtma yapılan her mahal için, istenen konfor şartını sağlayacak bir ısıtma sistemi projelendirilir. Her mahal bir veya birden fazla panel ihtiyacı gerektirebilir. Her panel ısıtıcı elemanların meydana getirdiği sıcak sulu veya elektrikli bir devreye sahip olmalıdır. Bu sistemlerle ilgili daha geniş bilgiler TS ...., Şekil 1'de verilmiştir.

Belirli kullanım maksadına haiz ve belirli iç konfor şartı taşıyan kapalı veya yarı kapalı her mahal ayrı ayrı projelendirilir. Bu projelendirmede, Madde 1.1'de açıklanan esaslar dahilinde hesaplanan döşemeden ısıtma yükünün, o mahale yerleştirilecek ısıtma sistemi ve gerekirse destek ısıtıcı(lar) ile karşılanmasına çalışılır.

Kollektörlerden, ısıtılan döşemeye (panele) geliş ve gidiş borularında meydana gelen kaçınılmaz ısı kayıpları, panel devresinin kazan (trafo) ısı yükünün hesaplanmasında (panel devresinin ısı kaybı, Q_k) gözönünde tutulur.

1.2.1. Isıl Konfor Şartı

Bir mahalde ısıl konforun gerçekleşmesi aşağıdaki şartın sağlanmasına bağlıdır:

$$(Q_{Top} + Q_D) \geq Q_h \tag{6}$$

$$Q_{Top} = \sum_{i=1}^{n_p} \frac{q_{Yi} \,.\, A_{pi}}{1000} = \sum_{i=1}^{n_p} Q_{Yi} \quad \text{kW} \tag{7}$$

Eğer bir mahalde birden fazla panel var ise ($n_p > 1$) (6) ve (7) numaralı formülleri sağlamak üzere ve mümkün olduğunca homojen bir ısıtma ve sıcaklık dağılımı olacak şekilde her panelin, döşemede kaplayacağı yüzey alanı ve birim ısıtma kapasitesi bulunur. Bu amaçla q_{Yi} değerleri birbirlerine yakın seçilir:

NOT : $$q_{Yi} = \frac{Q_{Yi}}{A_{pi}} \times 1000 \quad (i : \text{mahaldeki panel numarası, } 1, \ldots.. n_p) \tag{8}$$

0.2.16. Isıtılmayan Döşemelerden olan Isı Kayıpları (Q_n)

Isıtılmayan döşemelerden olan ısı kayıpları, döşemeden ısıtılan bir hacimde ısıtılmayan döşeme kısımlarından olan ısı kaybıdır. Birimi kW 'dır.

0.2.17. Diğer Tarifler

Döşemeden ısıtma sistemleri ile ilgili diğer tarifler TS......... ve TS 2164'de[1] verilmiştir.

0.3. KAPSAM

Bu standard, döşemeden ısıtma sistemlerinin projelendirilmesinde ve ısı yüklerinin hesabında uyulacak tasarım esaslarını kapsar.

0.4. AMAÇ

Bu standardın amacı, döşemeden ısıtılan mahallerde ısıtma yükünün hesaplanmasına ait özel usüllerin belirtilmesi, ayrıca bu sistemlerin projelerinin hazırlanmasında takip edilecek hesap ve tasarım temel esaslarını ve diğer kuralları açıklamaktır.

0.5. UYGULAMA ALANI

Bu standart, en az % 60 oranında ışıma yolu ile bir hacmin ısıtılmasına yönelik döşemeden ısıtma sistemlerinin projelendirilmesinde kullanılır.

1. Özel Kural ve Tasarım Esasları

1.1. Döşemeden Isıtmada Mahal Isı Yükünün (Q_h) Hesabı

Döşemeden ısıtmada mahal ısı yükünün hesaplanması için gerekli malzeme ısı iletim ve taşınım katsayıları TS 388, TS 825 ve TS 2164'den alınır. Ancak TS 2164'de verilen iç yüzey ısı taşınım katsayısı, (α) değerleri 0.75 ile çarpılır.

Panel ısı kayıpları (Q_A) mahal ısı yüküne katılmaz. Bu kayıplar ve varsa panel devresinin ısı kaybı (Q_k) panel devresinin kazan (trafo) ısı yüküne eklenir.

Döşemeden ısıtmada mahal ısı yükü aşağıdaki formül yardımıyla bulunur.

$$Q_h = C \cdot (Q_T + Q_L + Q_n) \qquad \text{kW} \qquad (1)$$

Madde 1.1.1, Madde 1.1.2 ve Madde 1.1.3'e göre yapılan hesaplar sonucu mahalin ısı yükü bulunur. Her mahalin ısı yükü ayrı ayrı bulunarak Çizelge 14 'de verilen föy üzerinde gösterilir. Mahalin ısı iletimi ve taşanımı ile meydana gelen ısı kayıpları çizelgeye işlendikten sonra, aynı çizelge üzerine, Madde 1.1.2 ile bulunan (Q_L) değerleri ile Madde 1.1.3'e göre bulunan (Q_n) ısı kayıpları da, var ise işlenir.

(C) katsayısı Çizelge 1'de verilmiştir. Bu işlemden sonra o mahalin hesabı, yatay bir çizgi ile kapatılır. Çizelgede yer kalmışsa aynı çizelge üzerinde başka bir mahalin hesabına geçilir. Ancak, bu hesap aynı çizelge üzerinde bitirilemeyecek ise o mahal için yeni bir çizelge kullanılır.

1.1.1. Isı İletimi ve Taşınımı ile Meydana Gelen Isı Kaybı (Q_T)'nın Hesabı

(Q_T) aşağıdaki formül hesaplanır.

$$Q_T = (1 + Z_H) \cdot Q_o \qquad \text{kW} \qquad (2)$$

(Q_o) zamsız ısı kaybı, aşağıdaki eşitlikle hesaplanır.

$$Q_o = \sum_{i=1}^{n_y} k_i \cdot A_i \cdot \Delta T_i \qquad \text{kW} \qquad (3)$$

Burada n_y mahaldeki ısı kaybına maruz yüzey sayısıdır.

---

1) Bu standard metninde atıf yapılan standardların numaraları, yayım tarihleri, İngilizce ve Türkçe isimleri kapak arkasında verilmiştir.

0.2.10. Eşdeğer Isı İletim Katsayısı ($k_{eş}$)

Eşdeğer ısı iletim katsayısı, toplam kalınlık (T_{top}) içerisinde yer alan malzemelerin kalınlıkları oranında meydana getirdikleri ve yatay yöndeki ısı iletim katsayısıdır. Birimi W/mK 'dır.
NOT : Döşemenin en üst kaplaması (örtüsü), panel yüzeyini kısmen örtüyor ise, bu kaplamanın (örtünün) eşdeğer kalınlığı, gerçek kalınlığının kapladığı yüzeyin panel alanına oranı ile çarpılarak bulunur.

0.2.11. Kanatçık Verimi (η)

Kanatçık verimi, döşeme içerisine yerleştirilmiş sıcak sulu ısıtıcı elemanlar veya döşeme katmanları arasına serilmiş elektrikli ısıtıcı elemanlar ile döşeme toplam kalınlığının meydana getirdiği varsayılan kanatçığın verimidir (Şekil 1).

0.2.12. Birim Elektrik Direnci (R_L)

Birim elektrik direnci, birim uzunluktaki elektrikli ısıtıcı elemanın, tasarım sıcaklığındaki direncidir. Birimi ohm/m 'dir.

0.2.13. Panel Elektrik Direnci (R_e)

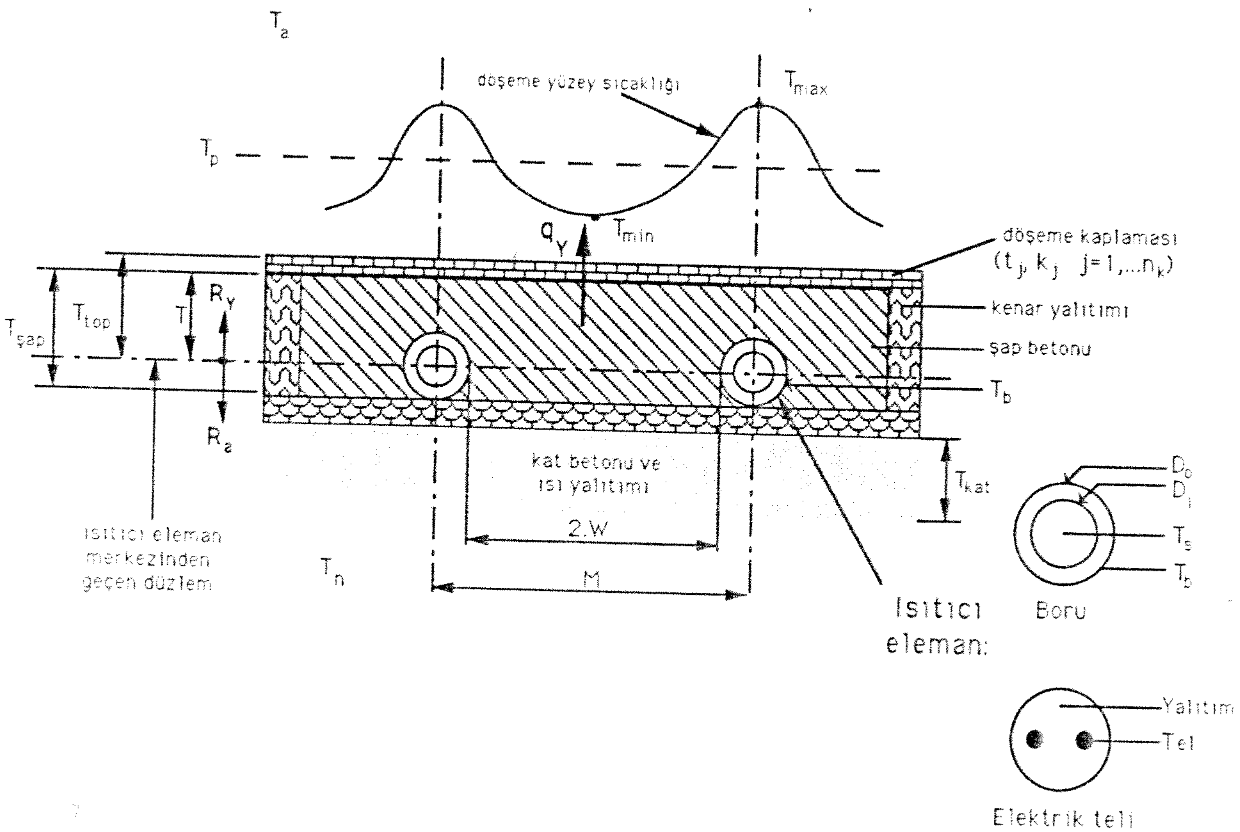
Panel elektrik direnci, elektrikli döşemeden ısıtma sisteminde, bir panelin toplam direncidir. Birimi ohm 'dur.
NOT : $R_e = L_T \cdot R_L$.

0.2.14. Özgül Isıtma Kapasitesi (q_e)

Özgül ısıtma kapasitesi, panel içerisindeki elektrikli ısıtıcı elemanın, tasarım şartlarında, ısıtılan mahale sağlayabildiği en fazla ısı miktarıdır. Birimi W/m^2 'dir.

0.2.15. Mahal Hacmi

Mahal hacmi, bir kapalı mekanı çevreleyen duvar, kapı, pencere, döşeme, tavan ve benzeri elemanların odaya bakan yüzeylerinden başlayarak yapılan ölçümlerle hesaplanan hacimdir. Birimi m^3 'dür.
NOT : Eğer açık bir cephe var ise, bu cepheye ulaşan yan duvar veya benzeri elemanların bitiminden çizilen doğru hat içersinde kalan hacim gözönünde tutulur.



Şekil 1. Döşemeden Isıtma Sisteminde Kanatcık Modeli.

0. KONU, TARİF, KAPSAM

0.1. KONU

Bu standard, döşemeden ısıtma sistemlerinin projelendirilmesinde uyulacak tasarım esaslarına dairdir.

0.2. TARİFLER

0.2.1. Isıtma Yükü Düzeltme Katsayısı (C)(1)

Isıtma yükü düzeltme katsayısı, döşemeden ısıtılan bir yapının, ısıl kütlesi, bulunduğu iklim bölgesi ve işletme rejimine bağlı olarak, yapı elemanlarında belirli bir miktar ısı depolanması sebebi ile, döşemeden ısıtmada mahal ısı yükü tasarım değerinin (Q_h) hesaplanmasında kullanılan bir katsayıdır. (Çizelge 1)

0.2.2. Döşeme Kaplaması (Örtüsü) Toplam Isıl Direnci (R_D)

Döşeme kaplaması toplam ısıl direnci, döşeme üzerinde bulunan bir veya birden fazla sabit kaplama (parke gibi) veya kaldırılabilir örtünün (halı gibi) meydana getirdiği toplam ısıl dirençtir. Birimi m^2K/W 'dır.
NOT : Döşemenin en üst kaplaması panel yüzeyini kısmen örtüyor ise, örtünün ısıl direnci kapladığı yüzey alanının panel alanına oranı ile çarpılır.

0.2.3. Yukarı Yöndeki Isıl Direnç (R_y)

Yukarı yöndeki ısıl direnç, ısıtıcı eleman merkezinden geçen düzlemden daha yukarıda yer alan tesviye betonu, şap gibi döşeme katmanlarının ve kaplama malzemelerinin, ısıl dirençleri ile döşeme yüzeyi ve iç mekan havası arasındaki taşınım (konveksiyon) ısı transfer direncinin toplamıdır. Birimi m^2K/W 'dır.

0.2.4. Aşağı Yöndeki Isıl Direnç (R_a)

Aşağı yöndeki ısıl direnç, ısıtıcı eleman merkezinden geçen düzlemden daha aşağıda yer alan bütün malzemelerin ısıl dirençleri ve alt yüzey (alt kat tavanı) ile komşu hacim havası arasındaki taşınım (konveksiyon) ısı transferi (toprak temaslı döşemelerde eşdeğer taşınım) direncinin toplamıdır. Birimi m^2K/W 'dır.

0.2.5. Şap Kalınlığı (T)

Şap kalınlığı, ısıtıcı eleman merkezi ile şap üst yüzeyi arasındaki mesafedir. Birimi m 'dir.

0.2.6. Toplam Kalınlık (T_{top})

Toplam kalınlık, ısıtıcı eleman merkezi ile en üst döşeme kaplaması yüzeyi arasındaki mesafedir. Birimi m 'dir.
NOT : $T_{top} \geq T$. (Şekil 1)

0.2.7. Toplam Şap Kalınlığı ($T_{şap}$)

Toplam şap kalınlığı, ısıtıcı elemanların içinde bulunduğu şapın alt ve üst yüzeyleri arasındaki mesafedir. Birimi m 'dir.

0.2.8. Kat Betonu Kalınlığı (T_{KAT})

Kat betonu kalınlığı, döşemeden ısıtma yapılan ara katlardaki beton kalınlığıdır. Birimi m 'dir.

0.2.9. Zemin Malzemesi Kalınlığı (T_z)

Zemin malzemesi kalınlığı, toprak temaslı döşemelerde, şapın üzerine döküldüğü malzemenin (zemin toprağı hariç) toplam kalınlığıdır. Birimi m 'dir.

---

(1) Bu standart metninde kullanılan sembollerin anlamları ve birimleri Ek-A 'da verilmiştir.

TÜRK STANDARDLARI

BİRİNCİ		TS
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TASARI

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TÜRK STANDARDLARI ENSTİTÜSÜ
Necatibey Caddesi, 112, Bakanlıklar / ANKARA

Hazırlayan : B. KILKIŞ

0.2.5.7.25 - En Küçük Boru Bükülme Çapı (D_{min})
Bükülebilir en küçük boru çapı, borunun, işletme şartlarında ve kullanım ömrü boyunca mekanik ve termo-mekanik yönden bozulmasına veya sererken kırılmasına, burulmasına, çatlamasına sebep olmayacak en küçük bükülme çapıdır. Birimi m'dir.

NOT - En küçük boru bükülme çapı, boru dış çapının katı olarak da ifade edilebilir.

0.3 - KAPSAM
Bu standard, döşemeden ısıtma sistemlerinde kullanılan terim ve tarifleri kapsar.

ATIF YAPILAN TÜRK STANDARDLARI

TS ..... (Döşemeden Isıtma Sistemleri Projelendirme Kuralları)
TS 2164
ISO 7730

ZA/

0.2.5.7.16 - Döşemeden Isıtma Plakası
Döşemeden ısıtma plakası, döşeme üzerine önceden yayılarak, panelde, belirli demet (düzen) veya demetlerde yerleştirilecek ısıtıcı elemanlar, sabitleyen özel profilli ve bünyesinde ısı, ses, elektrik yalıtım elemanı da bulunabilen modüler plakaların her birisidir.

0.2.5.7.17 - Su (Sıvı) Debisi (V)
Su debisi, tasarım şartlarında, panelin kazan yükünü karşılamaya yeterli ve varsa katkı maddesi veya özel akışkana ve işletme sıcaklığına göre düzeltilerek hesap edilen debidir. Birimi m^3/h'dır.

0.2.5.7.18 - Ortalama Su (Sıvı) Hızı (v_s)
Ortalama su hızı, aşağıdaki formüle göre hesaplanan hızdır. Birimi m/s'dir.

NOT - $$v_s = \frac{4.V}{3600.\pi D_i^2}$$

Burada;
V = Su debisi (m^3/h),
D_i=Boru iç çapıdır (m).

0.2.5.7.19 - Isıtıcı Eleman Uzunluğu (L)
Isıtıcı eleman uzunluğu, bir panel içersinde yer alan ısıtıcı elemanların toplam uzunluğudur. Birimi m'dir.

0.2.5.7.20 - İlave Isıtıcı Eleman Uzunluğu (ΔL)
İlave ısıtıcı eleman uzunluğu, kollektörden panel girişine ve panel çıkışından kollektöre kadar olan ısıtıcı eleman uzunluğudur. Birimi m'dir.

0.2.5.7.21 - Toplam Isıtıcı Eleman Uzunluğu (L_T)
Toplam ısıtıcı eleman uzunluğu, ısıtıcı eleman uzunluğu ile ilave ısıtıcı eleman uzunluğunun toplamıdır. (L_T=L+ΔL). Birimi m'dir.

0.2.5.7.22 - Birim Basınç Kaybı (R)
Birim basınç kaybı, düz borunun birim uzunluğunda işletme şartlarında meydana gelen basınç kaybıdır. Birimi Pa/m'dir.

0.2.5.7.23 - Özel Basınç Kayıpları (Z)
Özel basınç kayıpları, bir panel devresinde kollektör, kollektör ağzı ve varsa ayar vanası ile benzeri noktalarda meydana gelen ilave basınç kayıplarının toplamıdır. Boru kavislerindeki özel basınç kayıplarıda gözönüne alınarak belirlendiği değerdir. Birimi Pa'dır.

0.2.5.7.24 - Toplam Basınç Kaybı (H)
Toplam basınç kaybı, bir panel devresindeki birim basınç kaybının toplam ısıtıcı eleman (boru) uzunluğu çarpımı ile özel basınç kayıplarının toplamıdır ($H=R.L_T+Z$) Birimi Pa'dır.

0.2.5.7.7 - Isıtıcı Eleman Merkezleri Arasındaki Mesafe (Modül) (M)
Isıtıcı eleman merkezleri arasındaki mesafe (modül), panel içerisinde birbirine paralel konumdaki ısıtıcı elemanların merkezleri arasındaki mesafedir. Birimi mm'dir.

NOT - Bir panelde bu mesafe sabit değilse, belirli bir merkez arası mesafeye haiz her panel kısmının kapladığı yüzey alanı (A_1, A_2, ...) gözönünde tutularak ısıtma kapasitesi eşdeğeri ortalama bir değer hesaplanır.

NOT - Ortalama $M = \frac{A_p}{A_1/M_1 + A_2/M_2 + \ldots}$

0.2.5.7.8 - Isıtıcı Eleman Açıklığı (2W)
Isıtıcı eleman açıklığı, komşu iki ısıtıcı eleman arasında kalan net mesafedir. Birimi m'dir.

NOT - $2W = (M/1000 - D_o)$

0.2.5.7.9 - Isıtıcı Eleman Dış Çapı (D_o)
Isıtıcı eleman dış çapı, sıcak sulu döşemeden ısıtma sisteminde borunun işletme basıncında ve sıcaklığında ölçülen dış çapıdır. Birimi m'dir.

NOT - Elektrikli döşemeden ısıtma sisteminde elektrik, direncinin var ise yalıtım elemanı dahil olmak üzere ve işletme sıcaklığında ölçülen dış çapıdır. Birimi m'dir.

0.2.5.7.10 - Isıtıcı Eleman Çapı (D_i)
Isıtıcı eleman iç çapı, sıcak sulu döşemeden ısıtma sisteminde borunun işletme basıncında ve sıcaklığında ölçülen iç çapıdır. Birimi m'dir.

NOT - Bu şartlarda ölçme yapılamadığı takdirde işletme basıncı 1 atü alınabilir.

0.2.5.7.11 - Borunun Oksijen Geçirim Katsayısı
Borunun oksijen geçirim katsayısı, 1 m uzunluğundaki borudan standard atmosfer şartlarında, içinden akan suya bir gün zarfında geçen saf oksijen kütle miktarıdır. Birimi g/m^3.gün'dür.

NOT - Hacim hesabında, boru iç çapı D_i ve 1 m uzunluk esas alınır.

0.2.5.7.12 - Oksijen Bariyeri
Oksijen bariyeri, borunun oksijen geçirimini azaltmayı amaçlayan ve et kalınlığı içerisinde, boru boyunca uzanan özel malzeme zarfıdır.

0.2.5.7.13 - Boru Sıcaklık Dayanım Sınırı (T_{smaks})
Boru sıcaklık dayanım sınırı, sürekli rejimde, tasarım şartlarında ve işletme basıncında, borunun en az 30 işletme mevsimi (yıl) çalışabileceği en yüksek sıcaklıktır. Birimi °C'dır.

0.2.5.7.14 - Boru Demeti (Yerleşim Düzeni, Boru Modülasyonu)
Boru demeti, bir panel içerisindeki boruların yerleştirme düzenidir.

0.2.5.7.15 - Boru Tespit Kelepçesi
Boru tespit kelepçesi, borunun panel içerisinde belirli bir düzende yerleştirilmesini sağlayan ve boruyu sabitleyen elemandır.

0.2.5.5 - Panel Devresi
Panel devresi, paneldeki su (akışkan) veya elektrik devresidir.

NOT - Bu devre sıcak sulu sistemlerde kollektörlere, elektrikli sistemlerde elektrik panosuna kadar olan bölümü de ihtiva eder (Şekil-1).

0.2.5.6 - Malzeme Sıcaklık Dayanım Sınırı
Malzeme sıcaklık dayanım sınırı, döşeme veya kaplama malzemelerinin sürekli rejimde ve tasarım şartlarında, en az 30 ısıtma mevsimi (yıl) dayanabileceği sıcaklık sınırıdır. Birimi °C'dır.

0.2.5.7 - Sıcak Sulu Döşemeden Isıtma Sistemi Değişkenleri

0.2.5.7.1 - Su Giriş Sıcaklığı ($T_{giriş}$)
Su giriş sıcaklığı, panel devresinde suyun (sıvının) panel'e girdiği noktadaki sıcaklığıdır. Birimi °C'dır.

0.2.5.7.2 - Su Çıkış Sıcaklığı ($T_{çıkış}$)
Su çıkış sıcaklığı, panel devresinde suyun panelden çıktığı noktadaki sıcaklığıdır. Birimi °C'dır.

0.2.5.7.3 - Su Sıcaklık Farkı (ΔT_s)
Su sıcaklık farkı, su giriş ve su çıkış sıcaklıklarının farkıdır. Birimi °C'dır.

NOT - (ΔT_s:$T_{giriş}$-$T_{çıkış}$)

0.2.5.7.4 - Ortalama Su Sıcaklığı (T_s)
Ortalam su sıcaklığı, su giriş ve su çıkış sıcaklığının aritmetik ortalamasıdır.

Birimi °C'dır.

$$(T_s = \frac{T_{giriş}+T_{çıkış}}{2}).$$

0.2.5.7.5 - Kollektör
Kollektör, birden fazla penel devresinin giriş veya çıkış uçlarının bağlandığı elemandır.

0.2.5.7.6 - Kollektör Dolabı
Kollektör dolabı, kollektörlerin ve varsa ölçme, ayar ve kontrol elemanlarının muhafaza edildiği ve erişilebilir konumdaki kapalı hacimdir.

0.2.5.7.7 - Kollektör Ana Vanaları
Kollektör ana vanaları, giriş ve çıkış kollektörlerine suyun (sıvının) girmesini veya çıkmasını kontrol eden vanalardır.

0.2.5.7.8 - Panel Vanaları
Panel vanalar, kollektöre bağlı panel devrelerindeki su (sıvı) debisini ayarlayan veya su (sıvı) akışını kesen vanalardır.

).2.4.34 - Komşu Kat Tasarım Sıcaklığı (T_n)
:omşu kat tasarım sıcaklığı, döşeme altındaki mahalin iç hesap sıcaklığıdır. Birimi C'dır.

).2.4.35 - İç sıcaklık Farkı (ΔT_a)
ç sıcaklık farkı, mahal içerisinde düşey bir çizgi·üzerinde döşemeden 5cm yukarıda 'e tavandan 5 cm aşağıda ve tasarım şartlarında ölçülen sıcaklıklar arasındaki 'arkın, mahal içerisindeki en büyük değeridir. Birimi °C'dır.

).2.4.36 - Isıtıcı Eleman Dış Sıcaklığı (T_b)
sıtıcı eleman dış sıcaklığı, tasarım şartlarında, ısıtıcı eleman dış yüzeyindeki ıcaklıktır. Birimi °C'dır.

).2.4.37 - Hava hızı (V_h)
Iava hızı, bir mahalde tasarım şartlarında deneyle ölçülen veya sayısal metotlarla ıesaplanan ve söz konusu mahalde meydana gelen en yüksek hava hızıdır. Birimi ı/s'dir.

).2.5 - Döşemeden Isıtma Sistemi Değişkenleri
)öşemeden ısıtma sistemi değişkenleri, bir mahalde bulunan döşemeden ısıtma .isteminin tasarım şartlarını ve tasarım şartlarındaki durumunu ve işleyişini tarif :den, değerlendiren ve belirleyen değişkenlerin her birisidir.

).2.5.1 - Kontrol Prensibi
:ontrol prensibi, bir mahaldeki bütün panel devrelerini birlikte veya ayrı ayrı, .eğişen ısıtma yüküne göre ayarlayarak, konfor şartını yerine getiren kontrol isteminin dayandığı ana prensiptir.

.2.5.1.1 - Su Sıcaklığı Ayarı İle Kontrol
u sıcaklığı ayarı ile kontrol, sıcak su ile döşemeden ısıtma sisteminde su ıcaklığının değiştirilmesine dayanan kontrol prensibidir.

.2.5.1.2 - Açık-Kapalı (On-Off) Kontrol
çık-kapalı kontrol, sistemin fasılalı olarak çalıştırılmasına dayanan kontrol rensibidir.

.2.5.1.3 - Akım Şiddeti İle Kontrol
kım şiddeti ile kontrol, elektrikle döşemeden ısıtma sisteminde, akım şiddetinin eğiştirilmesine dayanan kontrol prensibidir.

.2.5.2 - Döşeme Katmanları (Tabakaları)
öşeme katmanları, ısıtma sistemini havi döşemeyi meydana getiren katmanların her irisidir.

.2.5.3 - Döşeme Kaplaması (Örtüsü)
öşeme kaplaması, döşeme üzerine serilen veya inşa edilen kaplamaların her irisidir.

.2.5.4 - Rejime Giriş Süresi
ejime giriş süresi, ısıtma sisteminin devreye alınmasından sonra, konfor şartını am olarak sağlayana kadar geçen süredir. Birimi saattır.

0.2.4.26 - Panel Devresinin Isı Kaybı (Q_K)
Panel devresinin ısı kaybı, panel ile ısıtıcı elemanların bulunduğu kollektör arasındaki parkurda ısıtıcı elemanların diğer mahallerden geçtiği durumlarda, bu gibi mahallerde meydana gelen birim zamandaki ısı kayıplarının toplamıdır. Birimi kW'dır.

0.2.4.27 - Tasarım Konfor Şartı
Tasarım konfor şartı, döşemeden ısıtmada mahal ısı yükü ile panel devresi ısı kaybının, o mahalde mevcut panellerin toplam ısıtma kapasitesi ve varsa destek ısıtma kapasitesinin toplamı ile karşılanabildiği durumdur:

NOT - $(Q_{Top}+Q_D) \geq (Q_h+Q_n)$

NOT - Q_n aynı mahal içersinde ısıtılmayan döşeme kısımlarından (A_y) olan ısı kaybıdır.

0.2.4.28 - Işıma İle Isıtma Oranı (I)
Işıma ile ısıtma oranı, bir mahalde ışıma yolu ile gerçekleşen ısıtma kapasitesinin toplam ısıtma kapasitesine oranıdır. Bu oran döşemeden ısıtılan bir mahalde en az %60 olmalıdır.

NOT - Mahalde destek ısıtıcı var ise destek ısıtma kapasitesi de hesaplara dahil edilir:

NOT - $$I = \frac{\sum_{i=1}^{n_p} \frac{q_{ri} \cdot A_{pi}}{1000} + Q_D \cdot 0,2}{Q_{Top}+Q_D \cdot 0,8}$$

NOT - Burada 0,8 ve 0,2 sayıları destek ısıtıcının sırası ile ısıma ve taşınım yolu ile ısıtma oranlarına ait yaklaşık değeridir.

0.2.4.29 - Ortalama İç Yüzey Sıcaklığı (T_K)
Ortalama iç yüzey sıcaklığı, döşemeden ısıtma yapılan mahali çevreleyen ve ısıtma yapılmayan bütün iç yüzeylerin alanları dikkate alınarak hesaplanan ortalama yüzey sıcaklık değeridir. Birimi °C'dir.

0.2.4.30 - Döşeme Yüzey Sıcaklığı (T_p)
Döşeme yüzey sıcaklığı, ısıtılan döşeme yüzeyinde sabit bulunduğu farz olunan sıcaklık olup, panelin birim ısıtma kapasitesinin hesabında kullanılır.

Birimi °C'dır.

NOT - Bu sıcaklık TS .....'de belirtilen ve oda fonksiyonuna bağlı sınırı aşamaz.

0.2.4.31 - En Yüksek Döşeme Sıcaklığı (T_{maks})
En yüksek döşeme sıcaklığı, tasarım şartlarında ısıtılan döşeme yüzeyinde, mevzii olarak yer alan en yüksek döşeme sıcaklığıdır. Birimi °C'dır.

0.2.4.32 - En Düşük Döşeme Sıcaklığı (T_{min})
En düşük döşeme sıcaklığı, tasarım şartlarında ısıtılan döşeme yüzeyinde, mevzii olarak gerçekleşen en düşük döşeme sıcaklığıdır. Birimi °C'dır.

0.2.4.33 - Yüzey Sıcaklığı Fark Katsayısı (t_v)
Yüzey sıcaklığı fark katsayısı, ısıtılan döşeme yüzeyindeki en yüksek ve en düşük döşeme sıcaklıkları farkının, döşeme yüzey sıcaklığına oranıdır.
$t_v=(T_{maks}-T_{min})/T_p$

0.2.4.17 - Panelin Birim Isıtma Kapasitesi (q_y)
Panelin birim ısıtma kapasitesi, bir panelin bulunduğu mahal içerisine tasarım şartlarında, sağlayabildiği toplam ısı akısı olup, taşınım ve ışıma ile birim ısıtma kapasiteleri toplamına eşittir. Birimi W/m²'dir.

$$q_y = q_c + q_r$$

0.2.4.18 - Panelin Isıtma Kapasitesi (Q_y)
Panelin ısıtma kapasitesi, bir panelin yüzey alanından, bulunduğu mahal içerisine tasarım şartlarında ve birim zamanda sağladığı en fazla ısı miktarıdır. Birimi kW'dır.

$$Q_y = q_y \times A_p/1000$$

0.2.4.19 - Panelin Isı Kaybı (Q_A)
Panelin ısı kaybı, bir panelin bulunduğu mahal içerisine sağladığı faydalı ısı dışında çevreye birim zamanda kaçan ısı miktarıdır. Birimi kW'dir.

0.2.4.20 - Panelin Kazan (Trafo) Yükü (Q_B)
Panelin kazan (trafo) yükü, bir panelin ısıtma kapasitesi ile ısı kaybının toplamıdır. Birimi kW'dır.

$$Q_B = Q_y + Q_A$$

0.2.4.21 - Panel Isıtma Verimi (X)
Panel ısıtma verimi, bir panelin, ısıtma kapasitesinin panelin kazan (trafo) yüküne oranıdır.

$$X = Q_y / Q_B$$

0.2.4.22 - Toplam Panel Yüzey Alanı (A_{pt})
Toplam panel yüzey alanı, bir mahaldeki panellerin meydana getirdiği toplam yüzey alanıdır. Birimi m²'dir.

0.2.4.23 - Toplam Isıtma Kapasitesi (Q_{Top})
Toplam ısıtma kapasitesi, o mahaldeki bütün panellerin ısıtma kapasitelerinin toplamıdır. Birimi kW'dır:

$$\text{NOT} - Q_{Top} = \sum_{i=1}^{n_p} \frac{q_{yi} \cdot A_{pi}}{1000} = \sum_{i=1}^{n_p} Q_{yi}$$

0.2.4.24 - Destek Isıtma Kapasitesi (Q_D)
Destek ısıtma kapasitesi, bir mahalde, kurulması gereken münferit veya merkezi sisteme bağlı destek ısıtıcı veya ısıtıcıların toplam ısıtma kapasitesidir. Birimi kW'dır.

0.2.4.25 - Birim Destek Isıtma Kapasitesi (q_D)
Birim destek ısıtma kapasitesi, destek ısıtma kapasitesinin o mahaldeki kullanıma müsait toplam panel yüzey alanına bölümüdür. Birimi W/m²'dır.

$D_{eş}:(4.A_r)/L_r$

Burada;

L_r=Mahal çevre uzunluğudur (m)

A_r=Mahal net alanıdır (m^2)

0.2.4.9 - Mahal Çevre Uzunluğu (L_r)

Mahal çevre uzunluğu, o mahali çevreleyen duvar veya benzeri elemanların mahale bakan yüzeylerinden yapılan ölçümle bulunan çevre uzunluğudur. Birimi m'dir.

NOT - Bir mahalde içe veya dışa açık cephe var ise; hesap sırasında, bu cepheye ulaşan yan duvarların bitiminden çizilen doğru hat gözönünde tutulur.

0.2.4.10 - Döşemeden Isıtma Yapılamayan Alan (A_y)

Döşemeden ısıtma yapılamayan alan, üzerinde sabit eşya, sıcaklığa hassas malzeme veya cihaz bulunması veya başka sebeplerle o mahalde döşemeden ısıtma yapılamayacak döşeme bölümlerinin toplam alanıdır. Birimi m^2'dir.

0.2.4.11 - Döşemeden Isıtmada Kullanılabilecek Net Alan (A_{pr})

Döşemeden ısıtmada kullanılabilecek net alan, o mahalde etkin ve verimli bir şekilde döşemeden ısıtma yapılabilecek döşeme yüzeyinin toplam alanıdır. Bu miktar, döşemeden ısıtma yapılamayacak yüzeyler düşüldükten sonraki mahal net alanına eşittir. Birimi m^2'dir.

$A_{pr}=A_r-A_y$

0.2.4.12 - Mahalin Birim Isıtma Yükü (q_{yr})

Mahalin birim ısıtma yükü, mahal ısı yükünün kullanıma müsait toplam panel yüzey alanına bölümü ile elde edilen ısıtma yüküdür. Birimi W/m^2'dir.

$$q_{yr}=\frac{Q_h}{A_{pr}} \times 1000$$

0.2.4.13 - Panel (Isıtma Yüzeyi) Sayısı (n_p)

Panel (ısıtma yüzeyi) sayısı, ısıtılan mahalde bulunan panellerin sayısıdır.

0.2.4.14 - Panel Yüzey Alanı (A_p)

Panel yüzey alanı, ısıtıcı elemanların kapladığı yüzeyin her kenarına ısıtıcı eleman merkezleri arası mesafenin (modülasyon) yarısı eklenmek suretiyle hesaplanan, bir panelin döşeme yüzeyinde etkili olduğu alandır. Birimi m^2'dir.

0.2.4.15 - Taşınım İle Isıtma Birim Kapasitesi (q_c)

Taşınım ile ısıtma birim kapasitesi, bir panelin, bulunduğu mahal içerisine, tasarım şartlarında, taşınım yolu ile sağlayabildiği en fazla ısı akısıdır. Birimi W/m^2'dir.

NOT - Bu kapasite mahal büyüklüğü ve yükseklik düzeltme katsayıları, F_1 ve F_2 ile düzeltilir.

0.2.4.16 - Işıma (Radyasyon) İle Isıtma Birim Kapasitesi (q_r)

Işıma ile ısıtma birim kapasitesi, bir panelin, bulunduğu mahal içersine, tasarım şartlarında, ışıma yolu ile sağlayabildiği en fazla ısı akısıdır. Birimi W/m^2'dir.

0.2.4.1 - Dış Hesap Sıcaklığı (T_d)
Dış hesap sıcaklığı, TS 2164[1]'de belirtilen ve mahal ısı yükünün hesabında kullanılan ve yapının bulunduğu yörenin ısıtma dönemine ait dış sıcaklık değeridir.

Birimi °C'dir.

0.2.4.2 - İç Hesap Sıcaklığı (T_i)
İç hesap sıcaklığı, döşemeden ısıtma sistemi dışında kalan, diğer merkezi veya münferit ısıtma sistemlerinin tasarım veya seçiminde kullanılan ve TS 2164'de belirtilen, mahal fonksiyonuna bağlı, ısıtma dönemine ilişkin iç sıcaklık değeridir. Birimi °C'dır.

0.2.4.3 - Eşdeğer Konfor İç Hesap Sıcaklığı (T_a)
Eşdeğer konfor iç hesap sıcaklığı, diğer ısıtma sistemleri için tasarımlanan iç konfor şartlarını aynen sağlaya ve ISO 7730'a uygun, deneylerle veya sayısal benzetim metotları ile tesbit olunan ve sadece döşemeden ısıtma sistemlerinin tasarımında kullanılacak olan iç sıcaklık değeridir. Birimi °C'dır.

NOT - Mahal fonksiyonuna bağlı olarak tesbit olunan bu sıcaklıklar TS ...'de verilmiş olup, genelde T_i değerlerinden daha düşüktür.

0.2.4.4 - Döşemeden Isıtmada Mahal Isı Yükü (Q_h)
Döşemeden ısıtmada mahal ısı yükü, TS .....'de belirtilen usullle hesaplanan ve sadece döşemeden ısıtma sistemlerinin tasarımında kullanılabilen ısı yüküdür. Bu yüke ısıtılan döşemelerdeki ısı kayıpları dahil edilmez. Birimi kW'dır.

0.2.4.5 - Mahal Net Alanı (A_r)
Mahal net alanı, mahali çevreleyen duvar ve benzeri elemanların iç yüzeylerinden başlayarak yapılan ölçümlerle hesaplanan net toplam döşeme alanıdır. Birimi m^2'dir.

NOT - Bir mahalde, açık cephe varsa, alan hesabında bu cepheye ulaşan yan duvarların bitiminden çizilen doğru hat içerisinde kalan alan gözönünde tutulur.

0.2.4.6 - Deniz Kotuna Göre Düzeltme Katsayısı (F_1)
Deniz kotuna göre düzeltme katsayısı, ısıtılan mahalde tabii taşınım yolu ile döşemeden gerçekleşen ısıtma kapasitesini, yapının denizden olan yüksekliğine bağlı olarak düzelten katsayıdır ($F_1 \leq 1$).

0.2.4.7 - Mahal Büyüklüğü Düzeltme Katsayısı (F_2)
Mahal büyüklüğü düzeltme katsayısı, ısıtılan mahalde tabii taşınım yolu ile döşemeden gerçekleşen ısıtma kapasitesini mahalin biçim ve büyüklüğüne bağlı olarak düzelten katsayıdır ($F_2 \leq 1$).

0.2.4.8 - Mahal Eşdeğer Çapı ($D_{eş}$)
Mahal eşdeğer çapı, mahal biçim ve büyüklüğünün bir ifadesi olup, aşağıdaki formül ile hesaplanır.

---

1) Bu standard metninde atıf yapılan standardların numaraları, yayım tarihleri, İngilizce ve Türkçe isimleri kapak arkasında verilmiştir.

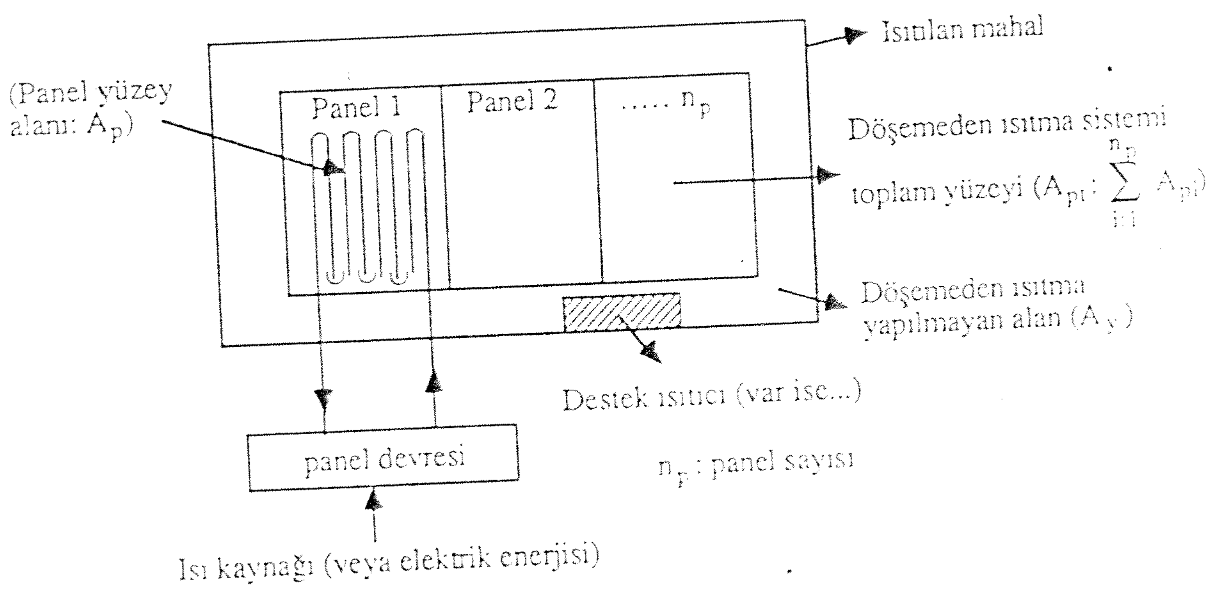
0.2.2.6.3 - Destek Sistem Isı Üretici Elemanı
Destek sistem ısı üretici elemanı, destek ısıtma sistemine enerji sağlayan sıcak sulu, sıcak havalı veya elektrikli, münferit veya merkezi ısı üreticisidir.

0.2.2.7 - Hissedici Eleman
Hissedici eleman, gerekli iç ve dış mekan değişkenlerini, diğer elemanların işleyişini, mahallin konfor açısından davranışını ve tesis ile sistemin emniyeti için ölçülmesi lüzumlu değişkenleri hisseden ve gerekli kontrol, düzenleme ve kaydetme görevi üstlenmiş cihazlara ileten elemanların her birisidir.

0.2.2.8 - Kontrol Elemanı
Kontrol elemanı, döşemeden ısıtma sisteminin önceden tespit edilen veya işletme sırasında yeniden ayarlanan sınırlar içersinde çalışmasını temin etmek üzere sistemi kontrol eden ve düzenleyen elemandır.

0.2.2.9 - Emniyet Elemanı
Emniyet elemanı, lüzumlu emniyet tedbirlerinin alınmasını, kontrolünü ve düzenlemesini, gerekirse düzeltilmesini sağlayan veya uyaran elemandır.

0.2.3 - Panel (Isıtma Yüzeyi)
Panel, aynı mahalin ısıtılmasında kullanılan, ancak herbirinin kendisine ait sıcak su veya elektrik devresi ve ayar tertibatı bulunabilen ısıtma yüzeyleridir (Şekil-1).



Şekil 1.

0.2.4 - Isıtılan Mahal Değişkenleri
Isıtılan mahal değişkenleri, o mahallin tasarım şartlarını ve tasarım şartlarındaki durumunu ve işleyişini tarif eden, değerlendiren ve belirleyen değişkenlerin her birisidir.

).2.1.2 - Döşemeden Isıtma Tesisi
)öşemeden ısıtma tesisi, döşemeden ısıtma sisteminin işletilmesi ve kontrolü için ;erekli olan bütün elemanlardan meydana gelen tesistir.

).2.2 - Elemanlar

).2.2.1 - Isıtıcı Elemanlar

).2.2.1.1 - Sıcak Sulu Isıtıcı Eleman (Boru)
;ıcak sulu ısıtıcı eleman, döşemeden ısıtma sisteminde gerekli ısı transferini ;ağlamak maksadı ile, içinde sıcak su (sıvı) dolaşan borudur.

).2.2.1.2 - Elektrikli Isıtıcı Eleman
]lektrikli ısıtıcı eleman, döşemeden ısıtma sisteminde gerekli ısı transferini ;ağlamak maksadı ile, üzerinden geçen elektrik akımını ısı enerjisine dönüştüren :ezistans elemanıdır.

).2.2.2 - Yalıtım Elemanı
'alıtım elemanı, sıcak su ile döşemeden ısıtma sistemlerinde su ve nem yalıtımı, ɜlektrikli döşemeden ısıtma sistemlerinde ise elektrik akımı ve gerektiğinde nem, ;tatik elektrik, elektromanyetik dalga yalıtımı için kullanılan elemandır.

).2.2.3 - Isı Yalıtım Elemanı
:sı yalıtım elemanı, döşemeden ısıtma sisteminde, ısıtılacak ortam dışına)labilecek ısı kayıplarını azaltmak amacı ile kullanılan elemandır.

).2.2.4 - Hidrolik Eleman
Hidrolik eleman, sıcak su ile döşemeden ısıtma tesisinde, sıcak sulu ısıtıcı ɜlamanlar haricinde dolaşımı sağlayan, düzenleyen, depolayan, havayı tahliye eden, ;u takviyesi veya boşaltması yapan elemanların her birisidir.

).2.2.5 - Elektro-Mekanik Eleman
]lektro-mekanik eleman, bir döşemeden ısıtma tesisinde malzeme taşıyıcı (katı yakıt :onveyörü gibi), tahrik edici (elektrik motoru gibi), tahliye edici (aspiratör ;ibi), kumanda edici (vana motoru gibi) veya elektrik akımını düzenleyici, ;ağlayıcı (trafo, kumanda panosu gibi) gibi görevleri üstlenen, sabit veya ıareketli, elektriksel, mekanik veya elektromekanik elemanların her birisidir.

).2.2.6 - Enerji Temin Elemanları

).2.2.6.1 - Sıcak Su Üreticisi
;ıcak su üreticisi, sıcak su ile döşemeden ısıtma sisteminde suyun ısıtılmasını ;ağlayan ve herhangi bir birincil enerji kaynağını kullanan (kazan, kombi şofben ;ibi) veya herhang bir alternatif enerji kaynağını kullanan (güneş kollektörü, ısı ›ompası, atık ısı eşanjörü gibi) eleman veya elemanlar topluluğu ile bunların :ipmanlarından meydana gelen cihazlardır.

·.2.2.6.2 - Kuvvet Merkezi (Trafo)
.uvvet merkezi, elektrik ile döşemeden ısıtma sistemine gerekli elektrik enerjisini ağlayan ve bu sistem için aynı mahalde kurulan elektrik trafosu veya kumanda ablosudur.

DÖŞEMEDEN ISITMA SİSTEMLERİ - TERİMLER VE TARİFLER

0 - KONU, TARİF, KAPSAM

0.1 - KONU

Bu standard, döşemeden ısıtma sistemlerinde kullanılan terim ve tariflere dairdir.

0.2 - TARİFLER

0.2.1 - Sistemler ve Tesis

0.2.1.1 - Döşemeden Isıtma Sistemi

Döşemeden ısıtma sistemi, döşeme içerisine yerleştirilmiş ısıtıcı elemanlar aracılığı ile bir mahalin ısıtılmasını sağlayan ısıtma sistemidir.

NOT - Döşemeden ısıtma sistemi, bazı durumlarda duvar veya tavana da uygulanabilir

0.2.1.1.1 - Sıcak Su İle Döşemeden Isıtma Sistemi

Sıcak su ile döşemeden ısıtma sistemi, döşeme içerisine yerleştirilmiş borular içerisinde sıcak su dolaştırılması sureti ile döşemeden ısıtmayı sağlayan sistemdir.

NOT - Bu tip sistemlerde sistemde dolaşan suya özel amaçlı katkı maddeleri ilave edilebildiği gibi bazı durumlarda da sıcak su yerine özel bir ısı transfer sıvısı kullanılabilir.

0.2.1.1.2 - Elektrik İle Döşemeden Isıtma Sistemi

Elektrik ile döşemeden ısıtma sistemi, döşeme malzemesi içerisine veya döşeme örtü katmanları içine veya arasına yerleştirilmiş, elektrik ve elektromanyetik yalıtımlı rezistanslardan elektrik akımı geçirilmek sureti ile döşemeden ısıtmayı sağlayan sistemdir.

0.2.1.1.3 - Çift Maksatlı Sistem

Çift maksatlı sistem, sıcak su ile döşemeden ısıtma sisteminde, ısıtmanın yanısıra mevsimine göre soğutma da yapılabilen sistemdir.

0.2.1.1.4 - Tek Maksatlı Sistem

Tek maksatlı sistem, sadece ısıtma fonksiyonu bulunan döşemeden ısıtma sistemidir.

0.2.1.1.5 - Destek Sistem

Destek sistem, döşemeden ısıtma sisteminin tasarım kısıtlamalarından dolayı yeterli olmadığı durumlarda münferit veya merkezi olarak, ısıtılan mekana (mekanlara) destek ısı enerjisi sağlayan sistemin bütünüdür. Bu sistem döşemeden ısıtma sistemine bağlı veya bağımsız olabilir.

0.2.1.1.6 - Destek Isıtıcı

Destek ısıtıcı, destek sistemin ısıtıcı eleman(lar)ıdır.

T Ü R K S T A N D A R D L A R I

B İ R İ N C İ		TS
B A S K I		UDK 621.577

DÖŞEMEDEN ISITMA SİSTEMLERİ-TERİMLER VE TARİFLERİ

FLOOR HEATING SYSTEMS - TERMINOLOGY AND DEFINITIONS

TASARI

TEKNİK KURUL

671g

TÜRK STANDARDLARI ENSTİTÜSÜ
Necatibey Caddesi, 112 Bakanlıklar/ANKARA

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EK-5

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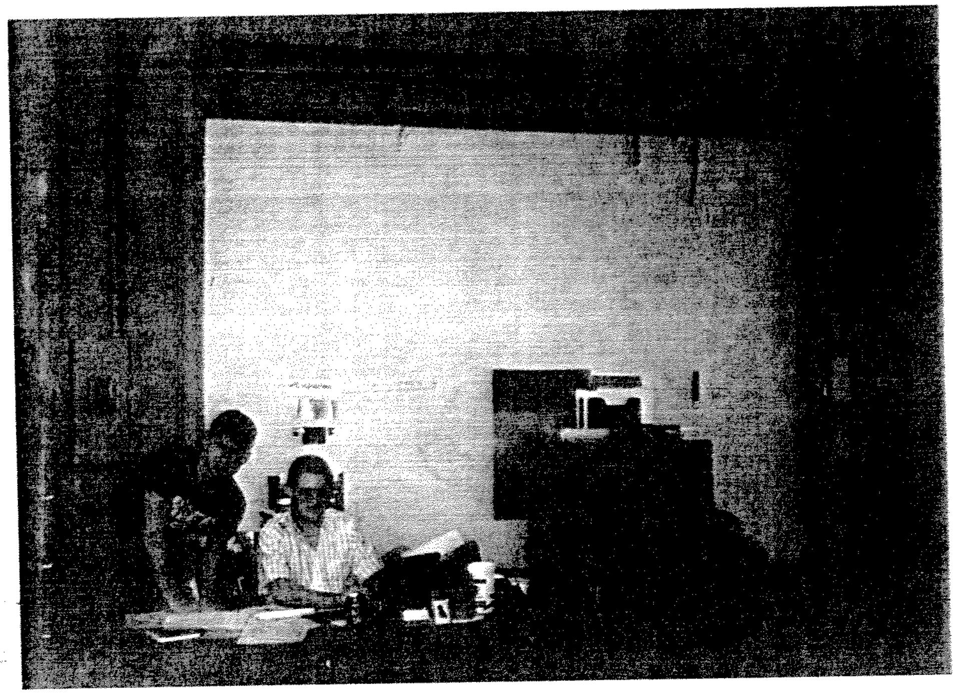
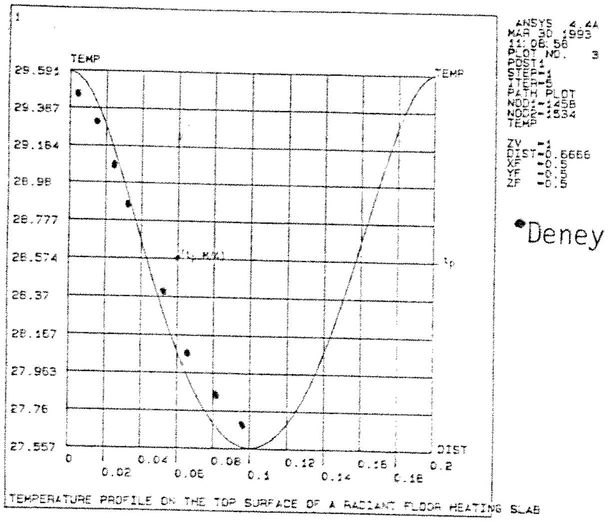
* Teknik Kurula sunuldu,TS numarası Ekim ayında verilecek.

Şekil 4-8 de ise tavana monteli duvar paneli ve proje çalışanları görülmektedir.



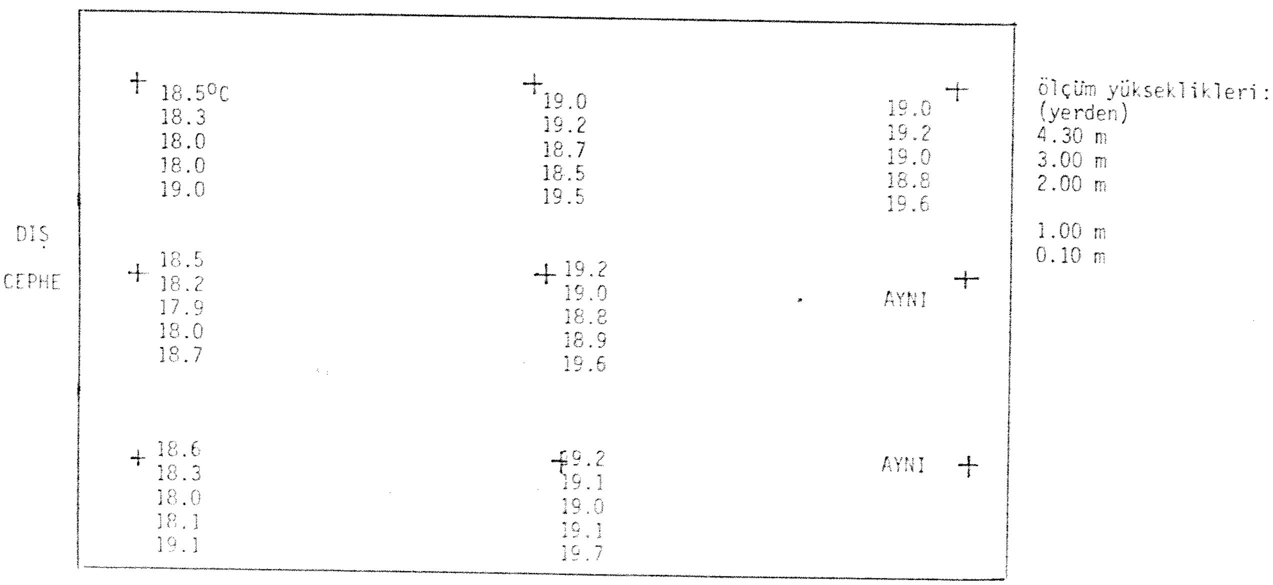
Şekil 4-8 Duvar Paneli

Geliştirilen analitik modelin sayısal doğrulanmasına ilaveten panel yüzey sıcaklık dağılımı ölçülmüştür.
Örnek Şekil 4-6 da verilmiştir.

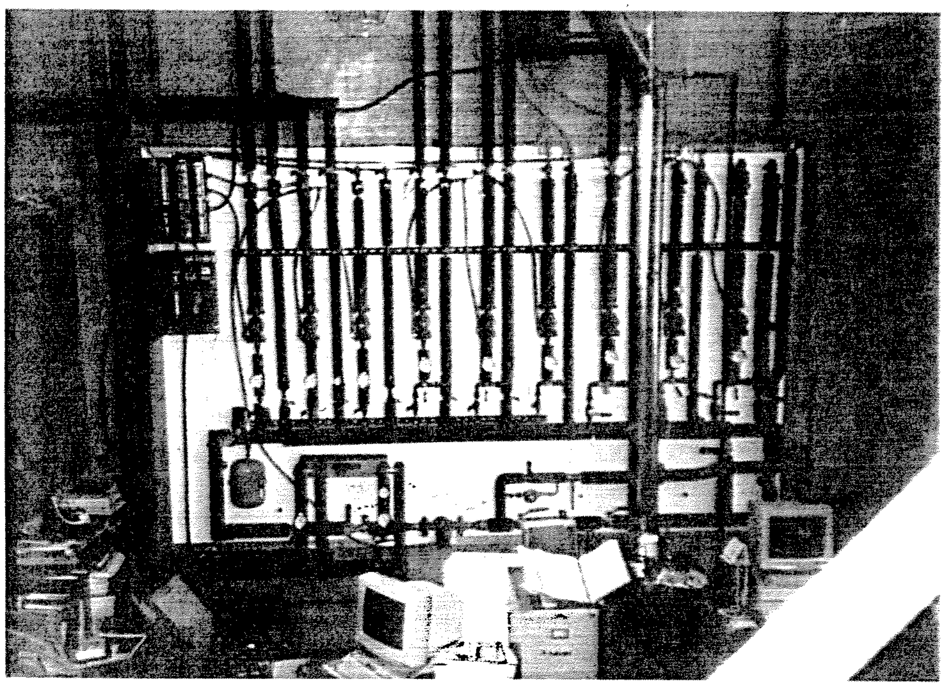


Şekil 4-6 Panel Yüzeyinde Sıcaklık Dağılımı

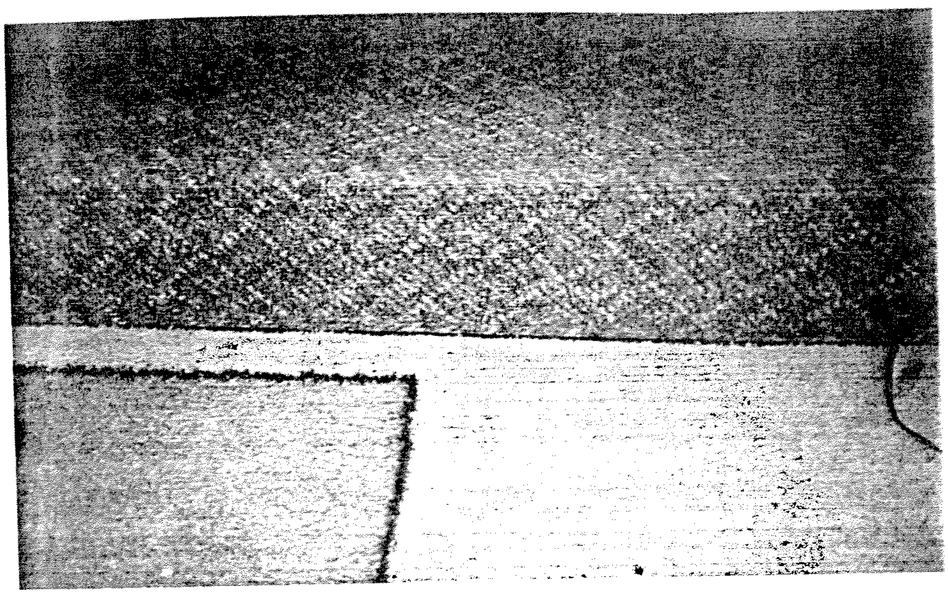
Oda içersindeki tipik sıcaklık dağılımı Şekil 4-7 verilmiştir.



Şekil 4-7 Odada Tipik Sıcaklık Dağılımı

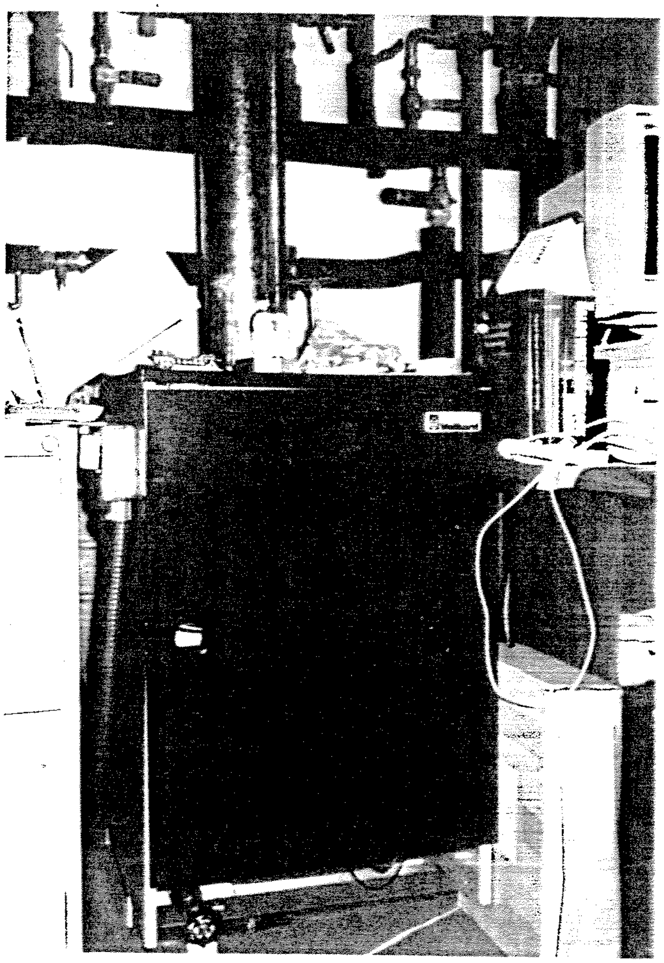


Şekil-4-4 Hidrolik Kontrol Paneli.

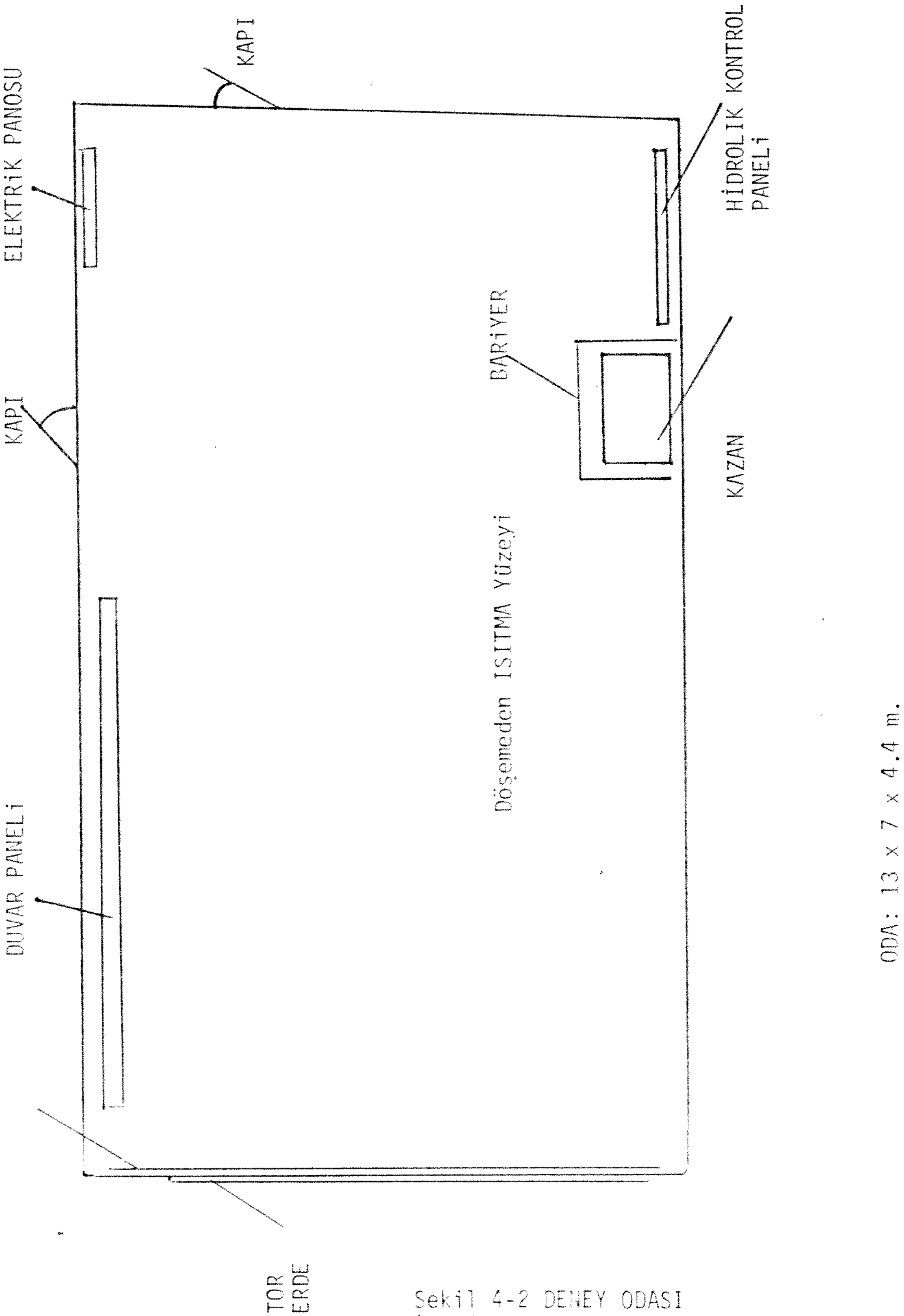


Şekil-4-5 Döşeme Paneli.

Şekil 4-3 de kazan,Şekil 4-4 de ise hidrolik kontrol paneli görülmektedir. Panelde bir genleşme tankı(Watts Model:ET15),her devrenin kendi sekonder sirkülasyon pompası (GRUNDFOSS Model UP 15-42 F),otomatik hava tahliye cihazı (Spirovent) ve basınç regülatörü (model:ITT C19) vardır. Her devrenin gidiş ve dönüş sıcaklıkları ise analog termometreler ile okunmaktadır.(NOSHOK - 20-120°C). Primer sirkülasyon ile sekonder sirkülasyon arasında düz levha plakalı bir eşanjör vardır(ALFA LAVAL:CB H50 50H). Kazan ile bu eşanjör arasındaki sirkülasyon bir pompa ile sağlanmaktadır(GRUNDFOSS Model: UP26 96 F).
Her devrenin kendi açma-kapama ve ayar vanası vardır(NIBCO Küresel vana).
Kontrol paneli ana boruları ve manifoldlar yalıtılmıştır.
Panel ise yükseltilmiş beton döşeme içine 25 mm ara ile yerleştirilen ENTRAN2 EPDM lastik borularından oluşmaktadır.
Boru iç çapı 10 mm dış çapı 15 mm dir. Isı iletim katsayısı 0.24 Kcal/hm°C olarak ODTÜ Makine Mühendisliği Bölümünde ölçülmüştür. Yükseltilmiş beton kalınlığı 13 cm olup üzeri makine halısı kaplıdır.(Şekil 4-5)



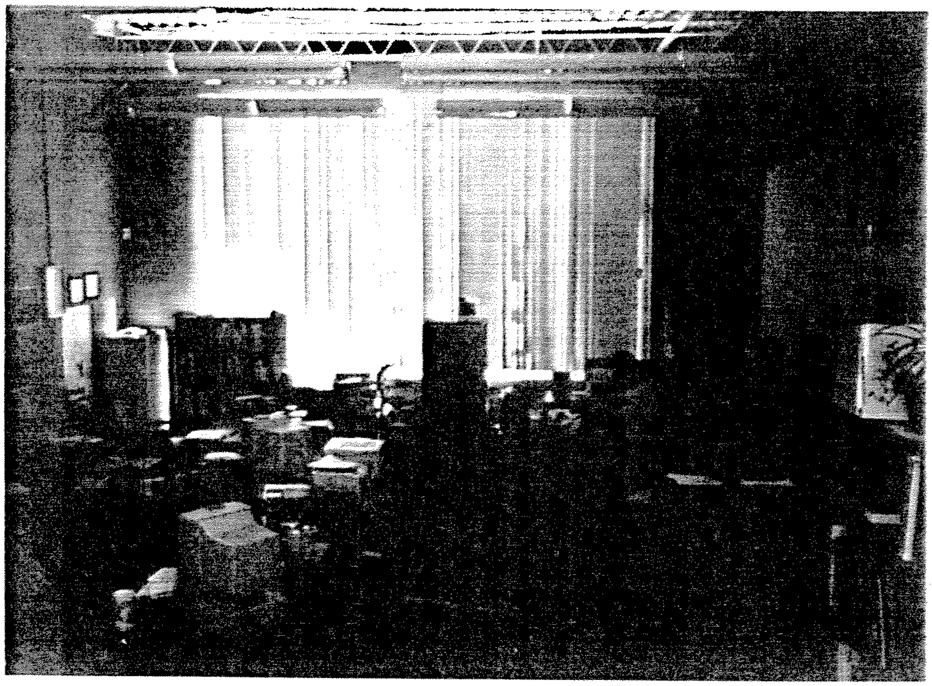
Şekil 4-3 Kazan



Sekil 4-2 DENEY ODASI

EK-4 DENEY DÜZENİ

Aslen analitik ve sayısal araştırmalar ile bilgisayar program yazılımlarının ağırlık taşıdığı projede bazı model ve hesapların doğrulanması,belirli mekanizmaların ve davranışların daha iyi anlaşılabilmesi için bir deney odası hazırlanmıştır. Bu deney odasınının her yönü ile gerçekçi olmasına çalışılmış ve her gün kullanılan bir fonksyona kavuşturulmuştur. Oda proje çalışmalarının belirli bir kımının yürütülmesinde ofis olarak fiilen kullanılmıştır. Odanın genel görünümü Şekil 4-1 de verilmiştir.



Şekil 4-1. Deney Odasının Genel Görümümü

Oda mevcut bir depo yapısından dönüştürülmüştür. Tek cephesi dışarıya bakmakta olup,tek katli normal pencere camı kullanılan saydam alanın yüzeyi 15m^2 dir. Oda boyutları 13x7x4.40 m. dir. Odanın hesaplanan ısıtma yükü 16000 Kcal/h dir.Isıtma 25000Kcal/h kapasitede bir doğal gaz kazanı VALLIANT marka) ve döşemeden ısıtma paneli ile sağlanmaktadır. Kazan başka bir odayı da beslemektedir.
Oda planı Şekil 4-2 de verilmiştir. Çatı saç olup yaklaşık 3cm strapor eşdeğeri yalıtıma sahiptir. Aydınlatma tavana bakar reflektörlü floresan lambalar iledir.

EK-3. Panel Isıl Dirençlerinin Formülasyonu

Kılkış (1993-c) ısıtıcı elemanların (hidrolik veya elektrik) panel içersine gömülü veya panele yapışık hatta panele bağlantılı ayrı konumlarda olabileceği üç hal içinde geçerli olacak toplam panel ısıl direncinin 4 elemana haiz olduğu belirtilmiştir. Buna göre:

$$r_u = r_s + r_t + r_p + r_a \qquad [3\text{-}1]$$

Burada r_s ısıtıcı eleman ile panel arasındaki temasın ısıl direncidir (Bakınız Ek-5, Yayın no: 19, Appendix 1).

r_t hidrolik sistemlerde boru ısıl direncidir. Akışkan tarafındaki film katsayısı içinde ortalama bir değer alınarak:

$$r_t \cong \{ 0.0237 \, (D_o / D_i) - 0.0219 \} / k_h \qquad 1.1 \leq D_o / D_i \leq 2 \qquad [3\text{-}2]$$

Burada k_h boru malzemesinin ısıl iletim katsayısıdır.

r_p panel gövdesinin ısıtıcı eleman merkezinden gövde yüzüne kadar olan kısmının ısıl direncidir. Bu bakımdan:

$$r_p = L/k \qquad \text{(Elemanlar panel dışında)} \qquad [3\text{-}3\text{-}a]$$

$$r_p = (L - D_o / 2) / k \qquad \text{(Elemanlar panele gömülü)} \qquad [3\text{-}3\text{-}b]$$

Burada L, panel karakteristik kalınlığıdır. Muhtelif L tarifleri Ek-5, 19 numaralı yayında (Chapter 6 bölümü) gösterilmiştir.

r_c ise panel yüzeyindeki kaplamaların ısıl direncidir.

$$r_{ci} : x_i / k_i \qquad [3\text{-}4]$$

Burada x_i ve k_i sırası ile i katmanının kalınlığı, k_i ise iletim katsayısıdır.

- Eşdeğer konfor şartları daha düşük iç mekan ıcaklıkları ile sağlanmaktadır. (örneğin 22°C erine 18°C),

)- Homojen ısı dağılımı ve daha az hava cere-'anı nedeni ile infiltrasyon ısı kayıpları azalmak-adır,

:- Kat betonu ısıtıldığından enerji depolama gö-evi yapmakta ve bu suretle günlük pik ısı yük-eri tıraşlanmaktadır. Bunun anlamı ise seçilen ‹azan (veya ısı pompası) ve ısıtıcı kapasiteleri-ıin daha düşük seçilebilmesidir.

3u durum, Şekil 2'de ayrıntılı bir biçimde izah edilmiştir.(a) ve (b) maddeleri ise doğrudan ısıt-ma için gerekli enerji sarfiyatında önemli bir azalma anlamına gelir. Örneğin tipik bir apat-man dairesinde günlük ısıtma enerji sarfiyatı % 35 ile %25 arasında azalır. Bu düşük enerji sarfiyatını karşılayacak olan ısı pompası ve yer-den ısıtma sistemi kapasiteleri ise pik saatteki (veya mevsimin en soğuk günündeki) yükün döşemede ısı depolaması sayesinde traşlan-ması nedeni ile daha düşük seçilebilmektedir. Bu katsayı da % 80 -% 85 arasındadır.

Isı yükünün ve seçilecek kapasitenin azalması ısı pompasının (ii) ve (iii) numaralı dezavantajla-rını oldukça hafifletmekte ve çoğu kez de orta-dan kaldırmaktadır. Bu ise ısı arz ve talep çizgi-lerinin kesişme noktalarının daha düşük dış sı-caklıklara doğru kayması ile gerçekleşmekte-dir.

Dış hava sıcaklığının artması ile birlikte ısı yükü-nün azalması, ısı pompasının bu ısı yükü eğrisi-ni takip etmesini gerekli kılar. Bu ise sabit devir ve kapasitede hatta dış hava sıcaklığı ile artan bir kapasitede çalışan ısı pompası için imkan-sızdır. Sadece ısı pompasını devreye alıp çıkar-mak (on/off) ise yerden ısıtma yapılan zeminler için ve devreye çok kısa girip çıkacak ısı pom-pasının performansı ve ömrü için zararlıdır. Ay-rıca böyle bir çözüm, ısı depolaması avantajını azaltacaktır. Bu nedenle bazı teknik çözümler gerekmekle birlikte, bu tür çözümler aslen pa-halı çözümler değildir. Ancak her uygulama özelinde analiz ve dikkatli bir tasarım gerekir. Böyle bir tasarımla örneğin İstanbul'da 25 000 kcal/h klasik ısıtma ısı yükü olan bir yapıda radyatörle ısıtma yapan ısı pompasının devre-den tamamen çıkma veya destek kazanın dev-reye girmesi için dış hava sıcaklığı +7.5°C ola-rak hesaplanmıştır. (Şekil 2, C noktası). Aynı binada döşemeden ısıtma yapıldığında hem daha küçük ısı pompası seçilebilmekte, hem de devreye girme/çıkma dış hava sıcaklığı (C)-5°C'a düşmektedir ki, aslen bu da İstanbul' un standart dış hava hesap sıcaklığının dahi altındadır. (Şekil 2).Ayrıca ek destek kazanına gerek kalmamaktadır. -5°C,-7°C arasında 4 kW'lık ek bir ısıtıcı yeterlidir (yeni E noktası -4.5°C). Kaloriferli sistemde E noktası + 1°C' dır. Döşemeden ısıtmanın en önemli bir diğer avantajı ise aynı borulardan serin su dolaştıra-rak ısı pompasının yazın, etkin bir duyulur (sen-sible) soğutma yapabilmesine imkan ve ortam sağlamasıdır. Soğutma yapılırken bir yandan da sıcak su elde edilebilmektedir. Soğutma yapıl-dığında ısı pompası ilk yatırımı sadece ısıtma mevsim yerine yılın bütün aylarına dağılması ile geri ödeme hızlanmaktadır. Böyle bir uygulama için hem ısıtma hem soğutma yapabilen ısı pompaları seçilmelidir.

ÖRNEK UYGULAMA

Kullanıcı bazında yıllık enerji masrafları yönün-den bir mukayese yapılmıştır. Bu mukayesede ısı pompası + yerden ısıtma sistemi ile bir yıllık parasal ısıtma gideri, klasik bir kazan (fuel-oil, doğal gaz, linyit kömürü seçenekleri için) ve radyatörlü sistemde beklenen yıllık gider ile ay-lık dökümler halinde mukayese edilmiştir. Isı pompası + kalorifer sistemi ise yukarıda izah edilen teknik sakıncalar nedeni ile inceleme dı-şı bırakılmıştır.

Kabuller

Yer = İstanbul, T_o = - 3°C, Klasik sisteme göre pik ısıtma yükü 25.000 kcal/h.

Yerden ısıtma kabulleri: Ortalama su sıcaklığı = 45°C,

Isı depolama katsayısı = 0.85,

Döşemeden ısıtma ısı yükü : 0.75 x Hesaplanan standart ısı yükü.

Aslen döşemeden ısıtma ısı yükündeki azalma % 35-% 40'a varmakla birlikte emniyetli olmak üzere % 25 değeri kullanılmıştır.

Yakıt fiyatlarında Aralık 1990 fiatları kullanılmıştır.

Çizelge 1. Aylık Toplam Isı Yükleri (İstanbul) kcal/ayx 10^6

	EKİM	KASIM	ARALIK	OCAK	ŞUBAT	MART	NİSAN	MAYIS
Yerden ısıtma	3.3	5.9	6.7	9.0	8.0	8.2	5.5	2.5
Kazan + Radyatör	4.4	8.0	9.3	12.4	11.2	11.7	7.7	3.5

Isı pompası seçiminde kullanılacak kapasite değeri ise % 25 emniyet zammı ile 20.000 kcal/h'dir.

Bu kapasiteye uygun ısı pompasının aylık toplam elektrik enerjisi sarfiyatı, yaklaşık olarak:

Çizelge 2. Isı Pompası Aylık Enerji Tüketimi (kW-h)

EKİM	KASIM	ARALIK	OCAK	ŞUBAT	MART	NİSAN	MAYIS
1050	2320	2400	3500	3100	3150	1870	780

Çizelge 3. Klasik Sistemde Aylık Yakıt Tüketimi (Fuel-oil ve kömürlü kazan verimi: 0.75, Doğal gazlı kazan verimi: 0.85

	EKİM	KASIM	ARALIK	OCAK	ŞUBAT	MART	NİSAN	MAYIS
Fuel Oil 5NO (kg)	580	1070	1240	1650	1500	1560	1026	470
Linyit (ton)	2.3	4.2	5	6.6	6	6.2	4.0	2.0
Doğal Gaz (m^3)	630	1140	1325	1770	1600	1670	1098	500

Yıllık Toplamlar:

Elektrik = 18170 kW-h
Fuel Oil = 9100 kg
Linyit = 36 ton
Doğal Gaz = 9730 m^3

Yıllık toplamlardan parasal maliyet, güncel fiatlarla bulunabilir. Ortalama Aralık 1990 fiatlarına göre yıllık enerji tüketim maliyetleri, yaklaşık olarak:

Isı Pompası = 4.550.000 TL
Fuel Oil = 9.100.000TL

Kazan Linyit = 7.200.000TL
Doğal Gaz = 8.297.500TL

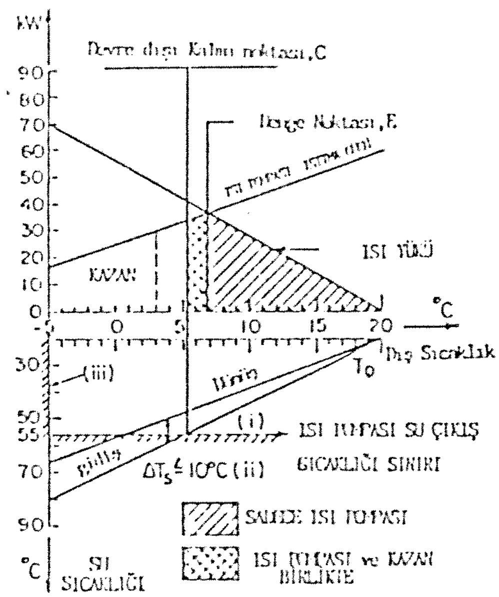
Görüldüğü üzere ısı pompası ile ısıtmada fuel oil ile ısınmaya göre tam % 50 tasarruf, linyit ile ısınmaya göre % 37 tasarruf sağlamaktadır.

Radyatörlü bir sisteme ısı pompası bağlansa idi yıllık maliyet 6.200.000 TL olacağı gibi, en az 15.000 kcal/h'lik bir klasik kazana gerek duyulacak, bu kazanda da örneğin 2.500.000 TL'lik linyit yanacaktı. Toplam enerji bütçesi bu durumda 8.700.000 TL olacağı gibi bir de 15.000 kcal/h'lik kazanın ilk yatırım külfeti olacaktır.

SONUÇ

Havadan-Suya ısı pompaları, ısıtmada teknik ve ekonomik yönden sadece döşemeden ısıtma ile avantajlı olabilmektedir. Böyle bir çözüm ile yıllık enerji giderlerinde tüketici yönünden en az % 40 kadar tasarruf gerçekleşmektedir. Ayrıca bugünkü şartlarda yerden ısıtma m^2 maliyeti İstanbul koşullarında zaten radyatörlü sisteme göre daha ucuz olmaktadır. Öte yandan soğutma yapabilen türde ısı pompası seçimi ile yazın bir yandan soğutma yapılırken bir yandan sıcak su elde edilebilmektedir. Bu da ısı pompasının ilk yatırımının geri ödenmesini hızlandıracak ve diğer sistemlere karşı rekabet şansını daha da arttıracaktır. Burada, kullanıcı açısından yapılan değerlendirme Ülke enerji bilançosu açısından tekrarlandığında ise ortaya değişik bir sonuç çıkmaktadır [1]: Elektrik, termik bir santralde üretiliyor ise ve bunun birincil enerjiye kıyasla verimi % 30 alınır ise, ortalama ısıtma tesir katsayısı 2.5 kabülü ile nihai birincil enerji tüketim verimi sadece 2.5x% 30 = % 70 eder ki bu da klasik bir kazanda elde edilebilecek verimden düşüktür. Bu mahzuru gidermek üzere ısı pompalarının elektrik yerine doğrudan primer enerji, örneğin doğal gaz ile tahriki gerekir. Bu ise teknik olarak mümkündür [2].

Şekil 3'de ise örnek çalışmaya ilişkin sistem önerisi verilmiştir. Bu sistemde bir kapalı genleşme kabı ve 4 kW'lık destek ısıtıcı bulunmakta, bu sayede kazan tamamen ısıtma sezonu boyunca devre dışı bırakılabilmektedir[3].



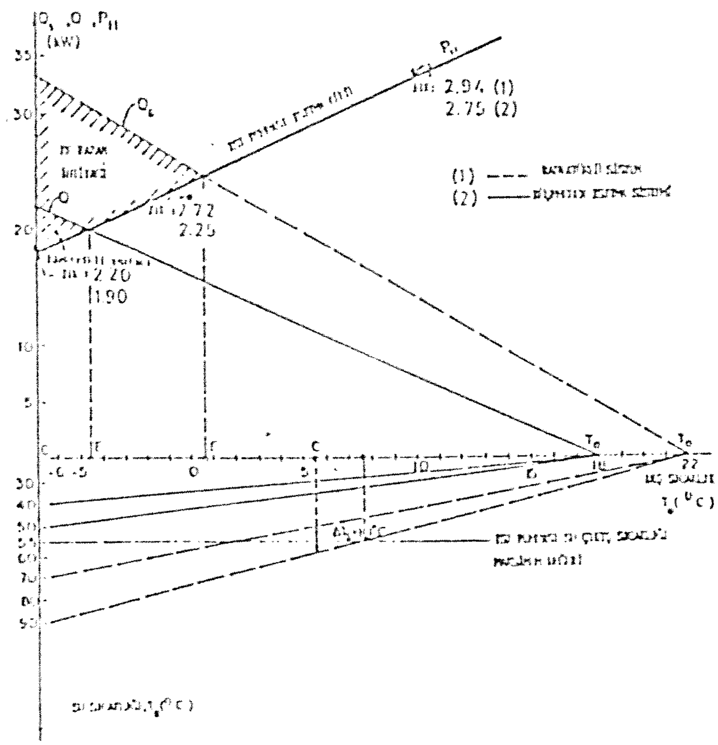
ŞEKİL.1. HAVADAN/SUYA ISI POMPASI PERFORMANSI İLE ISI YÜKÜNÜN MUKAYESESİ

Kısaltmalar

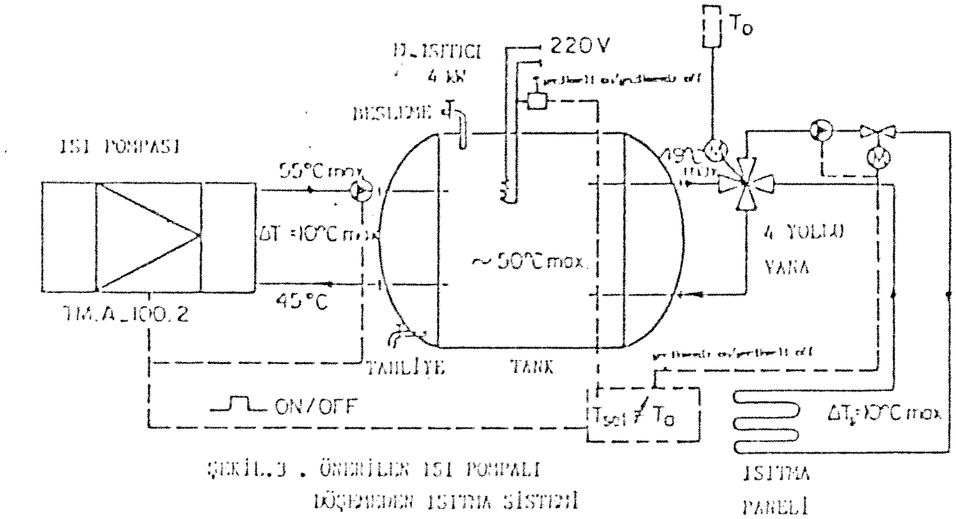
C Isı pompası devreden çıkma noktası
E Isı yükü-ısı arzı denge noktası
ITK Isıtma Tesir Katsayısı

SİMGELER

Q_s Kaloriferli sistem ısı yükü, kW
Q Döşemeden ısıtma yükü, kW
P_h Isı pompası ısıtma gücü, kW
T_a İç oda konfor sıcaklığı, °C
T_d İç mekan konfor sıcaklığı, °C
T_o Dış hava sıcaklığı, °C
T_s Ortalama su sıcaklığı, °C
T_{set} Oda sıcaklığı ayar değeri, °C
T_s Su sıcaklık farkı, °C



ŞEKİL. 2. ISI POMPASI PERFORMANSININ RADYATÖRLÜ ve DÖŞEMEDEN ISITMA SİSTEMİNDE MUKAYESESİ



ŞEKİL.3 . ÖNERİLEN ISI POMPALI
DÖŞEMEDEN ISITMA SİSTEMİ

KAYNAKÇA

1. Kılkış, B., "Floor Heating with Natural Gas Driven Heat Pumps: A New Alternative in Space Heating ", 4th. Int. Conf. on Solar Energy and Heat Pumpŝ, Proceedings, 7-8 April 1988, pp. 124-134, İstanbul, 1988.

2. Kılkış, B., "Computer Aided Analysis of Possibility of Increasing Heat Pump Performance by Application of Floor Heating Systems for Buildings at Adverse Climates", Int. Conference on Applications and Efficiency of Heat Pump Systems in Environmentally Sensitive Times, 1-3 October 1990, Munich, Fed.Rep. of Germany.

3. Kılkış,B., "Binaların Isıtılmasında Doğal Gaz Kullanımı; Tasarruf Olanakları ve Yeni Seçenekler", Ankara Sanayi Odası Dergisi, Mayıs-Haziran 1990, Yıl 14, Sayı 104, Ankara.

Ayrıca: 1. Ankara Doğal Gaz ve Isı Sistemleri Fuarı Dergisinde yayınlandı, 25-28 Ekim, Ankara,1990.

A TENTATIVE LIST OF CONTRIBUTIONS

TO

ASHRAE HANDBOOK - HVAC APPLICATIONS

CHAPTER 45

ON

SNOW MELTING

Review of snow melting heat load calculations.

Review of US/Canadian weather data.

-Determination of "snow fall Coincident" critical temperatures.

-Investigation of the most critical condition (a combination of outdoor temperature and snowfall rate) for US/Canadian cities..

-An investigation of the temperatures registered immediately after snow (usually there is a tendency of severe temperature dips and this may cause icing conditions before the surface gets completely dry.

Further investigation and comprehensive analysis of back and edge losses. Provide sufficient information and display a calculation algorithm in this Chapter.

Completely rewrite:

"Pipe System" section(including new figure(s) instead of Figure1.

This chaper must include analysis techniques customized to thermoplastic and rubber hoses. C. I and copper hoses have virtually vanished from snow-melting applications. The thermal performance of course is much different today. The existing information may be even misleading for a novice designer.The current procedure and information does not allow corrections for multiple surface coverings of different metal properties and thickness which is a common case in practice.

5- A new section must be be dedicated to thermo-plastic and rubber hoses explaining typical friction coefficients, singular losses due to the bends of the losses etc.
This section should enable the designer to arrive at an accurate prediction of the pressure drop with or without of antifreezing agents at a given proportion.

6 - Explicit equations to calculate the viscosity, density and specific heat of the fluid mixture must be either provided or referenced to appropriate ASHRAE publication(s) if possible.

7 - The designer must be made aware of thermal and mechanical expansion of hoses. Usually the loops are longer (like 400') and the fluid mixture expansion and hose expansion must be emphasized.Usually the expansion tanks are undersized.This causes mechanical problems in addition to corrosion problems(if there is no appropriate supply of the mixture.

ecial provisions for ice melting load must be provided. ike in car wash areas ice/sleet melting system in (amps).

ermal response of the slab and associated lag times ten pose operational problems that may deteriorate the rformance of the system. This also effects the necessary ling load for the selection of an optimum operational gime and control strategy. A new section for the prediction the thermal response time must be added or available infor- tion must be referenced.

order to provide the designer a simple design tool, design mograms together with appropriate correction factors must appended.

slab thermal analysis/design algorithm with necessary heat ansfer equations (most probably through fin efficiency eory) must be included.

e "Control" section may be expanded with some emphasis to iling monitoring strategies/control , "after snowfall" eration/control and possibly remote control.

ecific guidelines for heat exchanger design/selection st be provided in more detail.

imited experimental data in the literature dating back to nineteen fifties should be reviewed .Some more experiments may be performed if found necessary.

15 -The current design algorithm assumes a uniform surface temperature.This is not the case. The surface temperature distribution is usually a function of the snow melting load,wind conditions, outdoor temperature, depth of heat transferring hoses,properties of slab, and surface covering material(s) and their thicknesses, hose spacings, hose dimensions and tilt angle of the surface. A typical computer output about the surface temperature prediction is attached. As seen, usually a "below freezing" surface temperature exists at the midway of hoses.

16 -The necessary average fluid temperature can be calculated with relavant equations. In fact this temperature depends upon hose spacings, hose depth, slab/floor covering material(s), hose material, and surface temperature swing. Existing table on average the temperature is empirical and were originally developed for C.I. pipes. In addition, this temperature should be based upon the maximum temperature to be maintained on the surface right above the hose in order to keep minimum surface temperature "above" the freezing point.

Respectfully submitted to the members of ASHRAE T.C.6.1.

Prof.Dr.Birol KILKIS

mem.ASHRAE,ASME,ISES

cl:Appendix

APPENDIX 1

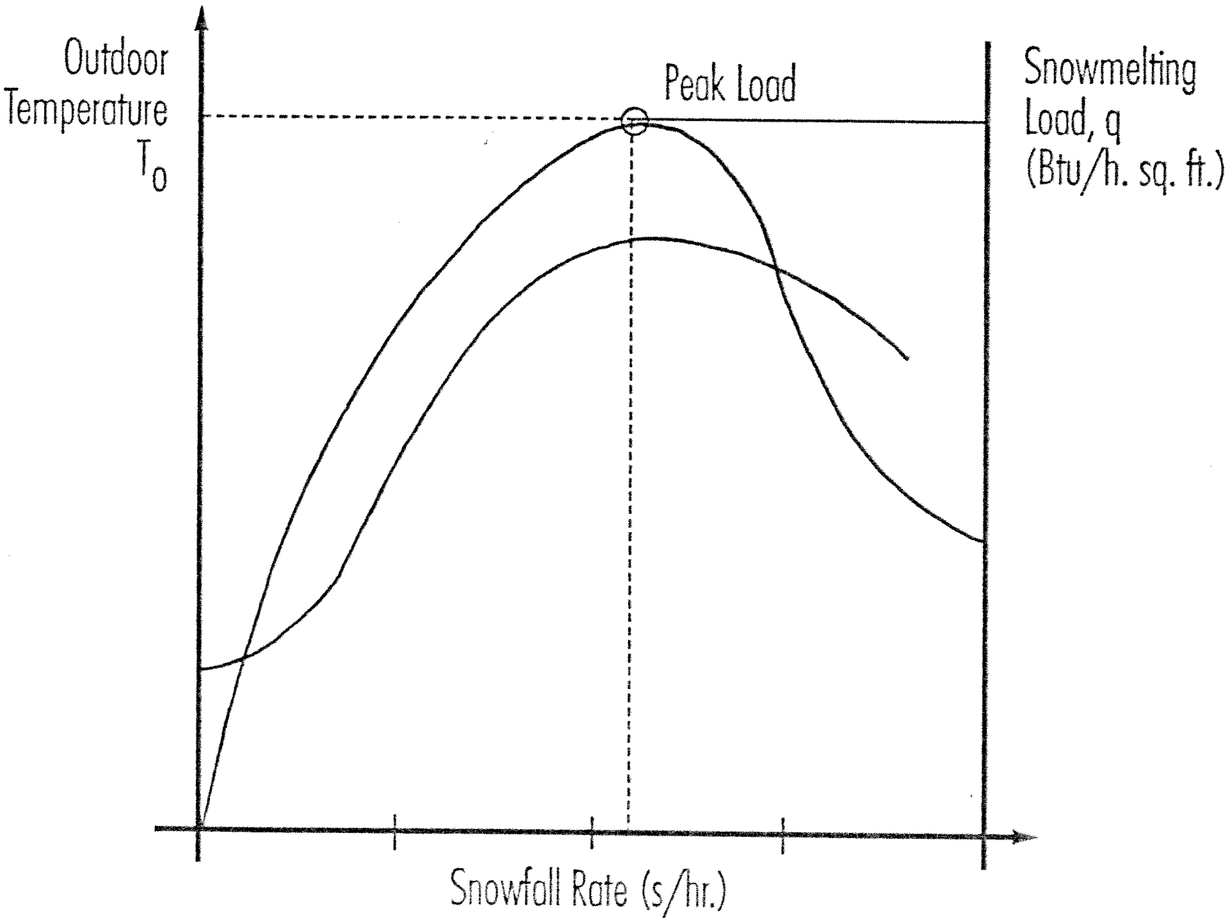
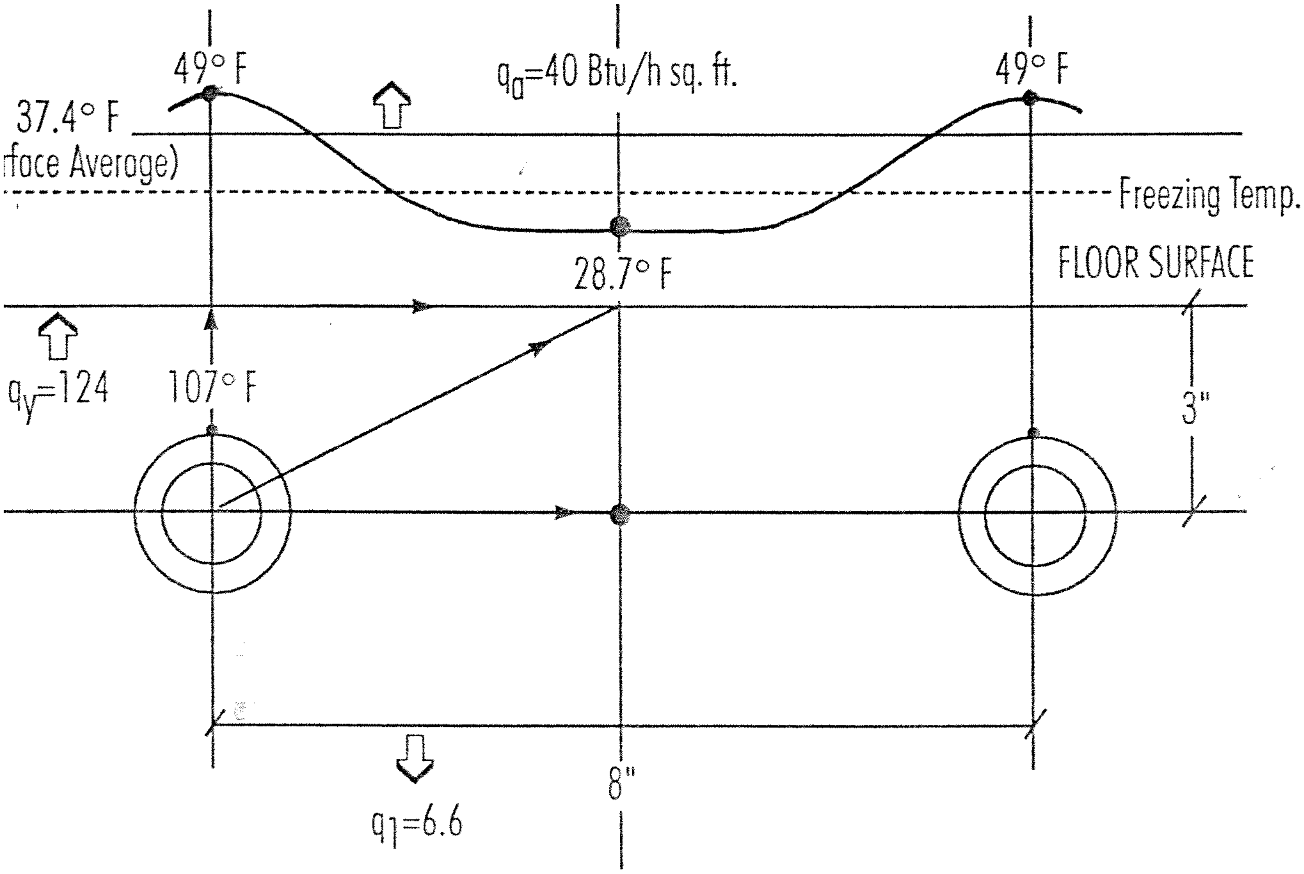


FIGURE 1: A Possible T_0, s and q relationship (Kilkis, 1991).

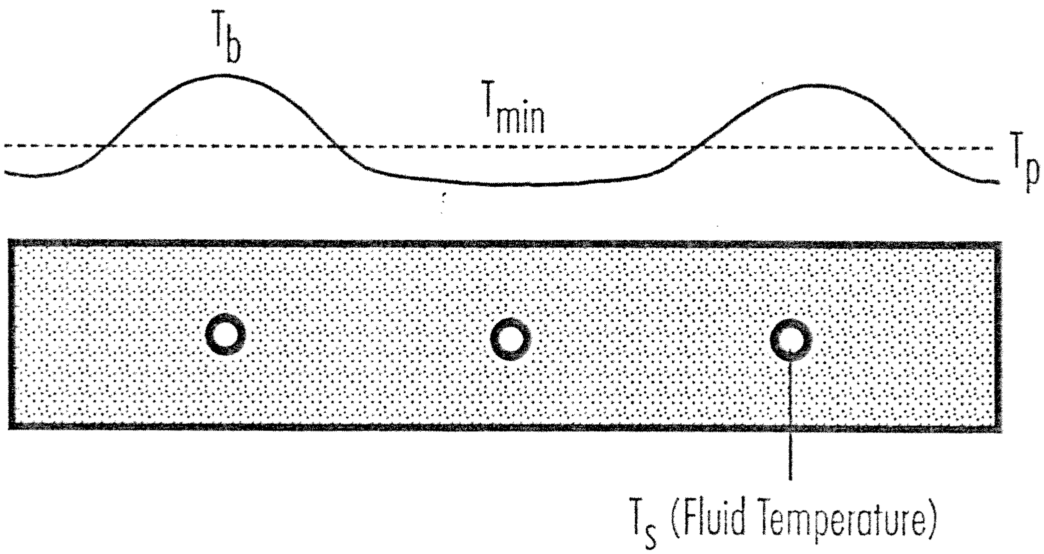
```
ect Name :SAMPLE
ation    :SPRINGFIELD
ent Name :
e        :01/22/1992
OR DESIGN PARAMETERS
wmelting Class = 2
               = 0.0
               =24804.2
               =200.00
               = 0.00 0.00
               = 3.0000
               = 1.0000
               = 20.0
               = 0.6250
 Efficiency of the Slab         =68.41[%]
ber of surface coverings        =1
e spacing between their centers = 8.000  [in]

IGN RESULTS:

 Temperatures:
rage surface temperature                  ,Tp =37.4 deg.F
imum surface temperature                  ,Tb =49.8 deg.F
imum surface temperature                ,Tmin =28.7 deg.F
e wall temperature                        ,Td =107.6 deg.F
rage Fluid temperature                    ,Ts =150.0 degF +/- 2

 If the system will be idled by
 Water Temperature modulation then ,Tsidle=135.1 deg.F

 probe set Temp.for HEATWAY CONTROL  ,     =33.0 deg.F

apacities and Loads:
al Heat Loss Intensity to Atmosphere ,ql = 40.858 [Btu/h-ft»]
al Snow Melting Intensity            ,qy =124.000 [Btu/h-ft»]
ing Heat Intensity                ,qidle = 78.000 [Btu/h-sq.ft]
w Melting Thermal efficiency           ,x =94.95 [%]
centage of Heat lost to  below    ,(1-x) = 5.05 [%]
ler Snow Melting Load                 ,BL =26118.2 [Btu/h]

ydraulics:
ID flow rate                           ,V =    0.676 [US GPM]
rage Fluid velocity in the hose       ,Vs =   0.71 [ft/s]
e Reynolds number                     ,Re =7517.627
id-Hose thermal convection coeff.   ,alfa =294.656 [Btu/h-ft»]
e friction coefficient             ,Lamda = 0.03714
rected Unit pressure drop             ,Dp =0.004 [psi/ft]
e length in the circuit                ,L =330.00 [ft]
rected Total pressure drop in circuit,Pt = 1.25 [psi]
rected Pump capacity for the circuit     =    0.676 [US GPM]/  2.79 [ft/hd]
E=Calculations were based upon  50% Glycol Water mixture
```



A Typical Temperature Distribution on a Snowmelting Slab

nple Computer Simulation Output for the Precision of the System Performance and ce Temperature Distribution.

T_b= Max. surface temperature, to be maintained in order to keep Tmin >32°F



A typical surface temperature swing with a "below-freezing" minimum temperature.

Note: When the design is carried on with T_p alone, T_{min} is usually less than 32° F. This leaves a snow/ice patch at the center of hose O.C. unless the snowmelting surface is inclined (ie: a ramp). See a typical output of the computer simulation.

it loss to the atmosphere

A. Radiation loss: $f(T_0$, sky conditions, angle of tilt [ie: ramps]).

B. Convective loss: $f(T_0, v)$.

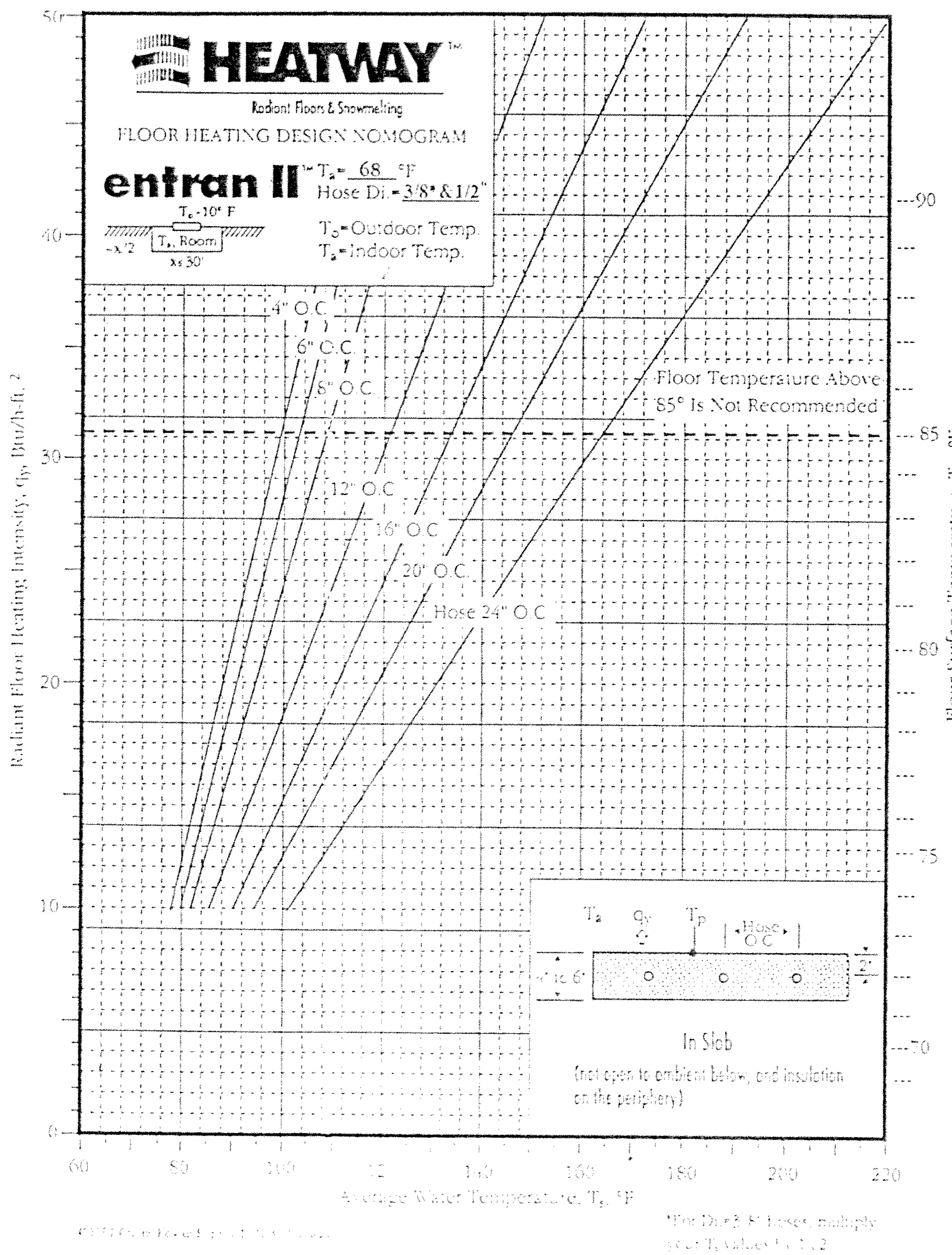
' must be individually expressed instead of a siingle equation.

uations for edge and back losses must be provided.

uations must allow for different insulation levels, material below the slab, sition of the slab (ie: above, or in the ground) etc..

ustomized design nomograms must be provided (see attached figure).

verage water temperature calculation procedure must be provided. This is also vailable on nomograms (see attached figure).

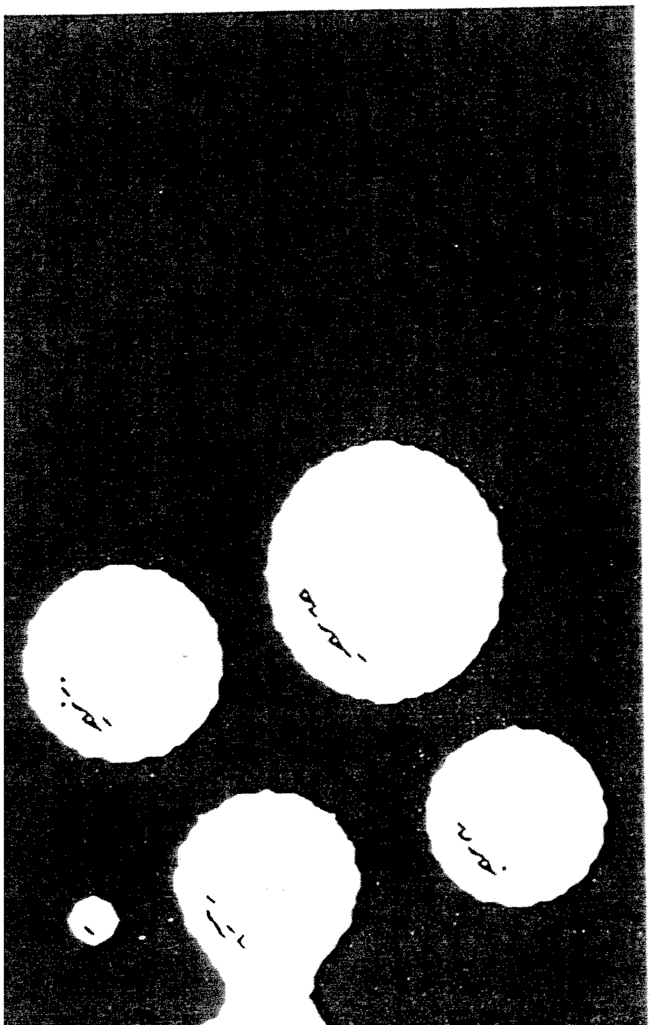


A Sample Design Nomogram (yet to be customized for snowmelting). (Kilkis, 1992)

There are seven specific entry points for oxygen into any radiant system. Even the best "barrier pipe" will only block one of these entry points. Permeation through radiant piping is, at most, a minor factor in oxygen based corrosion problems.

he O_2 FACTOR

r. Birol Kilkis
ike Chiles



HISTORY OF PERMEATION DISCUSSION IN EUROPE

For several years there has been a continuing discussion in Europe about the effect of coupling non-metallic radiant piping to ferrous components. A concern has been voiced about the results of oxygen molecules permeating through the walls of non-metallic piping.

This permeation concern was first raised by the European manufacturers of competing conventional hot water space heating equipment when they began to see a good part of their market go over to floor heating systems.

Due to the prevalence of thin-walled pressed steel radiators, European systems are particularly sensitive to corrosion processes. When a steel radiator would fail in a "mixed" system combining non-metallic radiant piping and steel radiators, it was natural for a radiator maker to automatically assign the blame to a competing product.

At this point in time many European pipe manufacturers took advantage of oxygen barrier technology developed by the food packaging industry to add barrier properties to their piping. This had the dual benefit of shifting blame for corrosion back to other parties; while enabling them to sell a "value added" pipe that their less well capitalized competitors could not produce.

These thin films used to slow down the flow of oxygen were designed as much as barriers to litigation from the radiator manufacturers, as they were barriers to oxygen. And in truth, the European barrier products have barred both oxygen permeation and litigation, while serving remarkably well as marketing vehicles for pipe, too.

These European marketing campaigns, designed for a heated and competitive overseas market, have done a disservice to the emerging North American market.

In North America, the unwise and misleading promotion of the "universal" merits of barrier products has instilled a false sense of security among some system specifiers.

An example of marketing hyperbole, which, if believed, will lead to expensive consequences for many hydronic system owners, is excerpted from a leading European manufacturer's own literature extolling the merits of his "barrier pipe".

"Boiler loop isolation heat exchangers are not needed. Neither are expensive and questionable corrosion inhibitors." (Wirsbo 1990)

There are seven specific entry points for oxygen into any radiant system system. Even the best "barrier pipe" will only block one of these entry points. As shown later, permeation is, at most, a minor factor in oxygen based corrosion problems. The six major sources of oxygen ingress must be addressed by trained, reputable design and installation professionals.

EXAMINING THE EUROPEAN LITERATURE ON OXYGEN PERMEATION

In sorting through all the various claims, it may be helpful to examine what one radiant pipe company (Wirsbo) has to say in their excellent book Water and Pipes. In addition we will look at what various scientific authorities have to say about this issue.

"In the case of surface heating using plastic pipes, we already have more than ten years of practical experience that we can use in evaluating the corrosion problem. The number of under-floor heating systems that are in operation today in Europe must be in the millions. Despite the large number of systems, there have only been a few cases where problems have developed in which oxygen permeation could not with certainty be ruled out as a contributing factor. Instead, most of the problems were caused by circulatory difficulties. These were eliminated by flushing out the system." [1]

On the next page of Water and Pipes we read, " The fact that, according to experience gathered up to the present time, the risk from problems caused by oxygen permeation is minimal anyway, should also be taken into account when deciding upon the material to use." [1]

German scientists have expressed a similar position that floor heating systems can be subject to corrosion problems, but that the problems do not arise from permeation. Dr. Scharman mentions "In compact floor heating systems, an oxygen diffusion caused by plastic pipes can be observed". According to the present data and calculation principles, however, this oxygen diffusion is not responsible for the frequently occurring strong corrosion process in the floor heating systems. [2]

Dr. Eagle concurs that there are observed problems with European floor heating systems. "Calculations using the coefficient of oxygen diffusion for a particular pipe, show that it is rarely possible for oxygen diffusion through the pipe wall itself to account for such high rates of corrosion. There must be clearly other sources of oxygen into the system." [3]

SUMMARY OF THE DIN 4726 STANDARD

Abstracted from another German publication is the following. "Plastic pipes, oxygen diffusion, and corrosion have raised discussions and stimulated proposals for solution both in the past and present. The state of the art is reflected in technical specifications and draft standards, e g the BVF Specifications Sheet No. 4, 'Corrosion prevention in floor heating systems using plasic pipes', and the DIN 4726 draft standard 'Plasic pipes for hot-water floor heating systems.' This standard specifies three measures to prevent corrosion from oxygen diffusion, which are listed in the article." [4]

One of these three different measures to deal with corrosion specifies the degree of oxygen barrier properties necessary to comply with the DIN standard 4726.

It should be pointed out that the DIN standard does not single out any one approach as the preferred solution. Two other measures are also acknowledged in the DIN standard as solutions to the problem of corrosion in European systems.

The other two DIN solutions acknowledged include:

1) Use of corrosion inhibitors in the entire system. DIN 4726
2) Use of corrosion resistant pumps, valves, manifolds and other components in the floor heating circuits with total separation of the floor heating circuit water from the boiler by means of a corrosion resistant heat exchanger. DIN 4726

As Wirsbo also concludes in their book "Water and Pipes". [1] "At this time there are several possibilities being discussed to lessen the risk of possible damage due to oxygen permeation. They include:

1) Addition of corrosion retardants to the heating water,
2) Separation of the heat source and the under-floor heating system by using a heat exchanger,
3) Using plastic tubing with a sheathing that is impervious to oxygen."

CORROSION PROBLEMS WITH EUROPEAN SYSTEMS

It should be emphasized that virtually all the problems in European floor heating systems are associated with the use of thin walled pressed steel radiators and welded steel radiant distribution manifolds. The steel distribution manifolds had the greatest difficulty in systems where little other ferrous materials in the radiant system were available to react with the available free oxygen in the system.

THE NORTH AMERICAN MARKET IS DIFFERENT

The prevalence of copper tube and thicker walled cast iron radiation in North America virtually eliminates the source of problems observed in Europe. Steel radiant distribution manifolds are a rarity.

No one can deny that some amount of oxygen does in fact move through the walls of any non-metallic pipe. The real question for each system is this:

Is the amount of oxygen moving through the walls of the radiant piping of economic significance? In other words, will this amount of oxygen moving through the pipe wall cause any individual component to fail before the end of its normal economic life?

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8. TEŞEKKÜR

Bu araştırmayı MİSAG-12 projesi ile desteklemekte olan Türkiye Bilimsel ve Teknik Araştırma Kurumu'nun yakın ilgi ve desteğinden dolayı yazar içtenlikle teşekkür eder.

avantajlarına karşın döşemeden ısıtma sistemlerinin başarısı detaylı ve bilinçli bir tasarıma ve eniyileme çalışmasına ihtiyaç göstermektedir. Ülkemizde eksikliği hissedilen standardizasyon çalışmalarına bu araştırma projesi çerçevesinde başlanmış olup iki adet tasarım standardı tasarısı hazırlanmış bulunmaktadır. Ayrıca döşemeden ısıtma borularının belirli standard özelliklere kavuşturulması ve mecburi standardlar ile denetlenmesi bu sistemin potansiyel yararlarının uygulamaya aktarılmasına geniş ölçüde katkıda bulunacaktır.

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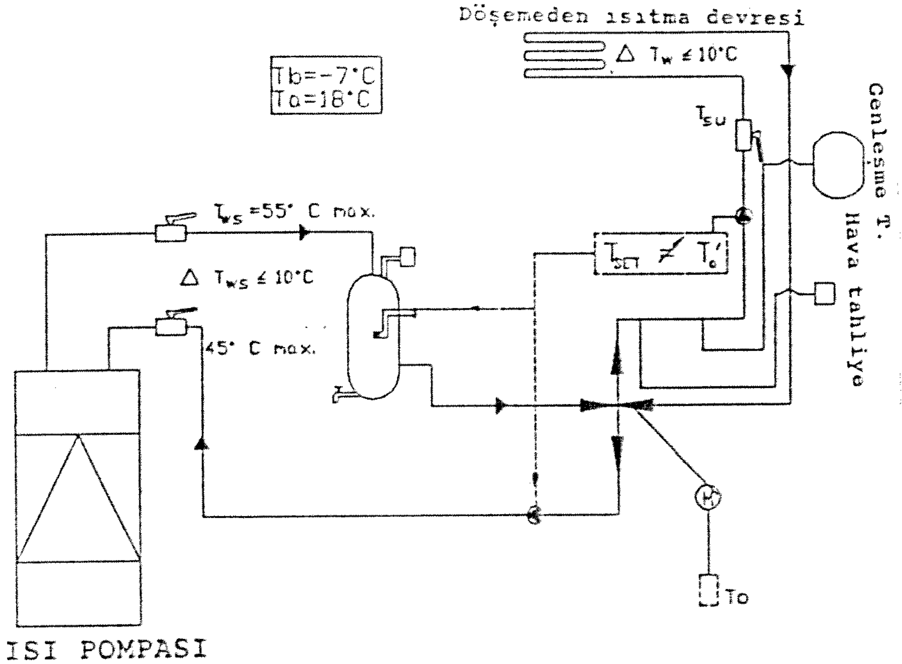
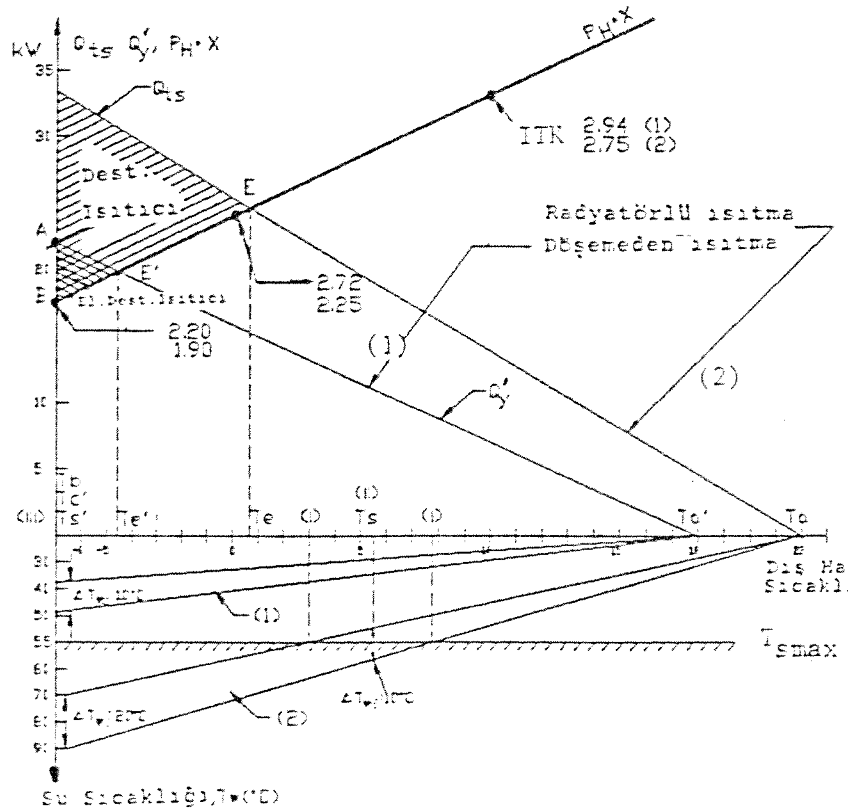


Şekil 6. Örnek Döşemeden Isıtma Sistemi.

Öngörülen döşemeden ısıtma sistemi Şekil 6 'da gösterilmiştir. Bu sistemde bir dört yollu vana ile ortalama su sıcaklığı ayarlanmakta, ısı pompası döşemeden ısıtma devresini bir depo aracılığı ile beslemektedir. Isı pompasının çalışmasını iç hava sıcaklık kontrolu düzenlemektedir. Bununla birlikte döşeme ortalama sıcaklığının belirli bir düzeyde tutulması da sağlanmaktadır.

6. SONUÇ VE TARTIŞMA

Yabancı ülkelerde inşa edilen yeni konutların yaklaşık üçte birinde ısı pompası kullanılmaktadır (Lannus, 1989). Isı yükünün azalması nedeni ile döşemeden ısıtma sistemleri önemli bir avantaj sağlamakta ayrıca ısı pompaları ile çalışmada gösterdikleri uyum çerçevesinde alternatif enerji kaynaklarının etkin kullanımı da o derece önem kazanmaktadır. Diğer bir aşama olarak elektrik motoru yerine doğal gaz motoru ile tahrik edilen ısı pompalarının kullanımı ile birincil enerji kaynaklarının kullanımındaki verimlilik daha da artacak ve kısıtlı kaynakların çok rasyonel bir biçimde kullanımı gündeme gelebilecektir (Kılkış, 1988). Konuya ısı pompalarının ekonomik ve teknik rekabeti yönünden bakılması durumunda ise döşemeden ısıtma sistemlerinin katkıları oldukça büyük boyuttadır. Özellikle destek ısıtma sistemlerinin devreden çıkarılması ilk yatırım ve işletme masraflarını azaltacak, ısıtmada birincil enerji kaynaklarının payı da aynı oranda azalacaktır. Ayrıca hava kirliliğine belirli bir ölçüde çözüm getirilmiş olacaktır. Tüm bu bir yüzeyde döşemeden ısıtma uygulanacaktır. Döşeme kaplamasının ısıl direnci 0.05 m^2K/W 'dir. 0.0095 m iç ve 0.016 m dış çapı olan özel lastik boru kullanılacaktır. Hazırlanan bilgisayar programı (Kılkış, 1992) kullanılarak en uygun boru merkezleri arası mesafe 0.20 m olarak seçilmiştir. Bu aralığa karşılık gelen ortalama su sıcaklığı ise indirimli pik yükte 42 °C olarak hesaplanmıştır. Bu nedenle maksimum su sıcaklık talebi 47 °C (42+10/2) olacaktır. Çizelge 1, sütun 3 ve 4 'de verilen değerlerin interpolasyonundan seçilen ısı pompası kapasitesinin -5 °C koşulunda ısıtma yükünden sadece 2 kW az olacağı görülmektedir. Aslında bu zahiri nokta gerçek dış hava tasarım değerinden düşük olduğu için uygulamada destek ısıtıcıya gerek olmayacaktır. Eğer aynı ısı pompası radyatörlü bir ısıtma sistemine bağlansa idi, ısı yükü Q_{ts} doğrusunu takip edecek, Denge Noktası sıcaklığı +1 °C 'a yükselecek ve destek ısıtıcı ihtiyacı da 9 kW (29 kW - 20 kW) olacaktı. İki sistem Şekil 5 'de mukayese edilmiştir. Görüldüğü üzere radyatör yüzeyleri büyük seçilse de kapatma noktası (+5 °C) nedeni ile destek ısıtıcı devreye denge noktasından daha önce girecek ve ısı pompasının mevsim boyunca etkinliği azalacaktır. Bazı ITK değerleri şekil üzerinde gösterilmiş olup bu değerler döşemeden ısıtma alternatifi için yaklaşık % 10 daha fazladır.



Şekil 5. Döşemeden ve Radyatör ile Isıtma Sistemlerinin Isı Pompası Performansı Yönünden Mukayesesi.

4.3. Bilgisayar Programı

Söz konusu algoritmayı uygulamak için bir bilgisayar programı hazırlanmıştır. Program sisteme ve tasarıma etken tüm değişkenleri göz önünde tutabilmektedir. Panelin çevre ve alt ısı kaçakları hesaplara dahil edilmekte, performansa etken kısıtlar kontrol edilmektedir. Program gerçek zamanda tasarım ortamı için hazırlanmış olup tasarımcının en iyi çözüme ulaşmasına etken gerekli uyarı ve önerileri yapmaktadır. Program ayrıca panel akışkan devresine ilişkin bütün çözümleri de vermektedir. Döşemeden ısıtmaya özgü ısı yükü ise geliştirilmekte olan başka bir program aracılığı ile hesaplanmaktadır. Tasarım kısıtları ise:

(i) $T_p <= 29$ °C (kullanılmayan mahallerde ve oda çevre bandında 31 °C)

(ii) $T_w <= 70$ °C (Özel lastik malzeme); $T_w <= 60$ °C (Termoplastik malzeme)

(iii) $A_p / M * 1.1 <= Z$

(iv) $Q_y / x \leq 5$ kW

(v) $0.5 <= v_s <= 1.5$ m.sn^{-1}

(vi) $M >= 6.D_o$ (Özel lastik malzeme); $M >= 10.D_o$ (Termoplastik malzeme)

(vii) $q_y <= 140$ W/m^2

(viii) $X >= 0.80$

(ix) $P <= 5000$ mm·ss (49 kPa)

(x) $T_b <= 45$ °C

(xi) $T_{max} <= 70$ °C

(xii) $Re > 6000$

(xiii) 0.025 m. $<= L <= 0.060$ m.

5. ÖRNEK TASARIM

Deniz seviyesindeki ve birinci iklim bölgesinde iki katlı yeni bir konutun ısıtma sistemi tasarımlanacaktır. Yapının standard ısı yükü 29 kW, dış hava tasarım sıcaklığı -3 °C 'dır. Tipik bir odada standard iç hava sıcaklığı 22 °C 'dır. Yapının toplam alanı 320 m^2 olup 275 m^2 'si döşemeden ısıtma uygulamasına müsaittir. Havadan suya tip ısı pompasının kullanımı düşünülmekte olup T_c değeri -7 °C 'dır. Isı pompasının etkinliğini bu sınıra kadar sınamak için dış hava tasarım değeri bu örnek için -7 °C 'a kaydırılmış ve ısı yükü bu yeni değere göre düzeltilmiştir. Bu değer 33.64 kW 'dır. Döşemeden ısıtmada su sıcaklığı ayarlanmakta ve sistem sürekli devrede tutulmaktadır. Bu nedenle döşemede ve komşu duvarlarda ısı depolaması söz konusu olup pik tasarım yükünde indirim mümkündür. Eşdeğer konfor için iç sıcaklık 18 °C olarak seçilmiştir (Kılkış, 1990). Bu nedenlerle döşemeden ısıtma tasarım ısı yükü 22.20 kW olmaktadır. Ortalama birim ısıtma yükü: (22.20 / 275) · 1000 = 81 W/m^2 'dir. Tasarımı yapılan bir odada 15 m^2 'lik

Bu model kullanılarak;

$$T_{max} \cong \frac{(T_p - T_a') \cdot M}{[2 \cdot W \cdot \eta + D_o]} + T_a' \quad [14]$$

Burada η kanatçık verimidir:

$$\eta = \tanh(m \cdot W)/(m \cdot W) \quad [15]$$

$$m = \{q_y/[k_{eq} \cdot T_{total} \cdot (T_p - T_a')]\}^{1/2} \quad [16]$$

$$k_{eq} = \frac{\sum_{i=1}^{n_k} k_i \cdot x_i + L \cdot k}{T_{total}} \quad [17]$$

Burada k_i (i) sayılı malzemenin ısı iletim katsayısıdır.
Yüzey sıcaklık dağılımı ise:

$$T_p'(x) = \frac{\cosh[m \cdot (M/2 - x)]}{\cosh[m \cdot W]} \cdot \{T_{max} - T_a'\} + T_a' \quad ; \quad D_o/2 <= x =< M/2 \quad [18\text{-}a]$$

$$T_p'(x) = T_{pmax} \quad ; \quad D_o/2 >= x \quad [18\text{-}b]$$

Boru dış yüzey sıcaklığı:

$$T_d = T_{max} + q_y \cdot \{R + \frac{(L - D_o/2)}{k}\} \quad [19]$$

Gerekli ortalama su sıcaklığı, T_w ise boru et kalınlığı boyunca ısı iletimi ve boru iç yüzeyindeki ısı taşınımı terimleri ile çözülür:

$$T_w = \frac{q_y \cdot M}{X \cdot \pi} \cdot \{\frac{1}{a \cdot D_i} + \frac{1}{2 \cdot k_h} \ln(D_o/D_i)\} + T_d \quad [20]$$

Burada a, türbülanslı rejimde su ile boru iç yüzeyi ısı taşınım katsayısıdır:

$$a = 1056\,[0.02 \cdot \{T_w + 273\} - 4.06] \cdot v_w^{0.8} / D_i^{0.2} \quad [21]$$

Burada ilk terim doğrusal bir ifade ile ışıma ısı transferini ikinci terim ise taşınım ile ısı transferini ifade etmektedir. İlk terimdeki r çarpanı doğrusallaştırma faktörüdür. Genel işletme koşulları aralığı içersinde bu çarpan T_p ve T_u 'nun bir fonksiyonu olarak ifade edilebilir:

$$r \cong 0.0105 \cdot (T_p+T_u)/2 + 0.7955 \quad 15\,°C <= (T_p+T_u)/2 <= 30\,°C \qquad [11]$$

C_D ise sadeleştirilmiş ışıma katsayısıdır:

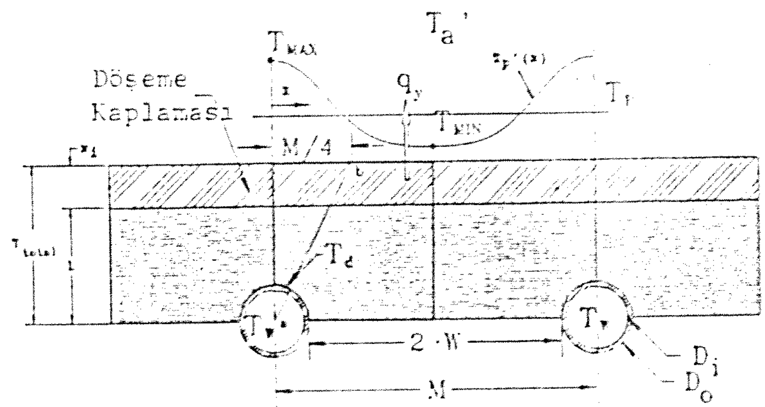
$$C_D = 5.67 \cdot e \qquad A_p/A_u <= 0.30 \qquad [12]$$

Taşınım ile ısı transferi genelde iç mekanın ölçülerine (D_{eq}) ve yapının deniz seviyesinden olan yüksekliğine (h) bağlıdır. Deniz seviyesinden olan yüksekliğin etkisi ile ısı taşınımı azalmaktadır (Oskay ve diğerleri, 1977). Aynı şekilde mekan boyutları büyüdükçe ısı taşınımı azalmaktadır.

$$D_{eq} = (4 \cdot A_r/L_r) \qquad [13]$$

4.2. Kararlı Rejimde Kompozit Kanatçık Modeli

Uygulamada ısıtma yükündeki değişim genellikle su sıcaklığının ayarlanması ile takip edilir, sistem devrede tutulur. Bu nedenle sistemin kararlı rejimde kaldığı kabul edilebilir. Geliştirilen kanatçık modeli Şekil 4 'de gösterilmiştir.



Şekil 4. Kompozit Kanatçık Modeli (Kılkış, 1992).

Pik yük indirim katsayısı, C günlük dış sıcaklık değişimine, yapı elemanlarının ısı iletim katsayılarına, panelin yapısına ve işletme rejimine bağlıdır. Kılkış (1990) bu konuda ilk kez üç ayrı iklim bölgesi ve üç ayrı tip yapı birim kütlesine göre bir çözüm önermiştir. Bu yaklaşım Çizelge 2 'de gösterilmiştir.

Çizelge 2. Pik Yük İndirim Katsayısı.

	C		
	Birim Yapı Kütlesi		
İklim Bölgesi	Hafif	Orta	Ağır
I	1.0	0.88	0.80
II	1.0	0.85	0.78
III	1.0	0.80	0.75

Döşemeden ısıtma verimi:

$$X = \frac{Q_y}{Q_y + \{Q_a + Q_e\}} = \frac{Q_y}{P} \qquad [6]$$

tarifi ile döşemeden ısıtılan iç mekanın kazan (tesis) yükü, P:

$$P = Q_y/X \qquad [7]$$

Eğer ısıtma tesisinde ısı pompası kullanılacak ise gerekli kapasite aşağıdaki iki koşuldan birine göre tesbit olunmalıdır:

$$P_H(T_b) >= P(T_b) \quad \text{destek sistem yok} \qquad [8]$$

$$P_H(T_b) + Q_D = P(T_b) \quad \text{destek sistem var} \qquad [9]$$

Burada Q_D destek sistemin gerekli ısıl kapasitesidir. Bir ısıtılan döşeme yüzeyinden ışıma ve iletim yolu ile iç mekana olacak ısı transferi aşağıdaki şekilde ifade edilebilir (Kılkış, 1990):

$$q_y = r \cdot C_D(T_p - T_u) + (1 - 2.22x10^{-5}h)^{2.627}(4.96/D_{eq})^{0.08} \cdot 2.67(T_p - T_a')^{1.25} \qquad [10]$$

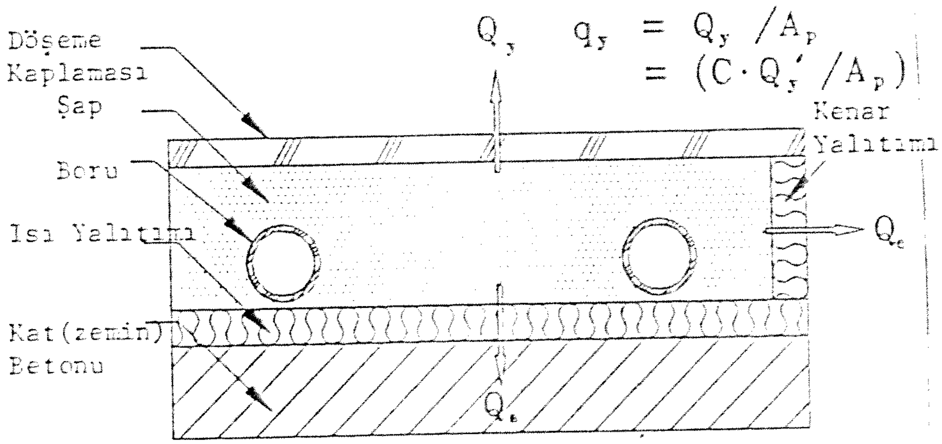
Belirli bir dış hava sıcaklığında, talep edilen su sıcaklığının düşmesi ile ITK artar. Bu örnekte T_{ws}, 55 °C seçildiğinde ve T_o = -5 °C koşulunda, ITK ancak 2 dolayındadır. Boru merkezleri arasındaki mesafeyi azaltarak talep edilen su sıcaklığı azaltılabilir. Örneğin T_{ws} 40 °C olarak tasarlığında ITK 2.5 olmaktadır. Devredeki su sıcaklığı dış hava kompenzasyonlu olarak sürekli değiştirildiğinde ITK değeri optimum değerde tutulabilir. Bu durum Şekil 2 'de kırık çizgiler ile gösterilmiştir. Böyle bir uygulama ile ITK değeri T_o = +5 °C koşulunda 3.3 olmaktadır.

4. TEORİ

4.1. Panel Isı Çıktısı

Döşeme betonu içersine yerleştirilmiş boruların oluşturduğu döşemeden ısıtma panel konstrüksyonuna bir örnek Şekil 3 'de gösterilmiştir. Ortalama birim ısı çıktısı q_y, birim alandaki döşemeden ısıtma yükünü karşılayacak düzeyde olmalıdır:

$$q_y >= Q_y'/A_P \cdot 1000 \qquad [4]$$



Şekil 3. Döşemeden Isıtma Paneli Örnek Konstrüksyonu.

Eğer döşemeden ısıtma sistemi kesintili kullanılmıyor ise panelde, çevre duvarlarda ve tavanda belirli bir ısı depolaması söz konusudur. Bu nedenle ısı yükünün günlük pik değerinde azalma olacaktır. Böyle bir azalmanın söz konusu olduğu durumlarda:

$$Q_y = C \cdot Q_y' \quad ; \qquad C \leq 1 \qquad [5]$$

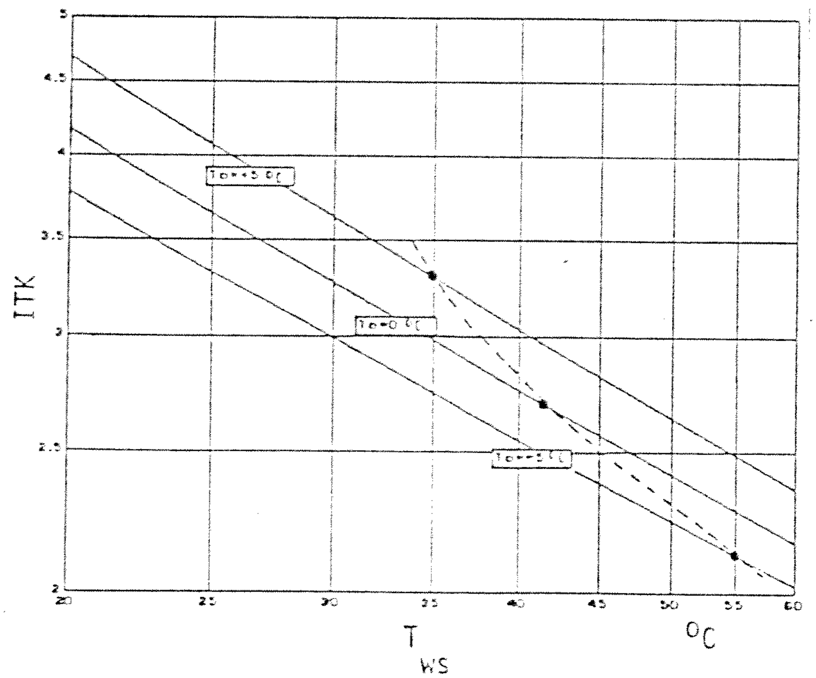
3. HAVADAN SUYA TİP ISI POMPALARI

Havadan suya tip ısı pompalarının Isıtma Tesir Katsayıları, ITK dış hava sıcaklığına duyarlıdır. % 100 dış hava ile çalışan bir ısı pompasının ITK değerleri Çizelge 1 'de verilmiştir (1990a). Örnek uygulama için seçilen ısı pompasının bazı özellikleri: Sıkıştırıcı (Kompresör): Hermetik, Pistonlu; Buharlaştırıcı (Evaporatör): Doğrudan Genleşmeli; Buzlanmayı Giderme Sistemi (Defrost): Ters Dolaşımlı; En Yüksek Su Sıcaklığı: 55 °C, ΔT_{ws}: 10 °C; T_c: -7 °C.

Çizelge 1. Isı Pompası Performansının T_o ve T_{ws} ile Değişimi.

	T_{WS}=25° C		35° C		45° C		50° C		55° C	
T_o	P_H	P_M	P_H	P_M	P_H	P_M	P_H	P_M	P_H	P_M
° C	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW
20	51,9	9,0	50,6	10,6	48,9	12,2	47,9	12,9	46,6	13,7
15	45,5	8,8	44,2	10,3	42.3	11,7	41.0	12,3	39,9	12,9
10	39,3	8,6	37,3	9,9	35,2	11,0	34,2	11,6	33,1	12,0
5	33,1	8,2	31,4	9,3	29,5	10,6	28,4	10,7	27,6	11,1
2,5	30,3	8,0	28,6	9,0	26,7	9,8	25,8	10,3	25,0	10,6
0	27,9	7,7	26,2	8,7	24,5	9,4	23,7	9,8	22,8	10,1
-5	23,7	7,2	22,4	8,0	20,9	8,8	20,2	9,0	19,7	9,3
-10	19,7	6,7	18,8	7,4	17,5	8,0	17,1	8,3	16,4	8,6
-15	16.0	6,1	15,1	6,0	14,3	7,3	13,9	7,5	13,4	13,7

Isı tesir katsayısının değişimi ayrıca Şekil 2 'de gösterilmiştir.



Şekil 2. ITK Değerinin T_{ws} ve T_o ile Değişimi.

mekandaki hava hareketlerinin daha zayıf oluşudur. Bu son faktör insan bedeninden taşınım yolu ile olan hissedilir ısı kayıplarının azalmasına katkıda bulunmaktadır. Bu faktör duvar ve özellikle pencerelerden olan ısı kayıplarının da bir miktar azalmasına yol açar.

2- Tasarım iç sıcaklığının (T_i') daha düşük seçilmesi ve iç hava hareketlerinin zayıf olması nedenleri ile ısı transferi ve infiltrasyon ısı kayıplarında önemli bir azalma söz konusudur. Ayrıca iç mekandaki hava sıcaklığı daha homojen bir dağılım sergiler. Bu ise tavandan olan ısı kayıplarını azalttığı gibi konfor koşullarını daha da iyileştirir (Raiss ve Roedler, 1969). Zmeurenau, Fazio ve Haghighat (1987) döşemeden ısıtılan bir iç mekanın sayısal modellemesine çalışmış ve klasik ısıtma birimleri ile mukayese etmişlerdir. Örnek çalışmalarında döşemeden ısıtma sistemi ile ısı yükünde % 38 azalma olacağı görülmüştür. Bu çalışmalar sonucunda ısı yükünün özel bir yöntemle hesabedilmesi gerektiğini savunmuşlardır. Bu görüşün tasarıma yönelik prensiplerini ise ilk kez Kılkış (1990, 1992) ortaya atmıştır. Bu görüşü destekleyen başka çalışmalar da yapılmıştır (Lafontaine, 1990), (Zuver, 1991), (Buckley, 1989). Bu görüşlere karşı olarak Dale ve Ackerman (1990) 49 m^2 'lik iki deney odasında elde edilen sonuçlara dayanarak döşemeden ısıtma sistemine özgü bir avantaja rastlamadıklarını belirtmişlerdir. Kılkış (1991) bu sonuca muhtemelen yetersiz döşemeden ısıtma tasarımının ve sıcak hava ile ısıtma yapılan mukayese odasında hava hızlarının normal uygulama sınırlarının dışında tutulmasının neden olabileceğini öne sürmüştür.

3- Sistemin ısıtma prensibinin bir özelliği olarak ortalama su sıcaklığı çok nadiren 50 °C 'ı aşar. Zaten döşeme yüzey sıcaklığının da 29 °C ile sınırlı olması su sıcaklığının daha yüksek seçilmesini engeller (Fanger ve diğerleri, 1980). Ayrıca döşemeden ısıtma boru malzemesinin ömrü-su sıcaklığı bağıntısı uyarınca ortalama su sıcaklığının ayrı bir sınırı vardır. Bu sınır termoplastik borular için 60 °C, özel lastik borular için de 70 °C 'dır. Bu nedenlerle sistem daha tasarım sırasında ısı pompasının su sıcaklığına ilişkin kısıtına uyar.

4- Devredeki su sıcaklığının düşük olması, standard su sıcaklık farkının da daha az olmasını gerektirir. Bu nedenle döşemeden ısıtma sistemlerinde bu fark 10 °C alınır. Bu prensip olarak (i) kısıtını kaldırır.

Bu özelliklerin genel etkisi Şekil 5 'de gösterilmiş ayrıca örnek tasarımda izah edilmiştir.

sırf su sıcaklıklarındaki uyumsuzluk nedeni ile devreye girmesi söz konusudur. Bunu önlemek için tasarım sırasında $T_e > T_s$ koşulunun sağlanması gereklidir.

Isı pompasının performansını arttıracak, ısı pompası ile daha uyumlu çalışabilecek ve daha verimli bir alternatif ısıtma sisteminin arayışı içersinde aşağıdaki noktalar göz önünde tutulmalıdır:

-Destek ısıtma sisteminin sistemden çıkarılması. Bu çözüm bir çok kuruluşun gündeminde olmakla birlikte klasik ısıtma sistemleri için henüz çözülmemiştir (Pietsch, 1991).

-Yukarıda sözü edilen kısıtların zayıflatılması veya ortadan kaldırılması. Tesiste bir destek sistem bulunsa bile destek ısıtıcının mevsim boyunca toplam devreye alınma süresinin azaltılması.

Bu amaçlar doğrultusunda çözüm yolları Şekil 1 'de aranabilir:

a- E noktası:

Bu nokta eğer ısıtma yükü (talep) azaltılabildiği taktirde daha düşük sıcaklıklara çekilebilir. Eğer T_s noktası T_b (Dış hava tasarım değeri) noktasından daha önde değilse böyle bir çözüm destek ısıtıcıyı sistemden bertaraf etmeye yeterli olabilir. Aksi durumda ise destek ısıtıcının gerek duyulan kapasitesi azaltılmış olacaktır. Klasik ısıtma birimlerinde T_s noktası her zaman T_b noktasından önde geldiği için destek ısıtıcının devreden çıkarılması, daha büyük bir ısı pompası seçilmedikçe teknik olarak mümkün değildir. Daha büyük ısı pompasının seçimi ise zaten ekonomik rekabet gücü zayıf olan ısı pompası çözümünü gündemden çıkartmaya yeterlidir.

b- S noktası:

- $T_{smax} <= 55$ °C koşulunu sağlayacak bir ısıtma sisteminin seçilmesi,
- $\Delta T_w <= 10$ °C koşulunu sağlayacak bir ısıtma sisteminin seçilmesi.

Bu durumda (ii) kısıtı T_b ile çalıştığı için bu kısıt ortadan kalkmış olacaktır.

c- $T_e > T_s$ koşulunun sağlanması.

Tüm bu amaçlar prensip olarak döşemeden ısıtma sistemleri ile gerçekleşebilir. Bunun nedenleri şu şekilde özetlenebilir:

1-Döşemeden ısıtma sistemlerinde iç mekan tasarım sıcaklığı T_ε, eşdeğer konfor koşulu için standard değerlerden 2 ila 3 °C daha düşük seçilebilir (Kılkış, 1990). Bunun iki ana nedeni sırası ile iç konforun daha çok ışıma ısı transferi ile gerçekleşmesi ve iç

-Kısıt (i): Isı pompasının yoğuşturucu tarafında sağlayabildiği en yüksek su sıcaklığı, T_{smax}:

Genellikle T_{smax} havadan suya tip ısı pompalarında 55 °C ile sınırlıdır. Şekil 1'de, T_{smax} = 55 °C doğrusu ile ısıtıcı birime giriş noktasında gerekli su sıcaklığı çizgisinin çakıştığı nokta T_s için birinci kısıt değerini verir (i). Radyatör ile ısıtılan bir mahalde tasarım şartlarında gerekli su sıcaklığı 90 °C 'a kadar ulaşabilir. Bu örnekte (i) kısıtının öngördüğü T_s +5 °C dolayındadır. Dış sıcaklık tasarım değeri ise -5 °C olarak gösterilmiştir.

-Kısıt (ii): Isı pompası yoğuşturucu tarafındaki su sıcaklık farkı, ΔT_{ws}:

Bu sıcaklık farkı genellikle 10 °C ile sınırlıdır. Halbuki tasarım şartlarında radyatörlü bir sistem için standard değer 20 °C 'dır. Bu sıcaklık farkı hidrolik sistem büyük seçilerek belirli bir ölçüde giderilse de böyle bir seçimin ekonomik olduğu rahatça söylenemez. Bu örnekte çift yakıtlı (Bivalent: örneğin fuel oilli destek kazanı ve elektrik motoru ile tahrik edilen ısı pompası) ve paralel yerleştirilmiş bir yedekli ısı pompalı tesis için kısıt (ii) +3 °C dolayındadır. Ard arda (tandem) dizilmiş bir bileşik sistemde bu kısıt, bir ısı değiştirici kullanılması ile zayıflatılabilse de daha düşük dış hava sıcaklıklarında ısı pompası veriminin zaten düşük olması böyle bir çözümün ekonomikliğini geniş ölçüde zedeler.

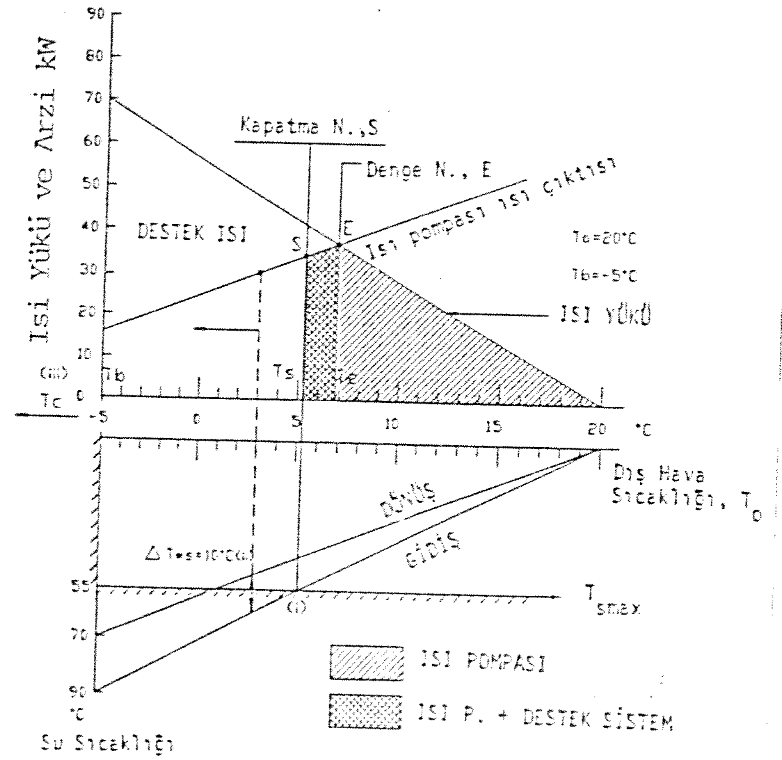
-Kısıt (iii): Dış Hava Sıcaklığı, T_o:

Son teknik gelişmelere rağmen havadan suya tip ısı pompalarında belirli bir devreden çıkma dış hava sıcaklığı T_c bulunmaktadır. Ters dolaşım defrost sistemi ile bu sınır genellikle -10 °C 'a kadar düşebilir (ASHRAE, 1988).

Bu üç kısıt şu şekilde özetlenebilir:

(i)	T_{smax} <= 55 °C	[1]
(ii)	ΔT_{ws} <= 10 °C	[2]
(iii)	T_b >= T_c (-10, -15 °C)	[3]

Bu örnekte ilk kısıt en erken oluştuğu için Kapatma Noktası S +5 °C 'da oluşur (T_s = +5 °C). Bu nedenle ısı pompası destek ısıtıcı ile beraber ancak +7 °C ile +5 °C gibi çok dar bir dış hava aralığında çalışabilecektir. Ard arda dizilmiş bir düzende bu sınır 0 °C 'a kadar inebilir. Uygulamada bu kısıtlardan her hangi birisinin denge noktasından bile önce gerçekleşmesi uygulamada mümkündür. Böyle bir durumda destek ısıtıcının denge noktasından daha önce yani henüz ısı pompasınının ısıtma kapasitesi yeterli iken karşılık ısı pompasının hem ısı çıktısı hemde devreye gönderdiği suyun sıcaklığı azalır. Dış hava sıcaklığına bağlı olarak ısı yükü ile ısı pompasının ısı arzı arasındaki bu uyumsuzluk nedeni ile belirli bir denge noktasından sonra (T_e) devreye bir destek ısıtıcının alınması kaçınılmazdır. Şekil 1 havadan suya tip bir ısı pompasının radyatörlü ısıtma sistemi ile birlikte kullanılması durumundaki ısı arzı ve talebi arasındaki çelişkiyi göstermektedir. Bu çelişki teknik nedenlerle giderilemediği için havadan suya tip ısı pompalarının klasik ısıtma sistemleri ile birlikte kullanımında ciddi kısıtlamalar devam edecektir. Bu kısıtlamaların hafifletilmesi ise ısı pompalarının daha uyumlu alternatif ısıtma birimleri ile eşleştirilmesine bağlıdır.



Şekil 1. Dış Hava Sıcaklığı T_o 'a Bağlı Olarak Isı Pompasının Isı Çıktısı ile Isı Yükünun Mukayesesi.

Bu örnekte ısı arz ve talebinin denge noktasındaki dış hava sıcaklığı +7 °C dolayındadır. Dış hava sıcaklığı tasarım değeri T_b 'nin denge sıcaklığından daha düşük olduğu yörelerde bir destek sistemin ısı pompalı ısıtma tesisine ilavesi gereklidir[1]. Destek ısıtıcının devreye girmesinden sonra ısı pompası bir süre daha ısıtmaya katkıda bulunabilir. Belirli bir dış hava sıcaklığının altında ise ısı pompasının kapatılması zorunlu hale gelir. Kapatma noktasındaki dış hava sıcaklığı T_s, aşağıdaki üç kısıttan en erken gerçekleşeni ile belirlenir:

[1] Isı Pompalarına İlişkin Terim ve Tarifler için Türk Standardı, "Isı Pompaları: Mekanik Tahrikli - Terim ve Tarifler"e bakınız.

T_{set} Termostat iç sıcaklık ayar noktası, °C

T_{smax} Isı pompasının sağlayabildiği en yüksek su sıcaklığı, °C

T_{total} Kanatçık (döşeme örtüleri dahil) toplam kalınlığı, m

T_u Isıtılan mekan zarfının alan-ortalaması iç yüzey (ısıtma yapılan döşeme yüzeyi hariç), sıcaklığı, °C

T_w Döşemeden ısıtma devresindeki ortalama su sıcaklığı, °C

T_{ws} Isı pompası yoğuşturucu tarafındaki ortalama su sıcaklığı, °C

v_w Döşemeden ısıtma borusu içersinde ortalama su hızı, $m \cdot sn^{-1}$

x Döşeme yüzeyindeki bir noktanın boru merkezinden geçen düşey düzlemden uzaklığı, m

x_i (i) sayılı döşeme örtüsünün kalınlığı, m

X Döşemeden ısıtma verimi, boyutsuz

W Komşu borular arası mesafe (açıklık) $(M-D_o)/2$, m

Z Standart boru uzunluğu, m

Δ Sıcaklık farkı, °C

ΔP Basınç kaybı, mm·ss, kPa

2. GİRİŞ

Primer enerjinin yaklaşık üçte biri binalarda tüketilmektedir. Ülkemizde bu payın önemli bir bölümü ise ısıtmada kullanılmaktadır. Bu nedenle atık ve alternatif enerji kaynaklarının ısıtmada kullanımı enerji verimliliği ve çevrenin korunması yönlerinden önemli bir potansyele sahiptir. Gerçekçi ve uygun bir tasarım yapıldığı taktirde döşemeden ısıtma sistemlerinde % 40 'a varan bir enerji tasarrufu mümkündür. Bu önemli avantajın yanısıra sistem, atık ve alternatif enerji kaynaklarının düşük sıcaklıklarında rahatça çalışabilmekte ayrıca ısı pompalarının daha verimli ve etkin çalışmasına büyük ölçüde katkıda bulunmaktadır. Bunun anlamı ise döşemeden ısıtma sistemlerinin uygulanması ile primer enerjinin daha rasyonel kullanımı ve dolayısı ile çevrenin daha iyi korunmasıdır. Atık ve jeotermal gibi alternatif enerji kaynaklarının döşemeden ısıtmada kullanımında genellikle ısı pompasına gerek duyulmaz. Ancak dış hava, toprak ısısı gibi daha düşük yoğunluk ve sıcaklıklardaki enerji kaynakları söz konusu olduğunda ısı pompalarının kullanımı gündeme gelmektedir. Isı pompaları ile daha iyi bir uyumun söz konusu olması nedeni ile döşemeden ısıtma sistemleri ısı pompası performansını önemli ölçüde arttırmaktadır. Bu nedenle birincil enerji kaynaklarının kullanımındaki verimlilik daha da artar.

Havadan suya tip bir ısı pompası ele alındığında dış hava sıcaklığının azalması ile ısı yükü artar. Bu yükü karşılamak için de su sıcaklığının arttırılması gerekir. Buna

k_{eq}	Döşeme ve döşeme örtülerinin (kaplamalarının) yatay yöndeki eşdeğer ısı iletim katsayısı, W/m K
k_h	Boru malzemesinin ısı iletim katsayısı, m
L	Döşeme malzemesi kalınlığı (döşeme örtüleri hariç), m
L_T	Isıtılan iç mekanın döşeme çevresi uzunluğu, m
m	Kanatçık katsayısı, m^{-1}
M	Boru merkezleri arası mesafe, m
n_k	Döşeme örtüsü sayısı, boyutsuz
P	Kazan (ısıtma tesisi) ısı yükü, kW
P_H	Isı pompasının ısı çıktısı, kW
P_M	Isı pompası tahriki için gerekli elektromekanik güç, kW
q_y	Panelin birim yüzey alanındaki ısı çıktısı, W/m^2
Q_a	Isıtılan döşeme altından olan ısı kaybı, kW
Q_D	Destek sistemin ısıtma kapasitesi, kW
Q_e	Isıtılan döşeme çevresinden olan ısı kaybı, kW
Q_{is}	Klasik bir ısıtıcı birim ile ısıtılan iç hacmin ısı yükü, kW
Q_y	Döşemeden ısıtılan bir iç hacmin pik yük indiriminden sonraki (eğer ısıtma rejimi uygun ise) ısı yükü, kW
Q_y'	Döşemeden ısıtılan bir iç mekanın ısı yükü, kW
r	Işıma ile ısı transferi denklemini doğrusallaştırma faktörü, $°C^3$
R	Döşeme kaplamalarının toplam ısıl direnci, m^2K/W
Re	Döşemeden ısıtma borusunda Reynolds sayısı, boyutsuz
T_a	İç mekan tasarım sıcaklığı, °C
T_a'	Döşemeden ısıtmada iç mekan tasarım sıcaklığı, °C
T_b	Dış hava sıcaklığı tasarım değeri, °C
T_c	Isı pompasının devreden çıkma dış hava sıcaklığı, °C
T_d	Döşemeden ısıtma borusu dış yüzey sıcaklığı, °C
T_e	Isı pompasının ısı çıktısı ile ısı yükü denge durumundaki dış hava sıcaklığı, °C
T_e'	Döşemeden ısıtma yapılan bir iç mekan için denge noktası dış hava sıcaklığı, °C
T_{max}	Maksimum panel yüzey sıcaklığı, °C
T_{min}	Minimum panel yüzey sıcaklığı, °C
T_o	Dış hava sıcaklığı, °C
T_p	"Homojen" panel yüzey sıcaklığı, °C
$T_p'(x)$	Döşeme yüzeyi yerel sıcaklığı, °C
T_s	Klasik bir ısıtıcı birim kullanıldığında ısı pompası kapatma noktasındaki dış hava sıcaklığı, °C
T_s'	Döşemeden ısıtma sistemi kullanıldığında kapatma noktasındaki dış hava sıcaklığı, °C

DÖŞEMEDEN ISITMA SİSTEMLERİNDE ENERJİ TASARRUFU: TEORİ, MODELLEME VE BİR ÖRNEK ÇALIŞMA

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ÖZET

Bu tebliğde döşemeden ısıtma sisteminin enerji tasarrufuna katkıları tanıtılmakta, alternatif enerji kaynaklarının ve özellikle ısı pompalarının ısıtma sektöründe daha etkin, yaygın ve verimli bir şekilde kullanımına bu yöntemin ne ölçüde yararlı olabileceği bir örnekle incelenmektedir. Bu sistemin enerji tasarrufu ve çevrenin korunmasındaki önemi, sistemi bütünü ile ele alan bir model ve standart tasarım algoritmasının bulunmayışı sonucu uygulamaya yeterince yansımamaktadır. Bu nedenle TÜBİTAK, MİSAG-12 sayılı proje çerçevesinde yeni bir model geliştirilerek bilgisayar programları yazılmış ve Türk Standardları hazırlanmıştır. Tebliğde hazırlanan model ve bilgisayar yardımı ile tasarım programı tanıtılmaktadır. Toplam 320 m^2 alana sahip, iki katlı bir yapıda, havadan suya tip bir ısı pompasının döşemeden veya radyatörle ısıtma seçeneklerinden birisi ile kullanılması arasındaki fark, ısı pompasının performansı, ekonomikliği ve enerji tasarrufu yönlerinden mukayese edilmiştir. Bu çalışma, döşemeden ısıtmada ısı pompasının destek ısıtıcıya gerek duymadığını, performansının önemli ölçüde arttığını ve genelde ısı pompalarının ekonomik ve teknik yönden ancak bu seçenekle avantajlı olduğunu göstermiştir.

1. SİMGELER

a	Su ile boru iç yüzeyi arasındaki ısı taşınım katsayısı, W/m^{2}K
A_p	Isıtmanın yapıldığı döşeme kesiminin (panel) yüzey alanı, m^2
A_r	Isıtılan iç mekanın toplam döşeme alanı, m^2
A_u	Isıtılan iç mekanı çevreleyen bütün yüzeylerin alanı, m^2
C	Pik ısı yükü indirim katsayısı, boyutsuz
C_D	Işıma ısı transfer katsayısı, W/m^{2}K^4
D_{eq}	Isıtılan iç mekan döşemesinin eşdeğer çapı, m
D_i	Döşemeden ısıtma borusu iç çapı, m
D_o	Döşemeden ısıtma borusu dış çapı, m
e	Yüzey yayınım katsayısı, boyutsuz
h	Deniz seviyesinden itibaren yükseklik, m
ITK	Isı pompası ısıtma tesir katsayısı, boyutsuz
k	Şap malzemesi ısı iletim katsayısı, W/mK

This is especially true when there is a need for a new boiler for the existing system which itself needs substantial overhauling or too expensive to operate as it is. Therefore over a large range of heating sector, floor heating is becoming a key element in enhancing the technical and economical feasibility of heat pumps. This is especially true for air to water type heat pumps where their feasibility might be already marginal. Due to its heat load reducing feature and other merits,floor heating systems can also be used in regions where an air to air type heat pump can not be operated successfully. The case study clearly demonstrated that it can also utilize heat pumps at higher COP values as well as eliminating the need for the supplementary boiler. These are important issues for the heat pump industry (Pietsch,1991). However in order to realize these benefits, a careful and detailed analysis of the radiant heating load,heat transfer,performance of the heat pump and the radiant floor has to be made. In order to maximize the heat pump output and optimize its match with the radiant floor heating system, it is also essential to simulate the indoors as well as the heating system on an hourly basis including the heat storage analysis. This definitely requires a sophisticated and flexible computer program. However it is definite that such an effort will be very rewarding and promising for the future of air to water heat pumps.

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7.ACKNOWLEDGEMENT

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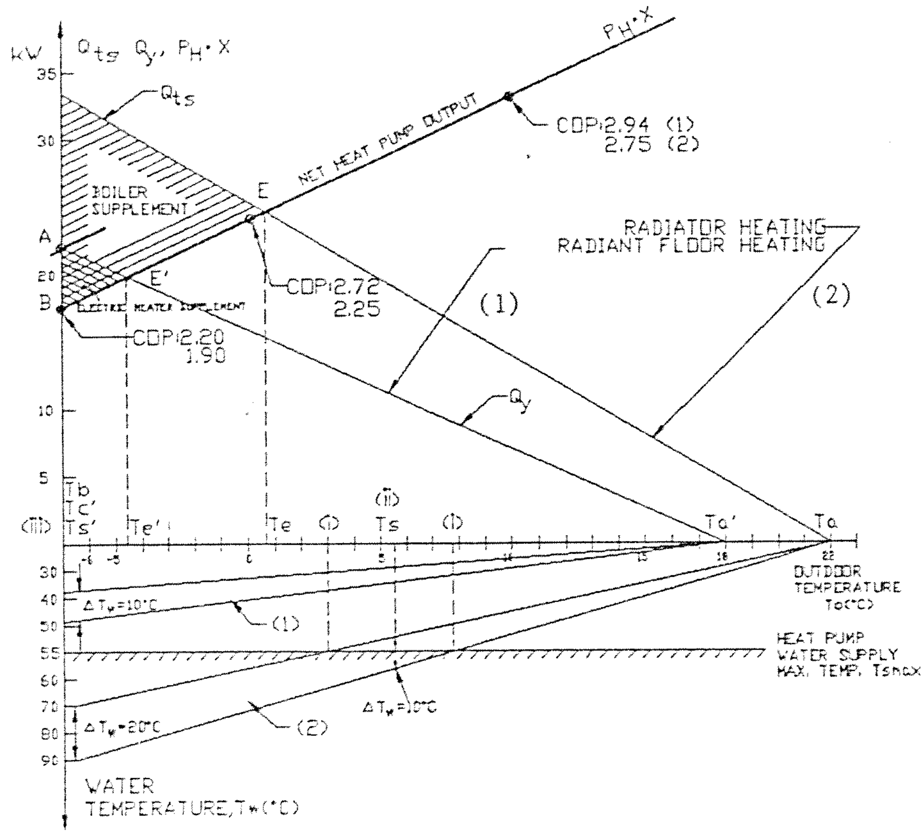


Figure 6. Comparison of the heating characteristics of radiant floor heating system and a radiator system.

can be eliminated. Otherwise it would be at around necessitating an early call of the supplementary r. Some typical COP values are also shown sponding to each heating system. As seen, COP s for radiant floor heating case are higher by as much 5%. T_{ws} has to be maintained at 55°C for the radiator ng case for most of the time.

re 7 shows the general layout of the radiant floor ng system. A four way valve continuously modulates vater temperature. The indoor air temperature also ols the heat pump on an on-off basis. However "off" ls are delayed through a preset timer so that the slab aintained at a temperature corresponding to a erate base load. Another way to maintain this condition directly monitor the slab temperature so that it can be above a certain temperature temperature.

ISCUSSION OF RESULTS AND CONCLUSION

Nearly one out of every three new single family homes in the United States today are equipped with an ric heat pump for heating and cooling (Lannus,1989). ldition to this,an increasing number of existing homes being retrofitted with heat pumps. If air to water type pumps operate with radiant floor heating systems,one ne key objectives which is the elimination of supplementary boilers can be easily realized . Besides new buildings, radiant floor heating can be a retrofit alternative for existing ones,because with thin slab or under subfloor type of applications, retrofitting is relatively easy and cheap.

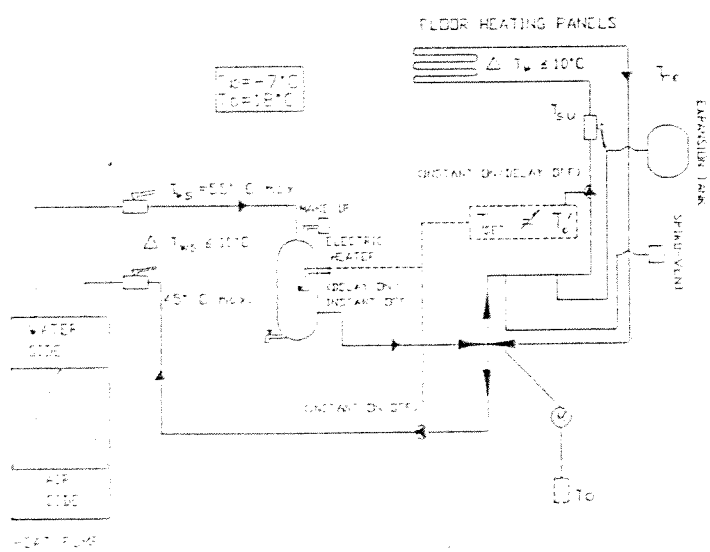


Figure 7. Design layout of the case study.

Table 2. Dependence of Heat Pump Performance on T_o and T_{ws}.

	T_{ws}=25°C		35°C		45°C		50°C		55°C	
T_o °C	P_H kW	P_M kW	P_H kW	P_M kW	P_H kW	P_M kW	P_H kW	P_M kW	P_H kW	P_M kW
20	51,9	9,0	50,6	10,6	48,9	12,2	47,9	12,9	46,6	13,7
15	45,5	8,8	44,2	10,3	42.3	11,7	41.0	12,3	39,9	12,9
10	39,3	8,6	37,3	9,9	35,2	11,0	34,2	11,6	33,1	12,0
5	33,1	8,2	31,4	9,3	29,5	10,6	28,4	10,7	27,6	11,1
2,5	30,3	8,0	28,6	9,0	26,7	9,8	25,8	10,3	25,0	10,6
0	27,9	7,7	26,2	8,7	24,5	9,4	23,7	9,8	22,8	10,1
-5	23,7	7,2	22,4	8,0	20,9	8,8	20,2	9,0	19,7	9,3
-10	19,7	6,7	18,8	7,4	17,5	8,0	17,1	8,3	16,4	8,6
-15	16,0	6,1	15,1	6,0	14,3	7,3	13,9	7,5	13,4	13,7

Information provided in Table 2 is also exhibited in Figure5.

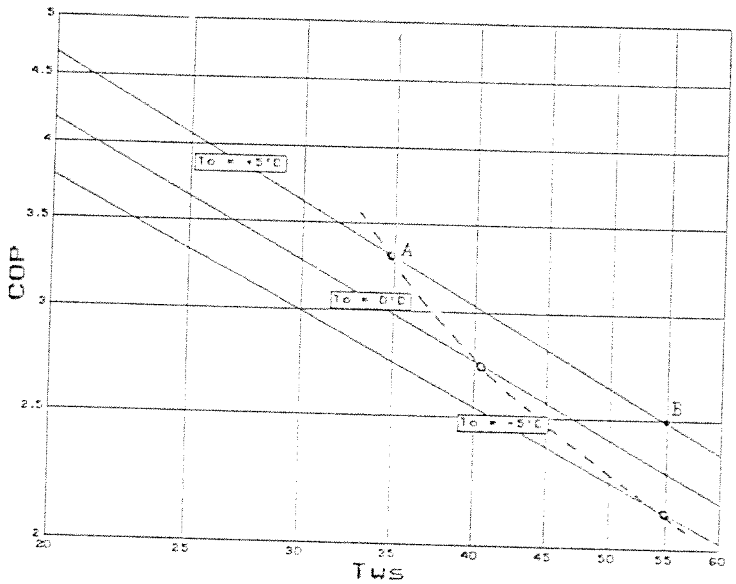


Figure. 5 Typical Dependence of COP on T_{ws} and T_o.

As expected and already known, any decrease in the supply water temperature requirement increases the COP value at a given outdoor temperature T_o. For the given heat pump,COP can barely reach 2 at T_{ws}=55°C and T_c=-5°C. By decreasing the hose spacing,T_{ws} can be decreased. i.e. if T_{ws} can be set at 40°C then COP will increase to 2.5. Being able to continuously adjust the water temperature in a radiant panel circuit with respect to outdoor climatic conditions,it is possible to obtain the maximum possible COP values as shown by the broken line in Figure 5 . With this strategy, the COP value at T_o=+5°C will reach 3.3 by modulating T_{ws} down to 35°C (Point A). Otherwise COP would only increase to 2.5 at T_{ws}= 55°C (Point B). Also it is evident that COP values beyond T_{ws}=60°C will be quite unsatisfactory.

3.4 Computer Program

A computer program was developed in order to exec the given algorithm. This program is sensitive to all releva slab construction data,information on back and ed insulation, temperature of the space or grou below,thermal properties of the hose,spacing of the ho and depth from the surface. It determines the surfa temperature distribution and calculates also the pressu drop,and required water flow rate as well as its avera temperature for given design conditions. Heat loads a calculated in accordance with the provisions described this article. By any suitable optimization technique it possible to determine the hose spacing for the best fit the heating load and the maximum COP (broken line Figure 5) lines. The next step in programming will be complete package for optimum sizing of the heat pump a the radiant floor heating system which will also determi the best operating and control strategies for a speci design at given meteorological conditions on an hou basis.

4. CASE STUDY

A new, two story residential house located in a sea si region is going to be heated in Winter. The building has conventional heat load of 29 kW at a design outdo temperature of -3°C. The standard indoor desig temperature is 22°C. There are 320 m2 floor space,and flo heating will be performed over 275 m2. In order to cover th entire operational range of the heat pump in the analysi design condition was extrapolated to the cut-off point at 7°C. At this outdoor temperature the conventional heat lo will be 33.64 kW. For an equivalent comfort, T_a' wa selected as 18°C (Kilkis,1990). The radiant heating load 25.23kW. Indoor thermal comfort is controlled by wat temperature modulation. This enables to shave off th radiant floor heating load by a C value of 0.88 (Se Table1). After peak load shaving through Equation 5, th design heating load reduces to 22.20 kW. This is 32% les than the conventional heating load. The average heat loa intensity will then be 22.20/275•1000 = 81 W/m2. A typica radiant floor heating circuit covers 15 m2 on the floo surface. Thermal resistance of the floor covering is 0.04 m2 K/W. Rubber hose with 0.0095 m I.D. and 0.016 m O was selected as energy transfer medium. The compute program revealed that the average water temperature w only be 42°C at the cut-off point of the heat pump(-7°C Sine Curve Model predicts the same temperature a 40.5°C. Optimum hose spacing was determined to be 0.2 m. The water supply temperature requirement, T_{ws} will b (42+10/2)= 47°C. A comparison of the characteristics of th two heating systems is shown in Figure 6. Interpolation o the values provided in Table 2 columns 3 and 4 reveals tha the selected heat pump capacity will be only 3 kW short o the radiant heat load at the cut-off point as shown in Figur 6 (Point A and B). As the cut off temperature T_c is below T supplementary heater is, in fact, not necessary for the actua outdoor design condition. If a conventional heating syster would be used, with the line Q_{ts}, the equilibrium point wi be at +1°C. The supplementary boiler requirement woul then be 9 kW (29kW - 20kW) at actual outdoor desig conditions, or about 12 kW at the cut off temperature. B oversizing the hydronic system at an extra expense, shut o

η is the fin efficiency:

$$\eta = \tanh(m \cdot W)/(m \cdot W) \tag{15}$$

$$m = \{ q_y/[k_e \cdot T_{total} \cdot (T_p - T_a')] \}^{0.5} \quad \text{for } T_p > T_a' \tag{16}$$

$$k_e = T_{total}/R \tag{17}$$

emperature profile is given by the following equation:

for $D_o/2 \le x \le M/2$:

$$(x) = \frac{\cosh[m.(M/2-x)]}{\cosh[m \cdot M/2]} \cdot \{T_{max} - T_a'\} + T_a' \tag{18-a}$$

for $x < D_o/2$:

$$(x) \cong T_{max} \tag{18-b}$$

ad of employing a piecewise expression for the erature profile,Equation 18-a may be activated over ntire range provided that D_o term is dropped in tion 14. However this generates a discontinuity at x=0. ollowing basic slab data was taken from a case design e details will be given in the following sections:

$_y$ = 81 W/m²	A_p = 15.0 m²	D_o = 0.016 m
$_a'$= 18.0°C	M = 0.20 m	L = 0.040 m
$_u$ = 17.0°C	k = 1.4 W/m K (concrete)	
$_p$ = 25.4°C	x_i = 0.005 m (one floor cover)	
	k_i = 0.116 W/m K (carpet)	

the computer program developed for this model, the ing results were obtained:

$_{ax}$ = 31.1°C $\quad T_{min}$ = 22.4°C

lot of Equation 18 is given in Figure 4.
bitrary sinusoidal temperature distribution curve is plotted for comparison purposes with the following dary condition:

$$T_p'(x) = T_p \quad \text{at } x = M/4$$

this Sine function becomes:

$$T_p'(x) = T_p - A \cdot \sin[\pi/2 + (2 \cdot \pi \cdot x)/M] \tag{19}$$

ler to assign a unique curve,the amplitude A must be d by using a second boundary condition which is not y available. Otherwise, there will be infinite number of ons including the trivial solution namely A=0 (Uniform e temperature). In order to compare the profile of the urves, the boundary condition $T_p'(M/2) = T_{min}$ was arily imposed. The piecewise temperature profile ied by the fin model is revised by a second degree omial with the following boundary conditions in the $0 <= x <= D_o/2$:

$D_o/4$; $T_p'(x) = T_{max}$ and at x=0; $dT_p'(x)/dx = 0$.

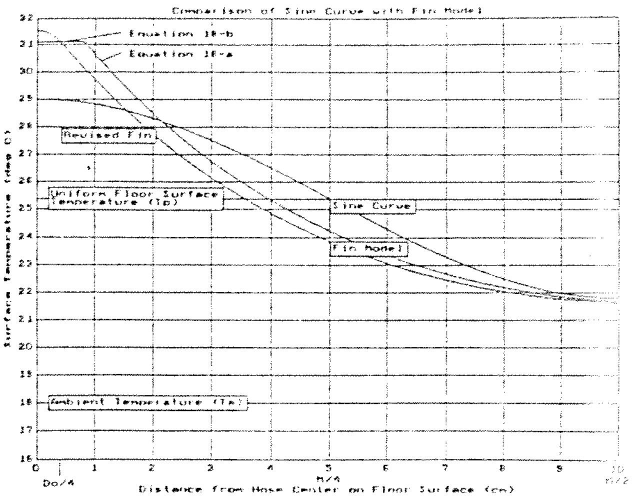


Figure. 4 Surface Temperatures Predicted by the Fin Model and the Sine Curve with an arbitrary boundary condition at x=M/2.

Using T_{max} and taking the shortest path for heat diffusion :

$$T_d \cong T_{max} + \frac{q_y \cdot \{ T_{total} - D_o/2 \}}{k_e} \tag{20}$$

Finally the required average water temperature,T_w will be.

$$T_w = \frac{q_y \cdot M}{X \cdot \pi} \cdot \left(\frac{1}{a \cdot D_i} + \frac{1}{2 \cdot k_h} \ln(D_o/D_i) \right) + T_d \tag{21}$$

Here a is the convection heat transfer coefficient between the water and hose wall. Usually hose Reynolds Number in radiant floor heating systems range between 10000 and 25000,therefore the Turbulent Flow model can be employed (Kreider and Kreith,1981):

$$a = 1056 \cdot [0.02 \cdot (T_w + 273) - 4.06] \cdot v_w^{0.8}/D_i^{0.2} \tag{22}$$

3.3 Air to Water Type Heat Pumps

Unless a waste heat like from the indoor exhaust air or an industrial combustion product is available,an air to water commercial heat pump relies on the outdoor air as the source of low temperature heat on its evaporator side. On its condenser side, heat is delivered to the water which is used as indoor space heating medium. The heating coefficient of performance,COP is pretty sensitive to the outdoor air temperature even though precautions like preheating the outdoor air by indoor exhaust air or a dual fuel system are taken. For 100 % outdoor air, COP values of a typical heat pump may be calculated from Table 2 (1990a). This heat pump has the following features:
Compressor:hermetic,reciprocating,
Evaporator: direct expansion type,
Defrost :cycle reversal,
Maximum supply water temperature, T_{smax} = 55°C, ΔT_{ws} = 10°C; T_c=-7°C

where the floor surface temperature $T_p'(x)$ is uniform, and equal to T_p, the amount of total heat that can be transferred to the indoor space from the unit floor surface area will be (Kilkis,1990):

$$q_y = r \cdot C_D \cdot (T_p - T_u) + (1 - 2.22 \times 10^{-5} h)^{2.627} (4.96/D_e)^{0.08} \cdot 2.67 (T_p - T_a')^{1.25} \quad (10)$$

r may be expressed by a linear function of T_p and T_u in the practical range of radiant floor heating:

$$r = 0.0105 \cdot (T_p + T_u)/2 + 0.7955 \quad \{15°C \le (T_p + T_u)/2 \le 30°\} \quad (11)$$

C_D may be simplified if the ratio of radiant floor area to total area of the indoor space envelope does not exceed 0.30 (Raiss and Roedler,1969):

$$C_D = 5.67 \cdot e \qquad \{A_p/A_u \le 0.30\} \qquad (12)$$

The second term in Equation 10 is the convective term with altitude (h) and room size (D_e) corrections.

$$D_e = (4 \cdot A_r / L_r) \qquad (13)$$

The average of the indoor surface temperatures of the partitions excluding the radiant floor surface, T_u is rather difficult to calculate. It primarily depends on T_o,the area ratio of indoor and outdoor partitions,thermal properties and dimensions of these partitions,position of the room in the building,and the outdoor wind conditions. An elaborate numerical algorithm was provided by Zmeureanu,Fazio and Haghighat(1987). Once T_a' and T_u are determined,T_p can be solved for a required heat output intensity. Equation 10 presumes that T_p is uniform which can only be true for infinite number of hoses. Therefore further elaboration is essential before calculating the required average water temperature to maintain the required heat output. Depending upon the slab construction, required heat output,indoor temperature as well as the spacing between the heat transfer hoses,the floor surface temperature definitely exhibits a periodic fluctuation. Before starting to calculate the heat transfer in the slab,this surface temperature profile has to be predicted. Models to explain and predict this temperature swing are few. One of the earliest model was given by Kollmar (1957). His model assumes that the slab operates as a fin and looses heat from the upper surface. Zhang and Pate (1986) developed a finite difference algorithm for ceiling heating panels in order to solve the steady and transient heat diffusion problem in the slab and predict the local surface temperatures. Krinninger (1989) and Leal and Miller (1972) claim that the temperature profile is simply a Sine curve. Krinninger (1989) states that lengthy numerical algorithms may improve the solution for T_w only by a few degrees Centigrade, and he simplifies the calculations by assuming that at a point $x = M/4$ on the floor, local surface temperature $T_p'(x)$ coincides with T_p (See Figure 3). However this assumption is insensitive to slab thickness, heat output intensity, indoor air temperature, thermal properties of the slab and floor coverings. Furthermore can not provide information like maximum and minimu surface temperatures because it lacks a boundary conditio for assigning a unique Sine Curve passing through th point ($M/4, T_p$). Kilkis developed a more elaborate fin mod which is useful both for steady state in-slab heat transfe calculations and prediction of the local surfac temperatures. This model is explained in the followin section.

3.2 Steady State Plate Fin Model

An actual radiant panel heating system frequentl operates in an unsteady manner where the wate circulation is interrupted by the indoor temperature se control. However if the peak load shaving feature is desire then the system must have a continuous circulation wit temperature modulation. In making such a choice,th amount that the peak design load may be shaved i important. This type of control strategy and the simplicity o the solution favor a steady state solution for desig purposes. Steady state plate fin model is shown in Figure 3 Part of the slab that remains above the hose centers wher hoses partially penetrate into it and floor covering(s) abov it establish a plate fin which looses heat from the to surface both by radiation and convection (Equation 10 Due to symmetry, there is no lateral heat flow across th mid plane between adjacent hoses. Edges are assumed t be totally insulated. Calculations do not need to conside back losses here,because they are accounted separatel through the definition of heating efficiency which effects th plant load (Equation 7) and required wate temperature(Equation 21). Hose surfaces which are i contact with the fin are assumed to be isothermal (T_d).

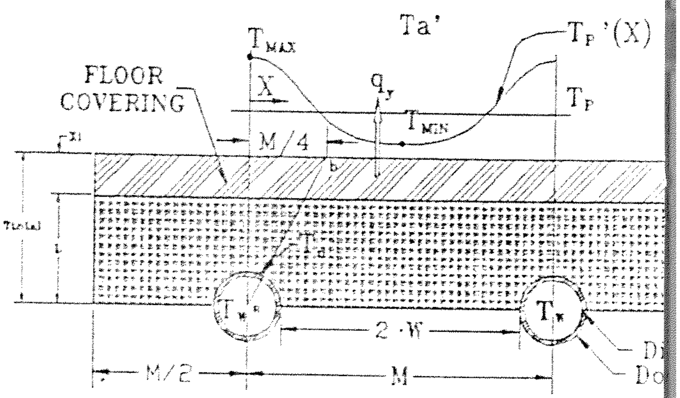


Figure 3. Plate Fin Model (Kilkis,1989).

Using this model:

$$T_{max} \cong T_a' + \frac{(T_p - T_a') \cdot M}{[2 \cdot W \cdot \eta + D_o]} \qquad (14)$$

This equation assumes that the floor surface temperature over an area which has a width of D_o is constant and the thermal resistance of the floor coverings does not exceed approximately 0.3 m^2 K/W.

ative. Their comparison indicated that radiant panel g is more economical and the peak load is lower by han for the conventional design. They concluded that diant panel heating load should be calculated in a nt manner. The first design oriented principles of this n were given by Kilkis (1990). There are other tical and experimental evidences supporting the opinion (Lafontaine,1990), (Zuver,1991), (Buckley, However Dale and Ackerman (1990) published their egarding two experimental homes having 49m² floor each and claimed that no substantial difference was between the warm air and radiant panel heating. (1991) argued that this conclusion may be the result ifficiency in the construction of the floor heating panel nusual air circulation rates maintained in warm air g case.

a design and operational requirement of the system, erage water design temperature in the circuit hardly ds 50°C due to large radiant surfaces and reduced oads. Another factor which already enforces such a limit is the comfort limit for the human foot. rmore, the sustained average water temperature may xceed 60°C for thermoplastic pipes and 75°C in ally formulated rubber hoses. Therefore the system y complies with the condition $T_{su} \leq 55°C$ at its design

to low water temperatures maintained in the circuit, nperature drop has to be small. Therefore it is already eral practice to keep ΔT_w at 10°C at design conditions. ill virtually eliminate constraint (i).

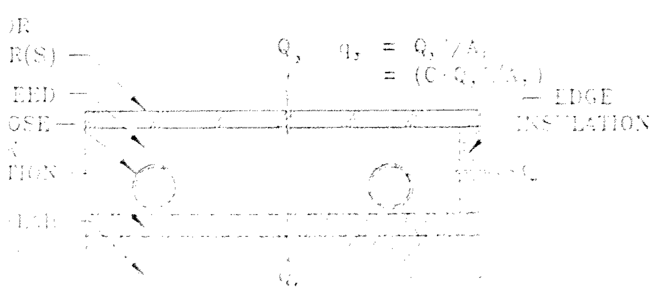
he combined effect of these attributes are shown in 6 and explained in the case study which is given in lowing sections.

EORY

eat Output From a Radiant Slab

cal In-Slab type radiant floor construction is shown in 2. Specially manufactured hoses made from rubber special composition,different types of thermoplastic or metal pipes can be used for the transfer of heat. aying the hose or pipe at a predetermined pattern and ng,screed is poured . In order to minimize heat losses the slab,back and edges are insulated in order to ize the required heat pump capacity. The required output intensity of an indoor space is:

$$q_y = Q_y'/A_p \cdot 1000 \qquad (4)$$



gure.2 A typical radiant floor heating slab

Radiant floor elements are designed to meet q_y. In calculating Q_y' all the attributes of radiant floor heating has to be taken into account. This reduced peak design load may be further shaved off by the heat stored in the radiant slab, and partly in enclosing walls, and the ceiling provided that the heating system is maintained at a base load. If this is the case, Q_y' may be reduced by the Peak Load Shaving Factor,C:

$$Q_y = C \cdot Q_y' \qquad (5)$$

This factor depends on many factors like the climate, typical daily outdoor temperature swings,heat storage capability of the building elements, in addition to the thermal properties and dimensions of the slab. By considering three distinct climatic conditions and distinguishing three different building thermal masses, Kilkis(1990) formulated an engineering rule for determining C as reproduced in Table1. A detailed analysis is a must in order to customize C for a given building and local climatic conditions. Therefore caution must be exercised in manipulating this table for specific designs. Here, light construction class corresponds to a building mass less than 6000 N/m² and heavy construction class corresponds to a building mass greater than 14000N/m².

Table 1. Peak Load Shaving Factor.

	C		
	BUILDING MASS		
LOCATION	Light Constr.	Regular Constr.	Heavy Constr.
Sea Side	1.0	0.88	0.80
Moderate	1.0	0.85	0.78
Cold, Mountain	1.0	0.80	0.75

Defining the Heating Efficiency, X as:

$$X = \frac{Q_y'}{Q_y' + \{Q_a + Q_e\}} \quad , \quad \text{or } X = \frac{Q_y}{Q_y + \{Q_a + Q_e\}} \qquad (6)$$

Plant load of the heated space at design conditions will be:

$$P = Q_y'/X \quad \text{or} \quad P = Q_y/X \qquad (7)$$

To meet this load, the heat pump may be sized according to one of the following criteria :

$$P_H(T_b) \geq P(T_b) \quad \text{without supplementary plant} \qquad (8)$$

$$P_H(T_b) + Q_D = P(T_b) \quad \text{with supplementary plant} \qquad (9)$$

Here Q_D is the supplementary plant (Boiler) capacity. The conditions given by Equations 8 and 9 are shown by points A and B respectively in Figure 6. In radiant floor heating case , T_s will always be below T_e. Therefore, there is the complete freedom to choose and implement any optimum load share between the heat pump and the supplementary boiler, or eliminate the boiler. The intensity of heat output of a given radiant slab depends on the hose spacing, average water temperature,size of the indoor space, altitude,and the measured indoor air temperature. For an ideal condition

Unless the hydronic system is oversized to match the 10°C limit, or a heat exchanger is placed between the heat pump and the heating system circuit in a parallel mode, constraint (ii) will determine the operating conditions. In this example constraint (ii) occurs at +3°C. In a tandem operation,this constraint will be ineffective, unless the elimination of the supplementary boiler is the primary design objective.

-Constraint (iii): Outdoor Design Temperature, T_b

In spite of several advances,a cut-off point C and a corresponding cut-off outdoor air temperature T_c still exists for the protection of the evaporator side from freezing. This limits the minimum outdoor temperature at which an air to water type heat pump can operate. With a reverse cycle defrosting (ASHRAE,1988) this limit may be as low as -10°C or even lower, -15°C. There are more favorable solutions like dual fuel heat pumps where a second primary energy source is utilized for supplementary heating and defrosting. In this case, constraint (iii) becomes the subject of economical parameters.

These three potential constraints on the heat pump operation can be summarized by the following set of inequalities:

(i) $T_{su}(T_c) \leq T_{smax}$ (~55°C) or $T_{re}(T_c) \leq T_{smax}$ (1)

(ii) $\Delta T_w \leq 10°C$ (ΔT_{ws}) (2)

(iii) $T_b \geq T_c$ (-10 to -15 °C) for reverse cycle defr. (3)

In this illustration,constraint (i) occurs first and determines the Shut Down Point S. For a parallel mode, it is at +5°C. For tandem mode,it occurs at +1°C. In all cases, the heat pump will be able to operate together with the supplementary boiler in a limited range, namely between +7°C and +5°C (or +1°C).

If any of these constraints occur to the right side of the Equilibrium Point, the heat pump has to be shut down because the water temperatures on its condenser side are not sufficient. In order to avoid this situation,heat pump sizing must insure that $T_e \geq T_s$ condition will always be satisfied.

Seeking an alternative and more suitable heating system to establish a better match with the operational characteristics of heat pumps and thus to enhance their performance and compatibility has the following objectives and justifications:

-Elimination of the supplementary boiler . One of the main objectives of many institutions involved in heat pump research and development is to eliminate the supplementary boiler from the system (Pietsch,1991). However such an attempt subjects the heat pump to constraints (i) and (ii) under their most conservative terms. For this reason, unless these constraints are completely waived,elimination of the supplementary boiler seems to be impossible.

-Enhancement of the heat pump performance.

With or without a supplementary boiler,any improvement of instantaneous and seasonal COP of the heat pump will increase the technical and economical feasibility.

-Minimization of the supplementary boiler size.

If for any reason the boiler can not be eliminated,any reduction in its size and seasonal contribution to heating will make the attributes of the heat pump more predominant.

-By achieving one or more of the above mentioned goal the overall utilization of primary energy sources ar corresponding environmental pollution will be decreased.

Figure 1 already reveals several ways of accomplishir these objectives:

a- Point E:

This point can be favorably shifted if the heating load lir moves towards the origin. Provided that T_s does n precede T_b, such a move may be sufficient to eliminate th need for a supplementary boiler. Otherwise the boiler siz can be minimized, only. For a conventional heating syste T_s will always precede T_b. Therefore it is virtually impossib to replace the supplementary boiler unless a bigger he pump is selected such that point B will coincide with point (see Figure 6). But such a move will be detrimental on marginally feasible case. Therefore elimination of th supplementary boiler remains one of the challengin problems unless a more suitable heating system i activated.

b-Point S:

Any heating system which already requires a T_{sm} value less than 55°C will waive constraint (i) and thu increase the chances of eliminating the supplementar boiler.

For any heating system which already calls for a ΔT value less than or equal to 10°C,constraint (ii) coincide with T_b and therefore it is virtually eliminated.

c-$T_e \geq T_s$ condition:

This condition must be maintained so that the heat pum can operate until T_e is reached with its usable water outpu temperatures.

All of these objectives can be realized by a radiant floo heating system which establishes a perfect match wit respect to the water temperature requirements an substantially reduces the heating load as compared to othe heating systems.

Its directly related attributes are:

1-Indoor air temperatures can be selected as 2 to 3°C lowe than the standard indoor design temperatures T_a, withou any sacrifice of the desired human comfort (Kilkis,1990) This is possible primarily due to two factors namely the existence of a higher mean indoor radiant temperature and a slower air movement inside a radiant floor heated space. The latter directly reduces the convective type of sensible heat losses from the human body and hence his/he dependence on the indoor air temperature for therma comfort. In addition,transmission type heat losses throug partitions slightly decrease with a decrease in the velocity o indoor air wetting such surfaces.

2-A lower indoor air temperature setpoint T_{set} and a slow indoor air movement decrease the infiltration and transmission type heat losses by as much as 30 % in total. In addition to this,air stratification is minimal (Raiss and Roeder,1969). This reduces the amount of the temperature gradients in the vertical direction in the space and thus eliminates extra heat losses which take place from the warmer air mass accumulated in the vicinity of the ceiling unless this is reversed by a ceiling fan. Zmeurenau,Fazio and Haghighat (1987) developed a numerical algorithm in order to predict the thermal performance of the radiant floor heating system. They also analyzed the warm air heating

Equilibrium temperature when a conventional space heater is used,°C
Equilibrium temperature when radiant floor heating system is used,°C
Maximum floor surface temperature,°C
Minimum floor surface temperature,°C
Outdoor air temperature,°C
"Uniform" radiant floor surface temperature,°C
:) Local surface temperature of the radiant floor,°C
Return water temperature in the heating circuit,°C
Shut down temperature when a conventional space heater is used,°C
Shut down temperature when radiant floor heating system is used,°C
Indoor air set temperature, °C
Supply water temperature in the space heating circuit,°C
ıx Maximum supply water temperature of the heat pump,°C
ı Thickness of the fin (including floor coverings), m
Average inside surface temperature of the partitions of the heated space (excluding radiant floor),°C
Average water temperature in the space heating circuit $(T_{su}+T_{re})/2$,°C
Water temperature leaving the heat pump condenser, °C
Average water velocity in the hose,m sec-1
Half of the net spacing between adjacent hoses $(M-D_o)/2$,m
Heating efficiency,dimensionless
Distance on the floor surface from hose center plane,m
Thickness of any floor covering,m

k symbols:

Temperature difference,°C
Fin efficiency,dimensionless

TRODUCTION

ṡpace conditioning of residences and commercial ings uses about one third of the primary energy ımed in the United States. Air to water type heat ɔs as well as other types which can generally be used for cooling and heating purposes round the year are a element in energy conservation and protection of the onment. In spite of their heating coefficient of ɔrmance figures well above unity,and recent opments which make them less sensitive to outdoor mperatures,the basic problem for air to water type heat ıs in space heating can not be solved: as the outdoors me colder,the heating load of an indoor space ases,but at the same time the heat pump output ıases. Figure 1 exhibits this situation for a central ng system using radiator type heating units. As this e contradiction can not be reversed, air to water type pumps will continue to face operational constraints as as they service conventional heating systems.

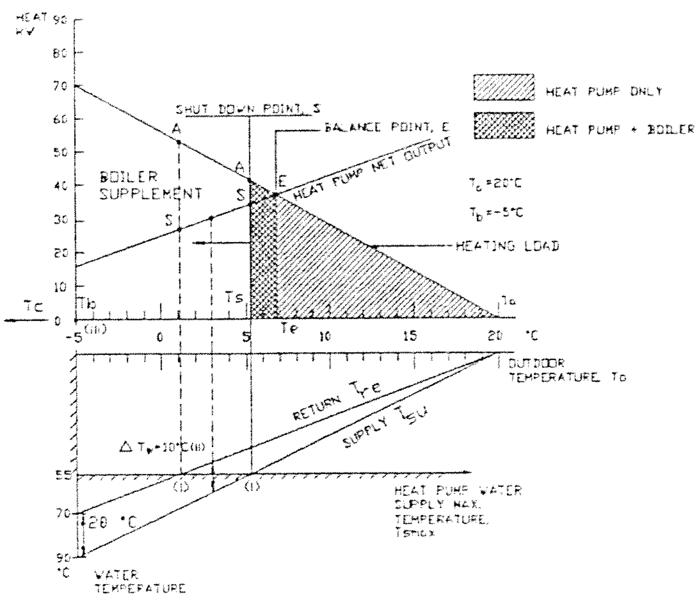


Figure 1. Heat output of an air to water type heat pump versus the heating load as a function of outdoor air temperature,T_o.

For any outdoor temperature below the intersection point of heat demand and supply lines (Balance Point E), a heat pump is unable to supply the entire heating load. In this specific example, T_e occurs at T_o= +7°C. Any design outdoor temperature T_b colder than T_e will require a supplementary plant such as a boiler. The plant can be selected to operate either in a parallel or tandem mode. In either case the plant will complement the heat pump until the latter is forced to shut down at an outdoor air temperature T_s (Shut Down Point S). Until this point is reached,both units will operate together. The supplementary plant will contribute to space heating by an amount corresponding to the area ASE in Figure 1. T_s is determined by one of the following constraints which comes first on T_o axis:

-Constraint (i): Maximum water temperature that can be delivered by the heat pump,T_{smax}

Generally T_{smax} is limited to 55°C. The intersection point of this line with the supply or return water temperature demand of the heating units determines the location of point (i). Typically a radiator unit at design conditions operates at 90°C/70°C supply and return water temperatures respectively. For the specific example illustrated in Figure 1, indoor design temperature T_a is 20°C,therefore supply and return water temperature lines originate at T_o = +20°C and diverge to 90°C and 70°C points at the design outdoor temperature which is -5°C. In this case,point (i) occurs around +5°C for a parallel mode. In a tandem mode,return water temperature governs the case. For a zero water temperature rise across the heat pump this lower bound is -1°C (T_{re}=T_{smax}).

-Constraint (ii): Condenser side water temp. drop, ΔT_{ws}

The maximum permissible temperature drop is generally limited to 10°C. This is in conflict with a radiator heating system where the design temperature drop ΔT_w is 20°C.

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ENHANCEMENT OF HEAT PUMP PERFORMANCE USING RADIANT FLOOR HEATING SYSTEMS

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ABSTRACT

According to the basic nature of radiant floor heating,it is possible to operate at an average water temperature as low as 35°C. The objective of this paper is to investigate how this feature when combined with other attributes of radiant floor heating may enhance the performance of an air to water type heat pump especially in adverse climates. A comparative design analysis between in-slab type radiant floor heating system and radiator heating was made for a 320m² residential home. It demonstrates that the overall performance and shoulder operating capabilities of the heat pump improves and the auxiliary boiler is eliminated when a radiant floor heating system is employed. Also included in this paper is a steady state heat transfer model based on an analogy between the heated slab and a flat plate fin. The advantages of employing radiant floor heating system in heat pump applications are discussed with special emphasis on primary energy conservation.

1. NOMENCLATURE

- A — Amplitude of floor surface temperature variation (for Sine curve),°C
- a — Water to hose convection heat transfer coefficient,W/m² K
- A_p — Radiant panel surface area,m2
- A_r — Total floor area in the heated space,m2
- A_u — Total surface area of the heated space enclosure,m2
- C — Peak load shaving factor,dimensionless
- C_D — Radiation coefficient,W/m2 K4
- COP — Heating coefficient of performance, P_H/P_M, dimensionless
- D_e — Equivalent diameter of the floor in the heated space,m
- D_i — Hose inside diameter,m
- D_o — Hose outside diameter,m
- e — Surface emissivity,dimensionless
- h — Altitude of the location above sea level,m
- k — Thermal conductivity of the slab,W/m K
- k_e — Eq. thermal conductivity of the fin and floor coverings,W/m K
- k_h — Thermal conductivity of the hose material,W/m K
- k_l — Thermal conductivity of the floor covering,W/m K
- L — Thickness of the fin (excluding floor coverings),m
- L_r — Inside perimeter of the floor in the heated space,m
- M — Hose spacing on centers,m
- m — Fin coefficient,m-1
- P — Plant heat load,kW
- P_H — Heat output of the heat pump,kW
- P_M — Mechanical power input to the heat pump,kW
- Q_ε — Back heat loss from the heated slab,kW
- Q_D — Supplementary boiler heating capacity,kW
- Q_e — Edge heat loss from the heated slab,kW
- Q_{ts} — Heat load when a conventional space heater is used,kW
- Q_y' — Heat load when the space is heated by radiant floor,kW
- Q_y — Radiant floor heat load after peak load shaving(if applicable) $C \cdot Q_y'$,kW
- q_y — Radiant floor heat load intensity (Q_y/A_r) or (Q_y/A_p),W/m2
- R — Thermal resistance of the heated slab along thickness T_{total},m2K/W
- r — Linearization factor for radiation heat transfer term,°C3
- T_ε — Indoor design temperature,°C
- T_ε' — Indoor design temperature for a radiant floor heated space,°C
- T_b — Outdoor design temperature,°C
- T_c — Outdoor cut-off temperature to protect evaporator of the heat pump from freezing,°C
- T_d — Outside surface temperature of the hose,°C

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…r Radius einer Kreisplatte …

…se einzusetzen und bei einer rechteckigen Platte die halbe Achse, …h der Luftstrom bewegt. Für größere Flächenabmessungen ist die … nicht geeignet.

…meübergangszahl gilt für den Wärmeübergang von unten nach oben, … Platte wärmer als die Luft ist, z. B. bei der (isolierten) Oberfläche …hängenden Strahlplatte.

…meübergangszahlen von Luft bei freier Konvektion an Rohren und …önnen auch graphisch mit dem DKV-Arbeitsblatt 2-15 (Kältetech-…, 1954, H. 9) bestimmt werden.

…ft- oder Gasdruck $p >$ 1 ata, dann ist das Ergebnis mit $\sqrt{p}$ (kg/cm²) …fachen.

$$Gr = \frac{d^3 \varrho^2 g \beta (t_W - t_L)}{\eta^2} \qquad (36)$$

$\varrho = \gamma/g$ in kg/s²m⁴ und $\beta = 1/T$ in 1/° K) vereinfachen zu

$$Gr = \varepsilon\, p^2 d^3 (t_W - t_L). \qquad (36a)$$

$$\varepsilon = \frac{\gamma^2_{1\,ata}}{g\, \eta^2\, T} \qquad (36b)$$

…r Zahlentafel 9 entnommen werden kann. Bei dieser Form der Gras-…der Luftdruck p bereits berücksichtigt.

…ng der Stoffwerte hat ebenfalls beim Temperaturmittelwert von Wand …erfolgen.

… Gleichungen 60 bis 62 zu beachten, daß sie laminare Luftströmung … Der Übergang in die turbulente Strömungsform erfolgt bei der kriti-…of-Zahl $Gr_{L} = 10^9$, die einer Reynolds-Zahl von Re = 44,5 ent-…, also bei einem wesentlich kleineren Zahlenwert als bei erzwungener …ntretende Turbulenz erhöht aber die Wärmeübergangszahl.

…zte Wandflächen

…en Wandflächen der Niedertemperatur-Strahlungsheizun-

…rme- und Stoffaustausch. Berlin/Göttingen/Heidelberg 1949, S. 119.

oder bei einem Heizstreifen von über 1 m bis zu etwa 2 m …

$$\alpha_K = 0{,}75 \sqrt[4]{t_D - t_L} \qquad (63a)$$

bei mäßiger, gerichteter Luftbewegung im Raum, z. B. durch eine Lüftungsanlage oder bei einem schmalen Heizstreifen bis zu etwa 1 m Breite

$$\alpha_K = 1{,}1 \sqrt[4]{t_D - t_L} \quad \text{nach Griffiths und Davis}^{3}) \qquad (63b)$$

Beheizte Wand

für Wandstreifen bis zu 1 m Höhe

$$\alpha_K = 2{,}2 \sqrt[4]{t_W - t_L} \quad \text{nach Nusselt}^{4}) \qquad (64)$$

bei einer Heizflächenabmessung von über 1 m Höhe

$$\alpha_K = 1{,}7 \sqrt[4]{t_W - t_L} \quad \text{nach Griffiths und Davis.} \qquad (64a)$$

Beheizter Fußboden

für Fußbodenstreifen nicht über 1 m in einer Richtung

$$\alpha_K = 2{,}8 \sqrt[4]{t_{FB} - t_L} \quad \text{nach Nusselt-Hencky}^{5}) \qquad (65)$$

bei einer Fußbodenfläche von über 1 m Breite und 1 m Länge

$$\alpha_K = 2{,}3 \sqrt[4]{t_{FB} - t_L} \quad \text{nach Griffiths und Davis.} \qquad (65a)$$

Die konvektiven Wärmeübergangszahlen nach den Gl. 64 bis 65a sind von den Wandheizflächenabmessungen nicht unabhängig, da die Temperatur der Raumluft vom Beginn des Anströmvorganges ansteigt und damit die wirksame Temperaturdifferenz sich ermäßigt. Mit zunehmender Größe der Heizflächenabmessungen wird jedoch der Randeinfluß bei schleichender Luftbewegung vernachlässigbar gering werden und damit die Wärmeübergangszahl unabhängig von den Heizflächenabmessungen.

Die vierte Wurzel aus der Temperaturdifferenz ($t_W - t_L$) in den Gleichungen 63 bis 65a läßt den Zusammenhang mit der Grashof-Zahl erkennen.

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DIE STRAHLUNGSHEIZUNG

FLÄCHEN-, STRAHLPLATTEN- UND INFRAROTHEIZUNGEN

VON

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VIERTE AUFLAGE

Mit 335 Abbildungen, 26 Arbeitsblättern,
40 Tabellen und Zahlentafeln

Prof. Dr. Birol KILKIŞ

range of temperature differ- and scattered considerably. on 13 and the slope of the in Fig. 4 may be slightly in

eral: A summary of the infor- on test apparatus, test condi- and convection equations for surfaces by various authori- given in Table 1.

hould be noted that the con- values reported here were ed in the Environment Labora- nder conditions of no infiltra- n an actual panel-heated room, tion would cause additional tion. For 1 air change per at 0 F in the test room, the tion coefficient increased about rcent for the heated ceiling at 120 F°, and increased about cent for the heated floor panel F°. However, the convection ients for the walls were not ed appreciably. More definite iation will be presented on this t in another paper.

racteristic Linear Dimension: the data for 3 room sizes cor- l well when the equivalent di- r was used in dimensionless ions 5 and 11 (See Figs 2a c), the characteristic linear di- on was concluded to be the lent diameter for the heated and ceiling panels. A similar ation for the convection to the as shown in Fig. 2b, indicated he characteristic linear dimen- f a wall is its height.

importance of the characteris- mension in Equation 4 depends the exponent of the equation. e exponent is 1/3, then the cteristic dimension appears on sides of the equation as $L =$ $^{1/3}$ and consequently L cancels, ting that the convection co- nt is independent of the size surface. It will be noted that or panel heating (Equation 5), ponent is nearly 1/3 and the of the dimension D_e is very (see Equation 6). The close agreement of the convection values obtained by other investigators for various sizes of plates as shown in Fig. 4, may be attributed to this fact. In contrast, however, the characteristic dimension is relatively important in determining convection heat transfer from a heated ceiling panel, as shown by Equation 12, since the exponent for this orientation is 1/4 (see Equation 11).

In Equation 7 for wall convection with a heated floor panel, the exponent of 0.32 indicates that the characteristic dimension (wall height) is not important. There is some evidence (Equation 13 and Fig. 4) that the coefficient is considerably different for wall convection with a heated ceiling, and that the wall height in this case may be an important factor. However, over the range of conditions covered in this paper it is suggested that Equation 8 be used for wall convection with either floor or ceiling heating.

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predicted that the restriction of air currents at the ceiling of a room would result in lower convection rates.

In order to investigate the effect of boundary conditions on the convection from a horizontal surface, and to verify the low measured value of convection from the ceiling, special tests were made with a horizontal free-edged heated plate 11½ in. square facing downward, suspended at the 60-in. level in the center of the room. The room air temperature was uniform because the floor, ceiling and walls of the room were controlled at one temperature while the small plate was controlled at a higher temperature. The heat flow was measured by a plate-type heat-flow meter cemented to the face of the heated plate. Resulting values of convection agreed well with those of previous investigations as shown in Fig. 8. The edge of the plate was then baffled by a wooden frame 2 in. wide and the convection reduced considerably as shown also in Fig. 8. When the heated plate was attached to and heated to the same temperature as the ceiling, the convection was further reduced. The area weighed value of measurements of several locations is shown in Fig. 8 to lie on the convection curve for the ceiling. In the Environment Laboratory or any actual room, the ceiling is in a sense baffled by the walls. Convection will be affected by this condition.

Convection, Floor Heating: In a floor-heated room with uniform environment the convection heat transfer from the floor panel to the room air was essentially the same as reported by other investigators for a small horizontal heated plate facing upward. The agreement is shown in Figs. 2a and 3.

Wilkes and Peterson's data, curve I of Fig. 3, are somewhat higher than all the other curves. Their data were obtained with large heated and cooled plates spaced only 4 in. apart. This close spacing may account for the difference.

Tests with the small heated plate were also made in the Environment Laboratory. The 11½ in. square heated plate previously described was faced upward and was located first at the 60-in. level and then at the floor level. The results from these tests are also shown in Fig. 3 and agree well with the other data.

The convection transfer from the air to the cooler ceiling in a space heated by a floor panel is shown in Fig. 5, and agrees well with the data

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floor, the boundary effects may also have contributed.

Convection, Walls: The values of convection to the walls both with floor heating and ceiling heating

CONCLUSIONS

The data reported in this paper were obtained with the entire floor area or ceiling area used as a heated panel, a uniform environment wherein all surfaces other than the heated panel were at a uniform temperature, relatively still air conditions (no infiltration), and an empty, unlighted room. Under these conditions the following conclusion may be drawn:

1. On the basis of research done at the ASHAE Laboratory, the following equations apply in calculating natural convection heat transfer at room surfaces.
   - A. In a floor-heated space
     1. Convection from floor: $q_c = 0.39\,(\triangle t)^{1.31}/D_e^{\,0.08}$
     2. Convection to walls: $q_c = 0.29\,(\triangle t)^{1.32}/H^{0.05}$
     3. Convection to ceiling: Same as convection from floor.
   - B. In a ceiling-heated space
     1. Convection from ceiling: $q_c = 0.041\,(\triangle t)^{1.25}/D_e^{\,0.25}$ ×
     2. Convection to walls: Same as for floor-heated space.
     3. Convection to floor: Same as convection from ceiling.
2. Natural convection data given by other investigators for small heated plane surfaces were found to be in good agreement with all of the equations listed in the first conclusion except the equation for natural convection from a heated ceiling. The convection coefficient for small free-edge plates may be 6 to 10 times as great as that for a heated ceiling.
3. Room size in a ceiling-panel-heated room has a significant effect on the unit convection from the ceiling and the floor. However, the convection heat transfer from a heated ceiling is small in comparison with radiation exchange, and therefore, the effect of room size on the total heat transfer is not important. In a floor-heated room, the effect of room size is not significant.
4. In a completely enclosed space with high emissivity room surfaces, the interchange factor between the heated panel and its enclosure may be approximated by the hemispherical emissivity of the panel surface.
5. The approximate combined film conductances (natural convection and radiation) for the heated panels based on the difference between panel temperature and the room air temperature for a normal size room with high emissivity surfaces are as follows:

   $h_{rc} = 2$ for heated floor panel at about 85 F.

   $h_{rc} = 1.1$ for heated ceiling panel at about 120 F.

of others. It differs only slightly from the data shown in Fig. 3, for heat transfer from a heated floor to cooler air. This slight difference may be attributed to the fact that the room air temperature on which the temperature differences were based was taken at a point 36 in. away from the ceiling but 60 in. away from the floor. Since the walls were at the same temperature as the ceiling but at a lower temperature than the agree substantially with the test values for a free-edge vertical-plate, as shown in Fig. 4. However the convection to the wall in a ceiling-heated room is a little lower than in a floor-heated room. This may be the result of less air motion in the room or because a higher radiation interchange factor was used when the ceiling is heated. Since all the data for convection to the walls with ceiling heating were taken within a *urnal Section*

ata for a free-edge small plate
o shown in the figure and will
cussed later.

convection data for the four
were rather scattered, especially
north wall where a door is
. From the average of values
other 3 walls, the convection
ents were obtained from 25
nd are expressed in dimen-
form as:

$$= 0.22 \; (N_{Gr} \; N_{Pr})^{[illegible]} \; \ldots\ldots (7)$$

Convection to Walls, Heated Floor

ta, in form of Nusselt number
e product of Grashof and
numbers are shown in Fig.

tion 7 may be simplified to

$$= 0.29 \; [(\Delta t)^{[illegible]} / H^{[illegible]}] \; \ldots\ldots (8)$$

Convection to Walls, Heated Floor

t data for one room size are
in Fig. 4.

onvection to the ceiling when
r is heated is shown in Fig.
he convection coefficient may
essed by

$$= [0.58 \; (\Delta t)^{[illegible]} / D_e^{[illegible]}] \; \ldots\ldots (9)$$

Convection to Ceiling, Heated Floor

lation of Convection Data, Panel Heating: Since the con-
heat flow from a heated ceil-
found to be relatively small,
rrors in radiation measure-
esulted in large errors in
on data. To reduce these er-
ne of the heat-flow meters
vered with polished alumi-
, thereby reducing the radi-
flow and making the con-
omponent a greater percent-
he total.

tion from the ceiling was
rmined directly by measur-
mperature gradient through
dary air film and computing
conducted through this film.
erature gradient was meas-
a fine thermocouple, the
f which could be precisely
d. The device used for
this temperature gradient
in Fig. 6. The ceiling dif-
l fluorescent lighting shown

in the figure were installed after the measurements reported in this paper were made. Typical experimental data obtained by this method are shown in Fig. 7. The conduction or the equivalent convection through this layer was then computed by the equation

$$q = -kA \; (dt/dX) \; \ldots\ldots\ldots (10)$$

where

k = the conductivity of the air.

dt/dX = the temperature gradient, Fahrenheit degree per foot.

Investigation of the boundary layer thickness was not attempted;

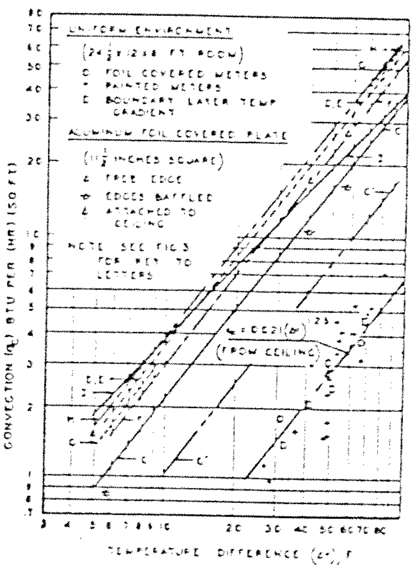

Fig. 8—Heat Transfer from Heated Ceiling by Natural Convection

however, in one test, measurements were made as far as 0.75 in. from the ceiling and the gradient was still linear. Although most measurements were steady, variation in thermocouple output increased in amplitude and frequency with increasing distance from the ceiling.

Since both the difference method and the direct method were judged to be equally accurate, data obtained by both methods were correlated. In dimensionless form, the correlation was found to be

$$N_{Nu} = [illegible] \; (N_{Gr} \; N_{Pr})^{[illegible]} \; \ldots\ldots (11)$$

Convection from Heated Ceiling

This relationship is plotted in Fig. 2c, and may be simplified to

$$h_c = 0.041 \; [(\Delta t)^{[illegible]} / D_e^{[illegible]}] \; \ldots (12)$$

Convection from Heated Ceiling

Convection heat transfer for a single room size is shown in Fig. 8. Experimental data determined with painted heat-flow meters are also plotted in Fig. 8 but were not used in determining the equation.

The convection coefficient for the walls was determined from the average test values for 3 walls and may be expressed as

$$h_c = 2.17 \; [(\Delta t)^{[illegible]} / H^{[illegible]}] \; \ldots\ldots (13)$$

Convection to Walls, Heated Ceiling

The convection data for one room size are plotted in Fig. 4.

In a ceiling-heated room, the temperature difference between the floor and room air was found to be less than 7 F. and the convection to the floor was less than one Btu per (hour) (square foot). This resulted in considerable scatter of the convection values when determined by subtracting the radiation exchange from the observed total heat flow. Another approach was made by plotting the observed total heat flow to the floor against the difference between the floor temperature and the area weighted average temperature of all surfaces excluding the floor. Based on the values of the total heat flow read from a smooth curve drawn through these points, the difference method was then used to get the approximate convection.

The convection values were found to lie between the extrapolated curves for free-edge plates and the convection from the heated ceiling shown in Fig. 8. Probably errors accumulated in the determination of the convection heat flow to the floor are of the same order of magnitude as the convection. Hence, no satisfactory equation has been developed. However, no significant error would result if Equation 12, convection from the heated ceiling, were used for calculating the convection to the floor as long as the range of temperature differences encountered in the tests is not exceeded.

Discussion

Convection, Ceiling Heating: The convection coefficient for the ceiling panel was found to be much smaller than that for a small horizontal heated plate facing downward with edges free, as shown in Figs. 2c and 8. Information concerning the latter has appeared in the literature[illegible]. Other investigators[illegible] serious. It is interesting to note that the painted surface appeared to behave as a highly specular rather than as a perfectly diffuse reflector (see Appendix †). Nevertheless the specular reflections became diffuse after 2 or 3 reflections from other surfaces.

Convection

In a space entirely enclosed by a ceiling, floor, and walls, with no infiltration, natural convection is brought about by differences in density of the air caused by temperature differences. The rate of convection for plane surfaces depends upon the temperature differential and the film coefficient. The film coefficient is related to the mean temperature of the surface and air, temperature differences, size and position of the surface, boundary conditions, and the relative positions of other surfaces.

For natural convection, the film coefficient or convection coefficient h_c

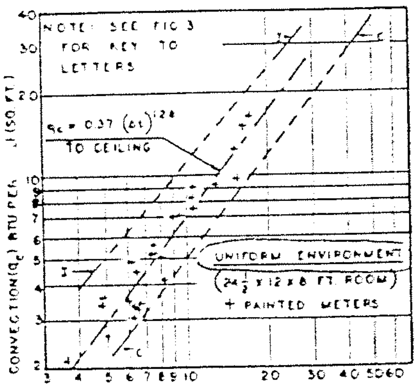

Fig. 5—Heat Transfer to Ceiling by Natural Convection (Room Heated by Floor Panel)

may be expressed in dimensionless terms by Nusselt, Grashof and Prandtl numbers[24] as follows:

$$N_{Nu} = a\ (N_{Gr}\ N_{Pr})\ c \qquad (3)$$

or

$$h_c L/k = a\ [(c_p\ \mu/k)(\beta\rho^2\ g\ L^3\ \Delta t/\mu^2)]\ c \qquad (4)$$

Once the constants a and c are evaluated from experimental data, the equation may be applied to similar problems as long as the shape and orientation of the system remain the same and the product of N_{Gr} and N_{Pr} is within the range of conditions used in the original evaluation.

†It is planned to publish Appendix with the paper in the ASHAE TRANSACTIONS 1956.

The physical properties of air, ρ, k, β, and μ are to be taken at the arithmetic mean of the surface and room air temperatures. In all cases in this report the room air temperature is the temperature at the 5-ft level at the center of the room.

The characteristic linear dimension L is one which relates the size of a surface of given shape and orientation to the convection coefficient. For example, for a long horizontal cylinder, the diameter affects the convection coefficient greatly, while the length has little effect. In the case of a horizontal square plane surface, the length of a side has been used as the characteristic dimension. For

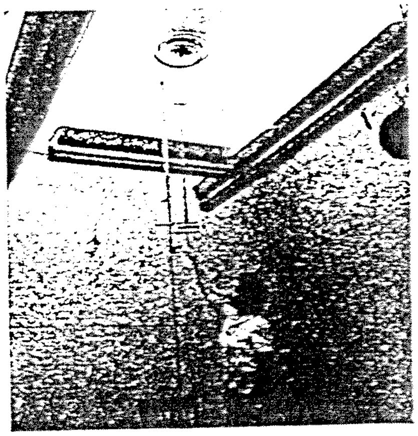

Fig. 6—Interior View of Test Room showing Thermocouple Probe Apparatus used for Boundary Film Temperature Measurements

rectangular horizontal plane surfaces, such as the ceiling or floor of a panel-heated space, the characteristic dimension was taken as the equivalent diameter, which is defined as 4 times the hydraulic radius or 4 times the area divided by the perimeter. For vertical plane surfaces, such as walls, the height was the characteristic dimension used.

Correlation of Convection Data. Floor Panel Heating: Convection heat flow was obtained by subtracting the calculated radiation output from the total panel output as described before. The convection from the heated floor was found to be greater near the walls than in the center of the room, as shown in Fig. 1. This was due to the fact that the air near the walls was cooler and at a higher velocity than that at the center of the

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room. The average area weighed heat flow was, therefore, used for correlation.

The dimensionless groups for 34 tests were evaluated‡ with the equivalent diameter as the characteristic dimension, and correlated by the

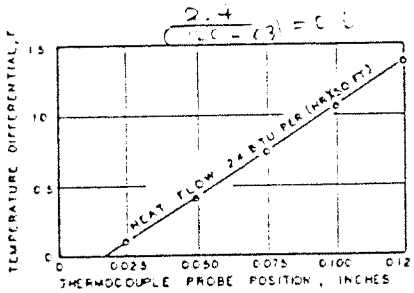


Fig. 7—Temperature Gradients in Laminar Boundary Layer Adjacent to Ceiling Panel ($5\frac{1}{2}$ feet from west wall, 120F ceiling, 65F AUST, no infiltration, uniform environment, $24\frac{1}{2} \times 12 \times 8$ foot room)

method of least squares. The analysis indicated that the convection from a heated floor may be represented by:

$$N_{Nu} = 0.33\ (N_{Gr}\ N_{Pr})^{[illegible]} \qquad (5)$$

Convection from Heated Floor

The test data for the 3 room sizes are shown in the top curve Fig. 2a.

As the temperature range occurring in panel heating systems is small, the physical properties of air may be considered as constants. For practical computation, therefore, the heat transfer coefficient for natural convection may be correlated for a given orientation by means of 2 variables, *viz*: the temperature difference between the heat-transfer surface and the room air, and the characteristic dimension. Equation 5 may therefore be simplified to

$$h_c = 0.39\ [(\Delta t)^{[illegible]}/D_e^{[illegible]}] \qquad (6)$$

Convection from Heated Floor

Equation 6 has been plotted in Fig. 3 for a room size $24\frac{1}{2} \times 12 \times 8$ ft ($D_e = 16.3$) and is compared with test data from that room. Curves of other investigators and additional

‡All air properties were taken from *Thermodynamic Properties of Air*, by Joseph H. Keenan and Joseph Kaye (John Wiley & Sons, Inc., New York, 1945) and the HEATING, VENTILATING, AIR-CONDITIONING GUIDE 1955 (published by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York).

urnal Section

·ity is independent of tempera-
.n interchange factor $F_{1\text{-}2}$ can
nd from the following equation
by Hottel[11]

$$= \frac{1}{\dfrac{1}{F_{1,2}} + \left(\dfrac{1}{e_1} - 1\right) + \dfrac{A_1}{A_2}\left(\dfrac{1}{e_2} - 1\right)} \cdots (2)$$

angular emissivity of the
surfaces in the Environment
tory was measured by Umur,
ee and Schutrum[6] for incident
up to 85 angular deg. This
eport gave the hemispherical
ity of the paint as 0.88 and
emissivity of 0.92.
the conditions existing in the
om average $F_{1\text{-}2}$ is unity for
r panel and its environment:
e_2 = 0.88; and A_1/A_2 =
84.2 = 0.34. Applying these
to Equation 2, $F_{1\text{-}2}$ becomes
hich compares favorably with
sured value of 0.876. Further-
more, the hemispherical emissivity of the panel surface, 0.88, is nearly the same as the observed interchange factor, 0.876. Therefore, the interchange factor for a panel in a completely enclosed space appears to be equivalent to the hemispherical emissivity of the panel surface ($F_{1\text{-}2} = e_1$). It is not known whether this is true for surfaces having low emissivities.

Although one would expect the interchange factor for the heated ceiling to be the same as for the heated floor panel, it was found to be 0.90, which is slightly higher than the 0.876 value for the floor. This may be due to the fact that meters installed on the ceiling were constructed with embossed aluminum foil instead of smooth foil as used on the floor meters. The added roughness of these surfaces would be expected to increase the radiation output.

The interchange factor for the polished foil covering on some of the meters was found to be 0.07 which is again in good agreement with Hottel's Equation 2, where A_1/A_2 approached zero, and the interchange factor $F_{1\text{-}2}$ equals the emissivity of the foil. The normal emissivity of highly polished aluminum foil is around 0.05[12] and the hemispherical emissivity would be about 0.06.

The interchange factor $F_{1\text{-}2}$, once established by measurement, was used thereafter to calculate the radiation exchange between the warm panel and its environment in the test room. The radiation transfer to the walls and the ceiling, from the heated floor panel, was obtained by distributing the net radiation output of the panel according to the shape factors between the panel and each of the other surfaces. This was possible because the ceiling and the walls were at the same temperature. The radiation exchanges with a heated ceiling were treated in a similar manner but were somewhat higher, since the interchange factor was 0.9 for the ceiling panel and 0.876 for the floor panel. This difference was judged not to be

able 1 — Comparison of Test Apparatus, Test Conditions, and Natural Convection Equations for Plane Surfaces

ne of esti- tor	Ref. No.	Appa- ratus	Size of Test Surface	Air Tempera- ture	Temp. Differ- ence	Convection Equations: Facing Upward	Convection Equations: Vertical	Convection Equations: Facing Downward
1	2	3	4	5	6	7	8	9
enden nd nders	(14)	Heated Plate	2 ft Square (Max.)	—	To Over 1000 F	Streamline Region		
						$h_c{}^* = 0.27\,(\Delta t/L)^{0.25}$	$h_c{}^* = 0.28\,(\Delta t/L)^{0.25}$	$h_c{}^* = 0.12\,(\Delta t/L)^{0.25}$
						Turbulent Region		
						$h_c = 0.35\,(\Delta t)^{0.25}$	$h_c = 0.30\,(\Delta t)^{0.25}$	—
enden nd nders dams)	(11)	Do.	Do.	—	Do.	Streamline Region		
						$h_c{}^* = 0.27\,(\Delta t/L)^{0.25}$	—	$h_c{}^* = 0.12\,(\Delta t/L)^{0.25}$
						Turbulent Region		
						$h_c = 0.22\,(\Delta t)^{0.25}$	—	—
fiths Davis	(15)	Heated Plate	3 to 4 ft sq	Room Tempera- ture	Up to 212 F	$h_c = 0.4\,(\Delta t)^{0.25}$	$h_c = 0.3\,(\Delta t)^{0.25}$	$h_c = 0.2\,(\Delta t)^{0.25}$
fiths Davis dams)	(21)	Do.	Do.	Do.	Do.	$h_c = 0.38\,(\Delta t)^{0.25}$	$h_c = 0.27\,(\Delta t)^{0.25}$	$h_c = 0.2\,(\Delta t)^{0.25}$
lman	(16)	Heated Disk / Heated Ribbon	3.47 to 9.97 in. / 0.50 in. Wide, to 2.108 in. Long	Air at Room Tempera- ture	Zero to 700 F	$h_c{}^{**} = \dfrac{C\,\Delta t^{0.266}}{D^{0.2}\,(T_{avg})^{0.245}}$		
						$C = 1.79$	$C = 1.394$	$C = 0.89$
lman cell)	(20)	Do.	Do.	Do.	Do.	$h_c{}^{**} = 0.32\,(\Delta t)^{0.25}$	$h_c = 0.25\,(\Delta t)^{0.25}$	$h_c{}^{**} = 0.16\,(\Delta t)^{0.25}$
ing	(17)	Correlated Data From Other Investigators				$h_c{}^{**} = 0.28\,(\Delta t)^{0.25}$	$h_c = 0.22\,(\Delta t)^{0.25}$	$h_c{}^{**} = 0.15\,(\Delta t)^{0.25}$
twelt Henky secke)	(19)	—	—	—	—	$h_c = 0.48\,(\Delta t)^{0.25}$	$h_c = 0.38\,(\Delta t)^{0.25}$	$h_c = 0.22\,(\Delta t)^{0.25}$
lies nd rson	(18)	Heated Plate	8 ft by 2 ft 8 in. With 4 in. Air Space Between Surfaces	Air at About Room Tempera- ture	8 to 28 F	$h_c = 0.83\,(\Delta t)^{0.25}$	$h_c = 0.5\,(\Delta t)^{0.25}$	$h_c = 0.38$
HAE arch b.		Environ- ment Lab.	24½ x 12 x 12 ft. (Max.) (See Paper)	Room Tempera- tures	Zero to 100 F	Floor Heated (From Floor or to Ceiling)	To Walls (Floor Heated or Ceiling Heated)	Ceiling Heated (From Ceiling or to Floor)
						$h_c = 0.39\,(\Delta t)^{0.31}/D_e^{0.08}$	$h_c = 0.29\,(\Delta t)^{0.32}/H^{0.05}$	$h_c = 0.641\,(\Delta t)^{0.25}/D_e^{0.25}$

Fishenden and Saunders' equations, L is the side length in feet; a mean value is taken for the side length if the plate is rectangular.
llman's (References 16, 20) and King's (Reference 17) values for horizontal heated plates facing upward and downward were based on the
ise distribution obtained by Griffiths and Davis (Reference 15). In Heilman's Equation, D is the diameter or length in inches, and T_{avg} is
e of surface and air temperature, Fahrenheit degrees absolute. Heilman suggested that, until further experimental work was done, 24 in. be taken
t for the size and height, and he assumed the heat loss becomes constant for heights greater than 24 in.

ished-if the net radiant energy ex-
ange can be evaluated. In radia-
on interchange between any 2 sur-
ces forming part of an enclosure,
e are 2 factors involved besides
well-known Stefan-Boltzmann law
r black-body radiations. These
e: (1) the view the surfaces have
each other; and (2) their emitting
d absorbing characteristics. The
rmer involves a shape factor or

where

$F_{1\text{-}2}$ = the interchange factor involving both items 1 and 2 as just stated.

When the reflections among 6 surfaces of a room are considered along with surface irregularities and the absorbing water vapor in the air, theoretically exact treatment of radiation exchanges is impractical. The procedure, then, was to rely on actual test measurements.

panel at the same temperature. The average net exchange between the entire surface of the heated panel and the surroundings was determined from net exchange measurements at various locations on the surface. The interchange factor between the floor

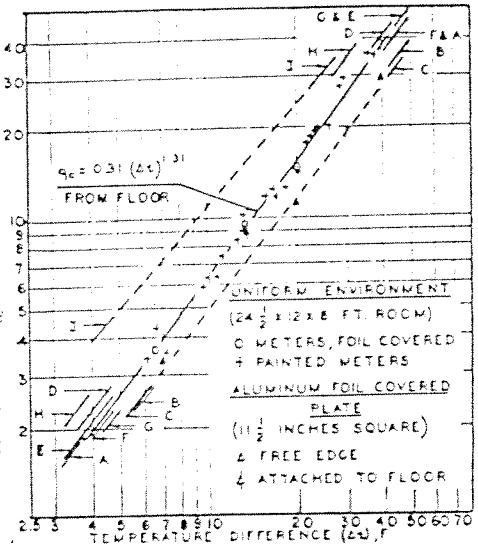


Fig. 3 — (left) Heat Transfer from Heated Floor Panel by Natural Convection

Notes for Fig. 3

. Fishenden & Saunders — Turbulent Region Ref. 14.
. Fishenden & Saunders — Turbulent Region (McAdams) Ref. 11.
. Fishenden & Saunders — Streamline Region (L = 0.96 Ft) Ref. 14.
. Fishenden & Saunders — Streamline Region (L = De = 16.3 Ft) Ref. 14.
. Griffiths & Davis Ref. 15.
. Griffiths & Davis (McAdams) Ref. 21.
. Heilman (Carroll) Ref. 20.
. King Ref. 17.
. Nusselt, Henky & Hettinger (Giesecke) Ref. 19.
Wilkes & Peterson Ref. 18.

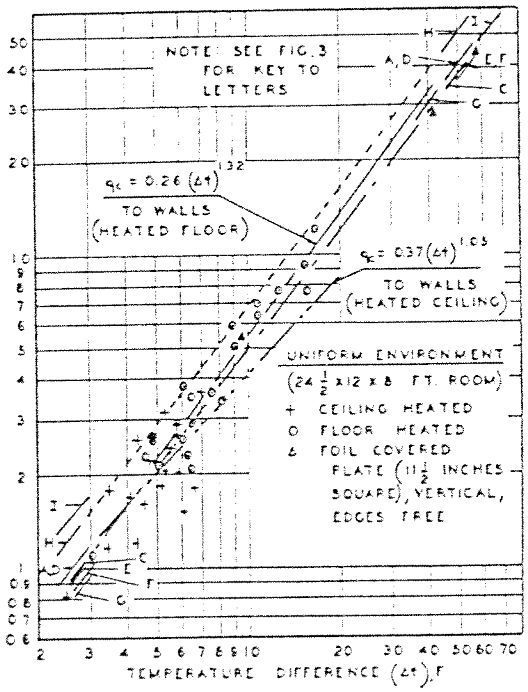


Fig. 4—(right) Heat Transfer to Walls by Natural Convection

For accuracy in evaluating the radiation exchange, the simplest configuration possible was used, namely, all room surfaces except the heating panel and its environment was found to be 0.876. If the surface is a perfectly diffusing emitter and reflector, i.e. obeys the cosine law, and if its

iew factor which takes into account oth the shape and the relative position of the surfaces. Information concerning this has appeared elsewhere.[5-11] The latter so-called emissivity factor or *effective* emissivity depends upon the relative sizes, relative positions, shapes, reflection haracteristics (diffuse or specular) nd the individual emissivity of the faces.

The general equation for radiation nterchange given by Hottel[12] is

$$q_r = F_{1\text{-}2}\,\sigma(T_1^4 - T_2^4) \;\ldots\ldots\ldots(1)$$

Nomenclature

A = area of a surface, square feet. A_1 refers to surface 1 (enclosed body). A_2 refers to surface 2 (enclosing body).

c = dimensionless exponent.

c_p = specific heat of air at constant pressure, Btu per (pound$_m$) (Fahrenheit degree).

D_e = equivalent diameter equals four times area divided by the perimeter of surface, feet.

ϵ = emissivity of surface for radiation, dimensionless; ϵ_1 of surface 1; ϵ_2 of surface 2.

F = overall interchange factor, dimensionless; $F_{1\text{-}2}$ from surface 1 to surface 2.

$\bar{F}$ = black-surface overall interchange factor, dimensionless; $\bar{F}_{1\text{-}2}$ from surface 1 to surface 2.

g = acceleration due to gravity, 4.17 x 10[8] ft per (hour)[2].

H = height of walls, feet.

h = coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree); h_c for natural convection from a surface to ambient air; h_{rc} for combined coefficient for radiation plus convection.

k = thermal conductivity of air, Btu per (hour) (square foot) (Fahrenheit degree per foot).

L = characteristic linear dimension for convection, feet.

N_{Gr} = Grashof number, $\rho^2\beta g L^3 \Delta t/\mu^2$, dimensionless.

N_{Nu} = Nusselt number, $h_c L/k$, dimensionless.

N_{Pr} = Prandtl number, $c_p\mu/k$, dimensionless.

q = rate of heat transfer, Btu per (hour) (square foot); q_c for natural convection, q_r for radiation.

T = absolute temperature, Rankine degrees; T_1, temperature of surface 1; T_2, temperature of surface 2.

X = length of conduction path, feet.

Greek

α = dimensionless constant.

β = coefficient of volumetric expansion of air, reciprocal Rankine degree.

σ = Stefan-Boltzmann constant, 0.173 x 10[-8] Btu per (hour) (square foot) (Rankine degree)[4].

Δt = temperature difference between surface and air, Fahrenheit degrees.

μ = viscosity of air, pounds$_m$ per (hour) (foot).

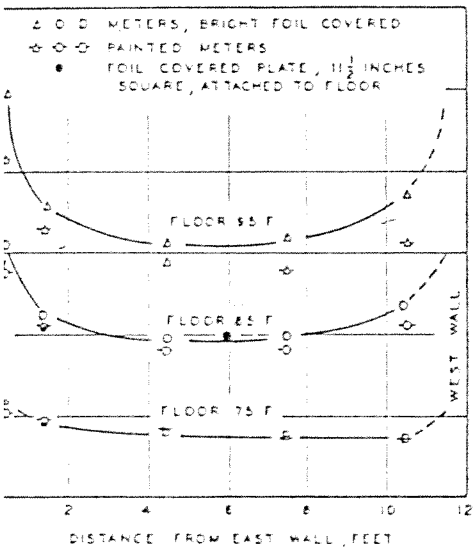
ρ = density of air, pounds$_m$ per cubic foot.

ents were made under steady- conditions with a uniform en- nent, wherein all room surfaces than the heated panel were at me temperature, and with the floor or ceiling heated. Other s on this program will follow.

Room and Apparatus

ASHAE Environment Labora- was described in a previous '. All 6 surfaces of the room are

—Convection from Floor Panel (uniform en- ent, no infiltration, AUST 65 F, 24½ x 12 x 8 oom)



sed of aluminum panels, the ratures of which are controlled quid circulation. The surfaces strumented for direct measure- of heat-flow rates and surface ratures, and for these tests were d with a semi-gloss gray paint 2 coats of primer.

radiation measurements, a ve radiometer[6] with its field w restricted to narrow cone was used.

dure

total exchange between a room e and its environment is the f the convection exchange be- the surface and the room air e radiation exchange between rface and the other room sur-faces which it can see. In all cases total heat exchange in the Environment Laboratory was measured by plate-type heat-flow meters[7] distributed over and fastened to the room-side surfaces of the walls, floor, and ceiling. The problem, therefore, was one of separating this total exchange into its convective and radiative components.

The total heat output of the panel surface was measured directly by plate-type heat-flow meters, and the radiation output was subtracted from the total panel output to obtain the convection heat flow. The convection coefficient was then evaluated by dividing the convection heat flow by the temperature differences of the surface and the room air. The radiation exchange was measured with the radiometer at a number of heat-flow meter locations by sighting at the meter surfaces and obtaining the total radiant energy leaving them at a given angle, and then sighting in the opposite direction and measuring the total radiant energy impinging on them. For proper summation of the net radiation exchange, readings were taken at a number of positions over the hemisphere. The heat-flow meters were painted the same as the surface on which they were located for most of the radiation measurement. However, for a few tests some of the meters were covered with highly polished foil.

Tests in the Environment Laboratory covered the full range of panel and unheated surface temperatures encountered in practice. Floor panel temperatures were varied from 75 F to 110 F and ceiling panel temperatures from 90 F to 150 F. During both series of tests, the AUST* was varied from 40 F to 70 F. Tests were also made with three different sized rooms, 24½×12×8 ft; 24½×12×12 ft; and 12×12×8 ft.

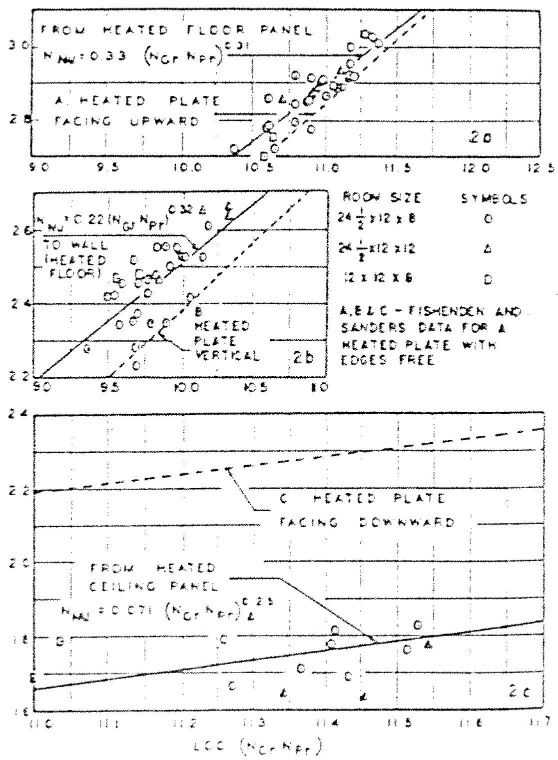


Fig. 2—Comparison of Natural Convection for Room Surfaces and Similarly Oriented Small Plates

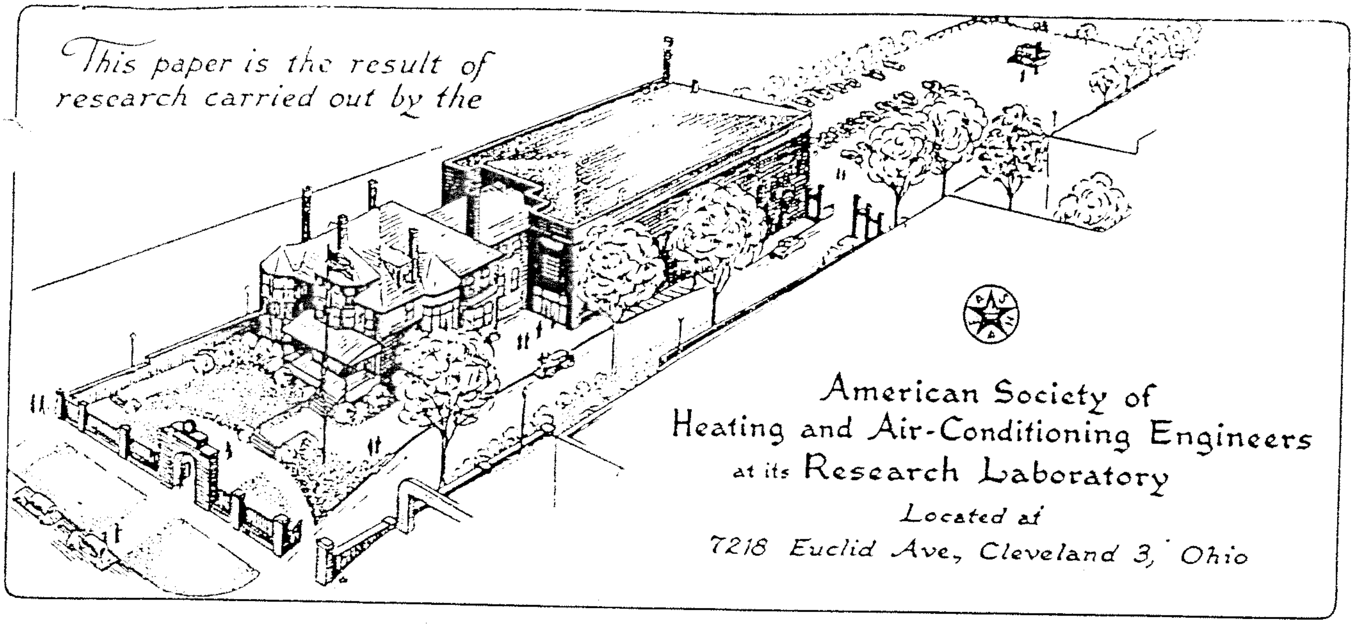
A few tests were made to determine the convection heat transfer from an 11½ in. square plate in various orientations, with free and with baffled edges.

In most of the tests, the relative humidity was low, around 20 percent, and the effect of water vapor absorption and reradiation was not evaluated.

Radiation

Separation of the convective component from the total heat exchange at a room surface can be accom-

*The area weighted average temperature of the unheated surfaces of the room. In a uniform environment, AUST equals the temperature of all surfaces other than the heated panel.



NATURAL CONVECTION AND RADIATION IN A PANEL-HEATED ROOM

By T. C. Min*, L. F. Schutrum**, G. V. Parmelee†, and J. D. Vouris‡, Cleveland, Ohio

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio

CONVECTION data previously available for panel heating design calculations are those obtained from experiments with heated wires, pipes, or small plates with free edges, or with parallel plates only a few inches apart. There has therefore been some uncertainty regarding the application of these data to panel heating. Under the guidance of the Technical Advisory Committee on Panel Heating and Cooling[6], research has been conducted in the ASHAE Environment Laboratory since November 1951 to develop basic information pertaining to panel heating and cooling applications.

SUMMARY — Natural convection coefficients and radiation interchange factors based on the experiments made in the ASHAE Environment Laboratory under floor panel and ceiling panel heating conditions are reported. Convection values obtained in the test room for the room surfaces and for a small free-edge plate, and values for small plates as reported by other investigators, are compared.

Convection from the heated ceiling panel was found to be considerably less than from a small downward-facing free-edge plate. Convection coefficients determined for other surfaces of the room compared favorably with values determined from similarly oriented, small heated plates.

Radiation exchanges between the heated panel and the room surfaces were measured with a radiometer and found to agree well with calculated values.

In four previous papers [1-4], the total heat flow rates from ceiling panels and floor panels under specific conditions were reported. The influence of various factors affecting these heat flow rates were also reported and discussed. In order to provide a more general approach to design, this paper presents an analysis of the convection and radiation components of the total heat exchange at all the surfaces of a room heated by warm panels. The results of this study, with certain limitations, may be useful in solving panel heating problems of various degrees of complexity. The convection and radiation coefficients can be combined for estimating panel outputs, or they may be used separately as coefficients in rational design procedures such as Hutchinson's, or in the thermal-circuit approach to unsteady-state problems as advanced by Nottage.

Data were obtained mainly from the previous studies but include results from recent experiments. Ex-

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[6]P. B. Gordon, Chairman; W. S. Harris, Vice Chairman; A. F. Agrer, W. P. Chapman, John [illegible], Jr., H. T. Gilkey, E. S. Howarth, N. Hunter, A. T. Jern, W. C. Kadow, G. D. [illegible], H. A. Lockhart, C. O. Mackey, R. L. [illegible], S. K. Smith, E. F. Snyder, Jr., W. F. [illegible], J. M. Van Nieuwerken, G. L. Wiggs.

For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Washington, D.C., June 1956.

DESIGN OF EMBEDDED SNOW MELTING SYSTEMS

PART 2. HEAT TRANSFER IN THE SLAB: A SIMPLIFIED MODEL

Dr.B.I.Kilkis*

TRACT

imple analytical technique was developed in order to predict the nsfer of heat in a snow melting slab. The primary objective was to elop a simple design tool while sufficient accuracy is retained. s analytical model is aimed to replace the rules given by ASHRAE delines which apply for a limited design range using metal pipes. model differentiates several performance classes of snow melting l load intensities for periods including before or after the snow. ults of sample designs were compared with finite element utions. Comparisons indicated sufficient accuracy for engineering culations.

INTRODUCTION

h the advent of plastics and rubber technology,heat pipes,and trol systems,hydronic and electric snow melting on critical faces by embedded heating elements are becoming popular and more ective. In spite of these developments,a practical but nprehensive design technique has not been developed yet although a ailed finite element analysis is already an option for engineers. researchers have attempted to develop a simple and accurate model ich is suitable for engineering calculations. However these efforts ually ignored the back and perimeter and edge losses and assumed a

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Member ASHRAE TC 6.4 & 6.5. Head,R&D Dept.Heatway,Springfield MO
ofessor,Middle East Technical University,Ankara,Turkey.

snow free surface (class 3 system).In addition, they did not recognize heat loads which are associated with adverse atmospheric conditions following the snow. Generally,the design of snowmelting systems still rely on intuition or simple rules. Consequently an optimization search which is crucial for the success of the system can not be properly made. On the other hand,ASHRAE guidelines do not provide design rules for systems using plastic or rubber hydronic elements. In a typical design,the primary objective is to optimize the system with respect to the arrangement and temperature of the heating elements,slab construction and the insulation,for minimum installation and operation costs. Any technique must recognize and sufficiently model the relevant relationships among all design variables. The heated slab must perform at design conditions such that the desired snow melting performance is accomplished.

2. THEORY

Heat diffusion in a heated slab is basically a two dimensional problem,if the temperature changes in the axial direction are neglected. Heat is transferred to the slab from the heating elements. If the system is adequately sized,heat delivered to the surface will melt the snow at the prescribed performance rate (class 1,2,or 3). At the same time,some of the heat is lost to the atmosphere from snow free surfaces by convection,evaporation and radiation. Depending on the degree of insulation,some heat will also be lost from the perimeter and back of the heated slab. Although finite element algorithms are now capable of predicting the heat diffusion very accurately,designers generally favor simplified and analytical hods with reasonable accuracy. In this respect, Okagaki and da (1968) developed an analytical formulation using nsformation techniques. They considered both steady and unsteady ditions with or without perfect insulation. This technique permits calculate the heat output from the heated slab at a given figuration and the surface temperature of the heating element. ever the iterative nature of the algorithm and complexity of the culations made it rather difficult to use. Schnurr and Rogers 970) developed an analytical algorithm to define the required ating element temperature in terms of the slab construction and ement spacing at steady state conditions. They were also able to edict the maximum surface temperature. Leal and Miller (1972) eloped a finite difference solution for transient temperature stribution in the slab. They assumed a perfectly insulated slab and snow free surface. They analyzed both the transient heating and oling of a typical slab for a given heating element temperature of 0°F and at the absence of snow fall. However their predicted rface temperature profile underestimates the extremes (t_{max} and in), mainly because they used the surface heat loss equation by apman and Katunich (1956) which was derived for snow melting nditions at $t_p \approx 33°F$ where this is not consistent with their mple solution with t_p about 90°F. Although above techniques are latively easy to implement on office computers or personal mputers, they did not gain much popularity. This necessitated to velop a more simplified and easy to use algorithm for hand lculations. In this respect the composite fin model proposed by lkis (1992) was employed for a steady state heat diffusion.

This model recognizes all of the three snow melting classes which are mentioned in ASHRAE guidelines (1991). Figure 1 shows the ideal surface condition for each performance class.

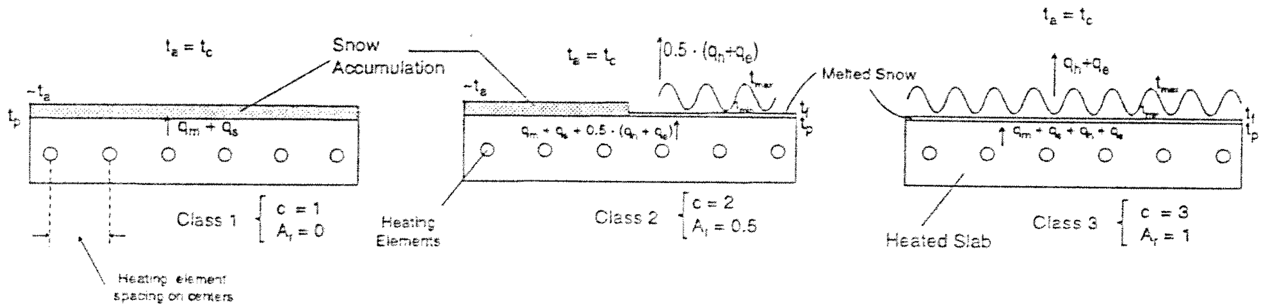


Figure 1. Surface Conditions for Different Performance Classes During Snow Fall.

In a class 1 system,entire surface is permitted to be covered with snow while it is snowing. This virtually eliminates heat losses from the surface to the atmosphere. In class 3, heat loss by convection, radiation and evaporation takes place from the surface. Class 2 is an intermediate case with a 50% snow free surface while it is snowing. Snow free area ratio (A_r) practically defines the performance class (See Fig.1). Figure 2 shows the composite fin model of the heated slab for steady state conditions. Here, t_f is the surface temperature of the water film,and heated surface temperature is t_p. Assuming a film thickness (x_i) like 1/16" (Adlam,1950),and treating this as a separate surface layer,by its definition, t_p becomes t_f. However in practice this film may be neglected.

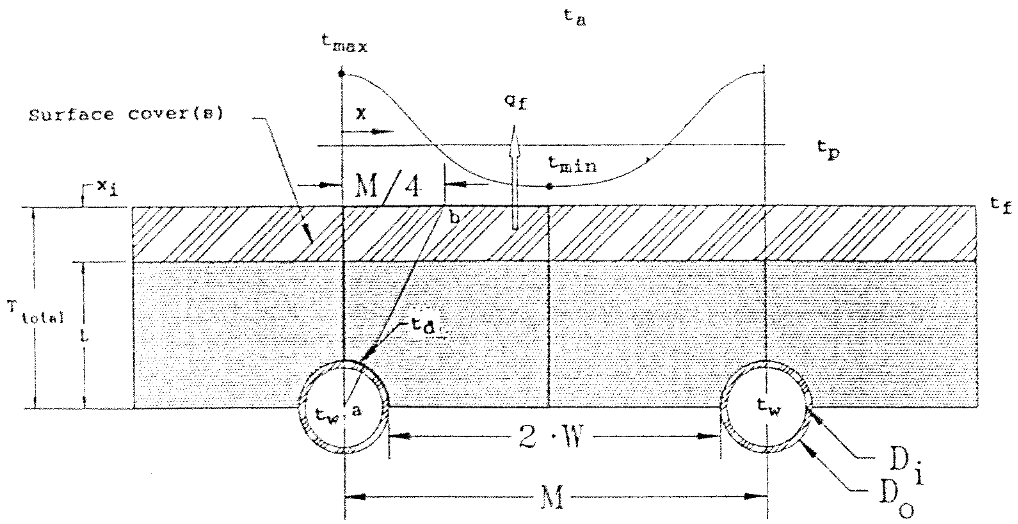


Figure 2. Composite Fin Model.

coefficient of the surface heat loss during the snow fall (h_f)
l be a function of the performance class and meteorological
ditions:

$$h_f = q_f/(t_p - t_a) = A_r \cdot (q_h + q_e)/(t_p - t_a) \quad [1]$$

a class 1 system ($A_r = 0$) surface heat loss,therefore h_f is
ctically zero.

class 1 and class 2 systems,the remaining snow needs to be melted
er the snow. Usually this phase is subject to more adverse
eorological conditions. Therefore the surface heat losses may be
her. If this is the case,for design purposes:

$$h_f = (q_{ha} + q_{ea})/(t_p - t_a) \quad [2]$$

the idling load during the freezing period is the critical
d,then:

$$h_f = q_i/(t_m - t_a) \quad [3]$$

is the air temperature. During idling and after the snow,it is
al to the design outdoor temperature (t_b).

For the snow fall conditions, t_d is replaced by the snow coincident temperature (t_c) :

$$t_c = t_b + \frac{(33 - t_b)}{(0.1 + 1.2 \cdot C)} \qquad [4]$$

Efficiency of the fin will be:

$$\cap = \tanh(m \cdot W)/(m \cdot W) \qquad [5]$$

where;

$$m = 3/4 \cdot [h_f/(k_e \cdot T_{total})]^{1/2} \qquad [6]$$

The factor 3/4 is the two dimensional heat diffusion correction for one dimensional slab model, which is correlated from finite element solution comparisons.

In class 1, the fin efficiency will be unity during the snow because h_f is zero. Here k_e is the equivalent thermal conductivity for lateral heat diffusion:

$$k_e = \frac{\sum_{i=1}^{n_k} (k_i \cdot x_i) + k \cdot L}{T_{total}} \qquad [7]$$

The corresponding maximum surface temperature is solved from the following equation:

$$(2 \cdot W \cdot \cap + D_o) \cdot h_f \cdot (t_{max} - t_a) = q_f \cdot M \qquad [8]$$

Therefore,

$$t_{max} = t_a + \frac{q_f/h_f \cdot M}{(2 \cdot W \cdot \cap + D_o)} \qquad [9]$$

re :

$$q_f/h_f = t_p - t_a \qquad [10]$$

ation 9 overestimates the temperature by about 5% when compared h finite element solutions.

ecially when the spacing between heating elements is wide,the imum surface temperature (t_{min}) at midway between adjacent ting elements,may drop below 32°F causing patches of ice remaining the surface. In order to avoid this condition, t_{min} must be cked with the condition $t_{min} \geq 32°F$:

$$t_{min} = (\cosh[m \cdot W])^{-1} \cdot (t_{max} - t_a) + t_a \qquad [11]$$

order to satisfy this condition, t_{max} must be adjusted. Noting t (m) stays relatively constant with t_f (see sample design) s can be directly accomplished by using Equation 11 for $_n \geq 32°F$. Following this operation, t_p may be adjusted by using ations 9 and 10:

$$t_p = t_a + \frac{(t_{max} - t_a) \cdot (2 \cdot W \cdot \cap + D_o)}{M} \qquad [12]$$

IRAE guidelines assume that t_p is 33°F and uniform on the surface. s assumption is not validated with the above statements.

er determining the surface temperatures,the required surface perature of the heating element can be calculated. A simple roach is to use the vertical path between the element and the face:

$$t_d \approx t_{max} + q_y \cdot [R_{cc} + (L - D_o/2)/k] \qquad [13]$$

Krinninger (1989) and Leal and Miller (1972) assumed that surface temperature profile is Sinusoidal. In this case the local surface temperature at $x = M/4$ coincides with t_p. Following this argument, one may use the temperature gradient along path a-b (see Fig.1) in order to calculate t_b.

The total design heat load intensity to be delivered by the heating elements to the surface is the highest of the following load intensities which correspond to different phases of snow melting operation:

$$
\begin{aligned}
q_y &= q_m + q_s + A_r \cdot (q_h + q_e) && \{ \text{during snow} \} \\
q_y &= q_{ha} + q_{ea} && \{ \text{after snow} \} \qquad [14] \\
q_y &= q_i && \{ \text{during idling} \}
\end{aligned}
$$

For a detailed review of these equations, one needs to refer to the first part of this study. It is not quite uncommon that the critical surface temperature profile and the critical load intensity may take place at different phases of operation. This means that each applicable phase for a given class should be considered in selecting the critical design variables for sizing the heat (power) source and the element surface temperature.

For a hydronic system, the required mean water temperature is:

$$T_w = \frac{q_y \cdot M}{X \cdot \pi} \cdot \left[\frac{1}{\alpha \cdot D_i} + \frac{1}{2 \cdot k_h} \cdot \ln(D_o/D_i) \right] + T_d \qquad [15]$$

Here α is the convection coefficient in the hose at the given fluid properties and flow rate. X is the thermal efficiency of the heated slab. It is a measure of the back and perimeter heat losses. These losses were overlooked in the literature.

y authors simply assumed it 40% of the heat delivered to the ted surface. Back heat loss intensity (q_b) can be approximated Krinninger's approach for the heat flow, namely by defining a aight path originating from the center of the heating element and ing in the ground at about 1 feet deep at an offset of (M/4) m the element center. Thermal resistances along this path are led. The corresponding temperature gradient is the difference ween t_d and the ground temperature at 1 feet below the surface $_g$). Comparisons with finite element solutions for typical cases licate an expected accuracy of 10% or better in terms of q_b. This inslates to about 3% accuracy in terms of thermal efficiency. The imeter heat loss intensity (h_a) can be approximated by following guidelines given by ASHRAE for floors on grade (1992). Then:

$$X = \frac{q_y \cdot \sum_{i=1}^{n_p} A_p}{\left[(q_y + q_b) \cdot \sum_{i=1}^{n_p} A_p \right] + Q_p} \qquad [16]$$

ere;

$$Q_p = h_a \cdot l_e \cdot ((t_d+t_p)/2 - t_a) . \qquad [17]$$

latter equation assumes that the bulk of the slab is at a mean nperature of ($t_d + t_p$)/2. h_a may be obtained from Table 7.1, in RAE Load Calculation Manual, Chap.7 (1992). This is for a snow e ground surface on the perimeter. In a snow melting design, this rface will be covered with snow which will reduce the perimeter loss. Therefore Equation 17 will render a conservative value for the thermal efficiency (see the numerical solution). It must also be noted that thermal efficiency will not stay constant during different phases of snow melting operation because of the changes in heat load intensities and the air temperature.

If the space at below is exposed to the atmosphere (Williamson,1967):

$$q_b = (1.4 + 0.13 \cdot U) \cdot (t_s - t_a) \qquad [18]$$

In a hydronic system,temperature of the heating element is important because it is primarily limited by the properties of the materials that it is made of. A typical limit on the mean fluid temperature is 170°F or less for a non-metallic pipe material. The temperature of the heating element is less significant in an electrical system which is used only for determining the electrical resistance of the cable at the operating temperature. However the temperature of the element may become more significant for any system if the temperature gradients in the slab are high enough that thermal stresses become critical. Shirakawa and et.al. (1985) have numerically investigated the principal stress distribution in a reinforced slab with 1.5% steel fiber content. They calculated that when the temperature difference between the slab surface and the heating element surface is 35°F,the tensile stress reaches a typical value of 420 psi. They have also concluded that the maximum tensile stress takes place in a direction perpendicular to the axis of the heating element. Using their data,one may correlate the tensile stress with the boundary temperatures along this axis:

$$\sigma_t \approx 12 \cdot (t_{max} - t_d) \qquad [19]$$

ation 19 may be used to determine the safe allowable temperature dient if the flexural cracking strength of the slab is known:

$$(t_{max} - t_d) \leq \sigma_c /(n \cdot 12) \qquad [19]$$

is suggested to apply a safety factor (n) not less than two.

SAMPLE DESIGN

lass 3 hydronic snowmelting system is to be designed. Dimensions the snow melting surface are 25 ft by 40 ft. Six identical ronic circuits serve the system. 3/4 inch I.D. rubber hose is used h an 11" spacing on centers. There is no insulation except the imeter. Constructional details are given in Figure 3.

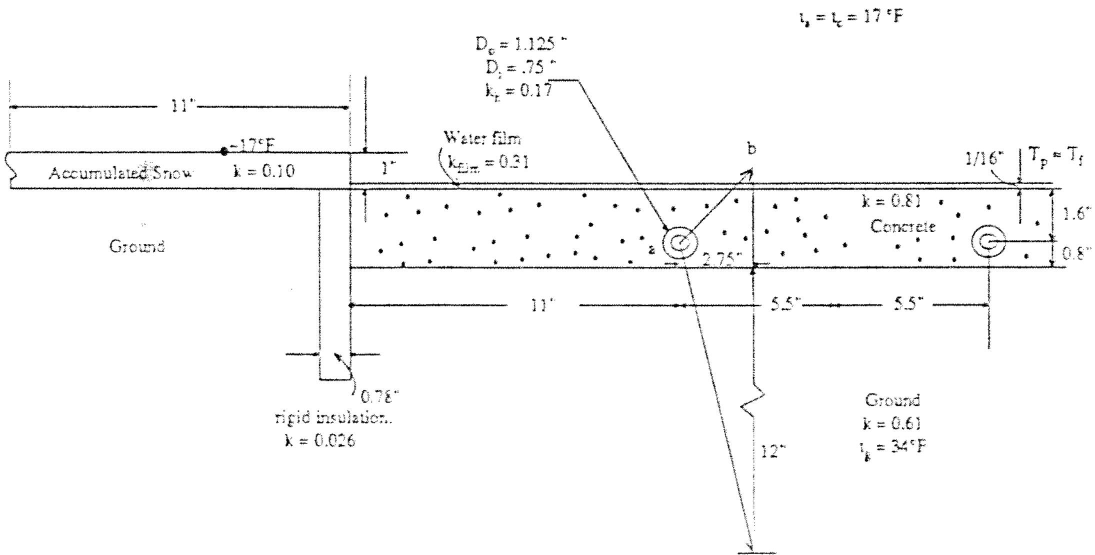


Figure 3. Constructional Details of the Sample Design.

er design inputs are:

formance Class : 3 ; (A_r : 1)

location : Open field/ground level surface.

ign Wind speed : 10 mph ; Adjusted wind speed : 8 mph

Heating degree-days : 7000 d-d

Rate of snow fall : 0.16 in. of H_2O/h

n_p : 3 ; Ap : (25 · 40)/3 ≈ 333 ft²

l_e : 130 ft. ; h_a : 0.90 Btu/h ft °F

t_b : 10.4°F (-12°C) ; t_m : 20°F ;

t_g : 34°F; (thermal cond. of earth : 0.6 Btu/h ft°F

d_i : (3/4)/12 ft ; d_o : (1.18)/12 ft

k_h : 0.17 Btu/h ft°F ; k : 0.81 Btu/h ft°F ; σ_c : 900 psi

[illegible] : 11/12 ft

Calculated inputs for the design are:

R_{co} = 1/16"/(12 · th.cond.of melted snow) = 0.016 h ft² °F/Btu

Step.1- First,assuming that t_p is equal to 33°F, snow melting load intensities were calculated (see the first part of this study):

q_m = 6.2 Btu/h ft²

q_s = 119.4 Btu/h ft²

q_e = 17.9 Btu/h ft² ; q_{ea} = 0

q_h = 42.0 Btu/h ft² ; q_{ha} = 0

q_y = 185.5 Btu/h ft² (Eqn.14)

q_i = 71 Btu/h ft²

Calculations:

t_c = 17°F (Eqn.4)

t_a = t_c (during snow fall)

h_f = (17.9 + 42)/(33 - 17) = 3.74 Btu/h ft² °F (Eqn.1)

k_e = 0.79 Btu/h ft°F (Eqn. 7)

W = 4.91/12 ft

m = 4.38 ft^{-1} (Eqn.6)

= 0.41 (Eqn.5)

x= 53.8°F (Eqn.9)

n= 28.9 °F < 32°F (Eqn.11)

urning to Step.1 with a second trial value for t_p = 40°F ;

= 6.2 Btu/h ft² (not effected)

= 119.4 Btu/h ft² (not effected)

= 27.7 Btu/h ft²

= 57.9 Btu/h ft²

= 211.2 Btu/h ft²

= 85.6 Btu/h ft² ; hf : 85.6/(40 - 17) = 3.72 Btu/h ft² °F

= 4.37 ft^{-1} (Eqn.6)

= 0.52

x= 63°F (Eqn.11)

n= 32°F

= 43°F (Eqn.12)

face temperature is close enough to the assumed value for design poses. This operation shows that by assuming t_p = 33°F, snow ting load would be underestimated by about 12% and there would be patches (unmelted snow) on the surface. It must be noted that surface temperature variation could be minimized by reducing the cing of elements,M (see Eqns. 5 and 9).

n;

= 88°F

same temperature is calculated to be 96°F by using Krinninger's roach (1989). Checking for thermal stress;

= 3 (Eqn.20)

= 31.6 Btu/h ft²

q_p = 5674 Btu/h (Eqn.17)

X = 0.85 (Eqn.16)

α = 400 Btu/h ft²°F (50 % P.Glycol/water mixture)

T_w = 187°F

Fluid temperature which seems to be high may be reduced by decreasing the spacing of heating elements.

5. NUMERICAL SOLUTION

The above example was modeled by a finite element analysis routine, namely ANSYS Engineering Package version 4.4A. The boundary conditions and the model is shown in Figure 4. The sides of the snow melting surface is assumed to be covered with snow with a surface temperature equal to t_c. This surface is permitted to exchange radiation heat with sky. The surface temperature variation is shown in Figure 5. The temperature distribution in the heated slab is shown in Figure 6.

Main results of this analysis were:

T_{max} : 59°F ; T_{min} : 34°F ; X : 0.82 ; T_w : 182°F

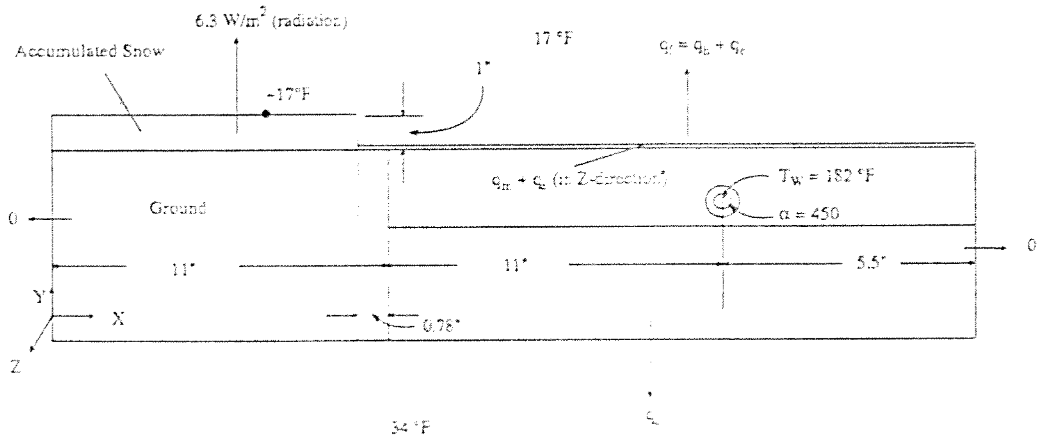
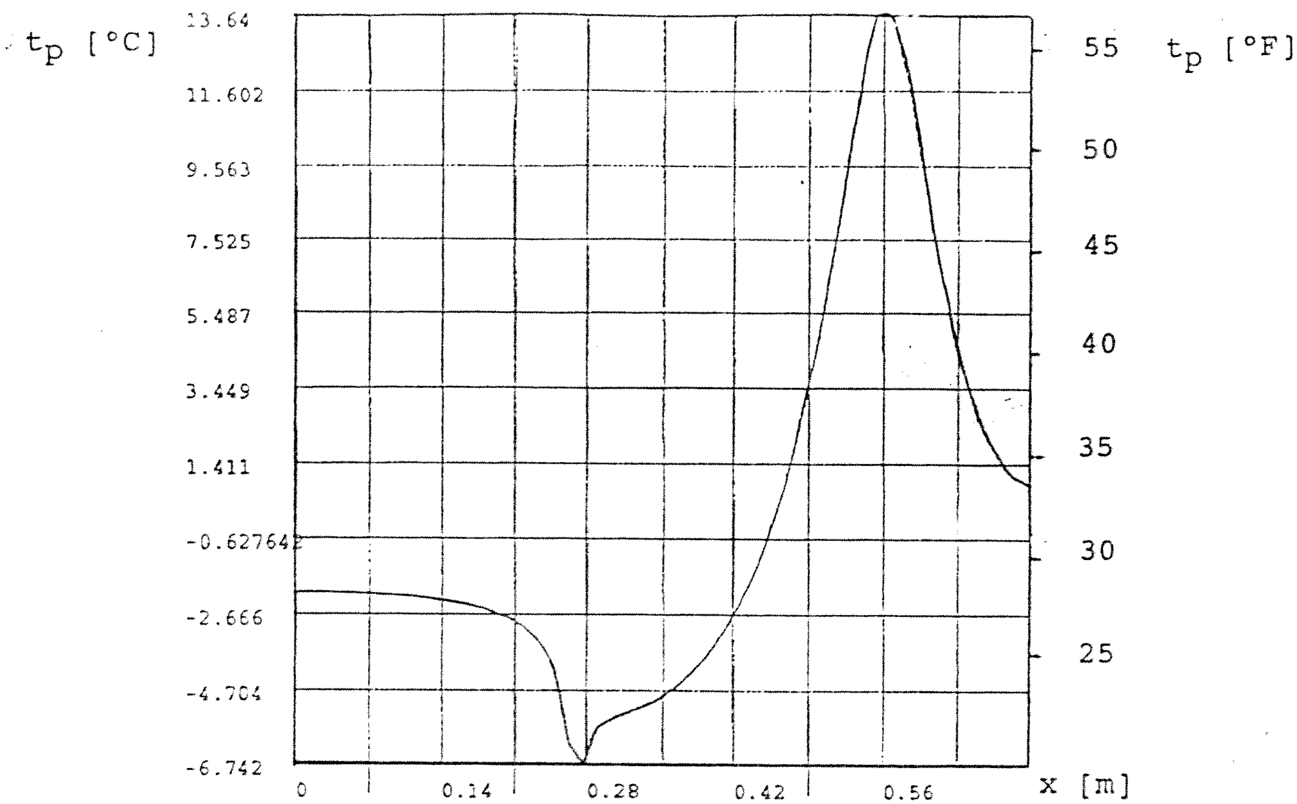


Figure 4. Idealization of the Problem for Finite Element Analysis.



igure 5. Temperature Distribution on the Snow Melting Surface.

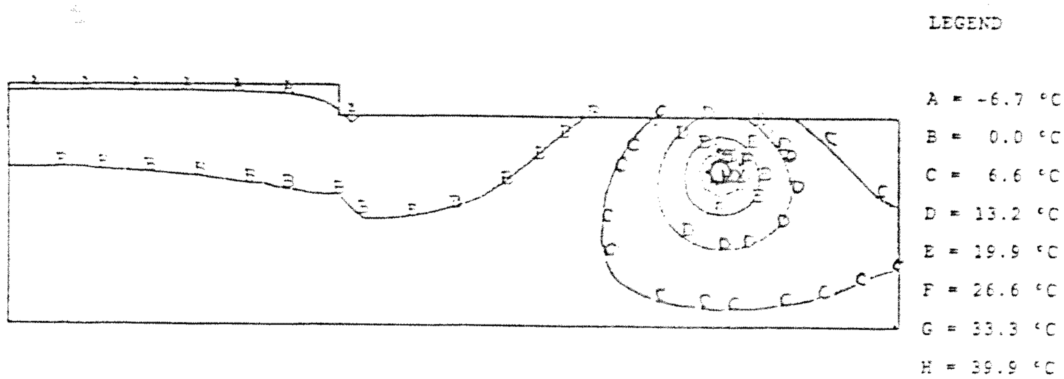


Figure 6. Temperature Distribution in the Heated Slab.

DISCUSSION of RESULTS

above case study and others indicated that the proposed algorithm accurate enough for engineering calculations for a snow melting b. The algorithm permits to adjust for different layers at top of slab and important parameters like spacing of the elements, hose ensions and properties, slab construction, insulation level can be changed in order to optimize the system. Furthermore it allows to predict the surface temperature variation and ensures that entire surface will be maintained above the freezing temperature.

7. NOMENCLATURE

A_p Surface area covered by a section of the heated slab (panel) which is served by a single circuit of the heating element, ft^2

A_r Snow free area ratio, dimensionless

C Snow melting performance class (1,2,3), dimensionless

D_i Inside diameter of the heat transfer hose, ft

D_o Outside diameter of the heat transfer hose (electric wire), ft

h_a Linear heat loss coefficient for the perimeter, Btu/h ft °F

h_f Coefficient of surface heat loss intensity, Btu/h ft^2 °F

k Thermal conductivity of the slab material, Btu/h ft °F

k_e Equivalent thermal conductivity of the composite slab for lateral heat diffusion, Btu/h ft °F

k_h Thermal conductivity of the hose material, Btu/h ft °F

k_i Thermal conductivity of each layer on the slab, Btu/h ft °F

l_e Perimeter of the snow melting slab, ft

M Heating element spacing on centers, ft

m Fin coefficient, ft^{-1}

n Mechanical safety factor against cracking of a concrete slab, dimensionless

n_k Number of layers at top of the heated slab, dimensionless

n_p Number of circuits (panels) in the snow melting system, dimensionless

Back heat loss intensity from the heated slab, Btu/h ft^2

Evaporation heat loss intensity during snow, Btu/h ft^2

Evaporation heat loss intensity after the snow, Btu/h ft^2

Total surface heat loss intensity, Btu/h ft^2

Surface heat loss intensity by radiation and convection during snow, Btu/h ft^2

Surface heat loss intensity by radiation and convection after the snow, Btu/h ft^2

Idling heat load intensity, Btu/h ft^2

Intensity of heat of fusion, Btu/h ft^2

Heat loss from the perimeter, Btu/h

Radiation heat loss intensity during snow, Btu/h ft^2

Intensity of sensible heat, Btu/h ft^2

Radiation heat loss intensity after the snow, Btu/h ft^2

Design heat load intensity, Btu/h ft^2

Total thermal resistance of layers above the slab, h ft^2°F/Btu

Air temperature, °F

Outdoor design temperature, °F

Snow fall coincident design air temperature, °F

Outside surface temperature of the heating element, °F

Melted snow film temperature, °F

Ground temperature, °F

Daily mean air temperature during freezing period, °F

x Maximum surface temperature, °F

n Minimum surface temperature, °F

Effective temperature on the surface, °F

t_s Surface temperature of the back of a heated slab if it is exposed to outdoor air, °F

T_{total} Thickness of the composite fin, ft

T_w Mean fluid temperature in the hydronic circuit, °F

U Adjusted wind speed, mph

W Half of the net spacing between adjacent heating elements $(M-D_o)/2$, ft

X Thermal efficiency of the heated slab, dimensionless

x_i Thickness of any layer on top of the heated slab, ft

Greek symbols

α Convection heat transfer coefficient between the fluid and the hose, Btu/h ft² °F

$\cap$ Fin efficiency of the composite fin, dimensionless

σ_c Flexural cracking strength of the slab material, psi

σ_t Tensile stress in the slab, psi

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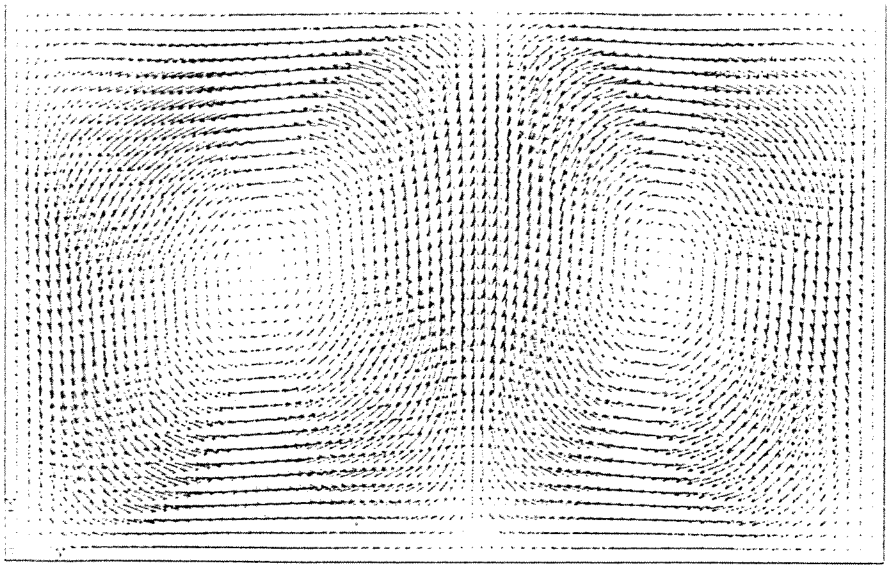

igure 1. Ideal Conditions without active surface cooling.

n buildings, wall surfaces are not at ideal conditions:There colder surfaces like windows and outdoor walls and warmer faces like indoor walls.These create additional convective loops. the test set up, no provisions were made to simulate a typical m.There were no window simulations,or simply cold surface ulations,and obviously no infiltration at all.

ir temperature was measured at 1.5 m level where under test ditions is was most probably the coldest spot due to incomplete circulation loops.This further decreased the calculated vective coefficient in the room,because of the calculated high perature gradient between the 1.5 m level and the ceiling face.They took some measurements at very close locations to the face.Although they claim that they used this data also in their relations,one can not explain this,because Figure 7 reveals much ner convection coefficients.This should be expected because the surements must be taken at 5 to 6 cm from the surface for ctical reasons. (They measured the temperature at about 2.5 mm n the surface).

reason ,in practice,heat loads can be met by heating only 50% up to 70% of the ceiling's total area. This kind of sufficiency,combined with the desire of creating cold strips on the ceiling for increasing the natural convection, there are always intentional or unintentional cold strips on the ceiling surface. This practical situation creates higher convective heat transfer rates.(See enclosed list of coefficients by Kollmar,1957).Min's experimental conditions do not reflect this practical situation and general design approach.

2-Walls were actively cooled:In order to maintain a constant and uniform surface temperature,walls and the floor was in fact cooled by the circulating liquid.This practice also cooled the air wetting the adjacent walls after being heated up by the ceiling panel.Being effectively and rather immediately cooled by the adjoining surfaces the indoor natural air circulation can not complete its loop.Incompletion of the circulation virtually halted the natural convection loops in the room. If the surfaces were not actively cooled there would be symmetrical loops in the room. A similar situation is plotted in Figure 1. Here there is no effective cooling. Figure clearly indicates a systematic and symmetric convective loop pair under ideal conditions.This also helps to increase the air temperature to normal levels at 1.5 m level in the center of the room.

NDIX 1

able reasons for low convection heat transfer measurements in and et.al's experiments.

enclosed paper of Min and et.al. provide the following rmation about their test rooms and procedures:

unheated surfaces were kept at the same and constant temperature.

constant surface temperatures were maintained by "Circulating uid" in the the walls and floor.

temperature measurements were primarily taken at 1.5 m. level

e film temperature measurements were also taken (Figure 7)

ire ceiling was heated

the walls consisted of solid surfaces.

experimenting in the test room with above mentioned configuration, rols ,testing and measuring methods, they derived very small ral convection heat transfer measurements. Reasons such a conclusion which contradicts with general literature are:

tire ceiling was heated:The natural convection from a heated ling panel largely depends upon natural convection to colder oining surfaces including"cold strips on the ceiling itself". addition to this,ceiling surface temperatures may be chosen higher there are no surface obstructions like in floor heating.For this

Table 1.

Ceiling Heating Convection Heat Transfer Coefficient: Summary of Literature.

Author	Expression	Comments
Dulong and Petit (Ref. 10)	$0.64 \bullet (T_p\text{-}T_o)^{1.25}$	Completely still air, whole ceiling is heated.
Kollmar (Ref. 14)	$0.87 \bullet (T_p\text{-}T_o)^{1.25}$	Ceiling is partly heated (1m spaced)
Griffiths and Davis (Ref. 15)	$1.28 \bullet (T_p\text{-}T_o)^{1.25}$	Ceiling panels consisting of 1m wide strips. Presence of cold portions.
Min. T.C., et al. Krause (Ref. 12)	$0.2 \bullet (T_p\text{-}T_o)^{1.25}$ between 0.15 and $0.6 \bullet (T_p\text{-}T_c)^{1.25}$	Experimental rooms.
Cigdemoglu (Ref. 11)	$1.33 \bullet (T_p\text{-}T_o)^{1.25}$	Theoretical
Zhang and Pate	$0.40 \bullet (T_p\text{-}T_o)^{1.25}$	Used in their numerical simulation (citing Min and Krause data)
Nusselt and Hencky (Ref. 3)	$0.22 \bullet (T_p\text{-}T_o)^{1.25}$ (IPS) $1.44 \bullet (T_p\text{-}T_o)^{1.25}$(SI)	Theoretical, P.S: This is in IPS (see system. In SI system coefficients is 1.44.**

See also other correlations cited by Min and et. al, in their paper, Table 1. (This paper is enclosed to this report).

Note: Most of the correlations cited here and in Table-1 of Min and et.al's paper confirm the wide acceptance that in practice, convection heat transfer from a heated ceiling is around half as much as a floor heating panel's convective heat output.

** The conversion factor from IPS to SI is:
$5.67 \bullet (1.8)^{0.25}$

Note: If De term is included, the conversion constant changes with the units of $(D_e)^b$

Comment on the effect of the room size (D_e)

For the size effect on natural convection, the relationship is in the form of:

$$\frac{\alpha_L}{\alpha} = \left(\frac{x}{x_L}\right)^b \tag{1}$$

Here α is the convection film coefficient for room with the typical length x. Subscripts "L" denotes a large room. Equation (1) is the original form found in the literature and is applicable for rooms sizes greater than 10 meters. Min and et. al's correlation confirms 0.08 for floor heating in their experimental room which is about 3.65x7.30 meters Replacing x and x_L by D_e and basing the correlation for Min's experiments:

D_e of Min's experimental room $\cong 4 \bullet (3.65 \bullet 6.10)/([3.65 + 7.30] \bullet 2)$
$\cong 4.86$ m (Min reports this as 4.96)

Equations 6 and 7 should have the term $(4.96/D_e)^b$ {b = 0.08 or 0.25}

Proposed Fields of Research

A 3-D simulation of the indoor air and enclosure including the panels. Such a simulation will yield valuable data on:

a- Natural infiltration

b- Local air movements leading to local film coefficients for convection. This will especially very helpful to determine accurately the convection heat transfer from panel surfaces.

c- Accurate local surface temperatures like walls, ceilings and windows.

d- Indoor air temperature distribution, pressure and velocity distribution for comfort simulations

e- (a), (b), (c) and (d) will enable to accurately predict heat losses from the room and panel. This algorithm will enable to develop the specific heat load calculation algorithm and set forth the guidelines that may be included in chapter 6.

f- By having a complete 3-D simulation,the performance predictions will be morecomprehensive and accurate. By trying different room configurations and positions, the dependence of heat output on these will be clarified (especially for ceiling heating). Further details are provided in "comments on Heating load".

g- Underfloor stapled sub-floor panel heating is a widely applied technique, but there is not any evident literature addressing this configuration (Kilkis, 1992). A computer simulation will reveal the design guidelines and sub-floor panel heat losses which has not yet been mentioned in chapter 6.

h- In conjunction with T.C.4.3., leakage coefficients can be simulated through the research topic mentioned in (a) through (e).

i- Regression of meteorological data to determine the daily peak load shaving factorsfor different localities, building types, and heating panels.

j- Positive impacts for Heat Pump performance enhancement (Kilkis, 1992).

k- Experimental data to supplement the above´mentioned topics of interest with specialemphasis to:

-Floor surface Temperature distribution :Comfort levels, effect on general comfort (With Dr. Fanger).

l- Surface temperature distribution in ceiling cooling with special emphasis to local condensation.

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With new thermoplastic and rubber technology, the majority of applications are performed with non-metallic pipes (hoses). There are so many variations of these materials that a general classification, quality requirement, test and acceptance procedure must be underlined. Although radiant floor heating pipes of non-metallic nature are not the main sources of oxygen ingestion, this issue seems to be of concern, for this reason this topic must be addressed and precautions be explained. (like corrosion proofers, etc.).

Figure 11.

Due to large water volume present in the system, expansion tank sizing is very crucial. Design guidelines must be provided its importance emphasized and shown in figure 11 with air venting systems. The locations of such elements are very important. For this reason it will be very useful to explain their location on figure 11.

Applications

Today, the majority of residential homes are floor heated through joisted sub-floors. This is one of the types which faces many design mistakes in the industry. Details and design guidelines of this panel type must be definitely given and explained at least as much as the others (see paper by Dr. Kilkis, submitted to ASHRAE)

Enclosed:

- References
- Proposed field of research
- Sample nomograms, charts
- Convection Heat Transfer Table, Table 1.
- Comments on Heating load
- Comments on the effect of the room size (D_e)
- Appendix 1.

Heating (Cooling load).

Due to the nature of heating and thermomechanical behavior of the space heating load particularly in floor heating and sensible cooling load in ceiling cooling are reduced. This fact is not addressed. Although the term "The room heating and cooling loads are calculated in the conventional manner" has been removed in chapter 6, how the loads will be calculated is still not clear and there is not any specific guideline given. Heat storage thus peak load shaving is not mentioned at all (Lafontaine 1990, Kilkis 1990). I strongly believe that this is an inter committee issue like among T.C.6.5, T.C.4.1, T.C.4.3 and T.C.6.4. By a cooperative research and literature survey, a specific heat load design guideline must be developed (see "research topics"). A report for T.C.4.1 has already been prepared and will be submitted during their committee meeting at ASHRAE meeting in Baltimore. A copy is enclosed which gives the details of this topic and emphasizes its importance.

Terminology

1. Tp: Panel temperature. This is in fact a fictitious temperature representing a plate (panel) surface with uniform temperature. particularly with large hose spacings, surface temperatures may appreciably change (Krinninger, 1989, Zhang 1986) and may cause local discomfort. This must be explained and sufficient tools for its analysis may be provided.
2. Indoor comfort air temperature: This temperature must be re-defined and standard design values for various space functions may be provided.

Pressure Drop

In most o the hydronic applications Reynolds Number is in the order of 10,000, therefore friction coefficients must be based upon Reynolds Number dependent equations. Furthermore upper and lower limits for water velocities must be specified for various materials. For this reason standard pressure drop charts using fully rough conditions ($Re > 10^6$) like in figure 3, chapter 33, Fundamentals Handbook, 1989 may not be referred. A specific pressure drop chart is enclosed which is prepared with Re dependent friction factors. Point 18 in page 6.9 must be elaborated in this respect.

Comfort Conditions

Specific comfort conditions and comfort zone must be more precisely determined and re-specified for panel heating/cooling. (With Dr. Fanger, a meeting is planned during the meeting). See also (N.A. Buckley, 1989)

Boiler Load (Heating Efficiency)

Panel heat losses may be as much as 35% depending upon the application and design. This effects the boiler load and pressure drop calculations (total heat to be delivered to the panel). A clear definition of the boiler load will be helpful and it will clearly remind that water flow rate should be based on the boiler load.

Material Standards

certain hydronic problems.

-"Ceiling" curve in figure 3 should be redrawn (see HVAC guide 1956).

B.

The term D_e in equation 6 and 7 should be retained in equation 9 and 10, because the influence of the room size becomes important for large spaces like hangars, enclosed sports complexes, etc. which are recently quite common cases in panel heating. i.e.: For a hangar having 30 ft. x 150 ft. dimensions, size effect on convection will be above 27%!

The reason for changing the constants of equations 6 and 7 in equation 9, 10 and 11 must be explained (in fact, D_e is taken to be 4.996 meters).(see the explanation enclosed). Therefore it is more appropriate to add the term $(4.96/D_e)^{0.08}$ instead of $(1/De)^{0.08}$ to the first set of equations. This will explain equations 9 and 10 and they may be eliminated from the text. A similar approach will be quite suitable for equations 8 and 11.

C.

Altitude correction due to density effect of buoyancy, high altitude applications may need a correction (Kilkis, 1990). i.e.: in 3500 ft, convection heat output is reduced by 7%.

<u>Figures 7 and 8.</u>

Mean water temperature is a function of:

-Panel heat losses,
-Hose (pipe or tube) diameter, thickness and material,
-Hose spacing O.C.,
-Heat delivered (thus convection on the water side),

In addition to the parameters been already accounted in these figures.

Especially hose type and hose spacing O.C. is very effective on the average water temperature. (Kilkis, 1992, Kilkis, 1990, Raiss, Roedler, 1969, Various U.S. manufacturer's data). A typical design nomogram is attached in order to underline this situation for a given hose material and type. In this nomogram it is quite clear how much hose spacing O.C. will effect the average water temperature.

Therefore:

(i) These figures must be removed.

(ii) Instead, governing equations for determining the average water temperature and others must be provided. This will in fact make the design more accurate and much simpler. (Kilkis, 1992, Kollmar, 1957, Krinninger, 1989).

(iii) The note stating that "manufacturer's data for specific system selected should be referred" may need rephrasing because with very few exceptions, like the one enclosed, no complete manufacturer's data is available in the industry. At the moment, ASHRAE must guide the industry through the information contained in chapter 6. Industry will then be able to generate reliable, respectable, and complete data. After this phase and completion of this mission, ASHRAE may consider to remove this material from the chapter.

paragraph states that Min. et. al. determined natural convection coefficient at a location 1.5m above the floor in the center of a 300 mm by 600 mm room (12x24 ft.). First of all, a 12x24 ft. room is not 300 x 600 mm but approximately (3657x7314 mm). Although Min. and et.al's corrections ended up with similar expressions for floor and wall panels, it largely deviates from available literature for ceiling heating. There are two main reasons for this:

1. In ceiling heating, warm air accumulates below the ceiling panel. Therefore, in ceiling heating, natural convection is only an indirect result of the action of cooling along the outside walls instead of direct convection downward. Therefore, this factor should be taken into account for corrections(Heating, Ventilating, Air Conditioning Guide, 1956). Apparently Min. and et. al.'s data represents a test room without any simulated cold outdoor wall or window. Therefore their data represents the worst case scenario where there is not any cooling action along the outside wall or walls which in fact causes the air movement. In practice, except inner rooms like the kitchen, reading room, etc., this secondary convection action is very important. It may be concluded that natural convection in ceiling heating is very sensitive to AUST, as well as the size and arrangement of the ceiling panel (i.e.: in strips, or full ceiling (Kollmar, 1957)), in contrary to wall and floor heating. For this reason, equations 6 and 9 are not comprehensive enough, and do not represent practical situations. To provide only Equation 6 in chapter 6 may be totally misleading. There are too many reported analytical and experimental reports of US or European originated authors including some of their papers published by ASHRAE publications (see table 1 and accompanying explanation). (see also Min's paper table 1.) These should also be referenced, and suitable equations with their limitations be reported in chapter 6. (see also appendix 1 attached to this text).

2. The airstream temperatures were measured at 1.5m height which corresponds to human comfort level. Such an approach may be justifiable for human comfort conditions and a common base comparison., However in this chapter the prime issue should be the balance of the heat loads and heat outputs, and definitely none of them correspond to this height.
   In ceiling heating due to strong thermal gradients in the vertical direction, the air temperature at 1.5 m of height is quite different than that of 50 to 65 mm below the panels. From the definition of convective film coefficient, it is therefore quite normal to obtain very low convection values. (see app. 1).
   The reason why equation 7 and 8 are in better agreement with the literature (i.e. in German literature, equation 7 reads: $2.67 \bullet (T_p - T_o)^{1.25}$) is due to the fact most air temperature does not change much in a vertical direction starting from the floor. Therefore error involved in measuring air temperature at 1.5m is much smaller than ceiling heating case. In addition, floor and wall convection primarily relies on direct convection from heated panel to the immediate air stream. For this reason correlations can represent the reality more satisfactorily and any such correction will be far too independent from AUST.
   Therefore:
   -Equations 6 and 9 should be critically reviewed and possibly replaced by more comprehensive and practical correlations, like the ones given in Table 1. Otherwise, equations 6 and 9 will effect the performance of a ceiling slab such that the panel will tend to output more than the design rating and this will cause

Suggestions and Preliminary Comments on Chapter 6., 1992 ASHRAE Systems Handbook (SI)

by
Prof. Dr. Birol Kilkis
Director of Research and Development
HEATWAY

Page 6.2 Equation 2:

(i) Here q_r^2 should be q_{r2} (indicating two-surface enclosure. Otherwise it maybe confused with the square of q_r).

(ii) The original radiation equation given in the literature is: $q_r = \sigma Fe[T_p^4 - T_r^4]$. Equation (2) should not have (1/100) terms. (Ref: Holman, Guyer)

Equation 5-a and 5-b:

(i) Tp is in °C therefore to convert it into Kelvin, constant is 273 not 460 (See chapter 8, pp. 8-4., Systems Handbook, 1984, Equations 3 and 3a)

(ii) The coefficient 0.15 is for (IPS) system. (See Chapter 8, 1984 Systems Handbook, Equation 1).

$$\text{Proof: } 0.1713 \text{ Btu/ft}^2 \bullet {}^{\circ}\text{F}^4 = 0.1713 \bullet 0.252/(0.3048)^2 \bullet (1.8)^4/.86$$
$$= 5.67 \text{ W/m}^2\text{ K}^4$$

This is confirmed by (Roedler and Raiss). Also see (Kilkis, 1992).
Therefore:
Equations 5a and 5b should read: (after 0.87 emissivity)

$$q_r = 4.93[((T_p+273)/100)^4-((AUST + 273)/100)^4]$$

(iii) Furthermore, it will be more convenient to express this term in a linear form. (See Guyer, pp. 1-98 and Kilkis, 1992). On the same page, a reference to equation 5a is made with IPS unit and quoted "0.152" (after Min. et. al.). In SI this must be "4.32").

Figures 1 and 2 are correct, and obey the above equation given in this text. This also proves that equations 5a and 5b need to be corrected.

Convection Heat Transfer

A.
Page 6.3, second paragraph, first sentence: This sentence is correct, but the data taken from Min et. al contradicts this statement.
i.e. The first sentence states that the most consistent measurements are obtained at about "50 to 65 mm below the panels", (In fact, this sentence should end like: below, above, or beside the panel depending on the location of the panel). The following

A preliminary Report Submitted Respectfully to

T.C.6.5

June 28th, 1992

by

Prof. Dr. Birol Kilkis

Heatway

ndix 2

Entran II™ Specifications and Test Results as reported by the Goodyear Laboratory.

#	Performance Specifications	Performance Test Results
380		
ide diameter:	+/- 0.6 mm	3/8" Entran II™: 9.5 +/- 0.6 mm 1/2" Entran II™: 12.7 +/- 0.6 mm
tside diameter:	+/- 0.8 mm	3/8" Entran II™: 16.8 +/- 0.8 mm 1/2" Entran II™: 21.0 +/- 0.8 mm
rst strength at 23 C. min- ım burst pressure:	800 PSI	839 PSI
igth change between 10 1 200 PSI:	+/- 10%	+ 5%
412 / D-380 nsile strength and elongation ubber.		
@ 23 C. Tensile Strength:	Tube - 5.0 MPa min. Cover - 7.0 MPa min.	Tube - 9 Mpa Cover - 8.5 Mpa
@ 23 C. Elongation %,	Tube - 200 min. Cover - 300 min.	Tube - 285% Cover - 433%
413 / D-380 hesion Tests		
@ 23 C. Tube to Yarn N/inch Cover to Yarn N/inch	27 minimum 27 minimum	52 N/inch 46 N/inch
471 mersion tests for change in lume and deterioration of hose ed for petroleum products. ote: working fluids can be substituted for ASTM oil.)		
immersion ASTM #3 oil 70 hrs. @ 100 C; Volume Swell	Tube - 25%	Tube - + 10.7%
rsion Distilled water 168 hrs	Cover - 100%	Cover - + 56.6%
00 C; Tensile % change Elongation % change Volume Swell %	- 25 - 40 - 0 to 25	+ 1.1% - 0.4 % + 20.1 %
rsion 50/50 Ethylene Glycol 168 hrs. 00 C; Tensile % change Elongation % change Volume Swell %	 - 25 - 40 0 - 25	 0.0% - 4.9% +16.1
380 w Temperature Test on mplete Hose. 5 hrs. Bend a circle,	diameter 10 times O.D. of hose.	No cracking
1149 ing tests of rubber @ 40 C. 70 hrs. partial essure of ozone	100 mPa.	No cracking

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it. Other factors such as strength, UV stability, temperature stabilization, etc. are more important." [3]

1) Strength: Entran products are designed and tested to withstand in excess of 800 psi at 70° F.

2) UV Stability: Plastic pipes begin to suffer irreversible losses of mechanical properties after as little as two weeks of exposure to UV from sunlight. Depending on the type of Entran product specified, sunlight exposures ranging from six months to more than five years will not result in any loss of mechanical properties.

3) High Temperature Resistance: A very real cause of boiler failure in non-metallic radiant systems results from the failure of boiler aquastats, flow switches, low water cutoff devices, and primary boiler pumps. A failure of any one of these components will cause a rapid rise in the outlet temperature of a hot water boiler, temporarily transforming a hot water boiler into a steam boiler.

Under these low pressure steam conditions polybutylene, polyethylene, and even cross linked polyethylene will quickly fail. The standard 30 psi relief valve found on most hot water boilers will not protect any of the nonmetallic radiant piping products, except a reinforced hose, from catastrophic heat related failure.

4) Mechanical Properties: Under unusual, but foreseeable, operating conditions, pressure surges in the system may take place. For this reason it is desirable to have a product with redundant properties. A reinforced hose is not susceptible to hidden damage from kinks, nor will it unpredictably fail at a later date from stress fractures caused by impacts sustained during the installation or transportation process.

Entran hose offers this safety factor. Engineering calculations reveal that an optimized hose design is subject to less than half the hoop stress of a typical unreinforced plastic pipe. Generally speaking, the lower the hoop stress, the longer the life to failure at a given pressure and temperature.

A reinforced composite hose is not subject to stress cracking, kinking, crushing, or the kind of mechanical damage that can lead to sudden failure of plastic pipe.

5) Summary of Entran Advantages: If radiant pipes can conceivably be exposed to temperatures in excess of 250 F, excessive pressures, or sunlight; then hose is the only acceptable choice.

If water might ever freeze in a slab, hose is the only acceptable choice.

The use of a reinforced composite hose is the conservative choice for construction professionals who are interested in a tough durable product that will withstand the kind of abuse found on typical construction sites.

Appendix 1

Corrosion Promoting Conditions [7,8]

1) Oxygenated water, due to system faults. These can include
   - a) Micro-air leaks at screwed/compression joints.
   - b) Incorrect placement of air venting devices.
   - c) Loose packing around valve stems.
   - d) Castings with micro-fractures.
   - e) Loose pump flange gaskets.
   - f) Failed or undersized diaphragm expansion tank.
   - g) Diffusion of oxygen through non- metallic piping

2) Naturally occurring impurities in the supply water such as chlorides, sulfates and peat acids. "Soft water", also, is generally more corrosive than "hard water".

3) Chemical impurities introduced into systems such as corrosive residues of soldering flux and sodium chloride from water softeners.

4) The presence of dissimilar metals and other conditions which create a "potential difference" leading to electrolytic corrosion, with water acting as the electrolyte, e.g.
   - a) Steel, cast ion, copper, brass and aluminum present in a variety of combinations
   - b) Copper shards or waste settling onto steel forming short-circuited electrolytic cells.
   - c) Copper plating out onto steel from flux residues or in areas of cupro-solvent potable water supplies.
   - d) Anodic/cathodic crystals or areas on the internal surface of steel components
   - e) Carburised steel in the vicinity of welds.
   - f) Carbon deposits formed as a result of welding oily or greasy steel.

5) Sediments or substances of any kind in contact with steel, leading to crevice corrosion.

6) The use of unsophisticated corrosion inhibitors which allow corrosion to concentrate on selected areas or spots. The use of "industrial type" corrosion inhibitors or water treatment agents which depend on daily or weekly monitoring is particularly unsuitable for radiant or other "domestic type" heating systems. These agents may cause accelerated corrosion if they are not maintained at very specific concentrations.

7) Stressed steel, as for example in pressed steel radiators.

8) The presence of organic matter, particularly anaerobic sulfate reducing bacteria.

9) Stray electric currents, caused by an earth leakage fault on site.

ied with this problem is the common misplacement of system mps so that they are pumping into the expansion tank, greatly lucing its capacity to absorb the expanding volume of heated ter without the system discharging through pressure relief ves.

Finally we have the issue of oxygen permeation through the lls of non-metallic piping itself. R. Scharmann [2] reports that y 10% of the oxygen necessary for the process of a medium ed corrosion may come through "oxygen permeable" piping. obahazi come to milar iclusion his ter. [6] should e here Dr. armann's ervation s not d for the y meable, -barrier DM ing sold in North America, often used for greenhouses and low projects. This material is not commonly used in Europe.

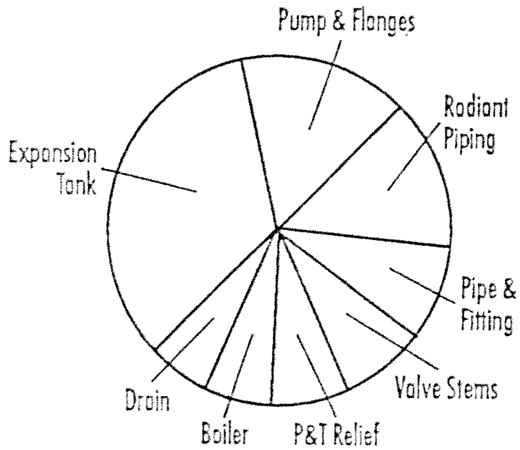

Figure 2. This pie chart shows approximate allocations of which some parts are responsible for oxygen ingress.

ortunately, the use of "barrier" pipe can often induce a false se of security in the minds of the both the owner and contrac- lessening the chance that regularly scheduled preventive ection and maintenance takes place.

LCULATION OF OXYGEN RMEATION IN A SYSTEM

s have been developed and procedures established to actually sure the amount of oxygen moving through the walls of ant piping at different temperatures. To actually calculate the ount of oxygen one must know two things: 1). The perature(s) at which the system operates, and 2) How many rs a year the system will be at these operating temperature(s). must then calculate the surface area of the radiant piping. By iplying this radiant piping surface area by the permeability ficient (for a given temperature), and dividing by the pipe wall kness, one can calculate the amount of oxygen moving ugh the piping on an hourly basis.

use there is a different piping permeability coefficient for i water temperature one must perform this calculation for ral temperatures, and assume how many hours the system will ate at each temperature to make a reasonably accurate ilation of the total oxygen moving into the system.

SUMMARY OF PERMEATION ISSUES

1) The amount of oxygen permeating through Heatway's™ Entran II™ piping ranges from 1% to 30% of the oxygen that normally infiltrates into any closed loop hydronic system. Another way of stating this is that if 100 units of oxygen would normally infiltrate into a baseboard type hydronic system, that the same system, with the substitution of Entran II™ for baseboard, would have a total of 101 to 130 units of oxygen a year infiltrating into the system.

By contrast, the substitution of plain EPDM tubing in this hypothetical system, would increase the total amount of oxygen present in the system to between 120 and 600 units.

2) Even using a product meeting the DIN 4726 oxygen permeability standard will by no means result in a corrosion-proof system. Plastic pipes meeting this standard have experienced continuing problems with delamination of the barrier layer, which can completely negate the benefits of the piping. This delamination process can begin with exposure to temperatures in excess of 140° F, or as little as two weeks of exposure to sunlight. Once begun, the process is irreversible, and negates any benefit the barrier may have had.

3) If the corrosion rate is to be reduced below the rates commonly referenced in the scientific literature the system must have corrosion inhibiting agents in the water, or the entire system must be constructed of naturally corrosion resistant materials.

4) Use of a isolating heat exchanger between the heating plant and distribution piping will eliminate any oxygen coming into the system through the radiant floor or distribution piping. This will protect the heating plant only from corrosion caused by oxygen originating in the distribution piping. Nevertheless, corrosion inhibiting agents or corrosion resistant materials must still be used in the distribution piping. Corrosion inhibiting agents should also still be used in the heating plant itself.

5) Even the practical elimination of all oxygen in the system, which is probably not economically achievable, will not eliminate all the other sources of corrosion referenced in Appendix 1. Therefore, the use of corrosion inhibiting agents is still strongly suggested.

A wise man once said "When you hear hoofbeats, look for the horses, don't look for Zebras". It's just as clear from the evidence in the preceding pages that in the unlikely event you were to find serious corrosion problems in a radiant hydronic system, you should look for the most likely cause first. This would include loose fittings, failed expansion tanks, and other problems common to all hydronic systems, as summarized earlier.

SELECTING THE CORRECT RADIANT PIPING

"The limited spread of these values show that the rate of oxygen diffusion for a given pipe should not be the sole reason to choose

CORROSION RATES IN NON-RADIANT HOT WATER HEATING SYSTEMS

Conventional heating systems are not immune from corrosion processes. It has been reported in the scientific literature that hot water heating systems (not incorporating floor heating piping) will experience significant corrosion. Typical corrosion rates reported vary from 1.5×10^{-3} to 5×10^{-2} inches of iron removal per year. Corrosion rates much higher than this have been reported in problem systems. [2]

Corrosion is caused by a wide variety of conditions. See Appendix 1 for a thorough list of corrosion promoting conditions. Section 1 from the Appendix is reprinted here for closer examination.

SEVEN SOURCES OF OXYGEN INGRESS INTO RADIANT SYSTEMS

These sources of oxygen ingress can include the following:

a) Micro-air leaks at screwed/compression joints.
b) Incorrect placement of air venting devices.
c) Loose packing around valve stems.
d) Castings with micro-fractures.
e) Loose pump flange gaskets.
f) Failed or undersized diaphragm expansion tank.
g) Diffusion of oxygen through non-metallic piping.

Note that with the advent of non-metallic piping in radiator systems, none of these sources of oxygen are peculiar to radiant floor heating systems. Any one, or all of these, can and does cause corrosion in conventional systems too. A degree of corrosion in conventional systems is simply accepted as a fact of life.

By their very nature, radiant floor heating systems are somewhat more susceptible to these system problems than other types of hot water systems. Because of this, radiant systems should be carefully designed, installed, and regularly serviced.

Unfortunately, as a result of competitive market pressures in Europe, oxygen diffusion has often been used as a kind of catch-all for what is, in actuality, poor system design and equipment selection.

Lets take a closer look at some of these sources of oxygen ingress:

a) "Round" plastic pipe always assume an oval shape when coiled. When this pipe is cold and stiff it is very difficult to make compression fittings that do not "open" up later when the system has operated for several weeks. Unless these fittings are tightened up at least twice after the system has operated these manifold compression fittings can admit far more oxygen than the pipe walls themselves.

Screwed joints anywhere in the system can hold pressure under pressure tests, but still admit air into the system, particularly on the

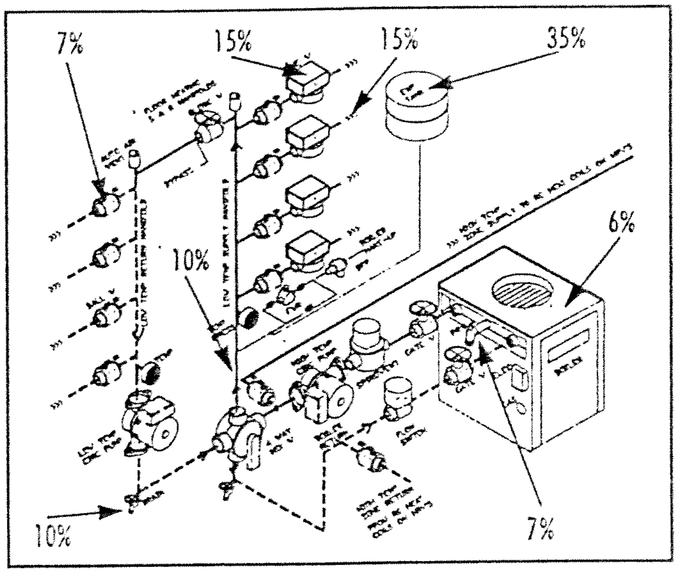

Figure 1. Shown here are typical points of oxygen ingress with approximations of oxygen entering at different classes of fittings.

suction side of system pumps.

b) Automatic air vents can admit tremendous amounts of air if they are improperly located on the suction side of a high head pump.

c) Similarly, loose packing around valve stems can admit large volumes of air, often with no noticeable sign of system problems.

d) Valve bodies, air scoops, pump heads, suction diffusers, and other hydronic components may have concealed micro-fractures that can admit atmospheric air, but still retain pressure during system testing. Often these fractures will only open up when the casting is heated and expands during normal system operation.

e) Gaskets on pump flanges, particularly in the higher head pumps often used in radiant systems can be a subtle but common way for atmospheric oxygen to be drawn into the system.

f) Probably the most common cause of corrosion in modern hydronic systems is the failure of the expansion tank. [5] Loss of air charge over the years caused by diffusion through the rubber bladder, or actual mechanical failure of the bladder has enormous potential for mischief. Upon failure of the expansive capability of the bladder, excess water pressure is discharged through the boiler relief valve. As the heating unit cycles down, the system fluid pressure drops. This may result in air being drawn into the system, but at the very least results in a gallon, sometimes several gallons, of fresh oxygenated water being drawn into the system with each boiler cycle. In addition to corrosion, this can result in scale formation on boiler surfaces.

This condition can persist for years until someone happens to be in the boiler room when the relief valve discharges, or until a system component fails from corrosion processes. Due to the thin walled steel construction of the tank itself, it is often the first component to suffer from this corrosive failure in a typical North American system.

Only regular inspection and maintenance of the expansion tank can guard against this most common kind of system problem.

Enhancement of heat pump performance by using radiant heating and cooling panels.

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Abstract

According to the basic feature of hydronic radiant panel heating and cooling, it is possible to establish a better tie-in with heat pumps. There are several attributes like moderate temperature requirement for the heat transfer fluid, low heating and sensible cooling loads for an equivalent human comfort, and peak load shaving. These, and other attributes enable the absorption and vapor compression type heat pumps to operate under more favorable conditions. Two design studies were carried out in order to reveal typical benefits of such a combination. The first design study involves an air to water type vapor compression heat pump used for indoor radiant heating in a 320 m^2 residential home. Results revealed that the design heating load is decreased by as much as 35%, supplementary boiler is eliminated, and the coefficient of performance is increased by 20%. The second design study involves a solar- absorption cycle heat pump used for sensible cooling of a solar house with 81 m^2 net living area. In this study, the exergetic analysis indicated an overall improvement of the performance of the heat pump and solar collectors. Sensible cooling is accomplished by radiant floor panels. A 3.5 kW water-ammonia absorption heat pump and 30m^2 flat plate collector area is required. The system also provides domestic hot water. The design analysis indicated that with radiant ceiling cooling, the peak cooling load will be shaved by about 25%.

1.INTRODUCTION

1.1. Radiant Heating

Figure 1 exhibits the contradiction between the heating demand of a central heating system using radiator type heating units and the air to water type heat pump. Therefore, air to water type heat pumps will continue to face operational constraints as long as they service conventional heating systems.

The supplementary plant will contribute to space heating by an amount corresponding to the area ASE in Figure 1. Shut down point is determined by one of the following constraints which comes first on T_o (outdoor temperature) axis:

-Constraint (i): Maximum water temperature that can be supplied by the heat pump.

Generally it is limited to 55°C. The intersection point of this line with the supply or return water temperature demand of the heating units determines the location of point (i).

-Constraint (ii): Condenser side water temperature drop, ΔT_{ws}.

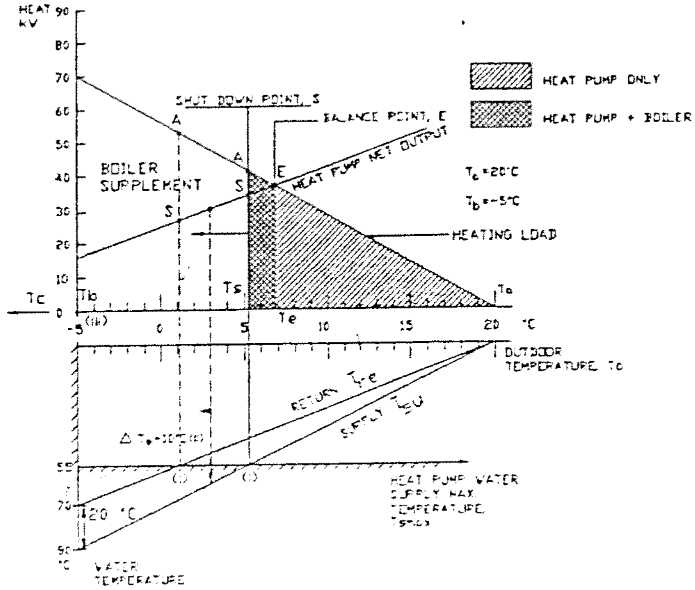

Figure 1. Heat output of an air to water type heat pump versus the heating load as a function of outdoor air temperature.

The maximum permissible temperature drop is generally limited to 10°C. Seeking an alternative and more suitable heating system to establish a better match with the operational characteristics of heat pumps and thus to enhance their performance and compatibility has the following objectives and justifications:

-Enhancement of the heat pump performance.

With or without a supplementary boiler,any improvement of instantaneous and seasonal COP of the heat pump will increase the technical and economical feasibility.

-Minimization of the supplementary boiler size.

If for any reason the boiler can not be eliminated,any reduction in its size and seasonal contribution to heating will make the attributes of the heat pump more predominant.

All of these objectives can be realized by a radiant floor heating system which establishes a perfect match with respect to the water temperature requirements and substantially reduces the heating load as compared to other heating systems.

1-Indoor air temperatures can be selected as 2 to 3°C lower than the standard indoor design temperatures T_a, without any sacrifice of the desired human comfort [1].

2-A lower indoor air temperature setpoint and a slow indoor air movement decrease the infiltration and transmission type heat losses by as much as 30 %.

3-As a design and operational requirement of the system, the average water design temperature in the circuit hardly exceeds 50°C due to large radiant surfaces and reduced heat loads.

4-Due to low water temperatures maintained in the circuit, the temperature drop,ΔT_w has to be small. Therefore it is already a general practice to keep ΔT_w at

10°C at design conditions. This will virtually eliminate constraint (i).
The combined effect of these attributes are shown in Figure 2 .

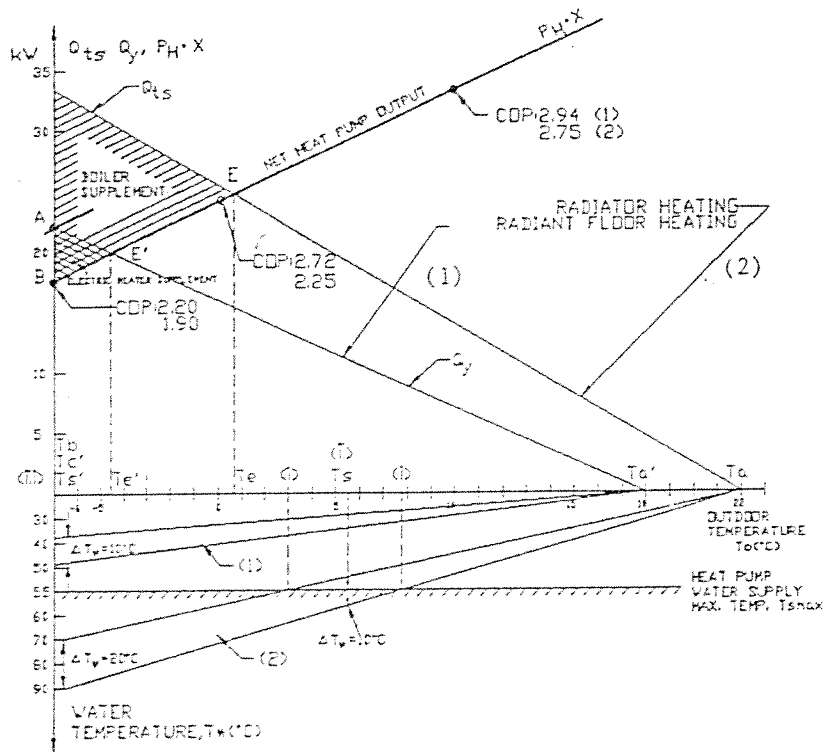

Figure 2. Comparison of baseboard heating and radiant floor heating with air to water type heat pump [1]

1.2. Heat Extraction by a Radiant Ceiling Slab (Sensible Cooling)

A typical in-slab type radiant ceiling cooling panel construction is shown in Figure 3. The slab is maintained at a colder temperature than the ambient by circulating chilled fluid through the embedded pipes.

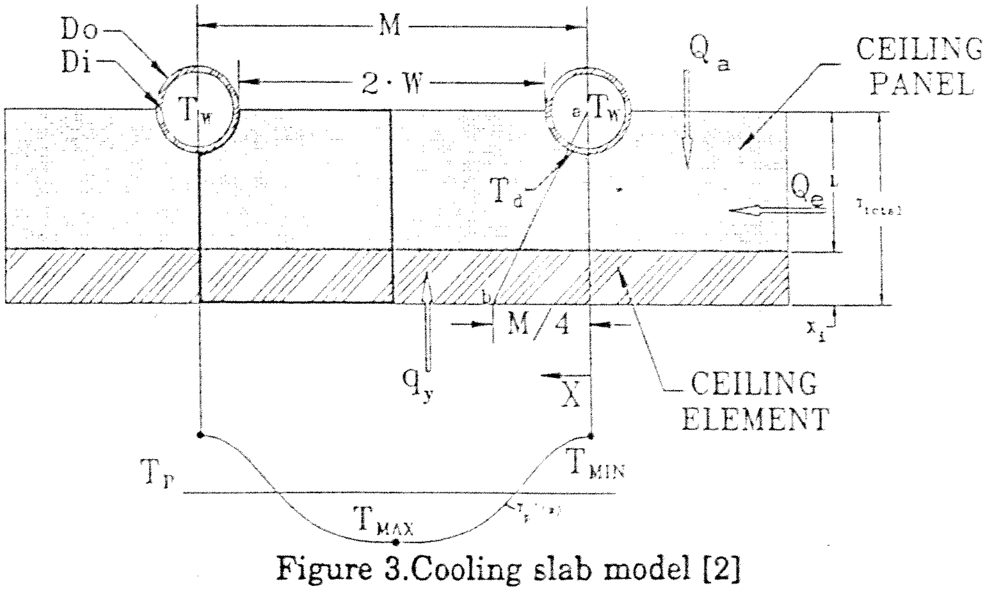

Figure 3.Cooling slab model [2]

Kiltis, E (3)

Absorption air-conditioning equipment has been in standard production for several decades. Water-ammonia systems also require relatively high solar collector water outlet temperatures like 115°C for optimum operation [3]. A water-lithium bromide system is simpler than the water-ammonia system and operates at a higher cooling ratio and smaller heat exchanger surfaces, but may involve more design problems [5]. A water-ammonia system is shown in Figure 4. Flat plate collectors provide the necessary heat, for energizing the generator. A storage tank may also be used. Evaporator provides the necessary heat extraction through the radiant ceiling panel circuit. Rejected heat from the condenser and the absorber heats the domestic water.

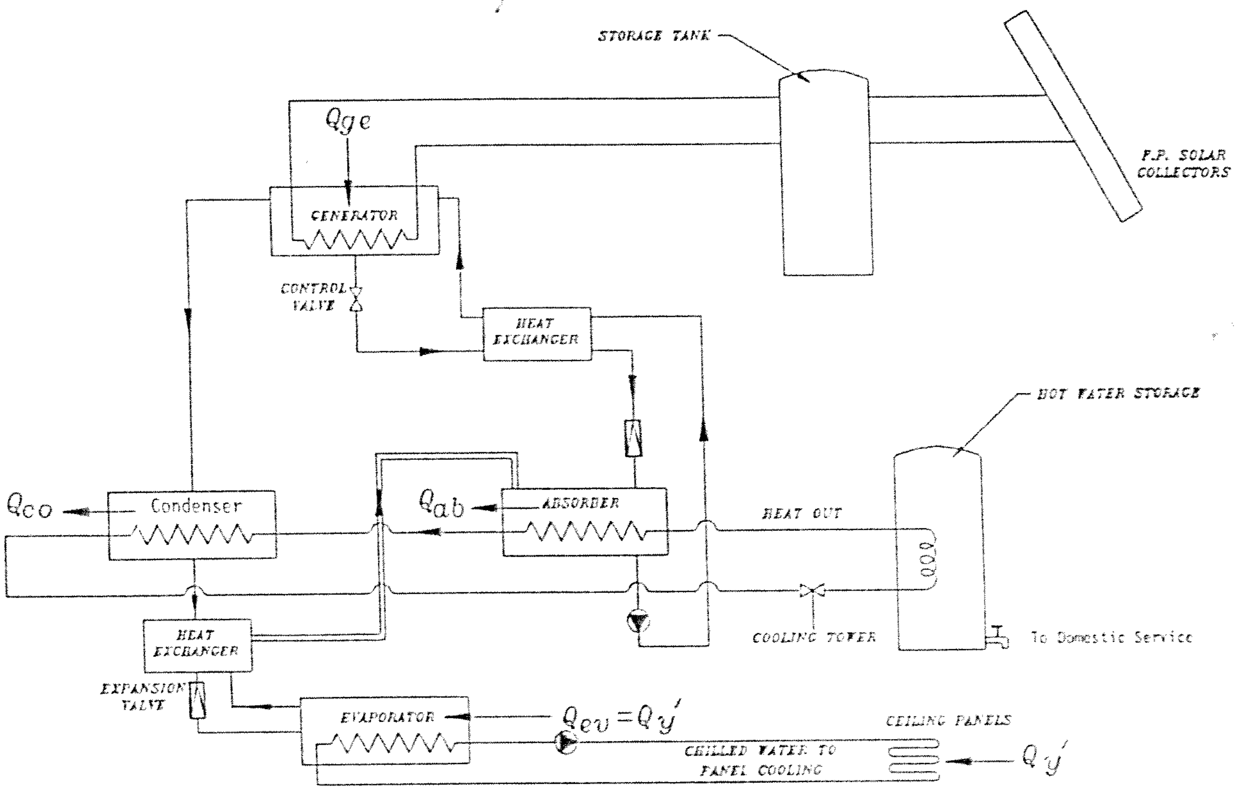

Figure 4. Solar absorption cooling principles [4]

In design and optimization of the performance of an absorption cooling system, the most important variable which has to be taken into account is the temperature of the generator.

Through the available literature, a typical relationship between $ECOP_h$ and T_{ge} is shown in Figure 5 for a water-ammonia absorption heat pump [6]. By extrapolation and interpolation on solid lines for Tev:10°C and Tev:0°C in Figure 5, it can be seen that an increase in the evaporator temperature from 7.5°C to 12.5°C decreases the optimum generator temperature from 80°C to about 67°C. This feature also eliminates the need for an auxiliary generator heater. This example also indicates that the maximum $ECOP_h$ increase from 0.48 to about 0.55. This increase may be enhanced if the mean chilled water temperature can be further increased to 15°C at an expense of reducing the spacing between the hoses. Figure 6 shows the variation of COP_r with T_{ev}.

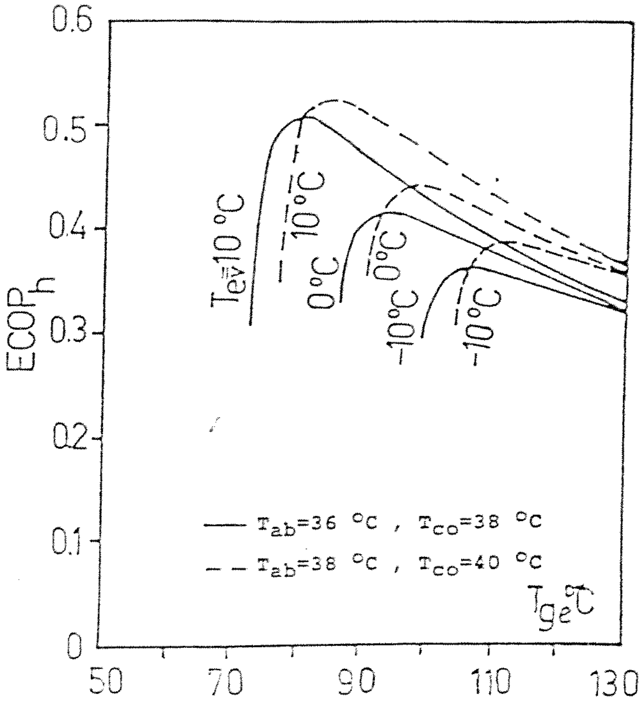

Figure 5. ECOP$_h$ in a water-ammonia system [6]

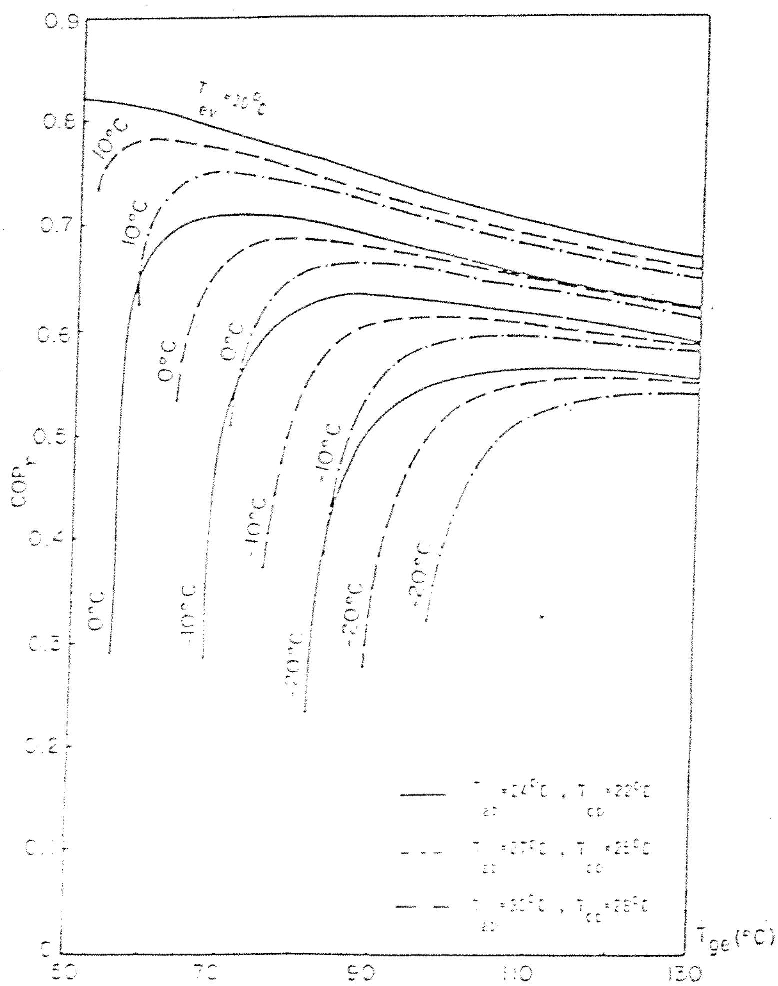

Figure 6. COP$_r$ in a water-ammonia system [6]

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2 CASE STUDIES

2.1. Space Heating

A new, two story residential house located in a sea side region is going to be heated in Winter. The building has a conventional heat load of 29 kW at a design outdoor temperature of -3°C. The standard indoor design temperature is 22°C. There are 320 m^2 floor space and floor heating will be performed over 275 m^2. In order to cover the entire operational range of the heat pump in the analysis, design condition was extrapolated down to the cut-off point at -7°C. At this outdoor temperature the conventional heat load will be 33.64 kW. For an equivalent comfort, indoor design temperature was selected as 18°C [2]. The radiant heating load is 25.23kW. Indoor thermal comfort is controlled by water temperature modulation. This enables to shave off the radiant floor heating load by 12%. The average heat load intensity will then be 22.20/275·1000 = 81 W/m^2. A typical radiant floor heating circuit covers 15 m^2 on the floor surface. Thermal resistance of the floor covering is 0.043 m^2 K/W. Rubber hose with 0.0095 m I.D. and 0.021 m O.D. was selected as energy transfer medium. The computer program revealed that the average water temperature will only be 42°C at the cut-off point of the heat pump(-7°C). The water supply temperature requirement, will be (42+10/2)= 47°C. A comparison of the characteristics of the two heating systems is shown in Figure 2. The selected heat pump capacity will be only 3 kW short of the radiant heat load at the cut-off point as shown in Figure 2 (Point A and B). As the cut off temperature T_c is below T_b, supplementary heater is, in fact, not necessary for the actual outdoor design condition. If a conventional heating system would be used, with the line Q_{ts}, the equilibrium point will be at +1°C. The supplementary boiler requirement would then be 9 kW (29kW - 20kW) at actual outdoor design conditions, or about 12 kW at the cut off temperature. By oversizing the hydronic system at an extra expense, shut off point can be eliminated. Otherwise it would be at around +5°C necessitating an early call of the supplementary boiler. Some typical COP values are also shown corresponding to each heating system. As seen, COP values for radiant floor heating case are higher by as much as 25%. T_{ws} has to be maintained at 55°C for the radiator heating case for most of the time.

Figure 7 shows the general layout of the radiant floor heating system. A four way valve continuously modulates the water temperature. The indoor air temperature also controls the heat pump on an on-off basis. However "off" signals are delayed through a preset timer so that the slab is maintained at a temperature corresponding to a moderate base load. Another way to maintain this condition is to directly monitor the slab temperature so that it can be kept above a certain temperature temperature.

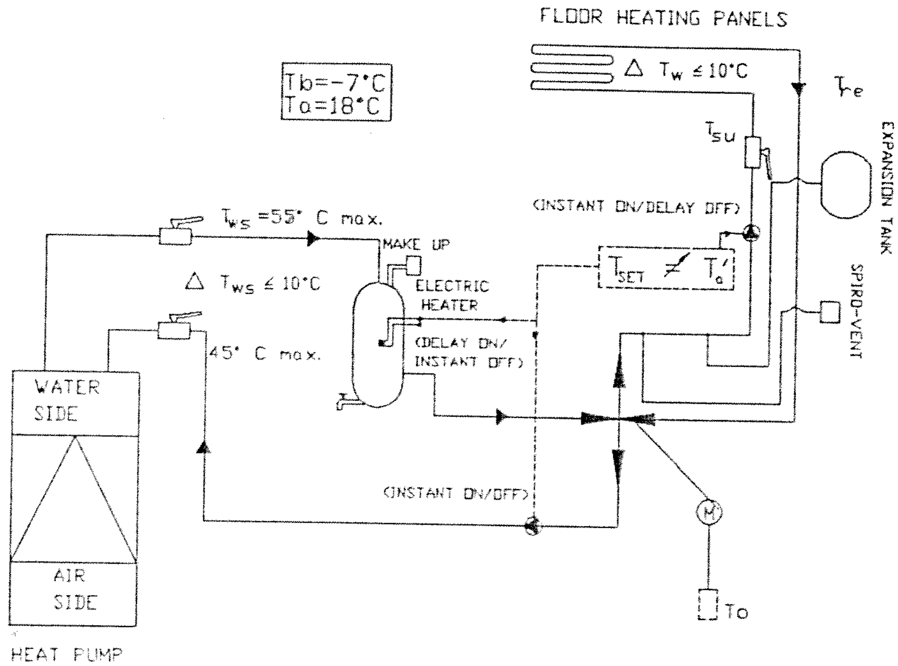

Figure 7. Design layout of the radiant floor heating system.

2.2. Radiant Panel Cooling

The net living area of the solar house at Middle East Technical University is 81 m^2 and there is a 19.8 m^2 greenhouse (see Figure 8). The heating and cooling volume is 320 m^3. The original design load for sensible cooling was 6.5 kW. The inclined roof can accommodate flat plate collar collectors up to a total collector absorber area of 40 m^2. Total ceiling area is around 120m^2. The average daily insolation in August is about 5.45kWh/m^2 day. As part of the ceiling cooling panel efficiency requirement, the ceiling was insulated by 0.025m thick styrofoam boards. This reduced the peak cooling load to 4.9 kW. With design provisions related to radiant ceiling cooling including the peak load shaving, the design cooling load reduced to 3.0 kW (10239 Btu/h) at a design indoor air temperature of 23.3°C and 33.3°C design outdoor temperature. At an expected solar collector water supply temperature of 80°C, the daily average collector efficiency was calculated to be 65% instead of the 55% rated efficiency for a locally available product. Dew point temperature is 15.4°C. Rubber hoses with about 0.0095 m inside diameter (3/8") will be stapled over 0.012 m gypsum board ceiling element. The ceiling is joisted wood type. The thermal conductivity of the hose material is 0.28W/m K. For the given cooling load and the dew point temperature, the maximum allowable hose spacing was calculated by the program as 0.15m. With this spacing, the mean chilled water temperature is calculated to be 11.1 °C. With an extra expense of hose material, if the hose spacing is reduced to 0.10m which is the minimum permissible hose spacing, the mean chilled fluid temperature increases to 14.4°C. In order to maintain a turbulent flow, drop in the chilled fluid temperature was selected to be around 2.7 °C at design conditions. This yields a design chilled fluid supply temperature of 13.0 °C. It must be noted that if the ceiling element was concrete which has a higher thermal conductivity than the gypsum board, the mean chilled fluid temperature would be even higher. For 0.1m thick concrete slab, the required mean fluid supply temperature will be 18.3 °C. With the required fluid temperature, the COP_r of a water-ammonia absorption heat pump with 3.5 kW cooling capacity was calculated to be around 0.70 at an absorber temperature of 38 °C. With this figure and an 80% efficiency for solar energy storage system including the power consumption of the circulating pump of the heat pump, and 65% collector efficiency, the necessary collector area will be:

$$\text{Solar Collector Area} = 60/(5.45 \cdot 0.70 \cdot 0.8 \cdot 0.65) = 30.2\ m^2$$

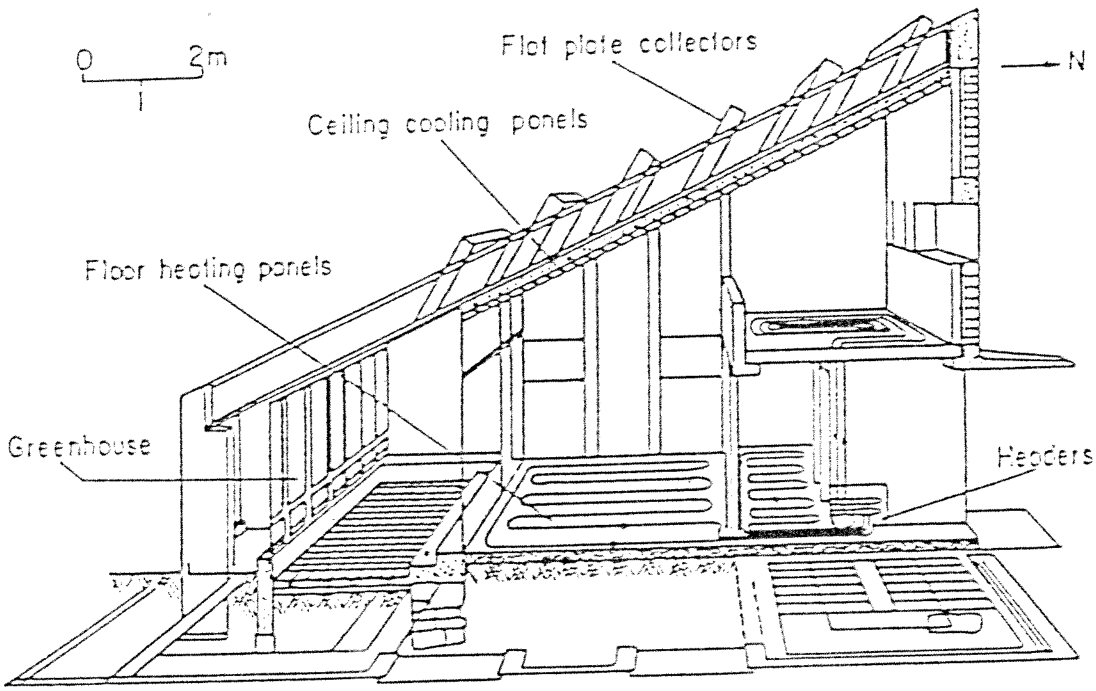

Figure 8. METU solar house

Twenty flat plate collectors with 1.5 m^2 absorption area each were selected, becoming a key element in enhancing the technical and economical feasibility of heat pumps. This is especially true for air to water type heat pumps where their feasibility might be already marginal. Due to its heat load reducing feature and other merits,floor heating systems can also be used in regions where an air to air type heat pump can not be operated successfully. The case study clearly demonstrated that it can also utilize heat pumps at higher COP values as well as eliminating the need for the supplementary boiler. However in order to realize these benefits, a careful and detailed analysis of the radiant heating load,heat transfer,performance of the heat pump and the radiant floor has to be made.

3. DISCUSSION OF RESULTS AND CONCLUSION

Nearly one out of every three new single family homes built in the United States today are equipped with an electric heat pump for heating and cooling. In addition to this, an increasing number of existing homes are being retrofitted with heat pumps. If air to water type heat pumps operate with radiant floor heating systems, one of the key objectives which is the elimination of supplementary boilers can be easily realized. Besides new buildings, radiant floor heating can be a retrofit alternative for existing ones, because with thin slab or under subfloor type of applications, retrofitting is relatively easy and cheap.

An analytical model for radiant ceiling cooling heat transfer under steady state conditions was developed. The computer program using this model revealed that the mean chilled fluid temperatures may be as high as 18 °C, especially if the slab material is concrete. This is a relatively high temperature when compared with other conventional cooling systems. When coupled with the cooling load reducing and shaving features of radiant ceiling cooling, this moderate chilled fluid temperature requirement guarantees noticeable improvements in the COP of absorption type heat pumps and solar collector efficiencies. In the same manner the total exergy loss of the system substantially decreases. These in turn increase the competitiveness and feasibility of such alternative cooling systems. Design calculations for the case study indicated that the performance of both the heat pump and solar collectors increase.

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AN ANALYTICAL MODEL FOR THE DESIGN OF RADIANT PANELS FOR HEATING AND COOLING

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ABSTRACT

Panel heating and cooling systems offer a variety of advantages in sensible conditioning of indoors. These include but not limited to substantial energy savings, and better tie-in capabilities with low temperature and low intensity energy sources like solar systems and heat pumps. In order to practice the attributes of panel systems, a comprehensive design model is necessary. With the objective of meeting this need, an analytical model was developed. This model includes the panel heated or cooled indoor space, with energy storage, heat transfer between the panel surface and the conditioned space, and the heat diffusion in the panel. A new guideline for calculating the sensible loads including the peak load shaving is proposed. The heat diffusion in the slab is solved by a composite fin model which can accommodate additional layers like the floor covers. This article summarizes the fundamentals of these models, explains the proposed algorithm, and gives sample design nomographs generated by the computer program.

1. Nomenclature

A_p	Panel surface area, m^2
A_r	Total floor (ceiling) area in the heated (cooled) space, m^2
A_u	Total surface area of the heated (cooled) space, m^2
AUST	Area-weighted average temperature of the unheated (uncooled) surfaces at design conditions (excludes panel surfaces), °C
c	Temperature gradient factor, °C
C	Peak load shaving factor in panel heating, dimensionless
C_c	Peak load shaving factor in panel cooling, dimensionless
d	Room position code, dimensionless
D_e	Equivalent diameter of the floor (ceiling), m
D_i	Inside diameter of the heat transfer hose, m
D_o	Outside diameter of the heat transfer hose, m
e	Surface emissivity,dimensionless
F_r	Radiation interchange factor,dimensionless
h	Altitude of the location above sea level, m
k	Thermal conductivity of the slab material, W/m K
k_e	Equivalent thermal conductivity of the composite slab for lateral heat diffusion, W/m K
k_h	Thermal conductivity of the hose material, W/m K
k_i	Thermal conductivity of each panel cover, W/m K

L Thickness of the slab, m
L_r Inside perimeter of the floor (ceiling), m
M Hose spacing on centers, m
m Fin coefficient, m^{-1}
n_k Number of panel covers,dimensionless
P Required plant load, kW
P_H Heat output (intake) of the heat pump, kW
Q_a Heat loss (gain) from (to) the panel from its back, kW
Q_D Required supplementary plant capacity, kW
Q_e Heat loss (gain) from the edge of the panel, kW
Q_y' Load when the space is panel heated (cooled), kW
Q_y Load after peak shaving, kW
q_r Radiant heat intensity, W/m^2
q_c Convective heat intensity, W/m^2
q_y Panel load intensity (Q_y/A_p), W/m^2
R Thermal resistance between the hose surface and the slab material, m^2 K/W
R_{co} Total thermal resistance of panel cover(s), m^2 K/W
r Linearization factor for the radiation heat transfer term, $°C^3$
T_a' Indoor design temperature for a panel heating (cooling) system, °C
T_b Outdoor design temperature, °C
T_d Outside surface temperature of the hose, °C
T_i Temperature of the space adjacent to the panel, °C
T_{max} Maximum panel surface temperature, °C
T_{min} Minimum panel surface temperature, °C
T_p Effective panel surface temperature, °C
$T_p'(x)$ Local panel surface temperature on the panel at position x, °C
T_{total} Thickness of the composite fin (including covers), m
T_w Mean fluid temperature in the circuit ,°C
U Overall heat transfer coefficient on the panel surface, W/m^2 K
U_c Convection heat transfer coefficient on the panel surface, W/m^2 K
U_r Radiation heat transfer coefficient on the panel surface, W/m^2 K
v_w Average water velocity in the hose, m sec^{-1}
W Half of the net spacing between adjacent hoses $(M-D_o)/2$, m
X Thermal efficiency of the panel, dimensionless
x Position variable on the panel surface, m
x_i Thickness of any panel cover, m

Greek symbols:

α Convection heat transfer coefficient between the heating fluid and the hose, W/m^2 K
η Fin efficiency of the composite fin,dimensionless

σ Stefan Boltzmann constant, $5.67 \cdot 10^{-8}$ W/m^2 K^4

Subscripts:

c Convective

r Radiant

2. Panel Loads

Usually panel heating (cooling) is accomplished at the floor or ceiling. Panel heating or cooling from the walls is also feasible. The sensible panel load needs to be customized. Major factors are:

-Indoor air temperature: for-equivalent human comfort, it is generally 2 to 3°C lower in heating or 2 to 3°C higher in cooling than standard indoor design temperatures (Kilkis,1990).

-Natural infiltration: natural infiltration taking place from external openings may be substantially lower, like by 50% due to the existence of a more uniform air pressure and velocity distribution, slow air movement, and little air stratification.

-Transmission losses: due to slow air movement, the film coefficient on the partitions will be small. Therefore transmission losses through outdoor exposed partitions, especially for regular glass windows and walls with low thermal resistance will be also smaller.

-Energy storage: energy is stored in the building structure provided that the system runs continuously with water temperature modulation or at frequent intervals. In this case the peak load will be shaved. Tables 1 and 2 provide the peak load shaving factors for sensible heating and cooling respectively.

Shaved-off sensible loads will then be:

$$-Q_y = C_c \cdot (-Q_y') \quad \text{in cooling} \quad (1)$$

$$Q_y = C \cdot Q_y' \quad \text{in heating} \quad (2)$$

3. Panel Heating

3.1 Heat output

A typical in-slab type floor heating construction is shown in Figure 1. Rubber hoses with special composition, different types of thermoplastic pipes or metal pipes can be used for the transfer of heat. After laying the hose or pipe at a predetermined pattern and spacing, screed is poured to form the slab. In order to minimize heat losses from the slab, back and edges are insulated. The required heat output intensity of an indoor space with available panel surface area A_p is:

$$q_y = Q_y/A_p \cdot 1000 \quad (3)$$

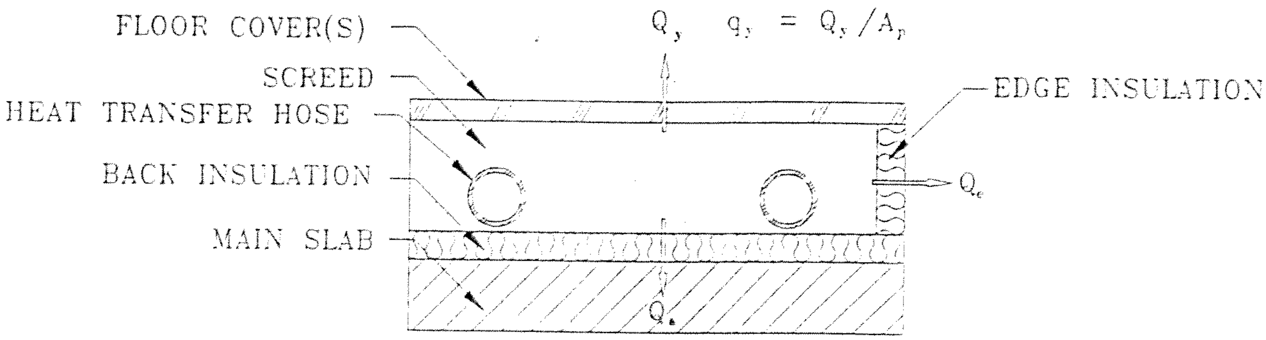

Figure.1 A Typical Heated Floor Slab.

The panel system is designed to meet q_y. All the attributes of panel heating system has to be taken into account in calculating Q_y'. Kilkis(1990) formulated an engineering rule for determining C as reproduced in Table 1. A detailed analysis is a must in order to customize C for a given building and local climatic conditions. Therefore caution must be exercised in manipulating this table for specific designs. Here, light construction class corresponds to a building mass less than 6000 N/m^2 and heavy construction class corresponds to a building mass greater than 14000N/m^2.

Table 1. Peak Load Shaving Factor In Panel Heating.

	C		
	BUILDING MASS		
LOCATION	Light Constr.	Regular Constr.	Heavy Constr.
Sea Side	1.0	0.88	0.80
Moderate	1.0	0.85	0.78
Cold, Mountain	1.0	0.80	0.75

Defining the thermal efficiency of the panel, at design conditions:

$$X = \frac{Q_y}{Q_y + Q_a + Q_e} \qquad (5)$$

Plant load of the heated space at design conditions will be:

$$P = Q_y/X \qquad (6)$$

If a heat pump will be used, it may be sized according to one of the following:

$P_H \geq P$ without supplementary source (7)

$P_H + Q_D = P$ with supplementary source (8)

Heat output of a floor panel depends upon the indoor air temperature, surface temperatures of all the unheated surfaces, air movement in the heated space and other surface characteristics like the emissivity. The convective part of the heat output depends also on the altitude and the size of the conditioned space. If these factors can be adequately correlated,the total heat output can be expressed only in terms of the panel surface temperature (ASHRAE,1992), (DIN 4725,1989). Once the relationship between the panel surface and the heated space is established, the heat diffusion in the slab and the required mean water temperature in the hydronic circuit may be calculated.
One of the earliest heated slab model was given by Kollmar and Liese (1957). They assumed that the slab operates as a plate fin and lose heat from the upper surface. Zhang and Pate (1986) developed a finite difference algorithm for ceiling heating panels for solving the steady and transient heat diffusion problem in the slab and predict the local surface temperatures. Kilkis (1990,1992) introduced a more elaborate fin model. Krinninger (1989) stated that lengthy numerical algorithms may improve the solution for the required mean fluid temperature only by a few degrees Centigrade. However this is not the case for certain combinations of the design variables. The total panel heat output intensity is the sum of the radiant and convective heat output intensities (Kilkis,1992),(ASHRAE,1992):

$$q_y = q_r + q_c \qquad (9)$$

where;

$$q_r = U_r \cdot (T_p - AUST) \qquad (10)$$

$$q_c = U_c \cdot (T_p - T_a') \qquad (11)$$

Here U_r and U_c are the radiation and convection heat transfer coefficients on the panel surface, respectively:

$$U_r = r \cdot F_r \cdot \sigma \tag{12}$$

r is the linearization factor for the radiation heat transfer:

$$r = 4 \cdot [(T_p + 273)/2 + (AUST + 273)/2]^3 \tag{13}$$

It may be further simplified in the usual design range of temperatures:

$$r \approx [0.0105 \cdot (T_p + AUST)/2 + 0.7955] \cdot 10^{-8} \quad \{15°C \le (T_p + AUST)/2 \le 30°C\} \tag{14}$$

F_r is the simplified radiation interchange factor for $A_p/A_u \le 0.30$ (ASHRAE,1992) :

$$F_r \approx e \tag{15}$$

$$U_c = (1 - 2.22 \cdot 10^{-5} \cdot h)^{2.627} \cdot (4.96/D_e)^{0.08} \cdot 2.67 \cdot (T_p - T_a')^{0.25} \tag{16}$$

Convection heat transfer coefficient is adjusted with the altitude (h) (Kilkis,1990) and the room size which is characterized by D_e (Min and et.al.,1956):

$$D_e = (4 \cdot A_r / L_r) \tag{17}$$

The area-weighted average temperature of the unheated surrounding surfaces,AUST is difficult to calculate. It primarily depends on the outdoor temperature, the area ratio of indoor and outdoor partitions,thermal properties and dimensions of the partition elements,position of the room in the building,and the outdoor wind conditions. An elaborate numerical algorithm was provided by Zmeureanu,Fazio and Haghighat(1987). An approximate expression is given by Kilkis (1990):

$$AUST \approx T_a' - d \cdot z \qquad \{14°C \le AUST \le 10°C\} \tag{18}$$

Here d is the room position code .i.e: d is one for an inner room, two for a room with one outdoor exposed side, and three for a room with two or more outdoor exposed sides. z is a function of T_b:

$$z = 15/(25 - |T_b|) \qquad \{T_b < 25°C\} \tag{19}$$

When T_a' and AUST are determined, T_p can be solved for a required heat output intensity by using Equations 9, 10 and 11. However these equations presume that T_p is uniform. Depending upon the slab construction, its heat output, indoor air temperature and the spacing between the heat transfer hoses, the panel surface temperature is in fact non-uniform. Models available for predicting this profile are few. Krinninger (1989), and Leal and Miller (1972) claim that this is a simple Sine curve. This assumption simplifies the calculations, because in this case at x = M/4, surface temperature always coincides with T_p (See Figure 2,path a-b). However this assumption ignores the effects of slab thickness,heat output intensity, indoor air temperature,thermal properties of the slab layer and floor coverings. Furthermore it can not determine the surface temperature distribution because of the insufficient number of boundary conditions for determining a unique Sine curve passing through the point (T_p, M/4). In this study, the panel surface temperature variation is predicted by the composite fin model (see Equations 32-a and 32-b).

Although DIN 4725 (1989) standard provides a simple panel heat output model in the following format:

$$q_v = 8.92 \cdot (T_p - T_a')^{1.1} \quad , \qquad (20)$$

It does not distinguish the radiant and convective heating split and is insensitive to surface emissivity,size of the heated space ,the altitude from the sea level and AUST. These factors may prove to be extremely important such that the actual heat output intensity may vary by as much as 50% depending upon specific combinations of these factors.

3.2 Composite fin model

A composite plate fin model was developed in order to predict the heat diffusion in a panel which is composed of layers with different thermal conductivities like the concrete slab,a wood floor and a carpet. A panel will most probably operate in an unsteady manner if a simple control is desirable, where the fluid circulation is interrupted by the indoor air temperature set control. However if the peak load shaving feature is desirable, then the system needs to be operated with temperature modulation. In prescribing the latter choice, the amount of peak load shaving is the key parameter with respect to the amount of energy consumed and the size of the heating plant required. This type of control strategy is more common in Europe. This fact and the simplicity of the solution favor a steady state assumption for design purposes. The composite plate fin model is shown in Figure 2. Part of the slab which is above the hose center plane and the floor cover(s) establish the composite plate fin which delivers heat from the top surface both by radiation and convection. According to the symmetry,characteristic fin length is M/2. In contrary to the existing models for plates and floor panels (Kollmar,1957), (Kreider and Kreith, 1981),this model treats the radiative and convective heat output of the composite fin separately. By superimposing these two, floor surface temperature at the base of the fin (T_{max}) is computed (see Equations 22 through 27). In order to accurately express the lateral heat diffusion in the composite fin with parallel layers of different thermal conductivities, an equivalent thermal conductivity in this direction is defined:

$$k_e = \frac{\left[\sum_{i=1}^{n_k}(k_i \cdot x_i) + k \cdot L\right]}{T_{total}} \tag{21}$$

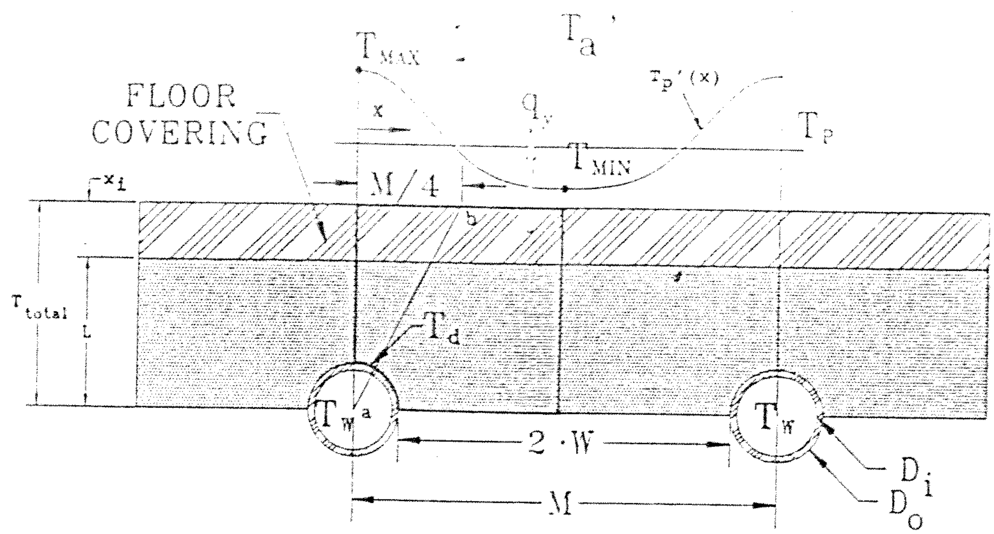

Figure 2. Composite Fin Model (Kilkis,1992)

Defining the radiation and convection fin efficiencies;

$$\eta_r = \eta_c = \eta \tag{22}$$

where; $\eta = \tanh(m \cdot W)/(m \cdot W)$ (23)

$$m = \sqrt{[U/(k_e \cdot T_{total})]} \tag{24-a}$$

and;

$$U = U_r + U_c \tag{24-b}$$

Total heat output of the fin surface between two adjacent hoses with a spacing M, and each having a unit length will be:

$$[2 \cdot W \cdot \eta_r + D_o] \cdot U_r \cdot (T_{max} - AUST) + [2 \cdot W \cdot \eta_c + D_o] \cdot U_c \cdot (T_{max} - T_a'] = q_y \cdot M \tag{25}$$

temperature gradients:

$$c = U_r / U \cdot [T_a' - AUST] \tag{33}$$

When T_a' equals AUST, c becomes zero.
The temperature profile may be revised with a second degree polynomial with the following boundary conditions:

$T_p'(x) = T_{max}$ at $x = D_o/4$, and $dT_p'(x)/dx = 0$ at $x = 0$
T_{min} can be calculated from Equation 32-a with $x = M/2$:

$$T_{min} = \cosh[m \cdot W]^{-1} \cdot (T_{max} - T_a' + c) + T_a' - c \tag{34}$$

Using T_{max} and taking the shortest path for the heat transfer, the required outside surface temperature of the hose at design conditions can be calculated :

$$T_d = T_{max} + q_y \cdot [\, R_{co} + (L - D_o/2)/k + R] \tag{35}$$

This derivation assumes that T_d is uniform around the hose perimeter. Thermal resistance between the hose and slab material, R may be neglected. R_{co} is the thermal resistance of floor covers above the slab:

$$R_{co} = \sum_{i=1}^{n_k} (t_i/k_i) \tag{36}$$

The required mean water temperature will be:

$$T_w = \frac{q_y \cdot M}{X \cdot \pi} \cdot [\, \frac{1}{\alpha \cdot D_i} + \frac{1}{2 \cdot k_h} \ln(D_o/D_i)\,] + T_d \tag{37}$$

where; $\alpha \cong 1056 \cdot [0.02 \cdot (T_w + 273) - 4.06] \cdot v_w^{0.8}/D_i^{0.2}$ (38)

For a turbulent flow using these equations, the required mean water temperature for a given hose spacing can be accurately determined. It is also possible to prepare design nomographs for manual calculations for typical applications and floor construction details (Figure 3).

Let;

$$A = [2 \cdot W \cdot \eta_r + D_o] \cdot U_r \, . \qquad (26\text{-a})$$

$$B = [2 \cdot W \cdot \eta_c + D_o] \cdot U_c \, . \qquad (26\text{-b})$$

Solving Equation 25 for T_{max} for the required q_y:

$$T_{max} = \frac{q_y \cdot M + A \cdot AUST + B \cdot T_a'}{A + B} \qquad (27)$$

Kreider and Kreith (1981) expressed the fin efficiency for a flat plate collector losing heat both by radiation and convection from its top surface to the ambient. They defined an overall heat transfer coefficient without regarding the split between the radiation and convection. Kollmar(1957) expressed a similar fin efficiency. Following these definitions with;

$$U' = q_y/(T_p - T_a') \qquad (28)$$
$$m' = [U'/(k_{eq} \cdot T_{total})]^{1/2} \qquad (29)$$
$$\eta' = \tanh(m' \cdot W)/(m' \cdot W), \qquad (30)$$

a simple expression for T_{max} may be obtained:

$$T_{max} \approx T_a' + \frac{(T_p - T_a')}{[2 \cdot W \cdot \eta' + D_o]} \cdot M \qquad (31)$$

This equation will become identical with Equation 27 only when AUST $\approx T_a'$ and $U_r \approx U_c$. These conditions may approximate the actual design conditions only for a limited combination of the design variables. For this reason, Equation 27 should be employed whenever a higher accuracy is desired. After computing T_{max},the surface temperature profile can be determined,noting that temperature gradients for radiation and convection are different:

$$T_p'(x) = \left(\frac{\cosh [m \cdot (M/2 - x)]}{\cosh [m \cdot W]} \right) \cdot (T_{max} - T_a' + c) + T_a' - c \quad \{D_o/2 \leq x \leq M/2\} \qquad (32\text{-a})$$

$$T_p'(x) = T_{max} \quad \{x \leq D_o/2\} \qquad (32\text{-b})$$

The c term represents the difference between the radiation and convection

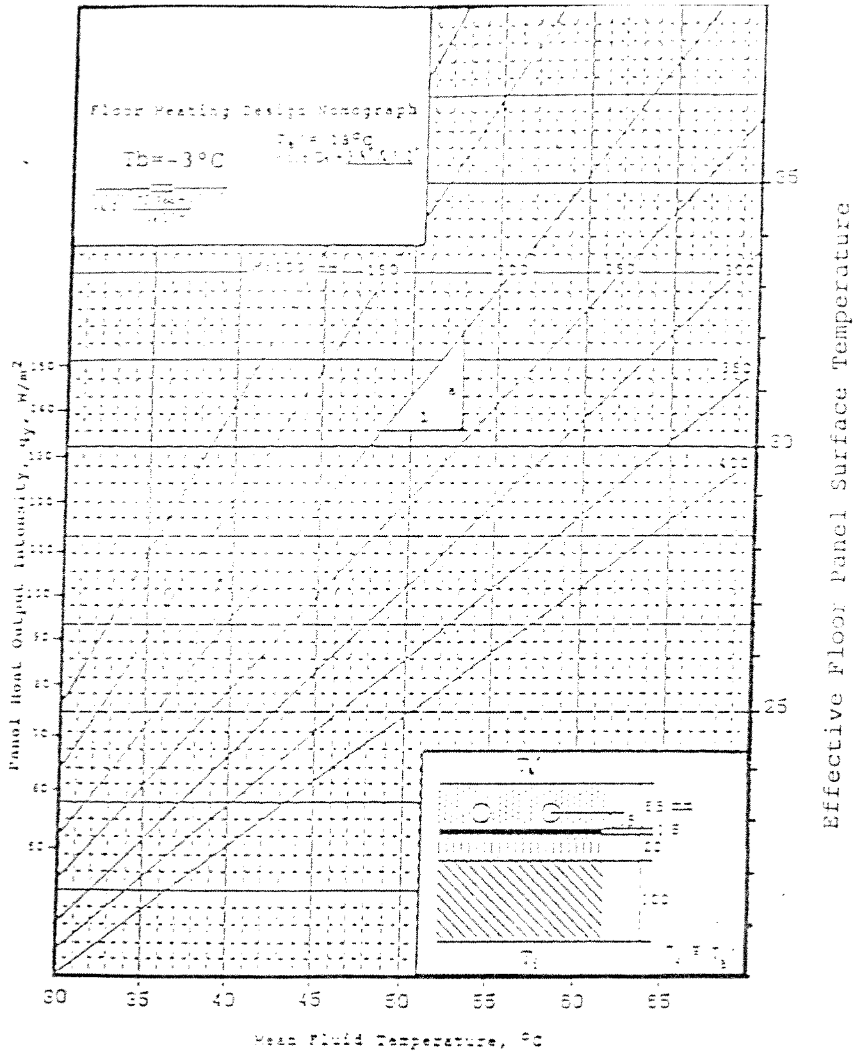

Figure 3. A Floor Panel Heating Design Nomograph

4. Panel Cooling

A typical in-slab type radiant ceiling cooling panel construction is shown in Figure 4. The ceiling slab is maintained at a colder temperature than the indoor air by circulating chilled fluid through the embedded pipes. In order to minimize heat gains from the back and perimeter of the cooled slab,its back and edges are insulated. The heat gain from the conditioned space establishes its sensible cooling effect. Designating this as the negative of the heat gained by the slab, the required cooling load intensity of the space to be conditioned is:

$$q_y = - Q_y'/A_p \cdot 1000 \tag{39}$$

Figure 4 shows a typical ceiling cooling panel.

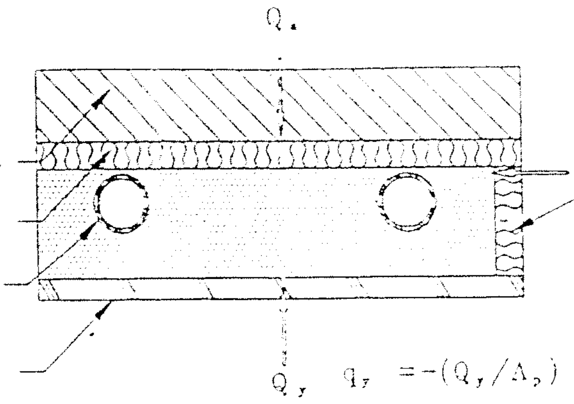

Figure 4. In-slab Type Ceiling Cooling Panel Construction.

Ceiling panel has to be designed to meet or exceed the magnitude of this load. The design load may be shaved-off due to cold storage in the cooled slab and the enclosing walls, provided that the system is not cycled with long periods. Then Q_y' may be reduced by the peak load shaving factor,(See Equation 1). This factor depends on many factors like the climate, typical daily outdoor temperature changes, and the cold storage capability of the building elements, in addition to the thermal properties and dimensions of the cooled slab. By considering five climatic conditions, Kilkis(1990,1991) formulated an engineering rule for determining C_c as reproduced in Table 2.

Table 2. Peak Load Shaving Factor in Panel Cooling

Climate/Location	C_c
Arid	0.70
Semi-arid	0.75
Moderate	0.80
Mountain	0.75
Seaside	0.85

The latent load may be handled by a hybrid cooling system where a radiant ceiling cooling panel is complemented by a small air handling unit (Wilkins and Kosonen,1992). This unit takes care of the necessary fresh air supply and the latent cooling load. The intensity of sensible cooling effect by the ceiling panel depends primarily on the hose spacing, mean fluid temperature, size of the indoor space, altitude, and the indoor design air temperature. When the ceiling surface temperature is assumed to be uniform, and equal to T_p, the amount of total heat that can be absorbed by the panel surface of unit area is the sum of radiation and convection terms:

$$q_y = q_r + q_c = U_r \cdot (T_p - AUST) + U_c \cdot (T_p - T_a') \quad \{q_y \leq 0 \text{ in cooling}\} \qquad (40)$$

where;

$$U_r = r \cdot F_r \cdot \sigma \qquad (41\text{-a})$$

$$U_c = (1 - 2.22 \cdot 10^{-5} \cdot h)^{2.627} \cdot (4.96/D_e)^{0.08} \cdot 2.67 \cdot |(T_p - T_a')|^{0.25} \qquad (41\text{-b})$$

r is the linearization factor (see Equation 13).
It may be further simplified by a linear function of T_p in the practical range of ceiling panel cooling:

$$r \cong [\,0.9659 + (T_p - 10) \cdot 0.00495\,] \cdot 10^{-8} \quad \{10°C \leq (T_p + AUST)/2 \leq 18°C\} \qquad (42)$$

Equation (41-b) gives the convective term with altitude (h) and room size (D_e) adjustments. There is not a direct correlation for the effect of the room size on convective cooling. However similarity between floor heating and ceiling cooling enables to employ the same min and et.al.'s (1956) data for floor heating.
The area-weighted average design temperature of the uncooled indoor surfaces, AUST is rather difficult to calculate. A simple approach was developed in order to calculate AUST (Kilkis, 1992):

$$AUST = T_a' + d \cdot z \qquad (43)$$

z is a function of the outdoor design temperature (Kilkis, 1993):

$$z \cong 7/(45 - T_b) \qquad \{26\,°C \leq T_b \leq 36\,°C\} \qquad (44)$$

When T_a' and AUST are determined, T_p can be solved for the required cooling intensity. Equations 40 and 41a and 41b presume that T_p is uniform which can only be true for an infinite number of hoses. Therefore further elaboration is essential before calculating the design mean chilled fluid temperature for maintaining the required heat extraction (cooling) intensity. Figure 5 shows the steady state composite fin model for ceiling panel cooling (Kilkis, 1993).

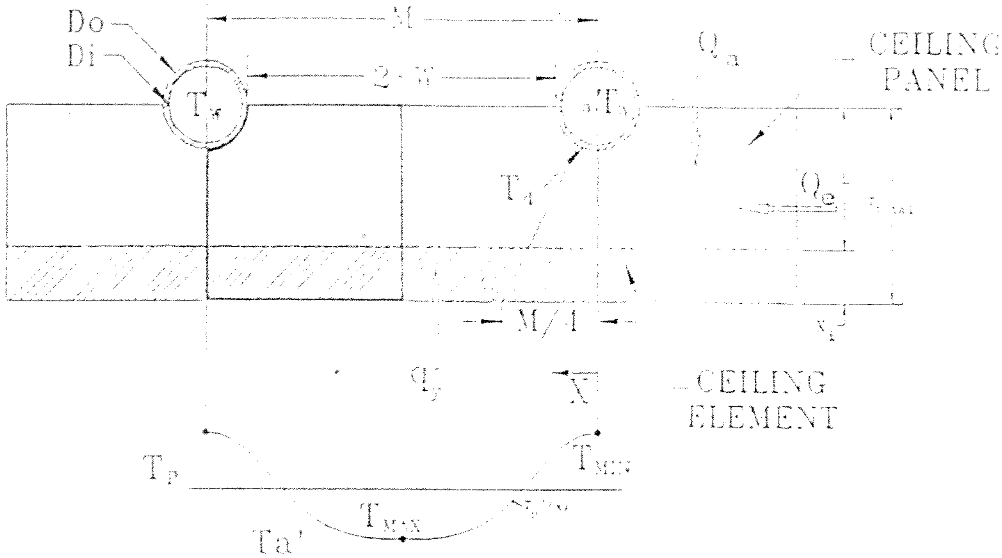

Figure 5. Composite Fin Model for Ceiling Panel Cooling [Kilkis 1993]

A ceiling panel cooling system most probably will operate in an unsteady manner if a simple control is required. However if the peak load shaving feature is desired, the system must have a continuous fluid (brine) circulation with temperature modulation. If a solar absorption heat pump is used, another reason is the fact that heat exchangers in absorption systems have long time constants and therefore cycling reduces the efficiency. There is a third reason for temperature modulation in solar systems since it allows to energize the generator at much lower temperatures, and thus the solar collector efficiency and the heat pump COP are both increased (Kreider and Kreith,1981). These facts and the simplicity of the solution favor a steady state solution for design purposes. Part of the slab that remains below the hose center plane and the ceiling cover(s) below it establish the composite fin. Edges are assumed to be totally insulated. Calculations do not need to consider the back heat gains at this stage, because they are accounted separately through the definition of panel cooling efficiency which effects the required capacity of the cooling plant. Hose surfaces which are in contact with the fin are assumed to be isothermal (T_d). The composite fin extracts heat from the space both by radiation and convection according to Equation 40. Defining the radiation and convection fin efficiencies in the similar manner with panel heating:

$$\eta_r = \eta_c = \eta \tag{45}$$

where;

$$\eta = \tanh(m.W)/(m.W) \tag{46}$$

and;

$$U = U_r + U_c, \tag{47}$$

$$m = \sqrt{\frac{U}{k_{eq} \cdot T_{total}}} \tag{48}$$

Superimposing the radiation and convection heat extraction intensities and solving for T_{min}:

$$T_{min} = \frac{q_y \cdot M + A \cdot AUST + B \cdot T_a'}{A + B} \tag{49}$$

where A and B were already given in Equations 26 a and b.

The panel surface temperature profile can be calculated by the following equation for $D_o/2 \leq x \leq M/2$:

$$T_p'(x) = \left(\frac{\cosh[m \cdot (M/2-x)]}{\cosh[m \cdot W]}\right) \cdot [T_{min} - T_a' + c] + T_a' - c \tag{50-a}$$

for $x \leq D_o/2$:

$$T_p'(x) \cong T_{min} \tag{50-b}$$

Here, c represents the difference between the radiation and convection temperature gradients (See Equation 33).

This piecewise temperature profile may be revised in a similar fashion:

$T_p'(x) = T_{min}$ at $x = D_o/4$

and;

$dT_p'(x)/dx = 0$ at $x = 0$

Using T_{min} and taking the shortest path for heat diffusion :

$$T_d \cong T_{min} + q_y \cdot [\, R_{co} + (L - D_o/2)/k + R \,] \tag{52}$$

Finally, the required mean chilled fluid temperature, T_w can be calculated by using Equation 37, by noting that $q_y < 0$ in cooling.

Usually hose Reynolds number in panel cooling systems is around 6000, and therefore the flow is marginally turbulent. In this case one may employ the same turbulent flow model for α which is given in Equation 38. Figure 6 shows a typical nomograph for ceiling panel cooling.

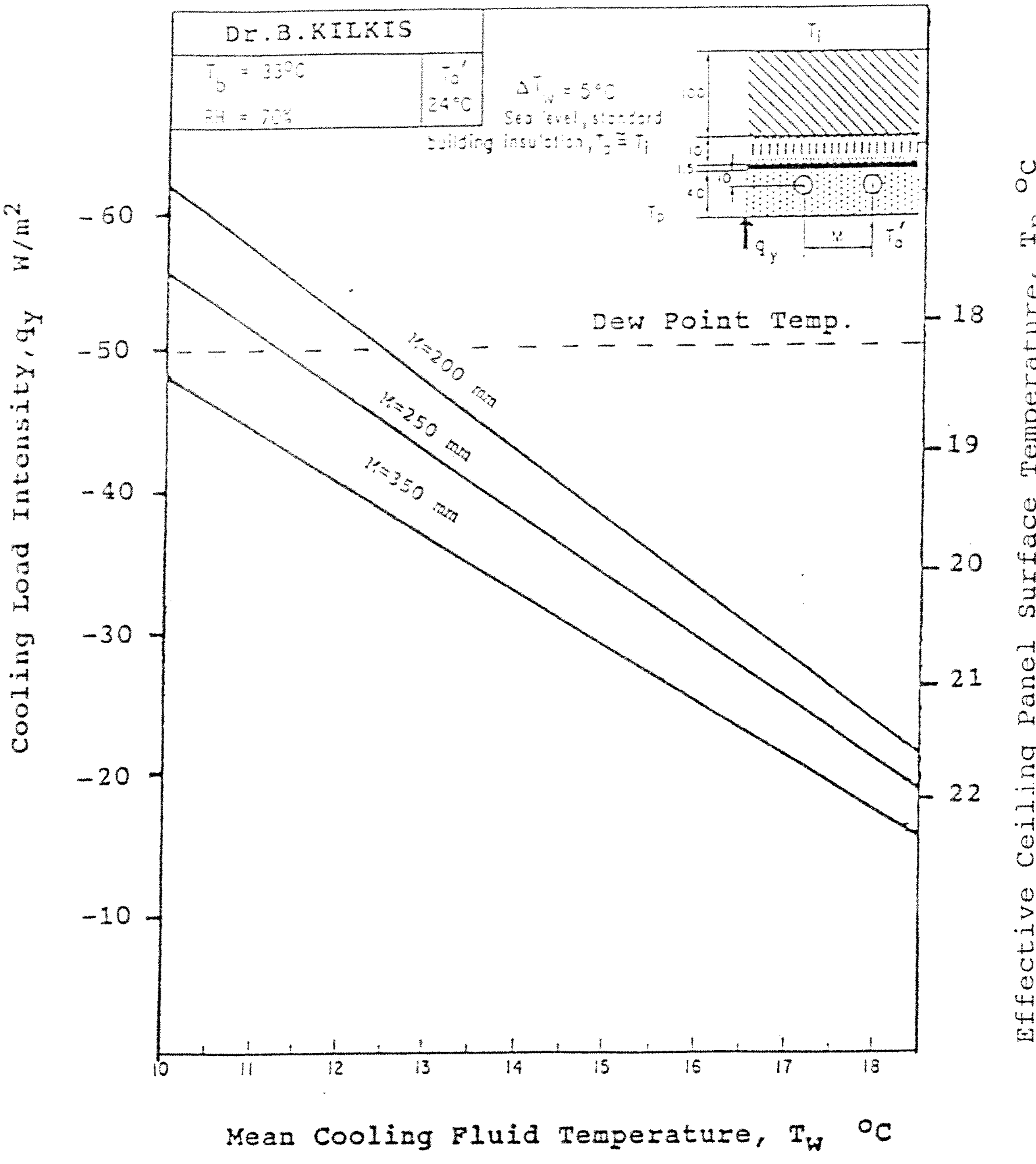

Figure 6. Design Nomograph for Ceiling Panel Cooling

5. Discussion of Results

Analytical models for panel cooling and heating under steady state conditions were developed. These models simplify the design calculations and can be effectively employed for computer programming. Models were tested and very close agreements with sophisticated finite element models for heat diffusion were

obtained. The computer program using this model revealed that the mean chilled fluid temperatures may be as high as 18 °C in panel cooling, especially if the slab material is concrete. This is a relatively high temperature when compared with other conventional cooling systems. When coupled with the cooling load reducing and shaving features of ceiling panel cooling, this moderate chilled fluid temperature requirement guarantees noticeable improvements in the coefficient of performance of absorption type heat pumps and solar collector efficiencies (Kilkis 1993). In the same manner, the total exergy loss of the system substantially decreases. These increase the competitiveness and feasibility of such alternative cooling systems. Due to the fact that panel cooling permits the designer to freely select the hose spacing in the technically feasible range, it is almost always possible to conform the temperature ratings of a given heat pump and thus optimize the overall performance. This flexibility also makes it possible to control the risk of condensation in a stand alone system. If the humidity has to be controlled, recent literature indicates that hybrid systems are feasible with minimum duct work and air handling unit requirements. In order to maximize the heat pump performance and collector efficiency, and achieve an optimal system, it is essential to simulate the indoors as well as the cooling system on an hourly basis including the cool storage analysis. This definitely requires a sophisticated computer program. However it is clear that such an effort will be very rewarding and promising for the future of absorption heat pumps whose COP figures are marginally acceptable.
Similar arguments also hold true for panel heating systems. A recent study revealed that employing panel heating systems eliminate the need for a supplementary boiler for air to water type heat pumps. (Kilkis, 1992)

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A SIMPLIFIED MODEL FOR THE DESIGN OF RADIANT IN-SLAB PANELS FOR HEATING AND COOLING

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Abstract

Panel heating and cooling systems offer various advantages in sensible space conditioning. These include but not limited to energy savings potential, and favorable tie-in capabilities with low temperature and low intensity energy sources like solar systems and heat pumps. In order to predict and implement the attributes of panel systems, a comprehensive, yet simple model is necessary so that designers will find it practical. Part of this model is the heat diffusion in the slab. With the objective of meeting these requirements, an analytical heat diffusion model was developed which is applicable for both heating and cooling. This article summarizes the fundamentals of this model, explains the proposed algorithm, and compares with finite element solutions.

1. Nomenclature

- **A_p** Panel surface area, m^2
- **A_r** Total floor (ceiling) area in the heated (cooled) space, m^2
- **A_u** Total surface area of the heated (cooled) space, m^2
- **AUST** Area-weighted average temperature of the unheated (uncooled) surfaces at design conditions (excludes panel surfaces), °C
- **c** Temperature gradient factor, °C
- **d** Room position code, dimensionless
- **D_e** Equivalent diameter of the floor (ceiling), m
- **D_i** Inside diameter of the heat transfer hose, m
- **D_o** Outside diameter of the heat transfer hose (or electric resistance), m
- **e** Surface emittance,dimensionless
- **F_r** Radiation interchange factor,dimensionless
- **h** Altitude of the location above sea level, m
- **k** Thermal conductivity of the slab material, W/m K
- **k_e** Equivalent thermal conductivity of the composite slab for lateral heat diffusion, W/m K
- **k_h** Thermal conductivity of the hose material, W/m K
- **k_i** Thermal conductivity of each panel cover, W/m K
- **L** Thickness of the slab in the fin, m
- **L_r** Inside perimeter of the floor (ceiling), m
- **M** Hose spacing on centers, m
- **m** Fin coefficient, m^{-1}
- **n_k** Number of panel covers,dimensionless
- **q_a** Heat loss (gain) intensity from (to) the panel from its back, W/m^2

q_e Heat loss (gain) intensity from the perimeter of the panel, W/m^2
Q_y Panel heating (cooling) load, kW,
q_r Radiant heat intensity, W/m^2
q_c Convective heat intensity, W/m^2
q_y Panel load intensity (Q_y/A_p), W/m^2
R Thermal resistance between the hose surface and the slab material, m^2 K/W
R_{co} Total thermal resistance of panel cover(s), m^2 K/W
r Linearization factor for the radiation heat transfer term, °C^3
t_a' Indoor design temperature of a panel heated (cooled) space, °C
t_b Outdoor design temperature, °C
t_d Outside skin temperature of the hose, °C
t_i Temperature of the space adjacent to the panel, °C
t_{max} Maximum panel surface temperature, °C
t_{min} Minimum panel surface temperature, °C
t_p Effective panel surface temperature, °C
$t_p'(x)$ Local panel surface temperature at position x, °C
T_{total} Thickness of the composite fin (including panel covers), m
t_w Mean fluid temperature in the circuit ,°C
U Overall heat transfer coefficient on the panel surface, W/m^2 K
U_c Convection heat transfer coefficient on the panel surface, W/m^2 K
U_r Radiation heat transfer coefficient on the panel surface, W/m^2 K
v_w Average water velocity in the hose, m sec^{-1}
W Half of the net spacing between adjacent hoses ($M-D_o$)/2, m
X Thermal efficiency of the panel, dimensionless
x Position variable on the panel surface, m
x_i Thickness of any panel cover, m
z Adjustment factor for AUST, °C

Greek symbols:

α Convection heat transfer coefficient between the heating fluid and the hose, W/m^2 K
η Fin efficiency of the composite fin,dimensionless
σ Stefan Boltzmann constant, $5.67 \cdot 10^{-8}$ W/m^2 K^4

Subscripts:

c Convective
r Radiant

2. Panel Heating

2.1 Heat output

A typical in-slab type floor heating construction is shown in Figure 1. Rubber hose with special composition, different types of thermoplastic or metal pipes can be used for the delivery of heat. In order to minimize heat losses from the slab, back and edges are insulated. The required heat output intensity of an indoor space with available panel surface area A_p is:

$$q_y = Q_y/A_p \cdot 1000 \qquad (1)$$

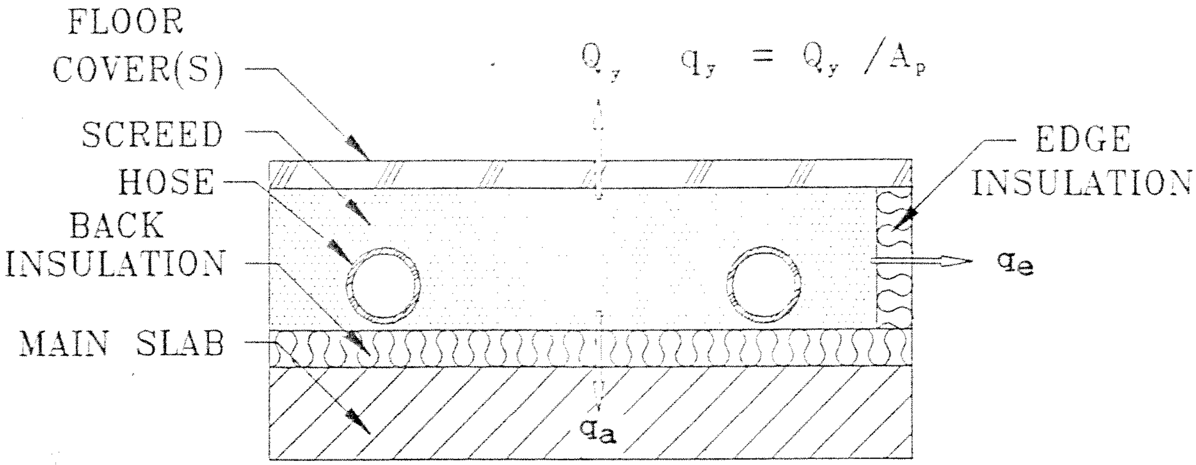

Figure.1 A Typical Heated Floor Construction.

The panel is to be designed to meet q_y.

The thermal efficiency of the panel is:

$$X = \frac{q_y}{q_y + q_a + q_e} \qquad (2)$$

Heat output of a floor panel depends upon the indoor air temperature, surface temperatures of all the unheated surfaces, air movement in the heated space and other surface characteristics like the emissivity. The convective part of the heat output depends also on the altitude (Kilkis, 1990), and the size of the conditioned space. If these factors can be adequately correlated, the total heat output can be expressed in terms of the panel surface temperature only (ASHRAE,1992), (DIN 4725,1990). When the relationship between the panel surface and the heated space is established, the required mean water temperature in the hydronic circuit

may be calculated when the heat diffusion in the slab is solved.
One of the earliest heated slab models was given by Kollmar and Liese (1957). They assumed that the slab operates as a plate fin and lose heat from the upper surface. Zhang and Pate (1986) developed a finite difference algorithm for ceiling heating panels for solving the steady and transient heat diffusion problem in the slab and predicted the local surface temperatures. Kilkis (1990,1992) introduced a practical yet accurate fin model. Krinninger (1989) stated that lengthy numerical algorithms may improve the solution in terms of the mean fluid temperature only by a few degrees Centigrade.
The total panel heat output intensity is the sum of the radiant and convective heat output intensities (Kilkis,1992),(ASHRAE,1992):

$$q_y = q_r + q_c \tag{3}$$

where;

$$q_r = U_r \cdot (t_p - AUST) \tag{4}$$

$$q_c = U_c \cdot (t_p - t_a') \tag{5}$$

Here U_r and U_c are the radiation and convection heat transfer coefficients on the panel surface, respectively:

$$U_r = r \cdot F_r \cdot \sigma \tag{6}$$

r is the linearization factor for the radiation heat transfer:

$$r = 4 \cdot [(t_p + 273)/2 + (AUST + 273)/2]^3 \tag{7}$$

It may be further simplified in the practical temperature range:

$$r \approx [0.0105 \cdot (t_p + AUST)/2 + 0.7955] \cdot 10^{+8} \quad \{15°C \leq (t_p + AUST)/2 \leq 30°C\} \tag{8}$$

F_r is the simplified radiation interchange factor for $A_p/A_u \leq 0.30$ (ASHRAE,1992) :

$$F_r \approx e \tag{9}$$

$$U_c = (1 - 2.22 \cdot 10^{-5} \cdot h)^{2.627} \cdot (4.96/D_e)^{0.08} \cdot 2.67 \cdot (t_p - t_a')^{0.25} \tag{10}$$

Convection heat transfer coefficient is adjusted with altitude (h) (Kilkis,1990) and the room size which is characterized by D_e (Min and et.al.,1956):

$$D_e = (4 \cdot A_r / L_r) \tag{11}$$

The area-weighted average temperature of the unheated surrounding surfaces, AUST is rather difficult to predict. It primarily depends on the outdoor temperature, the area ratio of indoor and outdoor partitions,thermal properties and dimensions of the partition elements,position of the room in the building,and the outdoor wind conditions. An elaborate numerical algorithm was provided by Zmeureanu,Fazio and Haghighat(1987). An approximate expression is given by Kilkis (1990) for practical design purposes:

$$AUST \approx t_a' - d \cdot z \qquad \{14°C \leq AUST \leq 10°C\} \tag{12}$$

Here d is the room position code I.e.: d is one for an inner room, two for a room with one outdoor exposed side, and three for a room with two or more outdoor exposed sides. z is a function of t_b:

$$z = 15/(25 + t_b) \qquad \{ t_b \geqslant -20°C\} \tag{13}$$

When t_a' and AUST are determined, T_p can be solved for a required heat output intensity by using Equations 3, 4 and 5. However these equations presume that T_p is uniform. Depending upon the slab construction, its heat output, indoor air temperature and the spacing between the heat transfer hoses, the panel surface temperature is in fact non-uniform. Models available for predicting this profile are few. Krinninger (1989), and Leal and Miller (1972) claim that this is a Sine curve. This assumption simplifies the calculations, because in this case at $x = M/4$, surface temperature always coincides with t_p (See Figure 2,path a-b). It can not determine the surface temperature distribution because of the insufficient number of boundary conditions for determining a unique Sine curve passing through the point (t_p, M/4). In this study, the surface temperature variation is approximated by the composite fin model (see Equations 27 and 28).

Although DIN 4725 (1990) standard provides a simple panel heat output model in the following format:

$$q_y = 8.92 \cdot (t_p - t_a')^{1.1} , \tag{14}$$

it does not distinguish the radiant and convective heating split and is insensitive to surface emissivity,size of the heated space, the altitude from the sea level and AUST. These factors may prove to be extremely important such that the actual heat output intensity may vary by as much as 50% depending upon specific combinations of these factors.

2.2 Steady state fin model

A composite plate fin model was developed in order to predict the heat diffusion in a panel which is composed of layers with different thermal conductivities. The composite plate fin model is shown in Figure 2. Part of the slab which is above the hose center plane and the floor cover(s) establish the fin which delivers heat from the top surface both by radiation and convection. According to the symmetry,characteristic fin length is M/2. In contrary to the existing models for plates and floor panels (Kollmar,1957), (Kreider and Kreith, 1981),this model treats the radiative and convective heat output of the composite fin separately. In order to define the lateral heat diffusion along parallel layers of different thermal conductivities, an equivalent thermal conductivity is defined:

$$k_e = 2 \cdot \frac{\left[\left(\sum_{i=1}^{n_k} k_i \cdot x_i\right)' + k \cdot L \right]}{T_{total}} \tag{15}$$

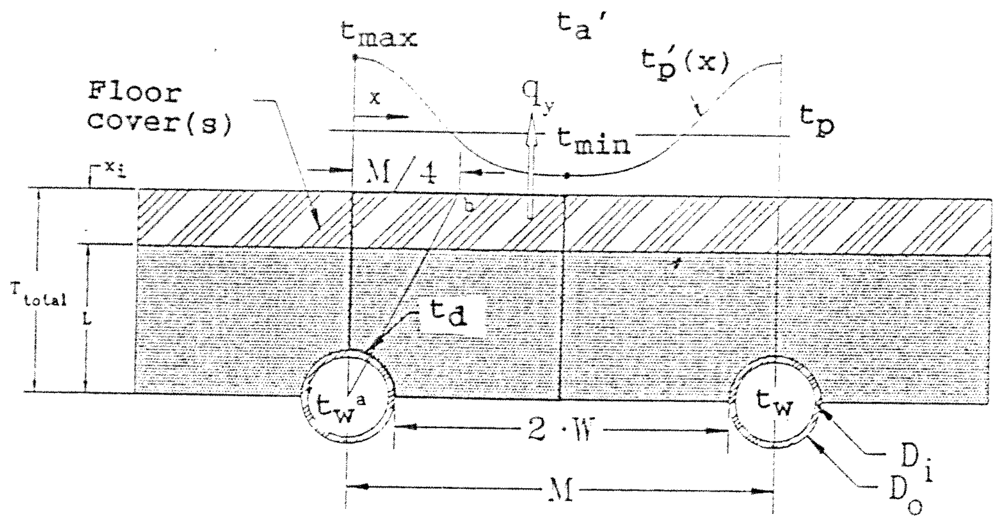

Figure 2. Composite Fin Model (Kilkis,1992)

Defining fin efficiency for radiation and convection;

$$\eta = \tanh(m \cdot W)/(m \cdot W) \tag{16}$$

where; $m = \sqrt{[U/(k_e \cdot T_{total})]}$ (17)

and;

$$U = U_r + U_c \tag{18}$$

Total heat output of the fin surface between two adjacent hoses with a spacing M and each having a unit length will be:

$$[2 \cdot W \cdot \eta + D_o] \cdot U_r \cdot (t_{max} - AUST) + [2 \cdot W \cdot \eta + D_o] \cdot U_c \cdot (t_{max} - t_a') = q_y \cdot M \qquad (19)$$

Let;

$$A = [2 \cdot W \cdot \eta + D_o] \cdot U_r , \qquad (20)$$

$$B = [2 \cdot W \cdot \eta + D_o] \cdot U_c , \qquad (21)$$

Solving Equation 19 for t_{max} for the required q_y:

$$t_{max} = \frac{q_y \cdot M + A \cdot AUST + B \cdot t_a'}{A + B} \qquad (22)$$

Kreider and Kreith (1981) expressed the fin efficiency for a flat plate collector losing heat both by radiation and convection from its top surface to the ambient. They defined an overall heat transfer coefficient without regarding the split between the radiation and convection. Kollmar(1957) expressed a similar fin efficiency. Following these definitions with;

$$U' = q_y/(t_p - t_a') \qquad (23)$$

$$m' = [U'/(k_{eq} \cdot T_{total})]^{1/2} \qquad (24)$$

$$\eta' = \tanh(m' \cdot W)/(m' \cdot W), \qquad (25)$$

a simple expression for t_{max} may be obtained:

$$t_{max} \approx t_a' + \frac{(t_p - t_a')}{[2 \cdot W \cdot \eta' + D_o]} \cdot M \qquad (26)$$

This equation will be identical with Equation 22 only when $AUST \approx t_a'$ and $U_r \approx U_c$. These conditions may approximate the actual design conditions only for a limited combination of the design variables. After computing t_{max},the surface temperature profile can be determined,noting that temperature gradients for radiation and convection are different:

$$t_p'(x) = \left(\frac{\cosh [m \cdot (M/2 - x)]}{\cosh [m \cdot W]} \right) \cdot (t_{max} - t_a' + c) + t_a' - c \quad \{D_o/2 \le x \le M/2\} \qquad (27)$$

$$t_p'(x) \cong t_{max} \qquad \{0 \le x < D_o/2\} \tag{28}$$

The term c represents the difference between the radiation and convection temperature gradients:

$$c = (U_r / U) \cdot [t_a' - AUST] \tag{29}$$

When t_a' equals AUST, c becomes zero.

t_{min} can be calculated from Equation 27 with x = M/2:

$$t_{min} = (\cosh[m \cdot W])^{-1} \cdot (t_{max} - t_a' + c) + t_a' - c \tag{30}$$

A simple solution for the required outside surface temperature of the heating element will be:

$$t_d = t_{max} + q_y \cdot [\, R_{co} + (L - D_o/2)/k + R] \tag{31}$$

This equation assumes a linear temperature distribution along the shortest path between the hose and the floor surface.
This derivation assumes that t_d is uniform around the hose perimeter. Thermal resistance between the hose and slab material, R may be neglected. R_{co} is the thermal resistance of floor covers above the slab:

$$R_{co} = \sum_{i=1}^{n_k} (\, x_i/k_i\,) \tag{32}$$

If an embedded electrical resistance heating system is used, then t_d will represent the required wire temperature at design conditions. (TSE, 1993)
For a hydronic system, the required mean water temperature will be:

$$t_w = \frac{q_y \cdot M}{X \cdot \pi} \cdot \left[\frac{1}{\alpha \cdot D_i} + \frac{1}{2 \cdot k_h} \cdot \ln(D_o/D_i)\right] + t_d \tag{33}$$

where for a turbulent flow of water in the heat transfer hose:

$$\alpha \cong 1056 \cdot [0.02 \cdot (t_w + 273) - 4.06] \cdot v_w^{0.8}/D_i^{0.2} \tag{34}$$

Back losses from a heated slab may be approximated by calculating the heat flow and corresponding surface temperature at the lower boundary along the path a-c (See Figure 5). The film coefficient at the boundary depends upon the adjacent space. If it is an indoor space, that boundary will be an unintentional ceiling heating surface. In this case:

$$q_a \cong \frac{1}{R_a + h_a} \cdot (t_d - t_i) \tag{35}$$

and $h_a = \sigma \cdot e \cdot F_r + 0.138 \cdot (t_p - t_i)^{0.25}$ (36)

The convective term is from Min and et. al's work (1956). Here t_p depicts the surface temperature at the lower boundary (ceiling of the adjacent room at below). The solution needs an iteration on $t_p(x)$. If the adjacent space is different than an indoor space, appropriate equations for h_a may be employed.
Perimeter losses may often be neglected except outdoor exposed or in the ground heated slabs.

3. Panel Cooling

A typical in-slab type radiant ceiling cooling panel construction is shown in Figure 3. The ceiling slab is maintained at a colder temperature than the indoor air by circulating chilled fluid through the embedded pipes. In order to minimize heat gains from the back and perimeter of the cooled slab,its back and edges are insulated. The heat gain from the conditioned space establishes its sensible cooling effect. Designating this as the negative of the heat gained by the slab, the required cooling load intensity of the cooling panel(s) will be:

$$q_y = - Q_y/A_p \cdot 1000 \qquad (37)$$

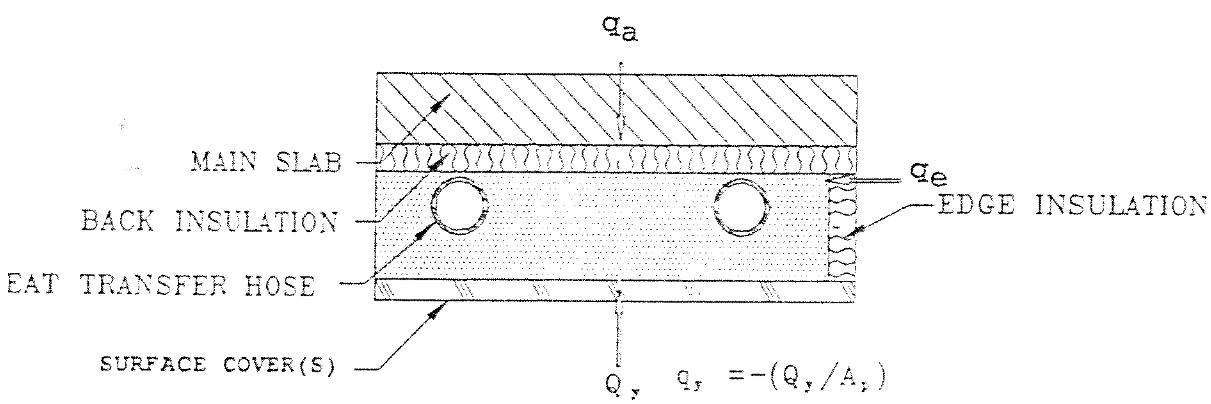

Figure 3. In-slab Type Ceiling Cooling Panel Construction.

All other heat hains are also reqarded as negative. The latent load may be handled by a hybrid cooling system where a radiant ceiling cooling panel is complemented by a small air handling unit (Wilkins and Kosonen,1992). This unit takes care of the necessary fresh air supply and the latent cooling load. It is interesting to note that all the fundamental equations for floor heating are applicable for ceiling cooling provided that q_y and Q_y are negative in cooling and t_{min} reciprocates with t_{max}. The amount of heat that can be absorbed by the panel surface of unit area is:

$$q_y = q_r + q_c = U_r \cdot (t_p - AUST) + U_c \cdot (t_p - t_a') \quad \{q_y \leq 0 \text{ in cooling}\} \qquad (38)$$

where;

$$U_r = r \cdot F_r \cdot \sigma \tag{39}$$

$$U_c = (1 - 2.22 \cdot 10^{-5} \cdot h)^{2.627} \cdot (4.96/D_e)^{0.08} \cdot 2.67 \cdot |(t_p - t_a')|^{0.25} \tag{40}$$

r is the linearization factor (see Equation 8).
It may be further simplified in the practical range:

$$r \cong [\,0.9659+(t_p\text{-}10) \cdot 0.00495\,] \cdot 10^{-8} \quad \{10°C \le (t_p+AUST)/2 \le 18°C\} \tag{41}$$

There is not a direct correlation for the effect of the room size on convective cooling. However dynamic similarity between floor heating and ceiling cooling enables to employ the same Min and et.al.'s (1956) data for floor heating.
A simple expression for AUST is given by Kilkis. (1992):

$$AUST = t_a' - d \cdot z \tag{42}$$

z is a function of the outdoor design temperature (Kilkis, 1993):

$$z \cong 7/(t_b\text{-}45) \qquad \{26\ °C \le t_b \le 36\ °C\} \tag{43}$$

When t_a' and AUST are determined, t_p can be solved for the required cooling intensity. Figure 4 shows the steady state composite fin model for ceiling panel cooling (Kilkis, 1993).

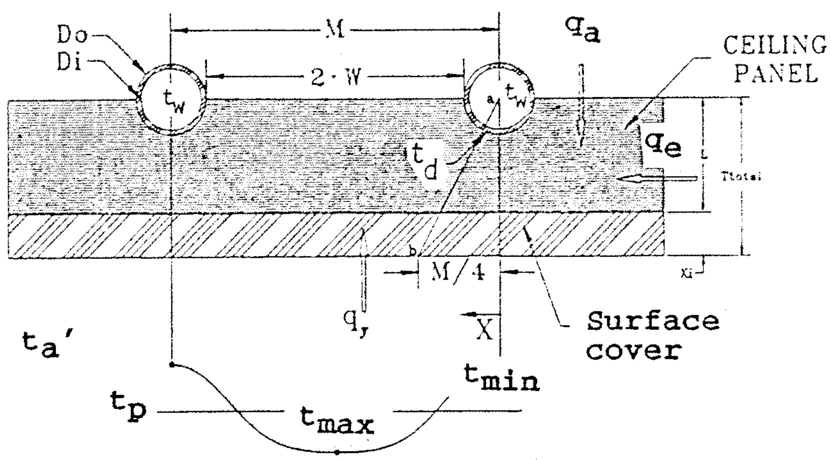

Figure 4. Composite Fin Model for Ceiling Panel Cooling [Kilkis 1993]
Part of the slab that remains below the hose center plane and the ceiling cover(s) below it establish the composite fin. Edges are assumed to be totally insulated. Superimposing the radiation and convection heat extraction intensities and solving for t_{min}, in a similar fashion with panel heating:

$$t_{min} = \frac{q_y \cdot M + A \cdot AUST + B \cdot t_a'}{A + B} \tag{44}$$

where A and B are given in Equations 20 and 21.

Panel surface temperature profile can be calculated with the same equations for panel heating (Equations 27 and 28) if t_{min} replaces t_{max}.
Using t_{min} and the corresponding path for heat diffusion:

$$t_d \cong t_{min} + q_y \cdot [R_{co} + (L - D_o/2)/k + R] \tag{45}$$

Finally, the required mean chilled fluid temperature, t_w can be calculated by using Equation 33, by noting that $q_y < 0$ in cooling.
Above equations indicate that heating model is directly applicable to radiant ceiling cooling, except secondary equations 41, 42 and 43.
Back heat gains may be predicted in a similar fashion like in floor heating. If the space at above is an indoor space, boundary film coefficients can be derived for floor heating equations.

4. Sample Calculations

In order to validate the practicality of the model, sample calculations were carried out and compared with finite element solutions. For this purpose, ANSYS software package, version 4.4A was employed.

Figure 5 shows the details of the floor cross-section.
Space at below is maintained at 15 °C. Hose spacing is 0.20 m O.C.
Other design inputs are:
Location Variables:
t_b = -3 °C; h = 0 (sea level)
Heated Space Variables:
Dimensions: 5m x 8m; number of outdoor exposed sides: 1 (d = 2) and fenestration is > 5%
t_a = 18 °C; Q_y = 3.1 kW; $A_p = 30 m^2$

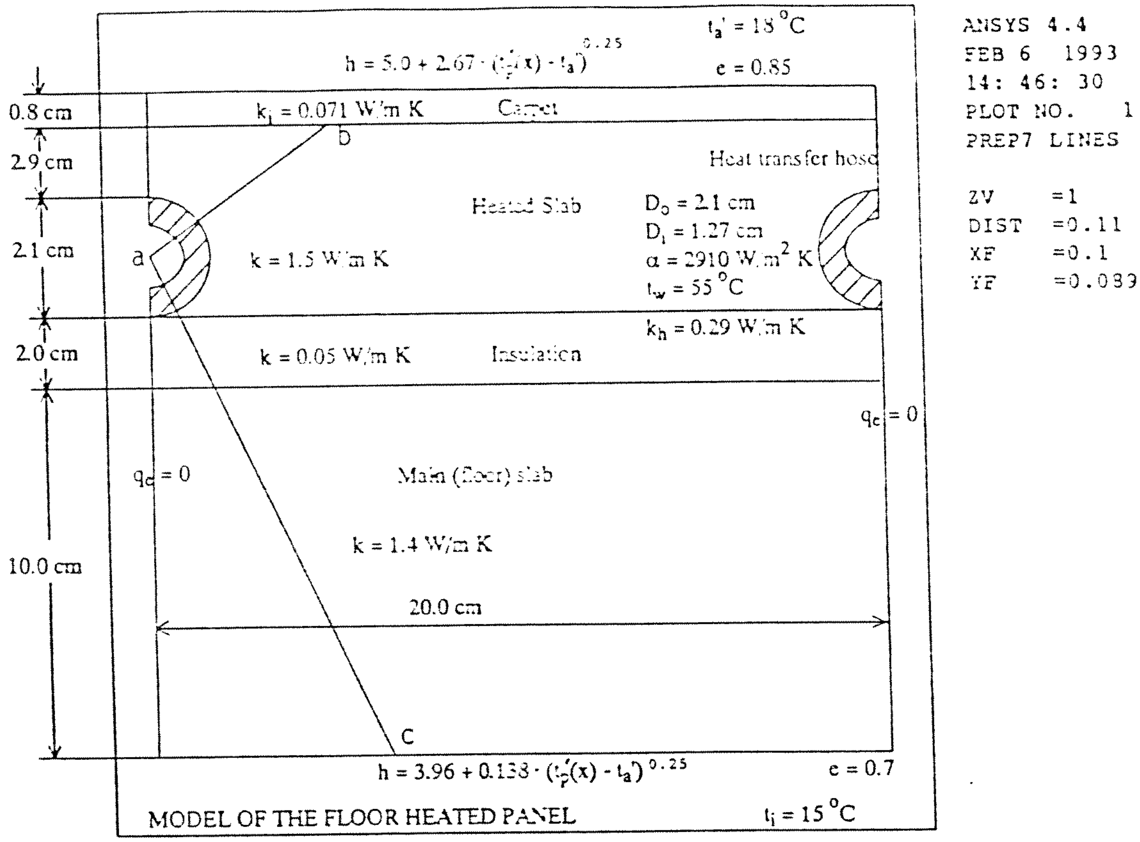

Figure 5. Floor Heating Slab Details for the Sample Solution.

Using these inputs:

q_y	$\cong 103.3$ W/m^2	(From Eqn. 1)
D_e	$= 5$ m	(From Eqn. 11)
U_c	$\cong 2.66 \cdot (t_p - 18)^{0.25}$	(From Eqn. 10)
z	$= 0.68$ °C	(From Eqn. 13)
AUST	$\cong 17.3$ °C	(From Eqn. 12)
r	$= (0.00525 \cdot (t_p + 17.3) + 0.7955) \cdot 10^8$	(From Eqn. 8)

Assume $t_p = 28$ °C to start iteration;

r	$= 1.033 \cdot 10^8$	
F_r	$\cong 0.85$	(From Eqn. 9)
U_r	$= 4.98$	(From Eqn. 6)

After iterating for t_p to meet the required load intensity given by Equations 3,4, and 5, and modifying by equations 4 through 8;

$t_p = 28.4$ °C

$U_r = 4.99$ W/m^{2}K

$U_c = 4.77$ W/m^{2}K

$U = U_r + U_c = 9.76$ W/m^{2}K

Then;

$k_e \ = 1.26$ (From Eqn.15)

$\eta \ \cong 0.83$ (From Eqs. 16,17 & 18)

$t_{max} = 30.1\ °C$ (From Eqs. 20,21 & 22)

$t_{min} \cong 26.8\ °C$ (From Eqs. 29 & 30)

neglecting R;

$t_d \ = 43.7\ °C$ (From Eqn. 31)

$q_a \ \cong 39\ W/m^2$ (From Eqs. 35 & 36)

$X \ = 0.72$

$t_w \ \cong 52\ °C$ (From Eqn. 33 with $\alpha = 2910$)

The same problem was discretized with the finite element package. Boundary conditions were:

top surface:

$$q_y \ = 4.98 \cdot (t_p'(x) - 17.3\ °C) + 2.66 \cdot (t_p'(x) - 15\ °C)^{1.25}$$

bottom surface:

$$q_a \ = 3.96 \cdot (t_p'(x) - 15)^{1.25} \cdot + 0.138\ (t_p'(x) - 15\ °C) \qquad \text{(with } e \cong 0.7\text{)}$$

$q_e \ = 0$ (totally insulated sides).

Figure 6 shows the output for the temperature distribution in the floor.

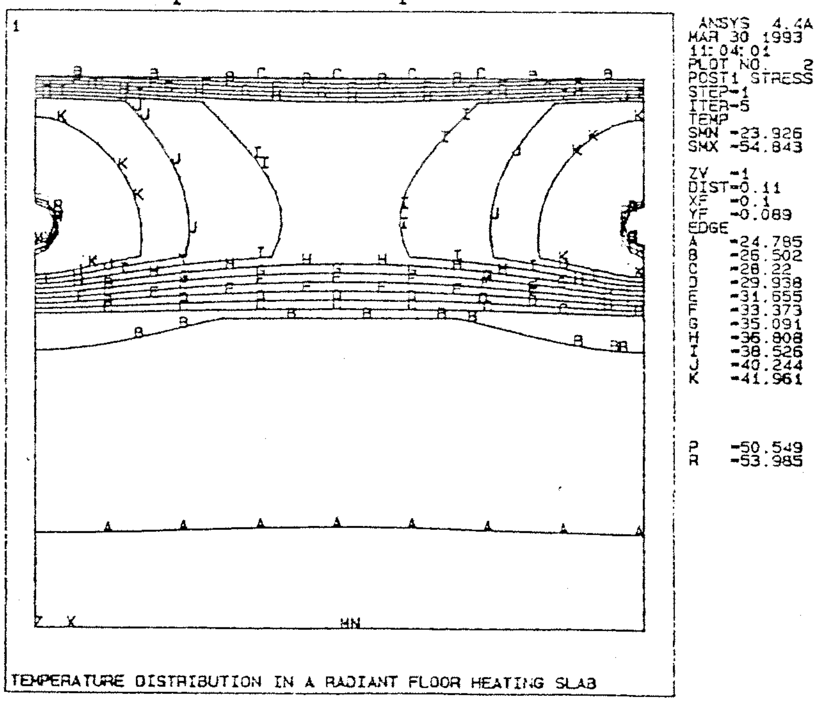

Figure 6. Finite Element Results For the Sample Solution

5. Comparison of Results and Discussion

Table 1 compares the results with the proposed algorithm.
Whenever applicable, predictions by DIN Standard(1990) and Krinninger (1989) are also cited.

Table 1. Comparison of results.

Variable	Proposed Algorithm	F.E. Solution	DIN and Krinninger
q_y [W/m2]	103.3	103.8	119.7
at tp[°C]	28.4	28.6	28.6(1)
t_{max} [°C]	30.1	29.6	--
t_{min} [°C]	26.8	27.6	--
q_a [W/m2]	39.0	37.5	--
q_e [W/m2]	neglected	neglected	neglected
X	0.72	0.73	--
t_w[°C]	52.0	55.0	59.9

(1). This value is selected arbitrarily for comparison purposes through Equation 14. The same equation yields t_p value of 27.3 °C for q_y = 103.8 W/m^2

This comparison shows that the proposed algorithm shows close agreements with respect to the required mean water temperature, thermal efficiency and the heat output intensity. Other two references overpredict the required mean water temperature and overestimate the heat output intensity for a given effective floor surface temperature.
However, the proposed algorithm predicts a bigger temperature swing on the floor surface, where this information in important only for human comfort for bare feet. Other references do not provide any means to predict t_{max} and t_{min}. With respect to main design variables like q_y, x and t_w, the proposed algorithm gave consistently agreeable results for several case designs. With this conclusion, it can be regarded as a practical and sufficiently accurate tool for design of radiant slabs. Panel cooling algorithm was also tested with finite element solutions and similar conclusions were drawn.

Figure 7 shows the temperature distribution in the slab.

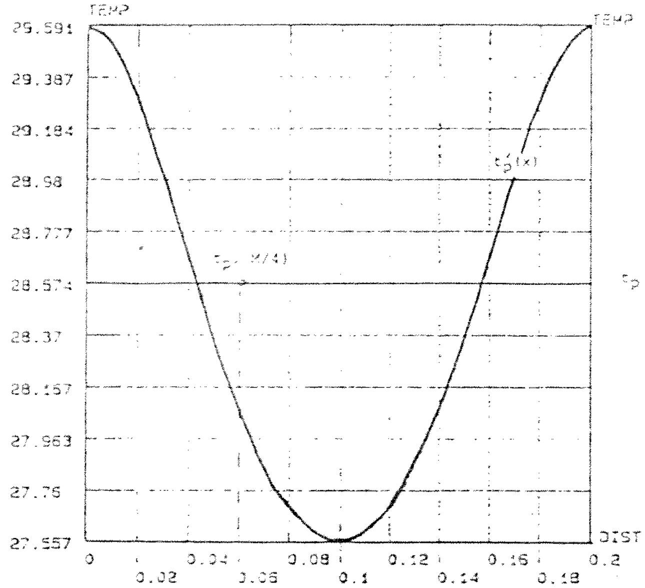

Figure 7. Temperature Distribution in the Heated Slab.

6. Conclusion

Analytical models for panel cooling and heating under steady state conditions were developed. These models largely simplify the design calculations where sufficient accuracy is retained and reasonable agreement within the practical design range were obtained. Although the fin model simplifies the heat diffusion into one dimension, sufficient accuracy is retained through the definition of equivalent thermal conductivity. Models were tested with finite element models. Sample calculations revealed that the mean chilled fluid temperatures may be as high as 18 °C in panel cooling, especially if the slab material is concrete. This is a relatively high temperature when compared with other conventional cooling systems. This moderate chilled fluid temperature requirement guarantees noticeable improvements in the coefficient of performance of absorption type heat pumps and solar collector efficiencies (Kilkis 1993). In the same manner, the total exergy loss of the system substantially decreases. These increase the competitiveness and feasibility of such alternative cooling systems. Due to the fact that panel cooling permits the designer to freely select the hose spacing in the technically feasible range, it is almost always possible to conform the temperature ratings of a given heat pump and thus optimize the overall performance. This flexibility also makes it possible to control the risk of condensation in a stand alone

the risk of condensation in a stand alone system. If the humidity has to be controlled, recent literature indicates that hybrid systems are feasible with minimum duct work and air handling unit requirements.
Similar features also hold true for panel heating systems. A recent study revealed that employing panel heating systems may eliminate the need for a supplementary boiler for air to water type heat pumps. (Kilkis, 1992)

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8. Acknowledgement

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DESIGN OF EMBEDDED SNOW MELTING SYSTEMS

PART 1. HEAT REQUIREMENTS : AN OVERALL ASSESSMENT AND RECOMMENDATIONS

Dr.B.I.Kilkis*

ABSTRACT

Although ASHRAE Handbook is the single reference for snow melting systems in North America, it provides design information only for 33 cities in 27 states. Its fundamental requirement of a detailed snow fall frequency analysis often forces the designers to count on intuition or limited experience for other locations and countries. Objections were also raised regarding its heat load predictions in comparison with field observations and other correlations. The aim of this study is to develop a universal and simple technique which does not require extensive manipulation of the meteorological data. In this repect, a new design algorithm is proposed which requires only design air temperature, wind speed, and the maximum recorded daily snow fall in order to calculate the heat requirement. The classification of the snow melting performance is directly associated with the snow free area concept, and a snow fall coincident air temperature is defined. Comparisons indicated a close agreement with other reports, and a limited agreement with ASHRAE guidelines for class 1 and class 2 systems. Results and other reports suggest that ASHRAE guidelines overestimate especially class 3 loads.

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1. INTRODUCTION

Embedded snow melting systems rely on a heated surface which is designed to melt the snow within a prescribed performance. The surface can be heated to the required effective temperature (t_p) either by hydronic piping or an electric resistance system (Potter,1967). In a hydronic system, rubber,thermoplastic,or metal piping may be used. A conventional,waste,or alternative energy source,or ground heat with heat pipes (Lee and et.al.,1984, Tanaka and et.al.,1981) may be utilized. The entire success of the system depends on an accurate prediction of the heat load. However,there are two difficulties: most of the empirical heat transfer correlations are limited to small scale tests,and generally the local meteorological data may be insufficient. In order to calculate the heat load,the desired system performance has to be prescribed. It is classified according to the permissible amount of snow accumulation during the snow,and how rapid it can be melted. Chapman (1957) established the rules of calculating the heat load intensity for each of the three performance class in terms of a snow fall frequency analysis. Therefore,at the absence of such an analysis,these rules will have a very limited use. Snow free area ratio (A_r), which is the ratio of snow free surface area to the total snow melting surface area (ASHRAE,1991) is indeed a good measure of the performance during the snow. Therefore,for practical purposes,it can be directly associated with Chapman's definitions of snow melting class. In terms of this relation:

Class 1 (residential): During the snow,entire surface is permitted to be covered with snow (A_r = 0). After the snow, system is expected to melt the accumulated snow,provided that an almost instant drop in the air temperature is recognized in sizing the system. Otherwise ice may form on the surface (Adlam,1950).

Class 2 (commercial): During the snow fall, 50% of the surface is permitted to be covered with snow (A_r = 0.5).

Class 3 (industrial): During the snow fall,the entire surface is kept free from snow accumulation (A_r = 1).

Figure 1 shows the ideal surface conditions according to this classification. In class 1,the temperature of the heated surface is uniform where accumulation of snow practically eliminates the heat loss to the atmosphere.

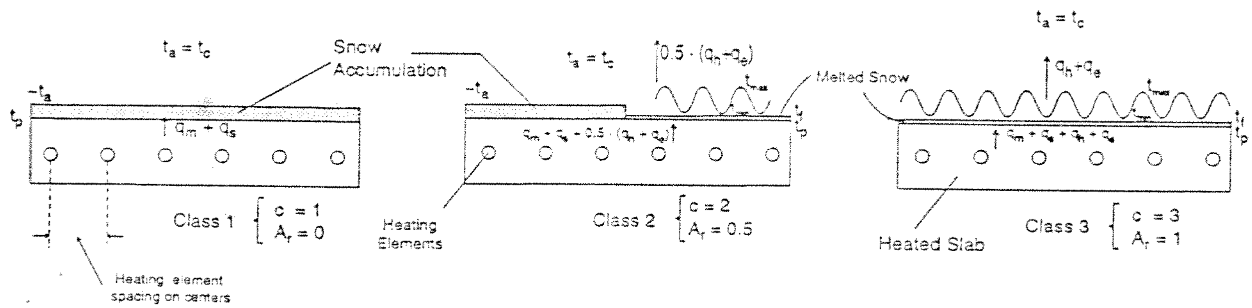

Figure 1. Surface Conditions During Snow for Each Performance Class.

In class 3, heat is lost from the surface of the melted snow to the atmosphere by radiation,convection,and evaporation.

Class 2 is an intermediate case.

On the snow free surfaces,temperature is not uniform. The minimum surface temperature (t_{min}) takes place at halfway between the embedded heating element positions. On surfaces with a temperature below 32°F, snow will not melt or eventually freeze (Adlam, 1950). This can be avoided by raising the effective surface temperature in order to maintain t_{min} above 32°F (Kilkis, 1992). There are four components of the total snow melting heat load intensity:

q_s = intensity of sensible heat transferred to snow,

q_m = intensity of heat of fusion,

q_e = intensity of heat of evaporation,

q_h = intensity of heat loss to the atmosphere.

q_s and q_m are independent of the snow free area ratio. Therefore the total heat load intensity during snow fall will be (Chapman, 1957):

$$q_o = q_s + q_m + A_r \cdot (q_e + q_h) \qquad [1]$$

2. THEORY

2.1. Heat Requirement During Idling

Usually the system is idled so that at the incipience of snow, the heated surface will be ready to function. During idling, the heated surface is free of snow, and the surface temperature has a similar distribution with a class 3 system (see Fig. 1). The following equation by Chapman (1957) defines the idling heat load intensity:

$$q_i = (0.27 \cdot U_{met} + 3.3) \cdot (t_p - t_m) \qquad [2]$$

In this expression, the radiation heat transfer coefficient is 0.8 Btu/h ft²°F, which agrees with the values used by Adlam (1950) and Willamson (1967). However, it does not differentiate the sky conditions and the temperature gradient. Williams (1967) noted that underpredicts the radiation heat transfer under a clear sky. r this reason it can not be used for conditions after the snow,or ring idling where in both cases the sky may be clear. llamson (1967) reported that when the temperature gradient is all,radiation coefficient will be again higher than Chapman's ediction. The convective term agrees with Adlam's (1950) perimental data where both authors used test surfaces up to 10 ft^2. erefore Equation 2 may reasonably predict the convective heat loss om a small surface like a driveway,but it will overestimate for ger applications. Williams (1967) already reported this dition and provided more elaborate data.

. Heat Requirement During Snow fall

ure 2 shows the qualitative progress of q_o for each performance ss. At the incipience of snow,the only load that exists is the : loss to the atmosphere (q_h), provided that the system was ed. Otherwise the heat load intensity prior to snow will be zero. ally snow fall is accompanied by a rise in the air temperature),and this study characterizes this phenomenon by the coincident erature (t_c) at design conditions.

lass 1,while blanketing effect of the accumulating snow gradually inates q_h ; load intensities,namely q_m and q_s are introduced. ough there may be some initial evaporation,it stops when the ace is completely covered with snow. Therefore,the total heat intensity may show an early peak,and then becomes ($q_m + q_s$). may be higher or lower than q_h depending upon current outdoor tions. When snowing ends, heat load intensity momentarily drops to zero. At this stage all the accumulated snow is practically at the melting point.

In class 2, the progress is similar except the additional evaporation, convection and the radiation heat losses to the atmosphere from the snow free surfaces. An initial peak due to an early evaporation from the entire surface is possible. Eventually the load intensity becomes $q_m + q_s + 0.5 \cdot (q_e + q_h)$. This may be higher or lower than q_h depending upon current outdoor conditions. When snowing ends, the remaining load intensity will be $0.5 \cdot (q_e + q_h)$.

In class 3, all elements of the total snow melting load are practically coincident. This cumulative load is practically effective during the entire snowing period. Then it becomes zero.

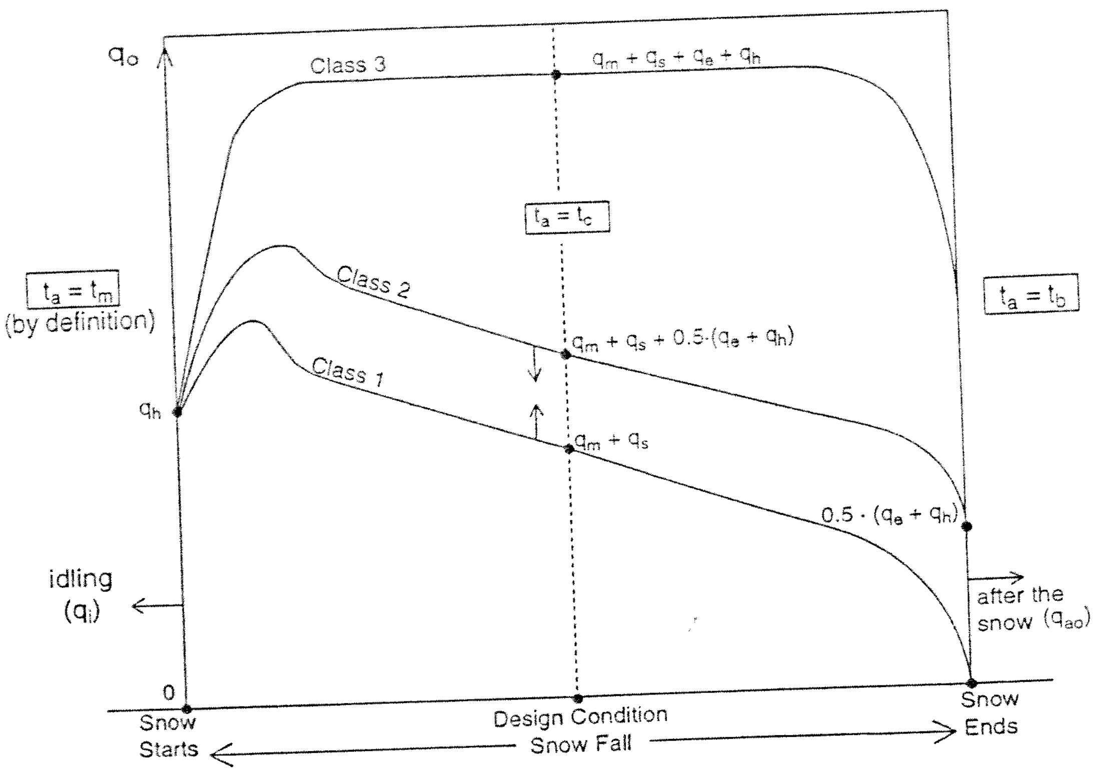

Figure 2. Progress of Snow Melting Load Intensity During Snow Fall.

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This figure indicates the essence of additional design checks to complement the ASHRAE guidelines:

In class 1 : $q_o = q_h$ if $q_h > q_m + q_s$,

In class 2 : $q_o = q_h$ if $q_h > q_m + q_s + 0.5 \cdot (q_e + q_h)$.

ASHRAE guidelines (1991) provide the following equations:

$$q_s = 2.6 \cdot s \cdot (32 - t_a) \qquad [3]$$

$$q_m = 746 \cdot s \qquad [4]$$

$$q_e = h_{fg} \cdot (0.0201 \cdot U_{met} + 0.055) \cdot (p_s - p_{av}) \qquad [5]$$

$$q_h = 11.4 \cdot (0.0201 \cdot U_{met} + 0.055) \cdot (t_f - t_a) \qquad [6]$$

p_s is 0.185 in. of Hg at 33°F. Chapman derived Equation 5 from his same idling test set-up,therefore it may overestimate the evaporation load for large surfaces. Equation 6 does not recognize the split between radiation and convection losses which are sensitive to different atmospheric factors. Instead,Chapman (1957) used his evaporation term and Mc.Adam's (1942) relationship between radiation,convection and evaporation to calculate q_h (Pentecost and et.al,1965). Indeed radiation loss is primarily a function of the cloudiness and time (day or night). The net long-wave radiation under cloudy sky (Williams,1976):

$$q_r = 10.3 + 8.14 \cdot 10^{-10} \cdot [(t_f + t_a) + 920]^3 \cdot (t_f - t_a) - \\ -7.68 \cdot 10^{-10} \cdot [255.2 + t_a/1.8]^4 \qquad \{ \text{cloudy sky} \} \qquad [7]$$

For clear sky (Williamson,1967):

$$q_r = 30.15 + 0.74 \cdot (t_f - t_a) \qquad \{ \text{clear sky} \} \qquad [8]$$

These equations have been adjusted for the emissivity of water film which is 0.95.

Although Williams (1976) investigated the effects of the size and the surrounding terrain on convection, there is no sufficient data to obtain a correlation with respect to the surface area. For this reason a conservative approach will be to use his data for a 16 ft^2 snow melting surface:

$$q_c = (0.14 \cdot U + 0.39) \cdot (t_f - t_a) \qquad [9]$$

Then:

$$q_h = q_r + q_c \qquad [10]$$

Although ASHRAE guidelines suggest that t_f may be taken 33°F, this does not quarantee that t_{min} will be above 32°F. Especially for wide heating element spacings, t_f, consequently t_p may need to be increased in order to maintain t_{min} above 32°F. However this adjustment effects the surface heat losses and an iterative solution becomes necessary. Usually, snow melting is performed at ground level with exceptions like roof tops and bridges. Chapter 14 in ASHRAE Fundamentals Handbook already recognizes that most of the wind data are recorded at 33 ft above the ground level (H_{ref}), and in open fields. Therefore the meteorological wind data must be adjusted with respect to surrounding terrain and height of the snow melting surface, H:

$$U = A_o \cdot (H/H_{ref})^a \cdot U_{met} \qquad [11]$$

A_o and a are the adjustment factors (ASHRAE, 1989).
This adjustment also applies for the idling load (see Eqn. 2).

For a snow melting surface at the ground level, H may be taken 6 ft in order to allow for the diffusion of evaporation and convection. Using Bowen's (1929) ratio R, the evaporation heat load intensity can be predicted in terms of the convective heat loss intensity (Williams,1976):

$$q_e = q_c/R \cdot (p_s - p_{av})/(t_p - t_a) \qquad [12]$$

$$R = 0.01 \cdot p_a/29.9 \qquad [13]$$

p_{av} may approximately be determined from the following correlation in the range of $-20°F \leq t_a \leq 35°F$:

$$p_{av} = \Theta\cdot[0.0371+1.964\cdot10^{-3}\cdot t_a + 5.235\cdot10^{-5}\cdot t_a^2 + 7.723\cdot10^{-7}\cdot t_a^3] \qquad [14]$$

or, roughly, in the range of $14°F \leq t_a \leq 32°F$, (Williamson,1967):

$$(p_s - p_{av}) \approx 0.01 / 1.8 \cdot (t_p - t_a) \qquad [15]$$

In Equation 13, p_a may be corrected for high altitudes from the sea level,(Oskay and et.al,1977):

$$p_a/29.99 = (1 - 0.687 \cdot 10^{-5} \cdot h)^{5.255} \qquad \{ 0 \leq h \leq 3600 \text{ ft} \} \qquad [16]$$

2.3. Heat Requirement After the Snow

ASHRAE guidelines do not recognize this phase,when in fact the remaining snow has to be melted. Usually a very low air temperature follows the snow fall,and sometimes the sky may become clear (Adlam 1950, Wilkinson,1967). Therefore a conservative design should assume a clear sky and t_a equal to t_b. According to this adverse atmospheric conditions,total heat load intensity may increase,in spite of the absence of fusion and sensible heat loads.

In this case, after the snow conditions will govern the design:

$$q_o = q_{oa} \qquad \text{if } q_{oa} > q_o$$

This is especially true for locations with a cold design outdoor temperature, and moderate rate of snow fall.

For class 1, while the surface is still covered with snow, q_{oa} is initially zero. The initial condition for class 2 and 3 is:

$$q_{oa} = (1 - A_r) \cdot (q_{ea} + q_{ha}) \qquad [17]$$

Figure 3 shows the ideal progress of after the snow load intensity.

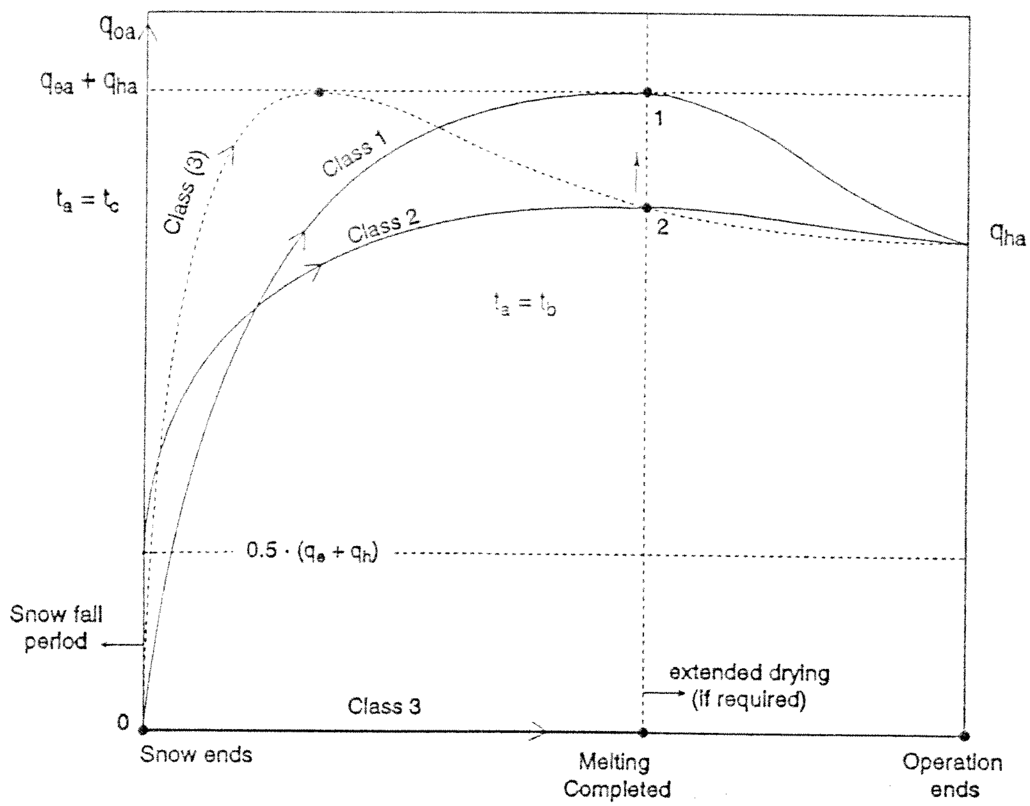


Figure 3 . Typical Progress of After the Snow Load Intensity.

Class 1: At the absence of sensible and fusion loads, the system will begin to evaporate the snow from the surface. Later, with the formation and expansion of the snow free area, the total heat loss by iation,convection and evaporation increases to ($q_{ea} + q_{ha}$). :he system is kept running long enough,surface may begin to dry eventually the evaporation load diminishes.

ss 2: According to Equation 17, the initial load intensity will be ·($q_{ea} + q_{ha}$). While the snow free area expands, part of the face may already begin to dry, especially at locations above the :ing element positions, if the drainage is sufficient and ective. Nevertheless, this can not be guaranteed. Expansion of :ed surfaces is faster than drying of the surfaces. Therefore nt 2 may approach or even meet point 1.

ss 3: The load is virtually zero,unless an extended evaporation cess is required in order to dry the surface completely before the :ed snow freezes due to poor drainage and,or a heavy snow. This dition is shown by the broken line in Figure 3.

conclusion,design value of q_{oa} will be:

$$q_{oa} = q_{ea} + q_{ha} \qquad \{ \text{ for class 1 and 2 } \} \qquad [18]$$

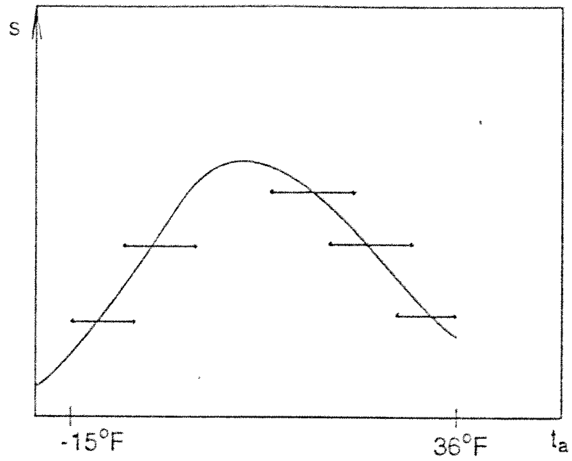
will also apply for class 3,if extended evaporation is required.

. Design Heat Load Intensity

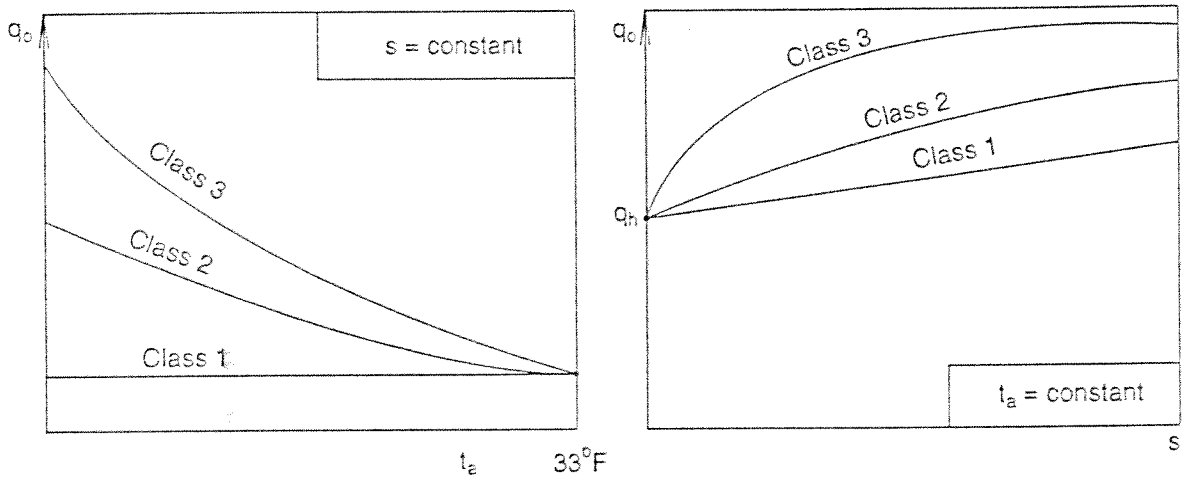
design heat load intensity (q_y) will be one of q_o, q_{oa}, or q_i, chever is higher. In sizing the system,expected back and perimeter : losses must also be considered. Many authors assume that this al is about 40%. This fraction depends upon the heated slab perature, outdoor conditions, degree of insulation and the struction. Therefore,it needs an elaborate calculation and will be cussed in the next article of this study.

2.5. Coincident Air Temperature

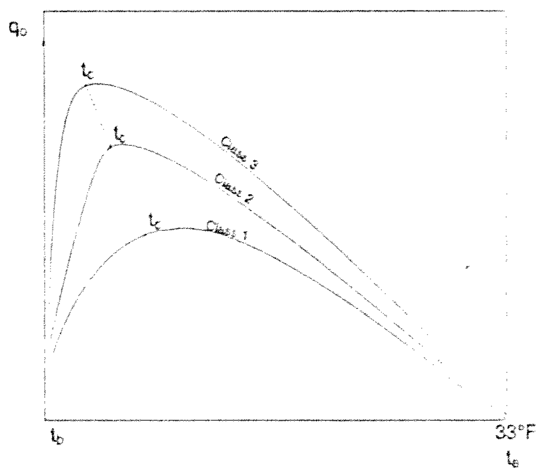
Due to small amount of humidity that cold air can hold, a heavy snow usually takes place at moderate air temperatures (Adlam, 1950). For this reason, the design outdoor temperature (t_D) and a heavy snow storm do not coincide. Therefore, any heat load equation will be meaningful and accurate as long as the air temperature corresponding to the design rate of snow fall is properly estimated. Otherwise a snow fall frequency analysis over an extended period may become necessary. In this case the validity of the equations involved in such an analysis is important. Using his equations, Chapman (1957) performed this analysis for some US cities; based on about 3700 readings per station, between 1940 and 1949. However it is difficult to perform the frequency analysis for every location, either due to the lack of sufficient data or its time intensive nature. In order to facilitate engineering calculations, a coincident air temperature was defined such that reliable results without any elaborate frequency analysis can be obtained. In deriving such an expression, the typical relationships between the air temperature, rate of snow fall, and the snow melting loads were considered. Figure 4 qualitatively shows a typical relationship between the air temperature and the rate of snow fall. Figures 5 and 6 show the relationship between the heat load intensity, air temperature and rate of snow fall, respectively. Combination of these figures define the coincident temperature (t_C) as the air temperature which corresponds to maximum snow melting load intensity at the design rate of snow fall for a given performance class and location.



igure 4. A Typical Relationship Between t_a and s.



igure 5. Variation of q_o with s and t_a.



igure 6. Coincident Temperature in Terms of Performance Class.

Comparative studies with snow fall frequency analysis revealed the following simplistic expression for t_c:

$$t_c = t_b + \frac{(33 - t_b)}{(0.1 + 1.2 \cdot C)} \qquad [19]$$

2.6. Design Rate of Snow Fall

In order to use the equations in ASHRAE guidelines, one needs to determine the rate of snow fall, but explicit data is available only for 10 locations. In order to provide a design tool for other locations and countries, the following expression was developed:

$$s' = \left[\frac{SF}{24} \cdot C \right] \cdot \frac{\Omega_s}{62.4} \qquad [20]$$

Here s' is the "design rate of snow fall" and is a function of class, C. Depending on the class, it is a multiple of the average hourly rate of snow fall (SF/24). Indeed, surveys about meteorological data indicate that the peak hourly rate of snow fall may be 2 to 3 times greater than the average hourly rate of snow fall during a stow storm (see Fig.7). SF is the maximum amount of snow fall in 24 hours recorded at a given location, and is the most common form of snow fall data in many countries. For class 1, the entire surface is permitted to be covered with snow, and therefore the average hourly rate of snow fall may be used provided that a possible ay in snow melting after a heavy snow storm can be tolerated. sity of snow (Ω_S) depends on the air temperature. Using the ilable meteorological data given by Adlam (1950) the following relation was obtained:

$$\Omega_S \approx 2.6 + 0.06 \cdot t_a + 0.0027 \cdot t_a^2 \qquad [21]$$

se equations were used in order to calculate the design rate of w fall for various locations. Table 1 gives both the calculated ults for class 2,and few data available in the ASHRAE Handbook, p.45,Table 4. SF values were compiled from Adlam's (1950) dies which reflect U.S. Weather Bureau data.

PROPOSED ALGORITHM AND CALCULATIONS

suggested technique for calculating the loads for a snow melting tem were compiled and a new algorithm was developed. This orithm is applicable both for hand calculations and computer gramming. Table 2 outlines the systematics of the new algorithm. omputer program was also developed by employing this algorithm. enables to calculate and compare different heat load intensities a given performance class during each phase of the applicable w melting operation. It also enables to change the heated slab struction variables so that the system can be optimized. le 3 gives typical results and compares them with the data given the ASHRAE guidelines.

TABLE 1. Calculated Design Rate of Snow Fall for Some U.S. Cities. (Class = 2 , terrain = suburban , snow melting surface = at ground level. s' values are rounded to the second digit.)

City	State	T_a [°F]	T_c [°F]	Ω_s snow density [lb/ft^3]	SF snow fall [in/24hr]	s' [in. H_2O/h] Calculated	ASHRAE
Colorado Springs	CO	2	14.4	4.04	~14	.08	-
Washington	DC	17	23..4	5.50	25.4	.18	-
Chicago	IL	-8	8.4	3.31	14.4	.06	-
Evansville	IN	9	18.6	4.66	20.0	.12	.08
Portland	ME	-1	12.6	3.80	21.0	.11	.16
Boston	MA	9	18.6	4.66	16.0	.10	-
Detroit	MI	6	16.8	4.38	11.0	.06	-
Sault Ste. Marie	MI	-8	8.4	3.31	12.6	.06	-
Duluth	MN	-16	3.6	2.86	28.7	.11	-
St. Louis	MO	6	16.8	4.38	20.4	.12	-
Lincoln	NE	-2	12.0	3.72	12.0	.06	-
Reno	NV	10	19.8	4.86	22.5	.15	-
Albuquerque	NM	16	22.8	5.39	6.0	.04	-
Albany	NY	-1	12.6	3.80	30.4	.15	.16
Buffalo	NY	6	16.8	4.38	17.4	.10	-
Asheville	NC	14	21.6	5.17	15.0	.10	.08
Bismarck	ND	-19	1.8	2.73	11.0	.04	.08
Cincinatti	OH	6	16.8	4.38	11.0	.06	.08
Columbus	OH	5	16.2	4.29	11.9	.07	-
Oklahoma City	OK	13	21.0	5.07	11.3	.08	-
Portland	OR	23	27.0	6.20	15.0	.12	-
Philadelphia	PA	14	21.6	5.17	21.0	.15	-
Pittsburgh	PA	5	16.2	4.29	15.0	.09	-
Rapid City	SD	-7	9.0	3.37	18.3	.08	-
Memphis	TN	18	24.0	5.61	18.0	.13	-
Amarillo	TX	11	19.8	4.86	13.5	.09	-
Burlington	VT	-7	9.0	3.37	15.1	.07	-
Seattle	WA	26	28.8	6.58	21.5	.19	-
Spokane	WA	2	14.4	4.04	9.6	.05	-
Madison	WI	-7	9.0	3.37	12.9	.06	.08
Cheyenne	WY	-1	12.6	3.80	14.0	.07	-
Cleveland	OH	5	16.2	4.29	12.0	.07	.08
Billings	MT	-10	7.2	3.18	~20	.09	.08
New York	NY	15	22.2	5.28	17.8	.13	-
Great Falls	MT	-15	4.2	2.91	~20	.08	-
Ogden	UT	5	16.2	4.29	~20	.11	-
Caribou	ME	-13	5.4	3.01	22.0	.09	-
Falmouth	MA	9	18.6	4.66	~14	.09	-
Hartford	CT	7	17.4	4.48	~25	.15	-
Minneapolis	MN	-12	6.0	3.07	~25	.10	-
Mt. Home	ID	12	20.4	4.96	~10	.07	-
Salina	KS	5	16.2	4.29	~12	.07	-

'ABLE 2. Proposed Algorithm for Snow Melting Load Calculations.

LOAD INTENSITIES			PERFORMANCE CLASS 1	2	3	COMMENTS
SNOW MELTING	q_s		Eqn. 3	Eqn. 3	Eqn. 3	$t_a = t_c$ (Eqn. 19),
	q_m		Eqn. 4	Eqn. 4	Eqn. 4	$s = s'$ (Eqns. 20, 21)
	q_e		-	Eqn. 12	Eqn. 12	Adjujst the wind speed (Eqn. 11)
	q_h	q_r	-	Eqn. 7	Eqn. 7	$q_h = q_r + q_c$
		q_c	-	Eqn. 9	Eqn. 9	
	q_o		$q_s + q_m$	$q_s + q_m + 0.5 \cdot (q_e + q_h)$	$q_s + q_m + q_e + q_h$	In Class 1, if $q_h > q_m + q_s$ then $q_o = q_h$. In class 2, if $q_h > q_m + q_s + 0.5 \cdot (q_e + q_h)$ then $q_o = q_h$
AFTER SNOW	q_{ea}		Eqn. 12	Eqn. 12	Eqn. 12	$t_a = t_b$
	q_{ha}	q_{ra}	Eqn. 7	Eqn. 7	0	$q_{ha} = q_{ra} + q_{ca}$ $t_a = t_b$ Adjust the wind speed (Eqn. 11)
		q_{ca}	Eqn. 9	Eqn. 9	0	
	q_{oa}		$q_{ea} + q_{ha}$	$q_{ea} + q_{ha}$	0	For class 3 if extended evaporation is required, then $q_{oa} = q_{ea} + q_{ha}$
IDLING	q_i		Eqn. 2	Eqn. 2	Eqn. 2	$t_a = t_m$ Adjust the wind speed (Eqn. 11)
DESIGN	q_y		$q_y \geq q_o$			if $q_i > q_o$ and q_{oa}; $q_y \geq q_i$ if $q_{oa} > q_i$ and q_o; $q_y > q_{oa}$

TABLE 3. Design Data for Three Classes of Snow Melting System.

Heat load intensity [Btu/h ft²]

City	q_i	q_{oa}	q_o Class 1 System	Class 2 System	Class 3 System
Albuquerque, NM	31	72	21/71	48/82	86/167
Amarillo, TX	37	81	44/98	90/143	139/241
Boston, MA	40	93	51/107	102/231	160/255
Buffalo, NY	40	89	55/80	101/192	159/307
Burlington, VT	58	111	44/90	86/142	140/244
Caribou, ME	72	122	60/93	107/138	171/307
Cheyenne, WY	50	105	42/83	84/129	138/425
Chicago, IL	50	114	42/89	79/165	137/350
Colorado Spr., CO	47	100	43/63	90/63	139/293
Columbus, OH	37	91	38/52	78/72	125/253
Detroit, MI	39	89	35/69	70/140	120/255
Duluth, MN	80	128	78/114	125/206	199/374
Falmouth, MA	48	83	45/93	91/144	140/165
Great Falls, MT	72	127	55/112	99/138	164/372
Hartford, CT	38	87	79/115	140/254	208/260
Lincoln, NE	59	117	36/67	80/202	136/246
Memphis, TN	26	66	62/134	119/144	180/212
Minn.-St.Paul, MN	70	120	70/95	116/120	184/204
Mt.Home, ID	35	79	33/50	74/90	115/140

Table 3 cont.

New York, **NY**	43	**80**	60/121	120/298	181/342
Ogten, **UT**	38	**93**	63/98	111/216	177/217
Oklahoma City, **OK**	41	**85**	37/66	84/81	131/350
Philadelphia, **PA**	28	**73**	71/97	134/229	195/263
Pittsburgh, **PA**	38	**91**	48/89	94/157	144/275
Portland, **OR**	18	56	54/86	103/97	163/111
Rapid City, **SD**	59	**113**	53/86	95/102	157/447
Reno, **NV**	34	73	73/98	127/154	191/155
St.Louis, **MO**	35	**89**	65/122	117/152	177/198
Salina, **KS**	42	**91**	38/85	79/120	126/228
Sault Ste.Ma., **MI**	62	**114**	37/78	79/144	129/213
Seattle, **WA**	20	50	77/92	154/128	231/133
Spokane, **WA**	32	**97**	30/87	65/127	112/189
Washington, **D.C.**	26	68	84/117	154/121	236/144

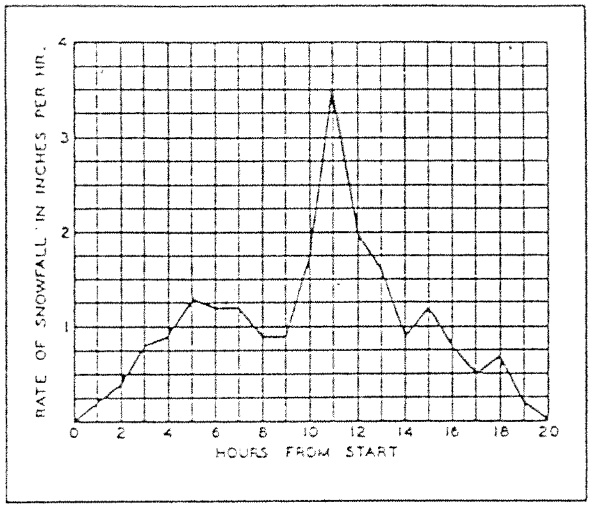
First figures in each row for q_o give the calculated result of the proposed algorithm. Second figures are ASHRAE predictions. Meteorological wind data is adjusted according to a ground level surface in a suburban terrain. For evaporation and convection, characteristic height is 6 ft. Any idling or after the snow load intensity is typed in bold numerals if it is higher than the snowmelting load intensities for one or more classes. SF and t_c values are taken from Table 1.

4. COMPARISON and DISCUSSION OF RESULTS

In every location listed in Table 3 shows that snow melting load for class 1 systems is surpassed by the corresponding after the snow load with two exceptions (Seattle,WA and Washington,D.C.). This is also true for class 2 loads in some locations. This implies that the installed capacity will deliver a better performance irrespective of the original choice of the class,when q_{oa} replaces q_o. This indicates that class 1 and 2 systems can not be sized according to the snow melting load alone. The new algorithm takes this argument and other several factors into account such that the designer can customize the design with respect to the local meteorological conditions and the surrounding. For example,in New York City if the terrain is open instead of suburban, the load intensities will be:

q_i : 57 ; q_{oa} : 106 ; q_o : 60 , 129 , 200 for class 1,2 and 3 respectively. In class 1 with A_r = 0, the snow melting load is independent of the terrain or elevation,but these factors definitely effect the other loads. When after the snow load replaces class 1 load,it can be seen that for majority of the locations, ASHRAE prediction may be short of after the snow load intensity. In other words, if sizing is made according to ASHRAE predictions for class 1, there will be a risk of poor performance following the snow. For class 2 and especially class 3 systems ASHRAE systematically overestimates the loads,except for Portland,OR, Seattle,WA, and Washington,D.C. Although wind speed is not adjusted,and the atmospheric heat losses are overestimated in ASHRAE guidelines,these factors can not explain the discrepancy completely.

example, Figure 7 shows the recorded most severe snow storm in York City.



gure 7. Rate of Snow Fall During Storm in New York City, December,19,1948.

ording to this figure,the instantaneous peak rate of snow fall is in. of snow/h. Corresponding to the calculated snow density of lb/ft^3 at design air temperature t_b : 15°F,this is equivalent to 3 in. of H_2O/h. Then, at the absence of any wind speed adjustment, t load intensity at 15°F outdoor design temperature,and a class 3 tem will be 278 Btu/h ft^2. ASHRAE prediction namely 342 Btu/h ft^2 responds to a rate of 0.31 in. of H_2O/h snow fall (≈ 4.7 in. of w/h) which has not been recorded so far at this location.

ter (1967) reported data for "successfully operating" systems 93 locations in the United States. None of these locations have a w melting capacity higher than 193 Btu/h ft^2 on the heated face. His data was compared with the calculated results and ASHRAE dictions. Typical results are given in Table 4.

Table 4. Comparison of Snow Melting Load Intensities with Reported Installations (back and edge losses are excluded)

City	q_y [Btu/h ft²] Class 1			Class 2			Class 3		
	1	2	3	1	2	3	1	2	3
Minneapolis,MN	81-145	**120**	95	116-145	**120**	155	104	**184**	254
Detroit,MI	77-116	**89**	69	116	**89**	140	116	**120**	255
Burlington,VT	97	**111**	90	97	**111**	142	145	**140**	244
Spokane,WA	58-77	**97**	87	58-87	**97**	127	-	**112**	189
Harford,CT	58	**87**	115	97	**140**	254	135	**208**	260
Salina,KS	68	**91**	85	95	**91**	120	-	**126**	228
St.Louis,MO	77-116	**89**	122	77-116	**117**	152	116	**177**	198
Pittsburgh,PA	58-77	**91**	89	58-87	**94**	157	91	**144**	275
Boston,MA	77-87	**93**	107	97-116	**102**	231	116	**160**	255

Legend: 1 = Installed; **2** = Calculated; 3 = ASHRAE

Calculated results have a closer agreement with the reported data. However they are somehow higher for class 1 and some of class 2 systems. This may be due to the fact that after snow loads which replace class 1 and some class 2 snow melting loads reflect the worst case scenario for the atmospheric conditions. For class 3 systems, a similar resemblance is observed. On the other hand it seems that ASHRAE guidelines substantially overestimate the loads for class 2 and class 3 systems.

t may be concluded that ASHRAE guidelines overestimate the loads due o three main reasons,namely the use of empirical equations which verestimate the loads,absence of wind speed and terrain adjustment, nd the way that snow fall frequency data is interpreted.

n addition, the guidelines do not recognize any after snow ondition,and therefore underestimate the system load for some class and even class 2 systems.

. NOMENCLATURE

. Wind speed adjustment factor w.r.t. height from the ground, dimensionless

$_{o}$ Wind speed adjustment factor w.r.t. terrain, dimensionless

$_{r}$ Snow free area ratio, dimensionless

: Snow melting performance class, dimensionless

$_{fg}$ Heat of evaporation at melted snow film temperature, 1074 Btu/lb

ı Elevation from sea level, ft

I Height of the snow melting surface from ground level, ft

I_{ref} Wind speed recording height from ground level, ft

I Embedded hose (electric wire) spacing on centers, in

$_{a}$ Atmospheric pressure, in.of Hg

$_{av}$ Vapor pressure of moist air, in.of Hg

$_{s}$ Vapor pressure of saturated air at film temperature, in.of Hg

$_{c}$ Convection heat loss intensity during snow, Btu/h ft^2

$_{ca}$ Convection heat loss intensity after the snow, Btu/h ft^2

$_{e}$ Evaporation heat loss intensity during snow, Btu/h ft^2

$_{ea}$ Evaporation heat loss intensity after the snow, Btu/h ft^2

$_{h}$ Heat loss intensity by radiation and convection during snow, Btu/h ft^2

q_{ha} Heat loss intensity by radiation and convection after the snow, Btu/h ft^2

q_i Idling heat load intensity, Btu/h ft^2

q_m Intensity of heat of fusion, Btu/h ft^2

q_o Total heat load intensity during snow, Btu/h ft^2

q_{oa} Total heat load intensity after the snow, Btu/h ft^2

q_s Sensible heat load intensity, Btu/h ft^2

q_r Radiation heat loss intensity during snow, Btu/h ft^2

q_{ra} Radiation heat loss intensity after the snow, Btu/h ft^2

q_y Design heat load intensity, Btu/h ft^2

R Bowen's ratio, in.of Hg/°F

s Rate of snow fall, in.of H_2O eqv./h

s' Design rate of snow fall, in.of H_2O eqv./h

SF Maximum recorded snow fall in 24 hours, in. of snow

t_a Air temperature, °F

t_b Design air temperature at 97.5 % frequency level, °F

t_c Snow fall coincident design air temperature, °F

t_f Melted snow film temperature, °F

t_m Daily mean outdoor temperature during freezing period, °F

t_{max} Maximum surface temperature, °F

t_{min} Minimum surface temperature, °F

t_p Effective temperature on the surface, °F

U Adjusted wind speed over the snow melting surface, mph

U_{met} Wind speed from local meteorological data, mph

Greek symbols

Ω_s Density of snow, lb/ft^3

Θ Relative humidity, dimensionless

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DESIGN OF EMBEDDED SNOW MELTING SYSTEMS

PART 2. HEAT TRANSFER IN THE SLAB: A SIMPLIFIED MODEL

Dr.B.I.Kilkis*

TRACT

imple analytical technique was developed in order to predict the nsfer of heat in a snow melting slab. The primary objective was to elop a simple design tool while sufficient accuracy is retained. s analytical model is aimed to replace the rules given by ASHRAE delines which apply for a limited design range using metal pipes. model differentiates several performance classes of snow melting load intensities for periods including before or after the snow. ults of sample designs were compared with finite element utions. Comparisons indicated sufficient accuracy for engineering culations.

INTRODUCTION

h the advent of plastics and rubber technology,heat pipes,and trol systems,hydronic and electric snow melting on critical faces by embedded heating elements are becoming popular and more ective. In spite of these developments,a practical but prehensive design technique has not been developed yet although a ailed finite element analysis is already an option for engineers. researchers have attempted to develop a simple and accurate model ch is suitable for engineering calculations. However these efforts ally ignored the back and perimeter and edge losses and assumed a

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now free surface (class 3 system). In addition, they did not recognize heat loads which are associated with adverse atmospheric conditions following the snow. Generally, the design of snowmelting systems still rely on intuition or simple rules. Consequently an optimization search which is crucial for the success of the system can not be properly made. On the other hand, ASHRAE guidelines do not provide design rules for systems using plastic or rubber hydronic elements. In a typical design, the primary objective is to optimize the system with respect to the arrangement and temperature of the heating elements, slab construction and the insulation, for minimum installation and operation costs. Any technique must recognize and sufficiently model the relevant relationships among all design variables. The heated slab must perform at design conditions such that the desired snow melting performance is accomplished.

2. THEORY

Heat diffusion in a heated slab is basically a two dimensional problem, if the temperature changes in the axial direction are neglected. Heat is transferred to the slab from the heating elements. If the system is adequately sized, heat delivered to the surface will melt the snow at the prescribed performance rate (class 1,2, or 3). At the same time, some of the heat is lost to the atmosphere from snow free surfaces by convection, evaporation and radiation. Depending on the degree of insulation, some heat will also be lost from the perimeter and back of the heated slab. Although finite element algorithms are now capable of predicting the heat diffusion very accurately, designers generally favor simplified and analytical ods with reasonable accuracy. In this respect,Okagaki and a (1968) developed an analytical formulation using sformation techniques. They considered both steady and unsteady itions with or without perfect insulation. This technique permits alculate the heat output from the heated slab at a given iguration and the surface temperature of the heating element. ver the iterative nature of the algorithm and complexity of the ulations made it rather difficult to use. Schnurr and Rogers 70) developed an analytical algorithm to define the required ing element temperature in terms of the slab construction and ent spacing at steady state conditions. They were also able to ict the maximum surface temperature. Leal and Miller (1972) loped a finite difference solution for transient temperature ribution in the slab. They assumed a perfectly insulated slab and ow free surface. They analyzed both the transient heating and ing of a typical slab for a given heating element temperature of F and at the absence of snow fall. However their predicted ace temperature profile underestimates the extremes (t_{max} and ı),mainly because they used the surface heat loss equation by man and Katunich (1956) which was derived for snow melting litions at $t_p \approx 33°F$ where this is not consistent with their le solution with t_p about 90°F. Although above techniques are tively easy to implement on office computers or personal outers,they did not gain much popularity. This necessitated to elop a more simplified and easy to use algorithm for hand ulations. In this respect the composite fin model proposed by is (1992) was employed for a steady state heat diffusion.

This model recognizes all of the three snow melting classes which are mentioned in ASHRAE guidelines (1991). Figure 1 shows the ideal surface condition for each performance class.

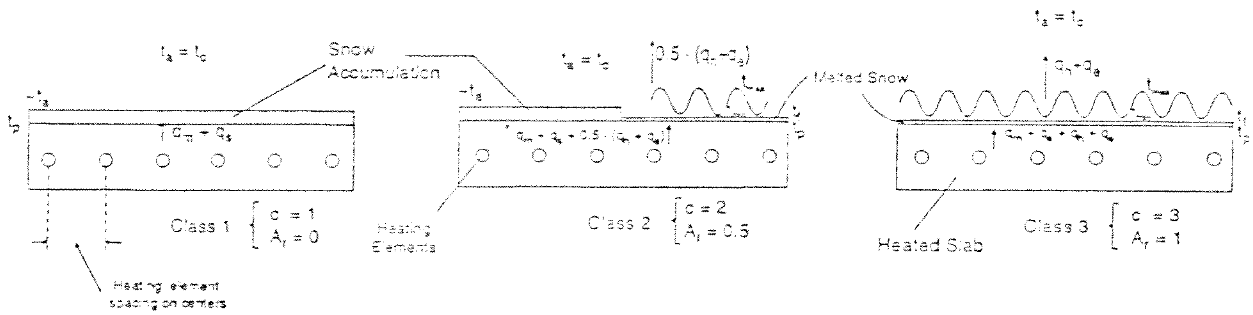
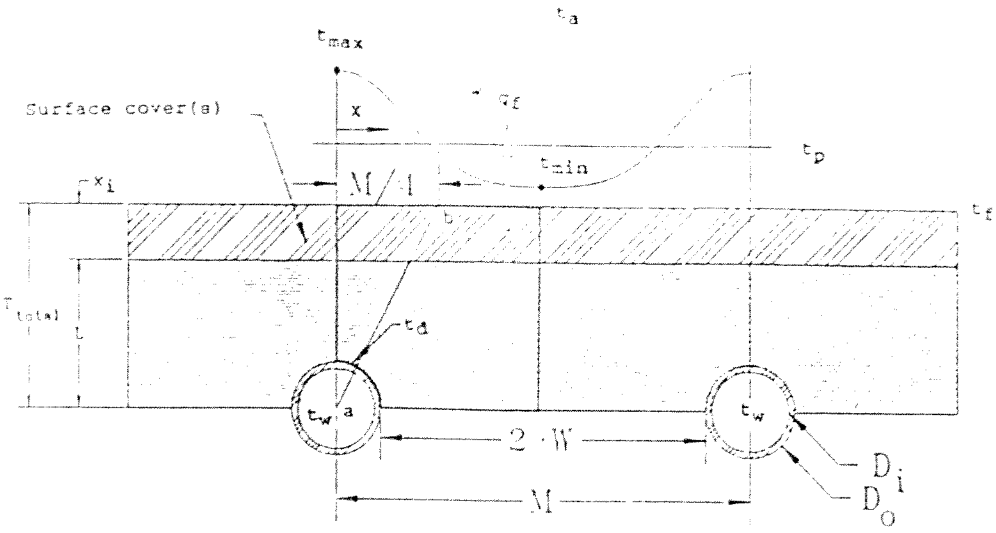


Figure 1. Surface Conditions for Different Performance Classes During Snow Fall.

In a class 1 system, entire surface is permitted to be covered with snow while it is snowing. This virtually eliminates heat losses from the surface to the atmosphere. In class 3, heat loss by convection, radiation and evaporation takes place from the surface. Class 2 is an intermediate case with a 50% snow free surface while it is snowing. Snow free area ratio (A_r) practically defines the performance class (See Fig.1). Figure 2 shows the composite fin model of the heated slab for steady state conditions. Here, t_f is the surface temperature of the water film, and heated surface temperature is t_p. Assuming a film thickness (x_i) like 1/16" (Adlam, 1950), and treating this as a separate surface layer, by its definition, t_p becomes t_f. However in practice this film may be neglected.



'igure 2. Composite Fin Model.

coefficient of the surface heat loss during the snow fall (h_f)
be a function of the performance class and meteorological
itions:

$$f = q_f/(t_p - t_a) = A_r \cdot (q_h + q_e)/(t_p - t_a) \qquad [1]$$

class 1 system ($A_r = 0$) surface heat loss,therefore h_f is
tically zero.
lass 1 and class 2 systems,the remaining snow needs to be melted
r the snow. Usually this phase is subject to more adverse
orological conditions. Therefore the surface heat losses may be
er. If this is the case,for design purposes:

$$h_f = (q_{ha} + q_{ea})/(t_p - t_a) \qquad [2]$$

he idling load during the freezing period is the critical
,then:

$$h_f = q_i/(t_m - t_a) \qquad [3]$$

s the air temperature. During idling and after the snow,it is
l to the design outdoor temperature (t_b).

r the snow fall conditions, t_d is replaced by the snow coincident
nperature (t_c) :

$$t_c = t_b + \frac{(33 - t_b)}{(0.1 + 1.2 \cdot C)} \qquad [4]$$

ficiency of the fin will be:

$$\cap = \tanh(m \cdot W)/(m \cdot W) \qquad [5]$$

ere;

$$m = 3/4 \cdot [h_f/(k_e \cdot T_{total})]^{1/2} \qquad [6]$$

e factor 3/4 is the two dimensional heat diffusion correction for e dimensional slab model, which is correlated from finite element ution comparisons.

class 1, the fin efficiency will be unity during the snow because is zero. Here k_e is the equivalent thermal conductivity for eral heat diffusion:

$$k_e = \frac{\sum_{i=1}^{n_k} (k_i \cdot x_i) + k \cdot L}{T_{total}} \qquad [7]$$

corresponding maximum surface temperature is solved from the lowing equation:

$$. W . \cap + D_o) . h_f .(t_{max} - t_a) = q_f . M \qquad [8]$$

refore,

$$t_{max} = t_a + \frac{q_f/h_f \cdot M}{(2 \cdot W \cdot \cap + D_o)} \qquad [9]$$

:e :

$$q_f/h_f = t_p - t_a \qquad [10]$$

ation 9 overestimates the temperature by about 5% when compared
ı finite element solutions.

ecially when the spacing between heating elements is wide, the
.mum surface temperature (t_{min}) at midway between adjacent
:ing elements, may drop below 32°F causing patches of ice remaining
:he surface. In order to avoid this condition, t_{min} must be
:ked with the condition $t_{min} \geq 32°F$:

$$t_{min} = (\cosh[m \cdot W])^{-1} \cdot (t_{max} - t_a) + t_a \qquad [11]$$

order to satisfy this condition, t_{max} must be adjusted. Noting
: (m) stays relatively constant with t_f (see sample design)
s can be directly accomplished by using Equation 11 for
ı $\geq$ 32°F. Following this operation, t_p may be adjusted by using
ations 9 and 10:

$$t_p = t_a + \frac{(t_{max} - t_a) \cdot (2 \cdot W \cdot \cap + D_o)}{M} \qquad [12]$$

RAE guidelines assume that t_p is 33°F and uniform on the surface.
s assumption is not validated with the above statements.

er determining the surface temperatures, the required surface
perature of the heating element can be calculated. A simple
roach is to use the vertical path between the element and the
face:

$$t_d \approx t_{max} + q_y \cdot [R_{co} + (L - D_o/2)/k] \qquad [13]$$

Krinninger (1989) and Leal and Miller (1972) assumed that surface temperature profile is Sinusoidal. In this case the local surface temperature at $x = M/4$ coincides with t_p. Following this argument, one may use the temperature gradient along path a-b (see Fig.1) in order to calculate t_b.

The total design heat load intensity to be delivered by the heating elements to the surface is the highest of the following load intensities which correspond to different phases of snow melting operation:

$$q_y = q_m + q_s + A_r \cdot (q_h + q_e) \quad \{ \text{during snow} \}$$
$$q_y = q_{ha} + q_{ea} \quad \{ \text{after snow} \} \qquad [14]$$
$$q_y = q_i \quad \{ \text{during idling} \}$$

For a detailed review of these equations, one needs to refer to the first part of this study. It is not quite uncommon that the critical surface temperature profile and the critical load intensity may take place at different phases of operation. This means that each applicable phase for a given class should be considered in selecting the critical design variables for sizing the heat (power) source and the element surface temperature.

For a hydronic system, the required mean water temperature is:

$$T_w = \frac{q_y \cdot M}{X \cdot \pi} \cdot \left[\frac{1}{\alpha \cdot D_i} + \frac{1}{2 \cdot k_h} \cdot \ln(D_o/D_i) \right] + T_d \qquad [15]$$

Here α is the convection coefficient in the hose at the given fluid properties and flow rate. X is the thermal efficiency of the heated slab. It is a measure of the back and perimeter heat losses. These losses were overlooked in the literature.

authors simply assumed it 40% of the heat delivered to the ted surface. Back heat loss intensity (q_b) can be approximated Krinninger's approach for the heat flow, namely by defining a aight path originating from the center of the heating element and ing in the ground at about 1 feet deep at an offset of (M/4) n the element center. Thermal resistances along this path are ed. The corresponding temperature gradient is the difference ween t_d and the ground temperature at 1 feet below the surface g). Comparisons with finite element solutions for typical cases icate an expected accuracy of 10% or better in terms of q_b. This nslates to about 3% accuracy in terms of thermal efficiency. The imeter heat loss intensity (h_a) can be approximated by following guidelines given by ASHRAE for floors on grade (1992). Then:

$$X = \frac{q_y \cdot \sum_{i=1}^{n_p} A_p}{\left[(q_y + q_b) \cdot \sum_{i=1}^{n_p} A_p \right] + Q_p} \qquad [16]$$

re;

$$Q_p = h_a \cdot l_e \cdot \left((t_d + t_p)/2 - t_a \right) \qquad [17]$$

latter equation assumes that the bulk of the slab is at a mean perature of (t_d + t_p)/2. h_a may be obtained from Table 7.1, in IRAE Load Calculation Manual, Chap.7 (1992). This is for a snow e ground surface on the perimeter. In a snow melting design, this face will be covered with snow which will reduce the perimeter loss. Therefore Equation 17 will render a conservative value for the thermal efficiency (see the numerical solution). It must also be noted that thermal efficiency will not stay constant during different phases of snow melting operation because of the changes in heat load intensities and the air temperature.

If the space at below is exposed to the atmosphere (Williamson,1967):

$$q_b = (1.4 + 0.13 \cdot U') \cdot (t_s - t_a) \qquad [18]$$

In a hydronic system,temperature of the heating element is important because it is primarily limited by the properties of the materials that it is made of. A typical limit on the mean fluid temperature is 170°F or less for a non-metallic pipe material. The temperature of the heating element is less significant in an electrical system which is used only for determining the electrical resistance of the cable at the operating temperature. However the temperature of the element may become more significant for any system if the temperature gradients in the slab are high enough that thermal stresses become critical. Shirakawa and et.al. (1985) have numerically investigated the principal stress distribution in a reinforced slab with 1.5% steel fiber content. They calculated that when the temperature difference between the slab surface and the heating element surface is 35°F,the tensile stress reaches a typical value of 420 psi. They have also concluded that the maximum tensile stress takes place in a direction perpendicular to the axis of the heating element. Using their data,one may correlate the tensile stress with the boundary temperatures along this axis:

$$\sigma_t \approx 12 \cdot (t_{max} - t_d) \qquad [19]$$

tion 19 may be used to determine the safe allowable temperature ient if the flexural cracking strength of the slab is known:

$$(t_{max} - t_d) \leq \sigma_c /(n \cdot 12) \qquad [19]$$

s suggested to apply a safety factor (n) not less than two.

AMPLE DESIGN

ass 3 hydronic snowmelting system is to be designed. Dimensions he snow melting surface are 25 ft by 40 ft. Six identical onic circuits serve the system. 3/4 inch I.D. rubber hose is used . an 11" spacing on centers. There is no insulation except the meter. Constructional details are given in Figure 3.

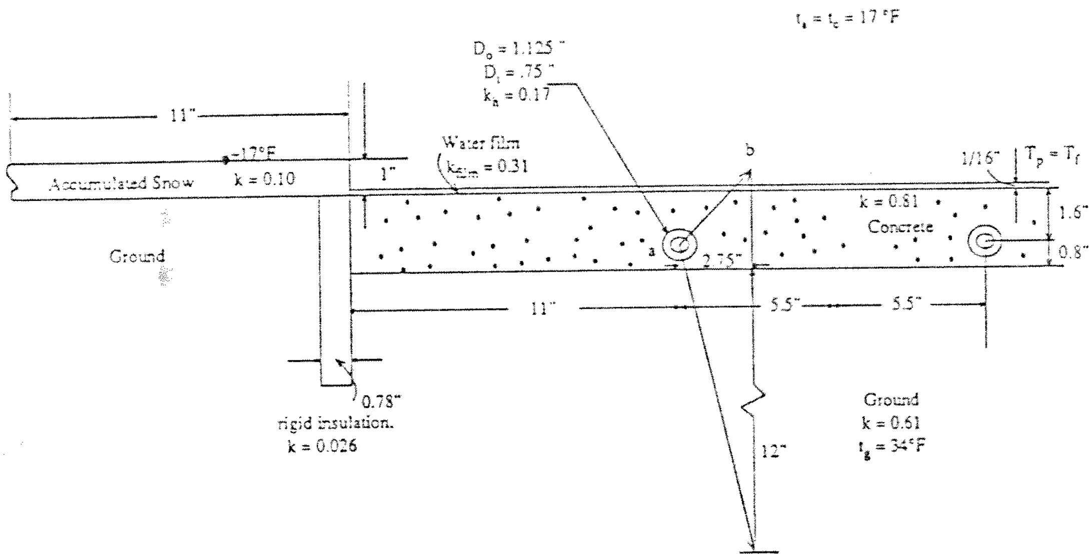


Figure 3. Constructional Details of the Sample Design.

er design inputs are:

Formance Class : 3 ; (A_r : 1)

location : Open field/ground level surface.

ign Wind speed : 10 mph ; Adjusted wind speed : 8 mph

Heating degree-days : 7000 d-d

Rate of snow fall : 0.16 in. of H_2O/h

n_p : 3 ; Ap : (25 · 40)/3 ≈ 333 ft²

l_e : 130 ft. ; h_a : 0.90 Btu/h ft °F

t_b : 10.4°F (-12°C) ; t_m : 20°F ;

t_g : 34°F; (thermal cond. of earth : 0.6 Btu/h ft°F

d_i : (3/4)/12 ft ; d_o : (1.18)/12 ft

k_h : 0.17 Btu/h ft°F ; k : 0.81 Btu/h ft°F ; σ_c : 900 psi

M : 11/12 ft

Calculated inputs for the design are:

R_{co} = 1/16"/(12 · th.cond.of melted snow) = 0.016 h ft² °F/Btu

Step.1- First,assuming that t_p is equal to 33°F, snow melting load intensities were calculated (see the first part of this study):

q_m = 6.2 Btu/h ft²

q_s = 119.4 Btu/h ft²

q_e = 17.9 Btu/h ft² ; q_{ea} = 0

q_h = 42.0 Btu/h ft² ; q_{ha} = 0

q_y = 185.5 Btu/h ft² (Eqn.14)

q_i = 71 Btu/h ft²

Calculations:

t_c = 17°F (Eqn.4)

t_a = t_c (during snow fall)

h_f = (17.9 + 42)/(33 - 17) = 3.74 Btu/h ft² °F (Eqn.1)

k_e = 0.79 Btu/h ft°F (Eqn. 7)

W = 4.91/12 ft

m = 4.38 ft^{-1} (Eqn.6)

= 0.41 (Eqn.5)

= 53.8°F (Eqn.9)

= 28.9 °F < 32°F (Eqn.11)

rning to Step.1 with a second trial value for t_p = 40°F ;

= 6.2 Btu/h ft² (not effected)

= 119.4 Btu/h ft² (not effected)

= 27.7 Btu/h ft²

= 57.9 Btu/h ft²

= 211.2 Btu/h ft²

= 85.6 Btu/h ft² ; hf : 85.6/(40 - 17) = 3.72 Btu/h ft² °F

= 4.37 ft^{-1} (Eqn.6)

= 0.52

= 63°F (Eqn.11)

= 32°F

= 43°F (Eqn.12)

ace temperature is close enough to the assumed value for design oses. This operation shows that by assuming t_p = 33°F, snow ing load would be underestimated by about 12% and there would be patches (unmelted snow) on the surface. It must be noted that surface temperature variation could be minimized by reducing the ing of elements,M (see Eqns. 5 and 9).

;

88°F

same temperature is calculated to be 96°F by using Krinninger's oach (1989). Checking for thermal stress;

3 (Eqn.20)

31.6 Btu/h ft²

q_p = 5674 Btu/h (Eqn.17)

X = 0.85 (Eqn.16)

α = 400 Btu/h ft²°F (50 % P.Glycol/water mixture)

T_W = 187°F

Fluid temperature which seems to be high may be reduced by decreasing the spacing of heating elements.

5. NUMERICAL SOLUTION

The above example was modeled by a finite element analysis routine, namely ANSYS Engineering Package version 4.4A. The boundary conditions and the model is shown in Figure 4. The sides of the snow melting surface is assumed to be covered with snow with a surface temperature equal to t_c. This surface is permitted to exchange radiation heat with sky. The surface temperature variation is shown in Figure 5. The temperature distribution in the heated slab is shown in Figure 6.

Main results of this analysis were:

T_{max} : 59°F ; T_{min} : 34°F ; X : 0.82 ; T_W : 182°F

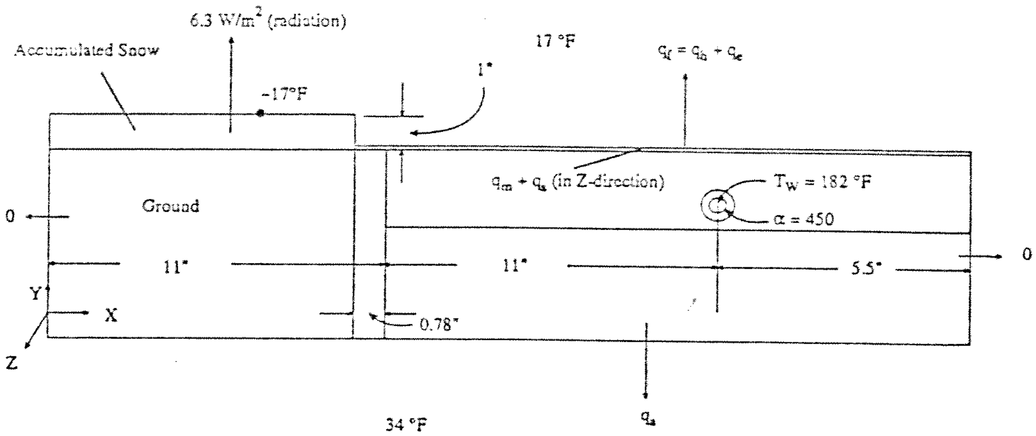


Figure 4. Idealization of the Problem for Finite Element Analysis.

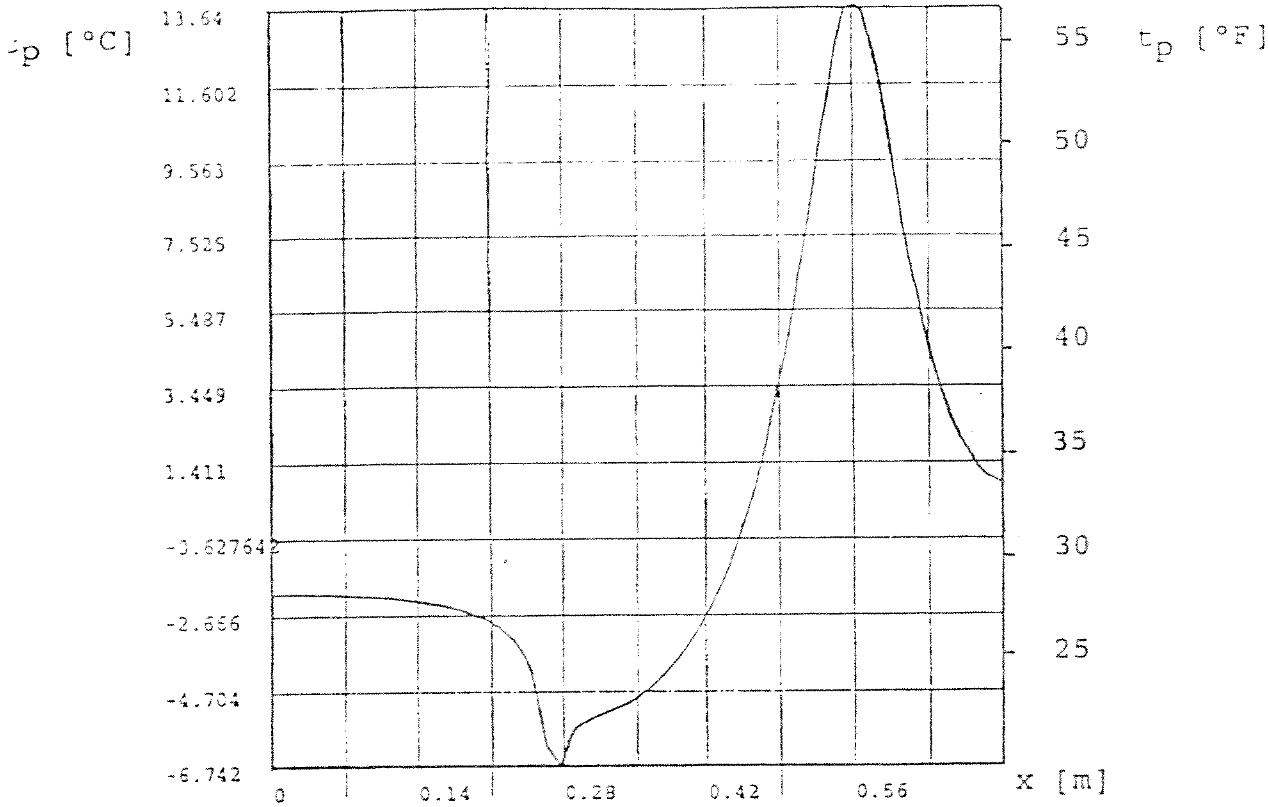


Figure 5. Temperature Distribution on the Snow Melting Surface.

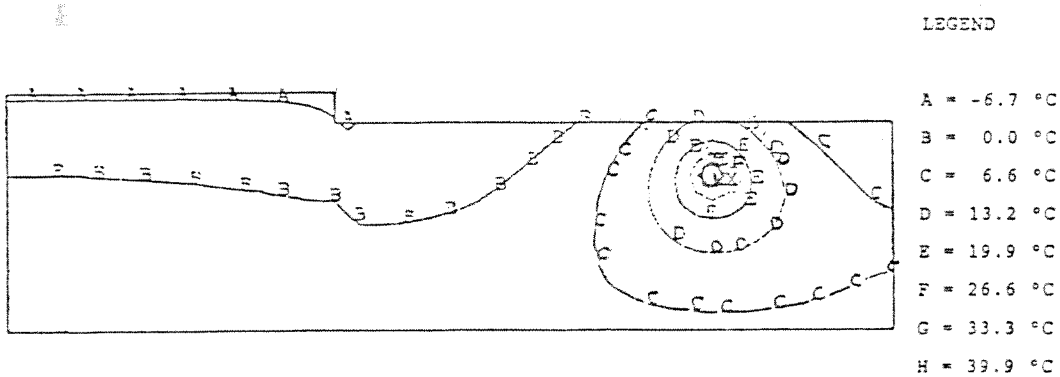


Figure 6. Temperature Distribution in the Heated Slab.

DISCUSSION of RESULTS

e above case study and others indicated that the proposed algorithm accurate enough for engineering calculations for a snow melting ab. The algorithm permits to adjust for different layers at top of e slab and important parameters like spacing of the elements,hose mensions and properties,slab construction,insulation level can be changed in order to optimize the system. Furthermore it allows to predict the surface temperature variation and ensures that entire surface will be maintained above the freezing temperature.

7. NOMENCLATURE

A_p Surface area covered by a section of the heated slab (panel) which is served by a single circuit of the heating element, ft^2

A_r Snow free area ratio, dimensionless

C Snow melting performance class (1,2,3), dimensionless

D_i Inside diameter of the heat transfer hose, ft

D_o Outside diameter of the heat transfer hose (electric wire), ft

h_a Linear heat loss coefficient for the perimeter, Btu/h ft °F

h_f Coefficient of surface heat loss intensity, Btu/h ft^2 °F

k Thermal conductivity of the slab material, Btu/h ft °F

k_e Equivalent thermal conductivity of the composite slab for lateral heat diffusion, Btu/h ft °F

k_h Thermal conductivity of the hose material, Btu/h ft °F

k_i Thermal conductivity of each layer on the slab, Btu/h ft °F

l_e Perimeter of the snow melting slab, ft

M Heating element spacing on centers, ft

m Fin coefficient, ft^{-1}

n Mechanical safety factor against cracking of a concrete slab, dimensionless

n_k Number of layers at top of the heated slab, dimensionless

n_p Number of circuits (panels) in the snow melting system, dimensionless

Back heat loss intensity from the heated slab, Btu/h ft^2

Evaporation heat loss intensity during snow, Btu/h ft^2

Evaporation heat loss intensity after the snow, Btu/h ft^2

Total surface heat loss intensity, Btu/h ft^2

Surface heat loss intensity by radiation and convection during snow, Btu/h ft^2

Surface heat loss intensity by radiation and convection after the snow, Btu/h ft^2

Idling heat load intensity, Btu/h ft^2

Intensity of heat of fusion, Btu/h ft^2

Heat loss from the perimeter, Btu/h

Radiation heat loss intensity during snow, Btu/h ft^2

Intensity of sensible heat, Btu/h ft^2

Radiation heat loss intensity after the snow, Btu/h ft^2

Design heat load intensity, Btu/h ft^2

Total thermal resistance of layers above the slab, h ft^2 °F/Btu

Air temperature, °F

Outdoor design temperature, °F

Snow fall coincident design air temperature, °F

Outside surface temperature of the heating element, °F

Melted snow film temperature, °F

Ground temperature, °F

Daily mean air temperature during freezing period, °F

Maximum surface temperature, °F

Minimum surface temperature, °F

Effective temperature on the surface, °F

t_s Surface temperature of the back of a heated slab if it is exposed to outdoor air, °F

T_{total} Thickness of the composite fin, ft

T_w Mean fluid temperature in the hydronic circuit, °F

U Adjusted wind speed, mph

W Half of the net spacing between adjacent heating elements $(M-D_o)/2$, ft

X Thermal efficiency of the heated slab, dimensionless

x_i Thickness of any layer on top of the heated slab, ft

Greek symbols

α Convection heat transfer coefficient between the fluid and the hose, Btu/h ft² °F

η Fin efficiency of the composite fin, dimensionless

σ_c Flexural cracking strength of the slab material, psi

σ_t Tensile stress in the slab, psi

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)IANT CEILING COOLING WITH .AR ENERGY: FUNDAMENTALS, DELING, AND A CASE DESIGN

is, Ph.D.
ASHRAE

CT

iim of this paper is to provide the fundamentals of sign algorithm for radiant panel cooling systems ial emphasis given to solar energy utilization and strate its feasibility with a case design. A steady- t transfer model was developed to determine the usion in a cooled ceiling slab and to predict its e performance in the conditioned space. More ie-in features of the radiant ceiling cooling system rption-type heat pumps and solar collectors were ' and explained in terms of the first and second ermodynamics. This analysis indicated an increase heating and cooling coefficients of performance f the absorption heat pump and a substantial de- the total exergy loss in addition to an increase in plate solar collector efficiency. The case design an experimental solar house with 81 m^2 net living nsible cooling and heating are designed to be shed by radiant ceiling and floor panels, respec- or cooling purposes, a 3.5-kW water-ammonia on heat pump and flat-plate collectors with 30 m^2 orber area are required. The same system will also domestic hot water. The design analysis indicated radiant ceiling cooling, the peak cooling load will d by about 25%.

)UCTION

ce conditioning of residences and commercial s uses about one-third of the primary energy d in the United States. Efficient utilization of ve energy resources including waste heat, with or the aid of heat pumps, is becoming the most nt issue in energy conservation and protection of the nent.

ive solar energy systems for space conditioning have been challenging alternatives for designers and makers. The fact that the heating and peak solar on seasons do not coincide often necessitates season- storage for space-heating purposes. On the other e need for an absorption cycle for solar cooling oefficient of performance is substantially lower than specially if used with conventional cooling units, l serious design and operational constraints on active stems.

Panel heating and cooling systems that rely on very moderate fluid temperatures establish a much better tie-in with active solar energy systems. Recent developments in plastics and rubber technology for heat transfer pipes also increased the reliability and life-cycle cost-effectiveness of such alternative conditioning systems in the last decade. The same is also true for heat pump technology. Electric utility companies are investing heavily in research and development on heat pumps and alternative space-conditioning systems with the primary objective of eliminating the need for a supplementary heating(cooling) source, leveling off the peak loads, and shifting the peak demand periods.

Although important advances have been made in this field, there are still unresolved problems, such as the frequent necessity for a supplementary cooling unit (Kilkis 1992; Pietsch 1991) when heat pumps are coupled to conventional conditioning systems. Combined with their other features, radiant panel systems seem to be the key element in resolving such techno-economic problems. Their main features in space conditioning (Kilkis 1991) are

1. *Decreasing the space-conditioning load.* Indoor air temperatures can be selected 2° to 3°C higher than the standard indoor design temperatures without any sacrifice of the desired human comfort (Kilkis 1990). This is possible because of two factors, namely, the establishment of a lower average indoor surface temperature and slow air movement in a radiant ceiling cooled space. The former enhances the heat loss from the human body by radiation, which is necessary for thermal comfort. The latter reduces the sensible heat gain of the human body through convection and, hence, dependence on the indoor air temperature for thermal comfort. The ability to maintain a higher indoor air temperature reduces the heat gains from transmission and infiltration. In addition to this, transmission-type heat gains through the exposed partitions, especially with low thermal resistance, decrease because of the increased surface film resistance as a result of the slower air movement wetting these surfaces. Slow indoor air movement also decreases the infiltration-type heat gains.
2. *Shaving off the peak loads by thermal (cool) storage.* Partitions that are partially cooled with radiant heat exchange with the ceiling cooling panel, as well as the

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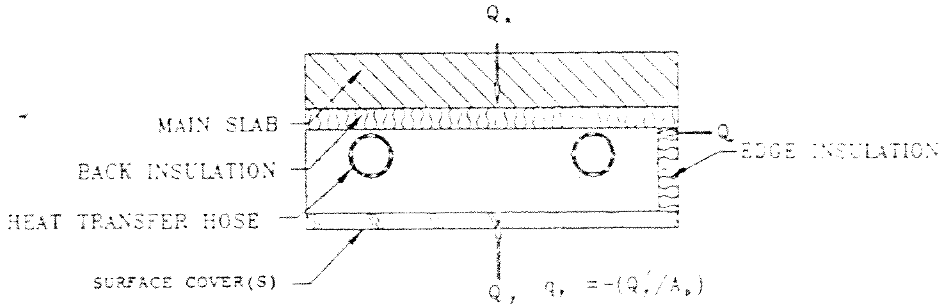


ure 1 *A typical ceiling cooling slab.*

panel itself, store cool and thus the daily swings of the cooling load are moderated.

Requiring moderate fluid temperature. Ceiling panels often require a very moderate chilled fluid (water or brine) temperature compared to conventional cooling systems. As a prime feature of the radiant system, the mean fluid design temperature hardly gets below 12°C due to large radiant surfaces and reduced cooling loads. With proper design and selection of a close hose spacing, radiant ceiling cooling systems can be used even without heat pumps, e.g., simply relying on seasonal cold storage in aquifers (Snijders 1992). A smaller fluid temperature drop across the panel circuit can also be chosen without any serious handicap, which further increases the required fluid supply temperature. As will be explained below, these features substantially improve the performance of an absorption cycle and solar collectors.

Although they are generally limited in scope, there are isting theoretical and experimental studies on radiant nel heating systems either from the ceiling, floor (Zhang d Pate 1986, 1987; Min etal. 1986), or snow melting)kagaki nd Okada 1968). Unfortunately less attention has en pai so far to radiant panel cooling systems, especially conjunction with active utilization of solar energy. lthough radiant panel cooling is limited to sensible oling, with the above-mentioned merits and potential lvantages, it is a viable alternative cooling system, as cently reported by many authors (Wilkins and Kosonen)92; Kilkis 1991).

HEORY

eat Extraction by a Radiant Ceiling Slab

A typical in-slab type of radiant ceiling cooling panel)nstruction is shown in Figure 1. The slab is maintained a colder temperature than the ambient by circulating illed fluid through the embedded pipes. Hoses manufacred from rubber compounds of special composition, fferent types of thermoplastic pipes, or metal pipes can be sed. In order to minimize heat gains from the back and perimeter of the cooled slab, its back and edges are insulated. The heat gain from the conditioned space establishes its sensible space-cooling effect. Designating this as the negative of the heat gained by the slab, the required cooling load intensity of the space to be conditioned is

$$q_y = -Q_y'/A_p \cdot 1000. \tag{1}$$

A radiant ceiling panel must be designed to meet or exceed this magnitude. In calculating Q_y', all the attributes of radiant ceiling cooling must be taken into account. This reduced peak design load may be further shaved off by the cold stored in the radiant slab and partly in enclosing walls, provided that the system is not cycled between long periods of time. In this case, Q_y' may be reduced by the peak-load shaving factor, C:

$$Q_y = C \cdot Q_y'. \tag{2}$$

This depends on many factors, such as the climate, typical daily outdoor temperature swings, and the cold storage capability of the building elements, in addition to the thermal properties and dimensions of the cooled slab. By considering five climatic conditions, Kilkis (1990) formulated an engineering rule for determining C, reproduced in Table 1. A detailed analysis is a must in order to customize C for a given building and local climate. Therefore, caution must be exercised in manipulating this table for specific designs. Defining the cooling efficiency of the panel, X, as

$$X = \frac{Q_y'}{Q_y' + (Q_a + Q_e)}$$

TABLE 1
Peak Load Shaving Factor for Radiant Ceiling Coolings

Climate/Location	C
Arid	0.70
Semi-arid	0.75
Moderate	0.80
Mountain	0.75
Seaside	0.85

or

$$X = \frac{Q_y}{Q_y + (Q_a + Q_e)}, \tag{3}$$

required capacity of the cooling unit at design conditions will be

$$P = Q_y^{\prime}/X \quad \text{or} \quad P = Q_y/X. \tag{4}$$

To meet this load, the absorption heat pump may be sized according to one of the following criteria:
without supplementary unit,

$$Q_{ev}(T_b) \geq P(T_b), \tag{5}$$

with supplementary unit,

$$Q_{ev}(T_b) + Q_D = P(T_b). \tag{6}$$

Here Q_D is the capacity required for the supplementary cooling unit. The latter equation represents a hybrid cooling system where a radiant ceiling cooling panel is complemented by a small air-handling unit (Wilkins and Kosonen 1992). This unit takes care of the necessary fresh air supply and the latent cooling load. The intensity of the sensible cooling effect by the radiant ceiling panel depends primarily on the hose spacing, mean fluid temperature, size of the indoor space, altitude, and the indoor design air temperature. For an ideal condition, where the ceiling surface temperature $T_p'(x)$ is assumed to be uniform and equal to T_p, the amount of total heat that can be extracted by the panel surface of unit area is the sum of radiation and convection terms (Kilkis 1990):

$$\begin{aligned} q_y = q_r + q_c &= U_r \cdot (T_p - AUST) \\ &+ U_c \cdot (T_p - T_a^{\prime}) \\ &\{q_y \leq 0 \text{ in cooling}\}, \end{aligned} \tag{7a}$$

where

$$U_r = r \cdot F_r \cdot \sigma, \tag{7b}$$

$$\begin{aligned} U_c = (1 - 2.22 \times 10^{-5} h)^{2.627} \cdot (4.96/D_e)^{0.08} \\ \cdot 2.67 \cdot |(T_p - T_a^{\prime})|^{0.25}. \end{aligned} \tag{7c}$$

r is the linearization factor for the radiant heat transfer:

$$r = 4 \cdot ([T_p + 273]/2 + [AUST + 273]/2)^3. \tag{8}$$

It may be further simplified by a linear function of T_p in the practical range of radiant ceiling cooling:

$$\begin{aligned} r \cong (0.9659 + (T_p - 10) \cdot 0.00495) \cdot 10^{-8} \\ \{10°C \leq (T_p + AUST)/2 \leq 18°C\}. \end{aligned} \tag{9}$$

F_r is the simplified radiation interchange factor for $A_p/A_u \leq 0.30$ (Raiss and Roedler 1969):

$$F_r \cong \epsilon \qquad \{A_p/A_u \leq 0.30\}. \tag{10}$$

Equation 7c gives the convective term with altitude (h) and room size (D_e) corrections. There is not a direct correlation for the effect of the room size on convective cooling. However, similarity between floor heating and ceiling cooling enables one to employ the Min et al. (1956) data for floor heating. D_e is the equivalent diameter of the room:

$$D_e = (4 \cdot A_r/L_r). \tag{11}$$

The area-weighted average design temperature of the uncooled indoor surfaces, $AUST$, is rather difficult to calculate. It primarily depends on T_b, the ratio of indoor and outdoor partitions' surface area, thermal properties and dimensions of the partitions, position of the room in the building, solar exposure, and the outdoor design wind speed. An elaborate numerical algorithm was provided by Zmeureanu et al. (1987), which, in principle, is also applicable to radiant cooling. A simple approach was developed for the computer program described below:

$$AUST = T_a^{\prime} + c \cdot w. \tag{12}$$

Here, c is the room position code, i.e., c is 1 for an inner room, 2 for a room with one exposed wall, and 3 with two or more exposed walls; w is a function of the outdoor design temperature:

$$w \cong 7/(45 - T_b) \quad \{26°C \leq T_b \leq 36°C\}. \tag{13}$$

Once T_a' and $AUST$ are determined, T_p can be solved for a required cooling intensity. Equations 7a, 7b, and 7c presume that T_p is uniform, which can only be true for an infinite number of hoses. Therefore, further elaboration is essential before calculating the design mean chilled-fluid temperature for maintaining the required heat extraction (cooling) intensity. Depending upon the slab construction, required cooling intensity, indoor temperature, as well as the spacing between the heat transfer hoses, the ceiling surface temperature distribution has a non-uniform profile. Before calculating the heat transfer in the slab, this surface temperature profile needs to be predicted. Models for explaining and predicting this temperature profile are few. One of the earliest models was given by Kollmar and Liese (1957) for heating. Their model assumes that the slab operates as a fin and loses heat from its surface. Zhang and Pate (1986) developed a finite-difference algorithm for ceiling heating panels in order to solve the steady and transient heat diffusion problem in the slab and predict the local surface temperatures. Leal and Miller (1972) stated that the temperature profile is simply a sine curve.

Steady-State Composite Fin Model

A radiant panel cooling system probably will operate in an unsteady manner if a simple control is required. However, if a peak load shaving feature is desired, the system must have a continuous fluid (brine) circulation with temperature modulation. Another important requirement emerges if a solar absorption heat pump is to be used: as the heat exchangers in absorption systems have long time constants, cycling reduces the efficiency. There is a third reason for temperature modulation in solar systems, since it allows one to energize the generator at much lower temperatures, and this, in turn, increases the solar collector efficiency and the heat pump COP (Kreider and Kreith 1981). These facts and the simplicity of the solution favor a steady-state solution for design purposes. The steady-state composite fin model is shown in Figure 2 (Kilkis 1992). Part of the slab that remains below the centerline of the hose and the ceiling-covering element(s) below it establish the composite fin. Due to lateral symmetry, the characteristic length of the fin is $M/2$. Due to the same symmetry, there is no lateral heat flow across the mid-plane between adjacent hoses. Edges are assumed to be totally insulated. Calculations do not need to consider back heat gains at this stage because they are accounted for separately through the definition of panel cooling efficiency, which effects the required capacity of the cooling plant (Equation 4) and the mean chilled-fluid temperature (Equation 22). Hose surfaces that are in contact with the fin are assumed to be isothermal (T_d). The composite fin extracts heat from the space both by radiation and convection according to Equation 7a. Defining the radiant and convective fin efficiencies,

$$\eta_r = \eta_c = \eta \qquad (14)$$

$$\eta = \frac{\tanh(m \cdot W)}{(m \cdot W)}, \qquad (15)$$

where

$$m = \sqrt{\frac{U}{k_e \cdot T_{total}}}, \qquad (16a)$$

and

$$U = U_r + U_c. \qquad (16b)$$

Here k_e is the equivalent thermal conductivity for lateral heat diffusion in the composite slab:

$$k_e = \frac{\sum_{i=1}^{n_k} x_i \cdot k_i + L \cdot k}{T_{total}}. \qquad (17)$$

Superimposing the radiant and convective heat extraction intensities and solving for T_{min},

$$T_{min} = \frac{q_y \cdot M + A \cdot AUST + B \cdot T_a'}{A + B} \qquad (18a)$$

where A and B are the radiant and convective heat extraction rates for the composite fin having a surface area occupied by two adjacent hoses of unit length each:

$$A = [2 \cdot W \cdot \eta_r + D_o] \cdot U_r, \qquad (18b)$$

$$B = [2 \cdot W \cdot \eta_c + D_o] \cdot U_c. \qquad (18c)$$

The floor surface temperature profile can be calculated by the following equation for $D_o/2 \le x \le M/2$:

$$T_p'(x) = \frac{\cosh[m \cdot (M/2 - x)]}{\cosh[m \cdot W]} \cdot [T_{min} - T_a' + c] + T_a' - c \qquad (19a)$$

for $x \le D_o/2$:

$$T_p'(x) \cong T_{min}. \qquad (19b)$$

Here, c represents the difference between the radiation and convection temperature gradients:

$$c = U_r/U \cdot [T_a' - AUST]. \qquad (20)$$

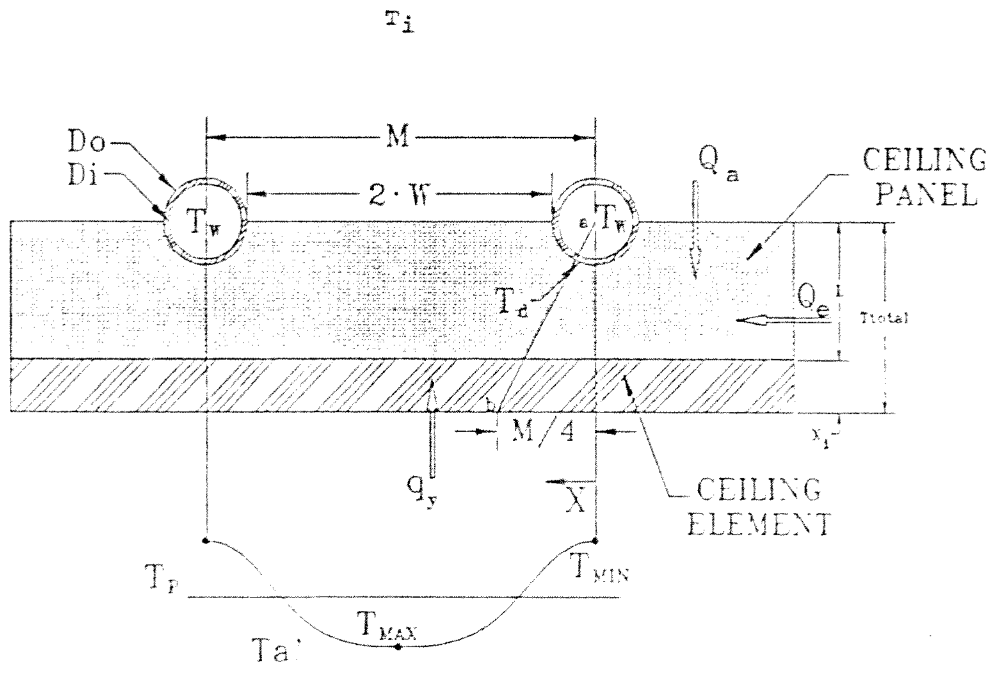


Figure 2 Composite ceiling fin model for cooling.

This piecewise temperature profile may be revised for $x \geq D_o/4$ by introducing a second-degree polynomial for $0 \leq x \leq D_o/4$ with the boundary conditions

$$T_p'(x) = T_{min} \text{ at } x = D_o/4$$

and

$$dT_p'(x)/dx = 0 \text{ at } x = 0.$$

Using T_{min} and taking the shortest path for heat diffusion,

$$T_d \cong T_{min} + q_y \cdot (R_{co} + (L - D_o/2)/k + R). \quad (21)$$

Here, R_{co} is the total thermal resistance of the ceiling elements. R is the thermal resistance between the hose surface and the slab material and may be neglected.

Finally, the required mean chilled-fluid temperature, T_w, will be

$$T_w = \frac{q_y \cdot M}{X \cdot \pi} \cdot \left[\frac{1}{\alpha \cdot D_i} + \frac{1}{2 \cdot k_h} \ln(D_o/D_i)\right] + T_d. \quad (22)$$

Usually, the hose Reynolds number in radiant ceiling cooling systems ranges between 6,000 and 10,000, thus the flow is marginally turbulent. Employing the turbulent flow model for water (Kreider and Kreith 1981):

$$\alpha = 1056 \cdot [0.02 \cdot \{T_w + 273\} - 4.06] \cdot v_w^{0.8}/D_i^{0.2}. \quad (23)$$

The required flow rate in the circuit is

$$V = \frac{Q_y'}{X \cdot \gamma \cdot C_p \cdot \Delta T_w}. \quad (24)$$

The oxygen permeation through rubber or plastic pipes does not exhibit any real corrosion potential, especially in radiant panel cooling systems, because the oxygen permeability of a rubber hose decreases almost linearly with the mean fluid temperature. Similarly, the oxygen permeability of a plastic pipe stabilizes at comparably low rates at chilled fluid temperatures. However, bacteriological infestation may be an important agent for metallic corrosion in the system. For this reason, cooling-dedicated panel circuits must be protected from bacteriological infestation by proper chemical agents available on the market.

Solar Cooling

There are several methods to achieve cooling by solar energy. The cooling effect can be produced directly by an absorption cycle (Wilbur and Mitchell 1975), a regenerative desiccant process (Lunde 1976; Roy and Gidaspow 1974), or a steam jet system. A conventional vapor-compression cycle may also be used in which the compressor is driven by a solar Rankine cycle or by an electric motor operated by photovoltaic cells. Absorption cooling systems essentially require only a heat source and have become one of the likely candidates for solar cooling. In the past, high initial investment was required for the large solar collector area and the heat pump itself. With the advent of the radiant ceiling concept, which increases collector efficiency and COP of the absorption heat pumps, this system seems to be economically feasible.

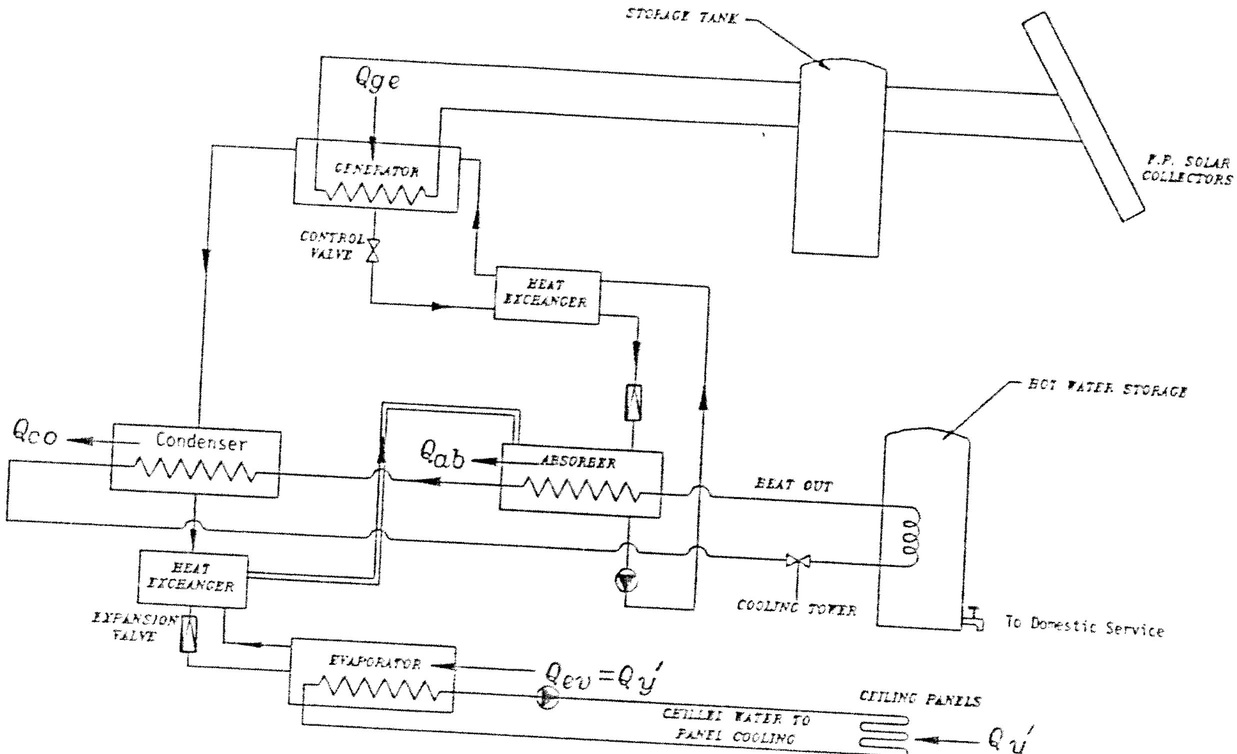
Absorption Heat Pumps

Absorption air-conditioning equipment has been in standard production for several decades. Residential water-ammonia and water-lithium bromide units have been used in gas-fired applications and have been modified for solar energy systems. These heat pumps have also been used in the energy recovery field for many years (Bjurestrom and Raldow 1981). In space-cooling applications, the presence of large quantities of ammonia in a building may violate fire and safety codes.

Water-ammonia systems also require relatively high solar collector water outlet temperatures (e.g., 115°C) for optimum operation (Shiran et al. 1982). A water-lithium bromide system is simpler than a water-ammonia system and operates at a higher cooling ratio and with smaller heat exchanger surfaces but may involve more design problems (Karakas et al. 1990). In addition, for the sole reason of minimizing the cost, the generator temperature needs to be relatively high (Alizadeh et al. 1979). However, this may impose serious limitations on the utilization of flat-plate collectors, although the relationship between the required flat-plate collector area and the generator temperature is reported to be weak (Alizadeh et al. 1979). At low levels of solar insolation, when the generator temperature is going to be too low, an auxiliary heater is required in order to increase the generator temperature. Due to the low regenerating temperatures required, silica gel-water absorption systems, as reported by Kluppel and Gurgel (1988), represent another alternative, especially for radiant ceiling cooling systems that already have established a suitable medium for low regeneration temperatures. In absorption heat pumps, the heating cycle is much better than the cooling cycle in terms of energy and exergy efficiencies (Cheng and Shih 1988), but their use in cooling is more popular because the use of waste heat or alternative energy has gained priority over the efficiency. A water-ammonia system is shown in Figure 3. An evaporator provides the necessary heat extraction through the radiant ceiling panel circuit. Flat-plate collectors provide the necessary heat for energizing the generator. A storage tank may also be used. Rejected heat from the condenser and the absorber heats the domestic water.

Performance Enhancement by Using Radiant Ceiling Cooling

In design and optimization of the performance of an absorption cooling system, the most important variable that



ure 3 *General arrangement of solar absorption cycle radiant cooling system.*

to be taken into account is the temperature of the rator because the other parameters of the system nd on the existing initial conditions and, consequently, fixed. On the other hand, the temperature of the rator directly affects the COP and ECOP figures of the m both for cooling and heating, as well as the flat-plate collector efficiency, when other design conditions are l. The required generator temperature, however, nds upon the temperature of the chilled fluid to be uced for cooling. For example, in a water-lithium ide system, producing a 5°C chilled-water temperature ad of 7.5°C would require an increase in the generator, heat source, temperature of about 5°C. Producing a chilled-fluid temperature would allow a decrease in at source temperature of approximately 5°C (Kreider reith 1981). This exhibits the key attribute of radiant g cooling: the system that requires very moderate d-fluid temperatures allows one to reduce the energiz- mperature at will. This reduction of energizing rature is very beneficial to the operation since COP, , and the solar collector efficiency will be higher:

$$\mathrm{COP}_h = \frac{Q_{ab} + Q_{co}}{Q_{ge} + W_p}, \tag{25}$$

ergetic coefficients of performance for heating and are

$$\mathrm{COP}_r = -\frac{Q_{ev}}{Q_{ge} + W_p}. \tag{26}$$

$$\mathrm{ECOP}_h = \frac{Q_{co}(1 - T_o/T_{co}) + Q_{ab}(1 - T_o/T_{ab})}{Q_{ge}(1 - T_o/T_g) + W_p}, \tag{27}$$

$$\mathrm{ECOP}_r = \frac{Q_{ev}(1 - T_o/T_{ev})}{Q_{ge}(1 - T_o/T_g) + W_p}. \tag{28}$$

Through the available literature, a typical relationship between ECOP_h and T_{ge} is shown in Figure 4 for a water-ammonia absorption heat pump (Ataer and Gogus 1991). The figure shows that there is an optimum generator temperature for maximum coefficient of performance. This temperature is common for COP and ECOP figures and decreases with the evaporator temperature, T_{ev}. By extrapolation and interpolation in Figure 4 on solid lines for T_{ev} = 10°C and T_{ev} = 0°C, it can be seen that an increase in the evaporator temperature from 7.5°C to 12.5°C decreases the optimum generator temperature from 80°C to about 67°C. This feature also eliminates the need for an auxiliary generator heater. This example also indicates that the maximum ECOP_h increases from 0.48 to about 0.55. This increase may be enhanced if the mean chilled-water temperature can be further increased to 15°C at the expense of reducing the spacing between the hoses. Figure 5 shows the variation of COP_r with T_{ev}. With T_{ab} = 30°C and T_{co} = 28°C in this figure, the optimum T_{ge} value drops from 75°C to 62°C for T_{ev} = 12.5°C. If absorber and condenser temperatures are higher, as listed in Figure 4, the reduced T_{ge} value will coincide with the 67°C value predicted from Figure 4. For the latter case, COP_r is about 0.70 and

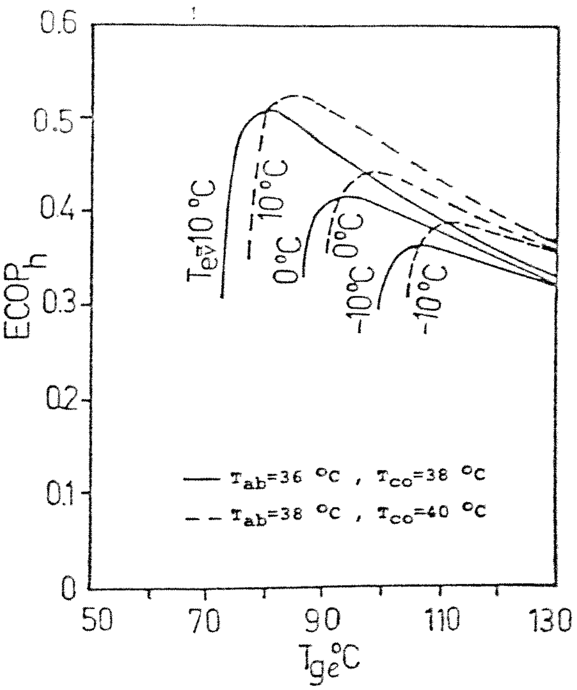


Figure 4 *Change of $ECOP_h$ with the generator and evaporator temperature.*

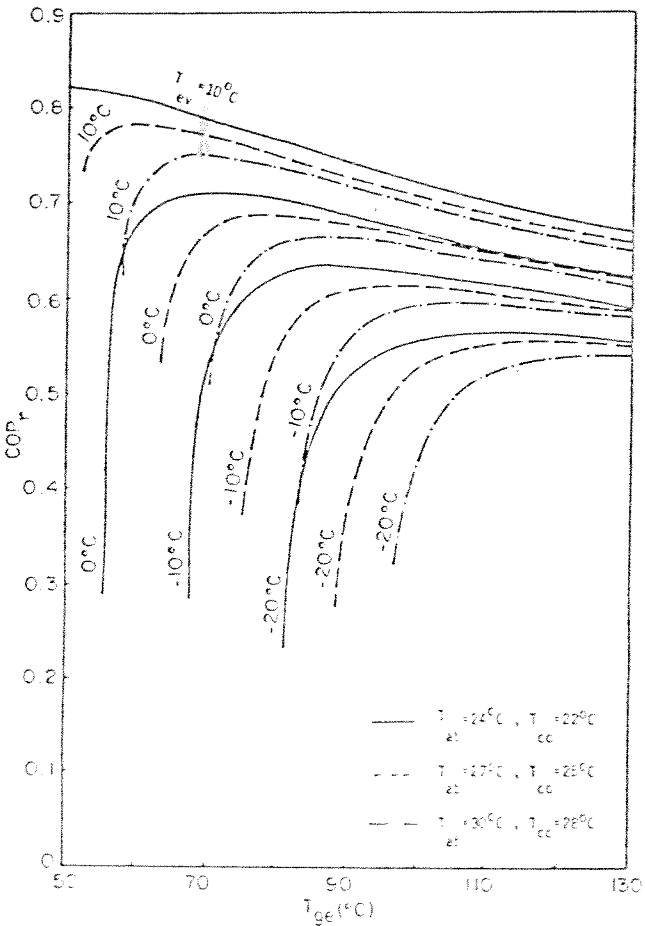


Figure 5 *Change of COP_r with the generator and evaporator temperature.*

indicates a noticeable increase of COP_r when compared with the T_{ev} = 7.5°C case ($COP_r \cong 0.65$). When $ECOP_r$ is defined as in Equation 28 and the reference temperature is 20°C, $ECOP_r$ decreases with T_e. However, the overall exergetic performance of the system, as measured by the amount of total exergy loss, improves as T_{ev} decreases. This relationship is shown in Figure 6 when the heat extracted from the condenser and absorber is not used (Ataer and Gogus 1990). In a solar absorption system, when absorber and condenser heat output is used for domestic water heating, as shown in Figure 3, the total exergy loss figures will be somewhat higher, but the general trend will be the same. This figure shows a substantial decrease in the total exergy loss with a decrease in T_{ev} at a given generator temperature and evaporator heat input (cooling output). Generally the temperature drop between the heat source and the generator is around 10°C (Alizadeh et al. 1979). Taking this into consideration, the exit temperature from the solar collectors may be as low as about 80°C for satisfactory operation. This, in turn, substantially increases the collector efficiency (Kreider and Kreith 1981):

$$\eta_{pl} = F \cdot [X_1 - X_2((T_{ge} + 10) - T_b)/I_{pl}]. \quad (29)$$

Here F is the collector heat removal factor and X_1 and X_2 are the collector design variables. This increase in collector

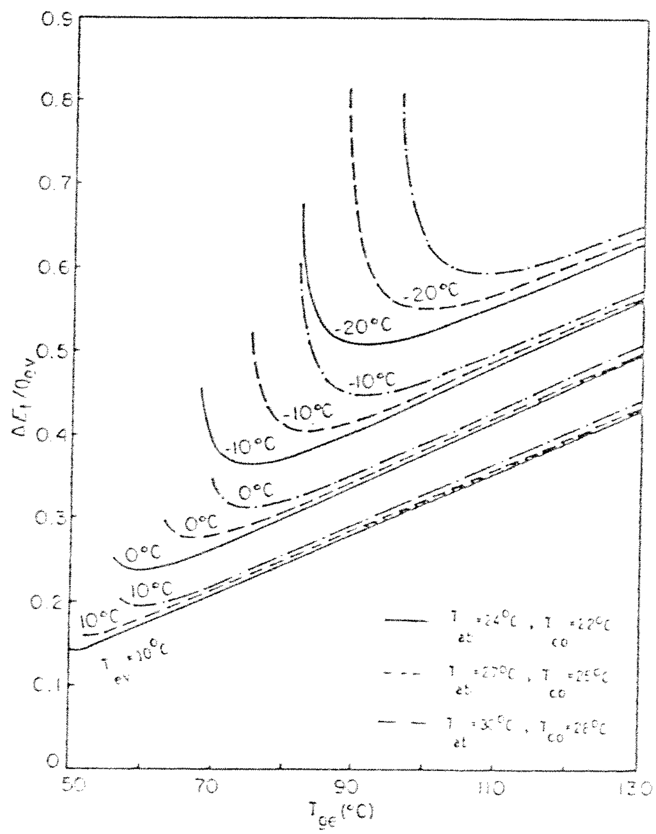


Figure 6 *Change of exergy increase with the generator and evaporator temperature.*

efficiency allows a decrease in the required collector area at a given solar insolation level I_{pl}, thus reducing the system cost.

Computer Program

A computer program was developed to execute the given design algorithm for in-slab radiant ceiling cooling systems. A second program was prepared for joisted ceilings. These computer programs are sensitive to all relevant slab construction variables, back and edge insulation, temperature of the space above, thermal properties of the hose, hose spacing, hose depth from the slab surface, and the position of the cooled space in the building. It determines the surface temperature distribution and also calculates the pressure drop and required chilled-fluid flow rate as well as the mean temperature for given design conditions. The cooling load is calculated in accordance with the provisions described in this article. By any suitable optimization technique, it is possible to optimize the overall cost. The dew-point temperature at given design climatic conditions is calculated and compared with the minimum ceiling surface temperature at design conditions such that there will be no condensation on the ceiling surface.

Computer programs recognize and continuously monitor the following design constraints:

$T_{min} > T_{dp}$	$T_{su} \geq 5°C$
$R_e > 6000$	$0.5 \leq v_w \leq 1.5 \text{ ms}^{-1}$
$X \geq 0.80$	$3 \leq \Delta T_w \leq 7°C$
$\Delta P \leq 50$ kPa	$A_p/M \cdot 1.1 \leq Z$

If the design is not technically feasible, user prompts are issued and "help" screens are provided. Design results are printed in detail. Using the same computer program, it is possible to generate design nomographs for any given design condition and the ceiling construction. These nomographs provide useful design information, such as the required mean fluid temperature and the average ceiling panel surface temperature at a specified hose spacing for a given cooling load intensity. A typical design nomograph for a concrete ceiling slab, which is 0.04 m thick, is shown in Figure 7.

CASE STUDY

Figure 8 shows the solar house at a university. The net living area is 81 m^2 and there is a 19.8-m^2 greenhouse. The

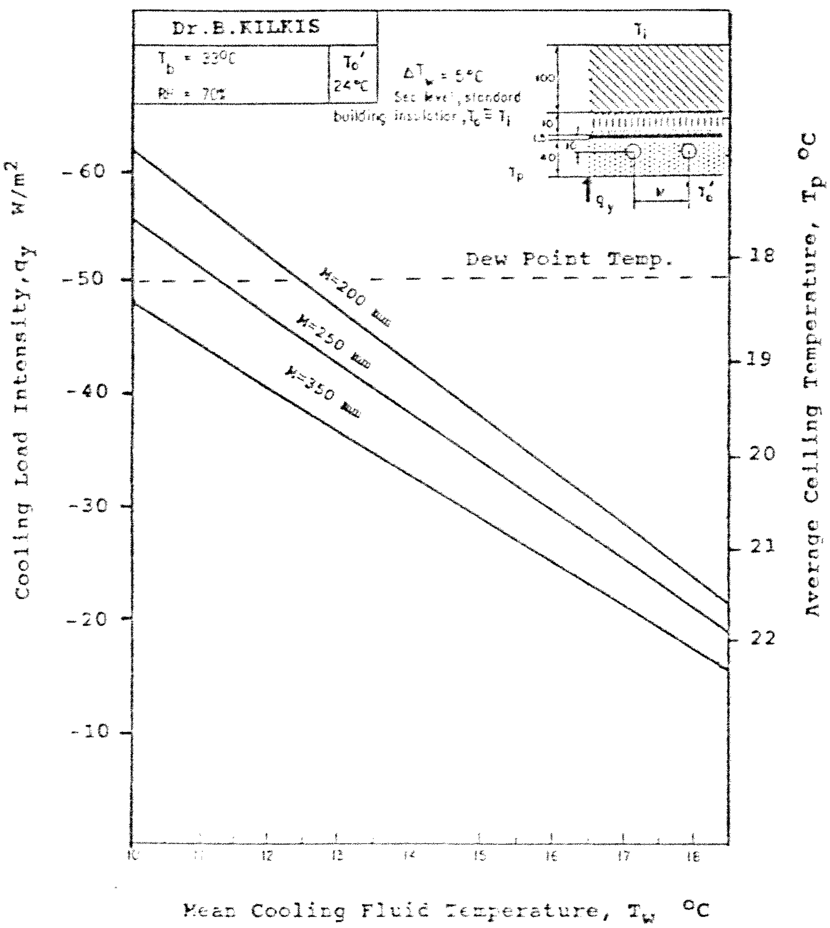


Figure 7 A typical design nomograph for radiant ceiling coolings.

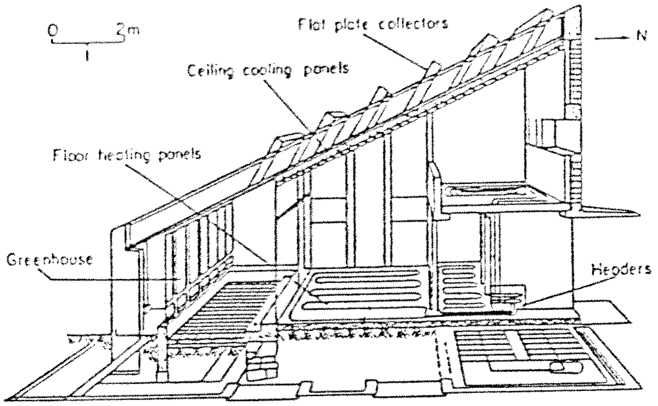


Figure 8 *Solar house.*

heating and cooling volume is 320 m^3. The original design load for sensible cooling was 6.5 kW (22,184 Btu/h). The inclined roof can accommodate flat-plate solar collectors up to a total collector absorber area of 40 m^2. Total ceiling area is around 120 m^2. The average daily insolation in August is about 5.45 kWh/m^2 a day (1,730 Btu/ft^2 a day). As part of the efficiency requirement for the ceiling cooling panel, the ceiling was insulated by 0.025 m (1 in.) thick styrofoam boards. This reduced the peak cooling load to 4.9 kW (16,723 Btu/h). With design provisions related to radiant ceiling cooling, including peak load shaving, the design cooling load reduced to 3.0 kW (10,239 Btu/h) at a design indoor air temperature of 23.3°C (74°F) and a design outdoor temperature of 33.3°C (92°F). At an expected solar collector water supply temperature of 80°C (176°F), the daily average collector efficiency was calculated to be 65% instead of the 55% rated efficiency for a locally available product. Dew-point temperature is 15.4°C (59.7°F). Rubber hoses with an inside diameter of about 0.0095 m (3/8 in.) will be stapled over a 0.012-m (1/2-in.) gypsum board ceiling element. The ceiling is a joisted wood type. The thermal conductivity of the hose material is 0.28 W/m·K (0.16 Btu/h·ft·°F). For the given cooling load and the dew-point temperature, the maximum allowable hose spacing was calculated by the program as 0.15 m (6 in.). With this spacing, the mean chilled-water temperature is calculated to be 11.1°C (52°F). With the extra expense of hose material if the hose spacing is reduced to 0.10 m (4 in.), which is the minimum permissible spacing, the mean chilled-fluid temperature increases to 14.4°C (58°F). In order to maintain turbulent flow, the drop in the chilled-fluid temperature was selected to be around 2.7°C (5°F) at design conditions. This yields a design chilled-fluid supply temperature of 13.0°C (55.4°F). It must be noted that if the ceiling element was concrete, which has a higher thermal conductivity than gypsum board, the mean chilled-fluid temperature would be even higher (see Figure 7), i.e., for 0.1 m (4 in.) thick concrete slab, the required mean fluid supply temperature would be 18.3°C (65°F). With the required fluid temperature, the COP_r of a water-ammonia absorption heat pump with 3.5-kW cooling capacity was calculated to be around 0.70 at an absorber temperature of 38°C. With this figure and 80% efficiency for the solar energy storage system, including the power consumption of the circulating pump of the heat pump, and 65% collector efficiency, the necessary collector area will be

$$A_{pl} = 60/(5.45 \cdot 0.70 \cdot 0.8 \cdot 0.65) = 30.2\,m^2.$$

Twenty flat-plate collectors with 1.5 m^2 absorption area each were selected. The computer printout for 0.15-m hose spacing is shown in Figure 9. A solar-assisted floor-heating system was also designed for the same solar house (Kilkis 1990)

DISCUSSION OF RESULTS

An analytical model for radiant ceiling cooling heat transfer under steady-state conditions was developed. The computer program using this model revealed that the mean chilled-fluid temperatures may be as high as 18°C, especially if the slab material is concrete. This is a relatively high temperature when compared with conventional cooling systems. When coupled with the cooling load reducing and shaving features of radiant ceiling cooling, this moderate chilled-fluid temperature requirement guarantees noticeable improvements in the COP of absorption-type heat pumps and solar collector efficiencies. In the same manner, the total exergy loss of the system substantially decreases. These, in turn, increase the competitiveness and feasibility of such alternative cooling systems. Due to the fact that the radiant ceiling cooling circuit permits the designer to freely select the hose spacing in the technically feasible range, it is possible to better suit the rated temperatures of a given heat pump and thus optimize the overall performance. This flexibility also makes it possible to control the risk of condensation in a stand-alone system. If the humidity has to be controlled, the most recent literature indicates that hybrid systems are feasible with minimum ductwork and air-handling unit requirements. In order to maximize heat pump performance and collector efficiency and truly optimize the radiant ceiling cooling system, it is essential to simulate the indoors as well as the cooling system on an hourly basis including the cool storage analysis. This definitely requires a sophisticated and flexible computer program. However, it is clear that such an effort will be very rewarding and promising for the future of absorption heat pumps whose COP figures are marginally acceptable.

A case study was carried out using available data in the literature for a water-ammonia-type absorption heat pump, where space cooling and domestic water heating are simultaneously accomplished. Design calculations indicated that the performance of both the heat pump and the solar collectors increases. This increase may be improved if the system is dedicated only to space cooling and thus the condenser and absorber temperatures are reduced. Similar improvements in system performance may be expected for other types of absorption heat pumps.

```
Project File Name   :solar
Location            :Ankara TURKEY
Project Description :METU Solab Project
Date                :9/4/1992
MAJOR DESIGN INPUTS
Summer Indoor Design Temperature        ,Ta  = 74.0 °F
Summer Outdoor Design Temperature       ,To  = 92.0 °F
Design Indoor Relative Humidity              = 61 %
Dew Point Temperature                 ,Tdew = 59.7 °F
Sensible Cooling Load                   ,Q   = -10200.0 Btu/h
Ceiling Cooling Panel Area in the Room,Ap   = 1199.96 ft^2
Gross Ceiling Dimensions of the Room    x,y = 40.00ft x 33.00 ft
Thickness of the Slab below hose centers,T = 0.5000 in
Back Insulation Thickness              ,XI = 1.0000 in
Temperature Drop                       ,DT =  5.0 °F
Hose Inside Diameter                   ,Di =  0.375 in
Fin Efficiency of the Ceiling Panel         = 21.28 %
Hose spacing between their centers          = 6.00  [in]
Number of identical circuits in the room,np=  7

DESIGN RESULTS:
A - Temperatures:
Average panel surface temperature       ,Tp = 69.4 °F
Minimum panel surface temperature      Tmin = 58.4 °F
Maximum panel surface temperature     ,Tmax = 74.0 °F
Av.temperature of uncooled surfaces,   AUST = 76.7 °F
Hose wall temperature                   ,Td = 54.9 °F
Mean chilled water temperature          ,Tw = 52.1+/-1 °F

B-Cooling Capacities and Loads:
Radiant cooling intensity               ,qr =  -6.054 [Btu/h ft^2]
Convective heating intensity            ,qc =  -2.447 [Btu/h ft^2]
Total cooling load intensity            ,qy =  -8.499 [Btu/h ft^2]
Radiant to total output ratio ,       qr/qy = 71.23   [%]
Convective to total output ratio ,    qc/qy = 28.80   [%]
Cooling efficiency                       ,x = 86.60   [%]
Heat Gain from the space  above      ,(1-x) = 13.40   [%]
Chiller [Heat Pump] load of the circuit ,CL =  1682.2 [Btu/h]
Total Chiller [Heat Pump] load         ,CLT = 11775.6 [Btu/h]

C-Hydraulics:
Water flow rate                          ,V = 0.672   [US GPM]
Average Water velocity in the hose      ,vw = 1.95    [ft/s]
Hose Reynolds number                    ,Re = 4475.8
Water-Hose thermal convection coeff.  ,alfa = 505.654 [Btu/h ft^2 °F]
Hose friction coefficient            ,Lamda = 0.04032
Corrected Unit pressure drop            ,Dp = 0.036   [psi/ft]
Hose length in the circuit               ,L = 359.99  [ft]
Corrected Total pressure drop in circuit,Pt = 13.12   [psi]
Corrected Pump capacity for the circuit     = 0.6724  [US GPM]/29.25 [ft/hd]
Total Pump Capacity [ circuits in one zone] = 4.7070  [US GPM]/29.25 [ft/hd]
```

Figure 9 Computer output for the design case.

ACKNOWLEDGMENT

The author gratefully acknowledges the kind support of the Turkish Scientific and Technical Council for developing the heat transfer model. He also acknowledges Middle East Technical University for partly supporting the design project for the retrofit of the METU Solar House.

NOMENCLATURE

A = radiant heat extraction term for the composite fin, W/m·K

A_p = surface area of the cooling panel, m^2

A_{pl} = total flat-plate solar collector area, m^2

A_r = total ceiling area in the cooled space, m^2

A_u = total surface area of the cooled space enclosure (excluding the ceiling panels), m^2

$AUST$ = Area-weighted average design temperature of the uncooled surfaces (excluding radiant ceiling), °C

B = convective heat extraction term for the composite fin, W/m·K

C = peak load shaving factor, dimensionless

COP_h = heating coefficient of performance, dimensionless

COP_r = cooling coefficient of performance, dimensionless

C_p = specific heat, kJ/kg·K

D_e = equivalent diameter of the floor in the cooled space, m

D_i = hose inside diameter, m

D_o = hose outside diameter, m

ϵ = surface emissivity of the ceiling panel, dimensionless

$ECOP_r$ = exergetic coefficient of performance in cooling, dimensionless

$ECOP_h$ = exergetic coefficient of performance in heating, dimensionless

F_r = radiant interchange factor, dimensionless

h = altitude above sea level, m

= thermal conductivity of the slab material, W/m·K

= equivalent thermal conductivity of the composite fin for lateral heat diffusion, W/m·K

= thermal conductivity of the hose material, W/m·K

= thermal conductivity of each ceiling element, W/m·K

= thickness of the slab below the centerline of the hose (excluding ceiling elements), m

= inside perimeter of the floor in the cooled space, m

= hose spacing on centers, m

= overall fin coefficient, m^{-1}

= number of ceiling covers, dimensionless

= cooling load of the plant (unit), kW

= heat transfer rate, kW

= cooling rate of the heat pump (evaporator heat transfer rate), kW

= heat gain of the cooled panel from its back, kW

= supplementary cooling air handler capacity, kW

= heat gain of the cooled panel from its edges, kW

= sensible cooling load of the space when cooled by radiant ceiling, kW

= sensible radiant ceiling cooling load after peak load shaving (if applicable) $C \cdot Q_y'$, kW

= radiant ceiling cooling intensity, W/m^2

= total thermal resistance of the ceiling elements, $m^2 \cdot K/W$

= linearization factor for the radiant heat-transfer term, K^3

= indoor design temperature, °C

= indoor design temperature for a radiant ceiling cooled space, °C

= outdoor design temperature, °C

= outside surface temperature of the hose, °C

= dew-point temperature, °C

= indoor temperature of the space above, °C

= maximum ceiling surface temperature, °C

= minimum ceiling surface temperature, °C

= reference temperature, °C

= uniform radiant ceiling surface temperature, °C

= local surface temperature of the ceiling panel, °C

= return fluid temperature in the cooling circuit, °C

= indoor air setpoint temperature, °C

= supply fluid temperature in the cooling circuit, °C

= thickness of the composite fin (including ceiling coverings), m

T_w = mean fluid temperature in the circuit $(T_{su} + T_{re})/2$, °C

U_c = convection heat transfer coefficient on the ceiling panel, $W/m^2 \cdot K$

U_r = radiant heat transfer coefficient on the ceiling panel, $W/m^2 \cdot K$

v_w = average water velocity in the hose, $m \cdot s^{-1}$

V = volumetric flow rate of the fluid in the cooling circuit, $m^3 \cdot s^{-1}$

W = half of the net spacing between adjacent hoses $(M - D_o)/2$, m

W_p = work rate into the pump(s), kW

X = panel cooling efficiency, dimensionless

x = distance on the ceiling surface from the vertical centerline of the hose, m

x_i = thickness of each ceiling element, m

Z = standard hose length, m

Greek Symbols

α = convective heat-transfer coefficient between the fluid and the hose wall, $W/m^2 \cdot K$

ΔE_t = total exergy loss rate, kW

γ = density of the cooling fluid, kg/m^3

ΔT_w = fluid temperature drop, °C

ΔP = fluid pressure drop, kPa

η = fin efficiency, dimensionlesss

σ = Stefan-Boltzmann constant, 5.67×10^{-8} $W/m^2 \cdot K^4$

η_{pl} = flat-plate solar collector efficiency, dimensionless

Subscripts

ab = absorber

c = convective

ev = evaporator

co = condenser

ge = generator

r = radiant

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MPUTER AIDED DESIGN AND ANALYSIS OF RADIANT FLOOR HEATING STEMS

)MPUTER BERECHNETER ENTWURF UND ANALYSE VON SSBODENHEIZUNGS-SYSTEMEN

Birol Kilkis*

MMARY

diant floor heating systems require moderate fluid temperatures and therefore ible an effective, and efficient utilization of low temperature, low intensity ernative energy sources. They may also reduce the space heating loads by as much 30%. In a successful design, it is essential to use a model which considers both thermal and hydrodynamical behavior of the system, and its interaction with the iditioned space. The primary aim of this study is to furnish a simple design tool 1 establish a standard design algorithm, using a composite plate fin model for ady state conditions. The article provides the essential information about this ign procedure, and describes the algorithm of the computer program that has n developed. A case design is presented and the results are discussed.

SAMMENFASSUNG

3boddenheizungssysteme benötigen gemäBtigte Flüssigkeitstemperaturen, um lurch einen effektiven und effizienten Gegrauch von alternativen Energiequellen driger Temperaturen und niedriger Intensität zu gewährleisten. Diese können :h die Heizleistung bis zu 30% verringern. Für den erstrebenswerten temententwurf ist es erforderlich, ein Modell anzuwenden, welches das thermische 1 das hydrodynamische Verhalten des Systems, und die Interaktion mit dem ieizten Raum berucksichtigt. Ziel dieser Studie liegt darin, ein einfaches rkzeug für den Entwurf zu liefern und, unter Anwendung eines aus Kühlrippen ammengesetzten Modells für gleichmaBige Zustandsbedingungen, einen ndartisierten Algorithmus für den Systementwurf zu erstellen. Der Artikel nhaltet die erforderlichen Informationen über diesen Entwurfsvorgang, und er chreibt den Algorithmus des Computerprogramms, welches entwickelt wurde. Anwendungsbeispiel für den Entwurf und eine Besprechung der Ergebnisse wird falls präsentiert.

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1. NOMENCLATURE

A Radiant heat output of the composite fin surface, over unit lengths of two adjacent hose, per unit temperature difference with the unheated surfaces W/m K

a Slope of the lines in the design nomograph (Figure 4), W/m^2 °C

A_p Total surface area of the floor panels in the heated space, m^2

A_r Total floor area of the heated space, m^2

A_u Total area of the surfaces enclosing the heated space (except the floor heating panels), m^2

AUST Area-weighted average temperature of the unheated surfaces at design conditions (excludes floor heating panels), °C

B Convective heat output of the composite fin surface, over unit lengths of two adjacent hose, per unit temperature difference with the ambient, W/mK

C Peak heating load shaving factor, dimensionless

C_p Specific heat, kJ/kg K

D_t Fluid temperature drop coefficient (see Equation 48), kJ/m °C

D_{eq} Equivalent diameter of the floor surface, m

D_i Inside diameter of the hose, m

D_o Outside diameter of the hose, m

e Surface emissivity, dimensionless

f Friction factor, dimensionless

F_r Radiation interchange factor, dimensionless

h Altitude from the sea level, m

k_{eq} Effective thermal conductivity of the composite slab for lateral heat diffusion, W/m K

k_h Thermal conductivity of the hose material, m

k_l Thermal conductivity of each floor cover material, W/m K

k_p Thermal conductivity of the slab material, W/m K

L Thickness of the slab above hose center line (excluding floor covers), m

l_j Total hose length in the jth. floor panel circuit, m

l_{eq} Pressure drop-equivalent hose length for a single circuit replacing parallel circuits served by the same supply and return manifolds, m

L_r Floor perimeter, m

M Hose spacing on centers, m

m Fin coefficient of the floor panel, m^{-1}

n_k Number of floor covers, dimensionless

n_p Number of panel circuits served by the same supply and return manifolds, dimensionless

P Heat load of the heating plant, kW

P_H Heat output of the heat pump, kW

Q_a Back heat loss from the floor panel, kW

Q_D Heating capacity of the boiler supplementing a heat pump, kW

Q_e Edge heat loss from the floor panel, kW

Q_y' Radiant floor heating load, kW

Q_y Radiant floor heating load after peak load shaving, kW

Intensity of the total heat output of the floor panels, W/m^2
Thermal resistance between hose outer skin and slab material, $m^2 K/W$
Total thermal resistance of floor cover(e), $m^2 K/W$
Reynolds number, dimensionless
Linearization factor for the radiant heat output intensity, K^3
Indoor design temperature, °C
Indoor design temperature for a radiant floor heated space, °C
Outside surface temperature of the hose, °C
Temperature of the space below the heated slab, °C
x Maximum panel surface temperature, °C
ı Minimum panel surface temperature, °C
Outdoor design temperature, °C
"Effective" floor panel surface temperature, °C
x) Local panel surface temperature, °C
Fluid temperature in the supply manifold, °C
ıl Thickness of the composite fin (including floor covers), m
Mean fluid temperature in the floor panel circuit, °C
y) Local fluid temperature in the floor panel circuit, at a distance y from the supply manifold, °C
Overall heat transfer coefficient on the panel surface, W/m^2K
Convection heat transfer coefficient on the panel surface, W/m^2K
Radiation heat transfer coefficient on the panel surface, W/m^2K
Fluid flow rate, $m^3 s^{-1}$
Average fluid velocity in the hose, $m s^{-1}$
Half of the net space between the adjacent hose (M-Do)/2, m
Heating efficiency, dimensionless
Position variable on the floor panel surface, m
Thickness of each floor cover, m
Available hose length, m

ek symbols:
Convection heat transfer coefficient between the fluid and the hose inner skin, $W/m^2 K$
Density of the fluid, kg/m^3
Temperature drop in the heating fluid (water), °C
Pressure drop in the panel circuit, kPa
Fin efficiency of the composite fin, dimensionless
Stefan Boltzmann constant, $5.67.10^{-8} W/m^2 K^4$
Pipe wall roughness, m
Kinematic viscosity of the heating fluid, $m^2 e^{-1}$

ɔscripts:
Convective
Radiant

2. INTRODUCTION

In spite of their substantial energy saving potential, there is not a complete model for predicting and simulating the overall performance of radiant panel heating systems. Existing models generally isolate the heating slab and analyze it under ideal conditions and overlook its thermal and hydrodynamical interaction with the conditioned space. In order to completely describe the behavior of the system, three coupled models need to be developed and interpreted. These are:

2.1. Thermal and Hydrodynamical Model of the Conditioned Space.

The natural convection taking place from the floor panel surface to the conditioned space, and the natural infiltration from external cracks and openings depend upon the velocity, temperature and pressure fields in the conditioned space. In addition, the moving indoor air controls the film coefficient on the partition surfaces . Thermal stratification also effects the ceiling heat loss as well as the infiltration heat loss. All these factors necessitate a comprehensive analysis of the pressure, temperature and velocity field. Calculation of the surface temperature distribution of the partitions is also essential for an accurate determination of the radiation heat exchange between the floor panels and these surfaces.

2.2. Slab Heat Output.

The total heat output of the floor panels depends upon the indoor air temperature, the surface temperatures of all the unheated surfaces, air movement in the heated space and other surface characteristics like the emissivity. The convective heat output also depends upon the altitude and the size of the conditioned space. If these factors can be adequately correlated, the total heat output can be expressed in terms of the effective panel surface temperature (ASHRAE, 1992), (DIN 4725, 1988).

2.3. Heated Slab Model.

Once the relationship between the panel surface and the heated space is established, the heat diffusion in the slab and the required mean water temperature in the hydronic circuit may be calculated.

One of the earliest heated slab model was given by Kollmar(1957). He assumed that the slab operates as a plate fin and loses heat from the upper surface. Zhang and Pate (1986) developed a finite difference algorithm for ceiling heating panels for solving the steady and transient heat diffusion problem in the slab and predict the local surface temperatures. Kilkis (1989, 1992-c) introduced a more elaborate fin model. DIN 4725 (1989) provides a simplified algorithm. Krinninger (1989) states that lengthy numerical algorithms may improve the solution for the required mean fluid temperature only by a few degrees Centigrade. However this may not necessarily be the case for certain combinations of the design variables.

3. THEORY

3.1 Heat Losses in a Floor Heated Space

Calculations of heat losses from a floor heated space needs to be customized because of the following factors:

i- Indoor air temperature:

Human thermal comfort in a conditioned space is established by controlling and maintaining the indoor air temperature and the average surface temperature of the enclosures. In a radiant floor heated space this average surface temperature which includes also the radiant floor panel will be higher when compared with conventional heating systems. Presence of a slow air movement also guarantees a lower film coefficient between the body and the ambient air. These reduce the dependence on the indoor air temperature for human comfort by as much as 3 to 4°C. Ability to set a low indoor comfort temperature directly reduces the transmission and infiltration heat losses. It is possible to calculate the customized gn indoor air temperature for an equivalent human thermal comfort for a radiant · heated space. A sample calculation is provided in Appendix A. In addition to the mal comfort through the thermal equilibrium between the human body and its ronment, the radiant floor heating system establishes a perfect balance among modes of this heat exchange: In a typical radiant floor heated space, the split ·een radiation and convection is about 70 % to 30 % respectively. A typical t body also tends to loose heat in a similar proportion where such a balance is available in other conventional heating systems where the radiant-convective is typically around 20 % versus 80 % respectively.

‘atural Infiltration

ıuse of the existence of a more uniform air pressure and velocity field, a tance against natural infiltration and internal draft builds up. This reduces the metric flow rate of infiltrating outdoor air through the external cracks and ings. According to the preliminary studies, this feature may be translated to the ;n by multiplying the standard leakage coefficients of exposed cracks by a factor 6 (Kilkis, 1992-c). Due to the lack of substantiative data at present, leakage icients for other infiltration ports may not be modified likewise.

' values of outdoor exposed partitions

use indoor air's wetting of the partitions is slow, the film coefficient is small. is especially important for outdoor exposed partitions with high U values like ows or uninsulated walls. For practical purposes, film coefficients in radiant heating may be taken about 75 % of the standard values (Kilkis, 1992-c).

'hermal storage (peak load shaving)

nding upon the available thermal mass exposed to the conditioned space, heat ɔred during off peak load periods provided that the system is idled with fluid erature modulation. In this case the peak design heating load may be shaved-off zing the panel:

$$Q_y = C \cdot Q_y' \qquad [1]$$

load shaving factor depends on many factors like the climate, typical daily ɔor temperature swings, heat storage capability of the building elements, in ion to the heated slab itself. By considering three distinct climatic conditions listinguishing three different building thermal masses, Kilkis(1990) formulated rst engineering rule for determining C.

able 1. Peak Load Shaving Factor.

	C		
	BUILDING MASS		
LOCATION	Light Constr.	Regular Constr.	Heavy Constr.
Sea Side	1.0	0.88	0.80
Moderate	1.0	0.85	0.78
Cold, Mountain	1.0	0.80	0.75

light construction class corresponds to a building mass less than 6000 N/m^2 heavy construction class corresponds to a building mass greater than)N/m^2.

3.2 Heat Output From a Radiant Slab

Provided that the back (Q_a) and edge heat losses (Q_e) are accounted in sizing the heating plant (see Equations 3, 4 and 37) and the mean fluid design temperature is accordingly modified (see Equation 35), the required net heat output of the panel(e) will be:

$$q_y \geq Q_y/A_p \cdot 1000 \qquad [2]$$

Defining the heating efficiency, X as:

$$X = \frac{Q_y}{Q_y + Q_a + Q_e} \qquad [3]$$

Total plant (boiler) load of the heated space, P will be:

$$P = Q_y/X \qquad [4]$$

If a heat pump is going to be used, it may be sized according to one of the following criteria :

$$P_H \geq P \quad \text{without supplementary plant} \qquad [5]$$
$$P_H + Q_D \geq P \quad \text{with supplementary plant} \qquad [6]$$

The total panel heat output intensity is the sum of the radiant and convective heat output intensities (Kilkis, 1990), (ASHRAE, 1992):

$$q_y = q_r + q_c \qquad [7]$$

where; $q_r = U_r \cdot (T_p - AUST)$ [8]

$$q_c = U_c \cdot (T_p - T_a') \qquad [9]$$

Here U_r and U_c are the radiation and convection heat transfer coefficients on the radiant floor panel surface respectively:

$$U_r = r \cdot F_r \cdot \sigma \qquad [10]$$
$$U_c = (1 - 2.22 \cdot 10^{-5} h)^{2.627} \cdot (4.96/D_{eq})^{0.08} \cdot 2.67 \cdot (T_p - T_a')^{0.25} \qquad [11]$$

Here r is the linearization factor for the radiation heat transfer:

$$r = 4 \cdot [(T_p + 273)/2 + (AUST + 273)/2]^3 \qquad [12]$$

It may be further simplified in the practical design domain:

$$r = [0.0105 \cdot (T_p + AUST)/2 + 0.7955] \cdot 10^8 \quad \{15°C \leq (T_p + AUST)/2 \leq 30°C\} \qquad [13]$$

the simplified radiation interchange factor for $A_p/A_u \le 0.30$ [RAE, 1992] :

$$F_r \approx e \qquad [14]$$

convection heat transfer coefficient is corrected with the altitude (h) (Kilkis,) and the room size (Min and et.al., 1956) which is characterized by D_{eq}:

$$D_{eq} = (4 \cdot A_r / L_r) \qquad [15]$$

area-weighted average temperature of the unheated enclosing surfaces, AUST is ult to calculate. It primarily depends on T_o, the area ratio of indoor and oor partitions, thermal properties and dimensions of the partition elements, ion of the room in the building, and the outdoor wind conditions. An elaborate erical algorithm was provided by Zmeureanu, Fazio and Haghighat(1987). An oximate expression was provided by Kilkis (1989):

$$AUST \approx T_a' - c \cdot w \qquad \{14°C \le AUST \le 19°C\} \qquad [16]$$

c is the room position code.i.e: c is one for an inner room, two for a room with utdoor exposed side, and three for a room with two or more outdoor exposed w is a function of T_o:

$$w = 15/(25 - |T_o|) \qquad \{T_o < 25°C\} \qquad [17]$$

ice T_a' and T_u are determined, T_p can be solved for a required heat output sity using Equations 7, 8 and 9. However these equations presume that T_p is m. Depending upon the slab construction, required heat output, indoor rature as well as the spacing between the heat transfer hoses, the floor surface rature exhibits a non-uniform profile. Models available for predicting this are few. Krinninger (1989), and Leal and Miller (1972) claim that the is a Sine curve. This simplifies the calculations, because at x = M/4 on the floor, e temperature always coincides with T_p (See Figure 1, path a - b). However ssumption is insensitive to slab thickness, heat output intensity, indoor air rature, thermal properties of the slab layer and floor coverings. Furthermore it ot determine the surface temperature profile because of insufficient number of ary conditions for determining a unique Sine Curve passing through the point [/4).

725 provides a simple panel heat output model (1989):

$$q_y = 8.92 \cdot (T_p - T_a')^{1.1} \qquad [18]$$

not distinguish the radiant and convective heating split and is insensitive to e emissivity, size of the heated space, the altitude from the sea level and AUST. factors may prove to be extremely important such that the heat output ity may vary by as much as 50% depending upon specific combinations of factors.

omposite Plate Fin Model

In order to establish a simplified analytical algorithm, for designers without sacrificing accuracy, a composite plate fin model was developed. The model is shown in Figure 1. Part of the slab that remains above the hose center line with the floor cover(e) establish the composite plate fin which looses heat from the top surface both by radiation and convection. According to the symmetry, characteristic fin length is W. In contrary to the existing models for plates and floor panels (Kollmar, 1957), (Kreider and Kreith, 1979), this model treats the radiative and convective heat output of the composite fin separately. By superimposing these two, floor surface temperature at the base of the fin (T_{max}) is computed (see Equations 20 through 25). The lateral heat diffusion in the composite fin with parallel layers of different thermal conductivities, is expressed in terms of an equivalent thermal conductivity:

$$k_{eq} = [\sum_{i=1}^{n_k} k_i \cdot x_i + k_p \cdot L\,]/T_{total} \qquad [19]$$

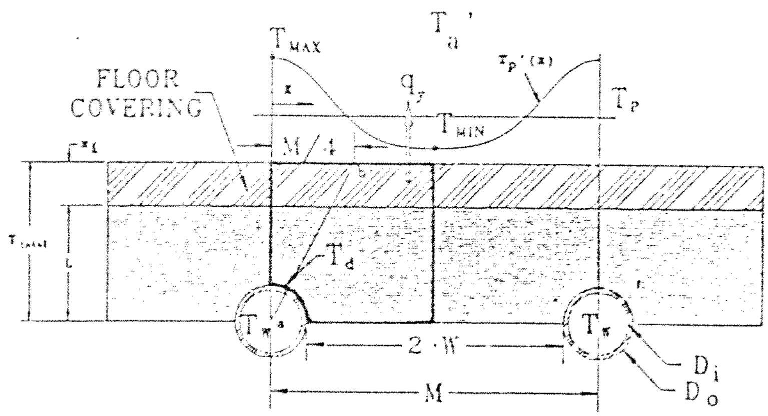

Figure 1. Composite Fin Model (Kilkis, 1992)

Defining the radiation and convection fin efficiencies;

$$\eta_r = \eta_c = \eta \qquad [20]$$

where; $\eta = \tanh(m \cdot W)/(m \cdot W)$ [21]

$$m = [\,U/(k_{eq} \cdot T_{total})]^{1/2} \qquad [22\text{-a}]$$

and;

$$U = U_r + U_c \qquad [22\text{-b}]$$

The total heat output of the fin surface covered by two adjacent hose spaced at M, and each having a unit length, will be:

$$[2 \cdot W \cdot \eta_r + D_o] \cdot U_r \cdot (T_{max} - AUST) + [2 \cdot W \cdot \eta_c + D_o] \cdot \cdot U_c \cdot (T_{max} - T_a') = q_y \cdot M \qquad [23]$$

rarily assigning;

$$A = [2 \cdot W \cdot \eta_r + D_o] \cdot U_r \, , \quad [24\text{-a}]$$

$$B = [2 \cdot W \cdot \eta_c + D_o] \cdot U_c \, , \quad [24\text{-b}]$$

solving Equation 23 for T_{max} for a required q_y:

$$T_{max} = \frac{q_y \cdot M + A \cdot AUST + B \cdot T_a'}{A+B} \quad [25]$$

der and Kreith (1981) expressed the fin efficiency for a flat plate solar collector ıg heat both by radiation and convection from its top surface to the ambient. defined an overall heat transfer coefficient without regarding the split between radiation and convection. Kollmar (1957) expressed a similar fin efficiency. owing these definitions with;

$$U' = q_y/(T_p - T_a') \quad [26]$$

$$m' = [U'/(k_{eq} \cdot T_{total})]^{1/2} \quad [27]$$

$$\eta' = \tanh(m' \cdot W)/(m' \cdot W), \quad [28]$$

ore simple expression for T_{max} may be obtained:

$$T_{max} \approx T_a' + \frac{(T_p - T_a')}{[2 \cdot W \cdot \eta' + D_o]} \, . \, M \quad [29]$$

s equation becomes identical with Equation 25 only with the conditions ST $\approx T_a'$ and $U_r \approx U_c$ .These conditions may be valid for a limited combination of pertaining variables. After computing T_{max}, the surface temperature profile can determined, noting that radiation and convection temperature gradients are erent:

$$x) = \frac{\cosh [m \cdot (M/2 - x)}{\cosh [m \cdot W]} \, . \, (T_{max} - T_a' + c) + T_a' - c \qquad \{D_o/2 \le x \le M/2\} \quad [30\text{-a}]$$

$$x) = T_{max} \qquad \{x \le D_o\} \quad [30\text{-b}]$$

term c represents the difference between temperature gradients for radiation l convection:

$$c = U_r/U \cdot [T_a' - AUST] \quad [31]$$

temperature profile may be revised for $x \ge D_o/4$ by introducing a second degree

polynomial for $0 \le x \le D_o/4$ with the boundary conditions:
$T_p'(x) = T_{max}$ at $x = D_o/4$ and $dT_p'(x)/dx = 0$ at $x = 0$
T_{min} can be calculated from Equation 30-a with $x = M/2$:

$$T_{min} = \cosh[m \cdot W]^{-1} \cdot (T_{max} - T_a' + c) + T_a' - c \qquad [32]$$

The following data represents the case study to be described in the following sections:

$q_y = 81\ W/m^2$	$A_p = 15.0\ m^2$; $D_o = 0.0162\ m$
$T_a' = 18.0\ °C$	$M = 0.20\ m$
AUST $= 16.6°C$	$x_1 = 0.006\ m$ (one floor cover)
$T_p = 25.4°C$	$k_1 = 0.071\ W/m\ K$ (carpet)
$L = 0.040\ m$	$k_p = 1.4\ W/m\ K$ (concrete)

By using the computer program which was developed in this study, surface temperatures were calculated. They are:

$T_{max} = 28.5°C$; $T_{min} = 23.5°C$

The surface temperature profiles are shown in Figure 2. A Sinusoidal temperature distribution with an arbitrary boundary condition was also plotted in order to compare the form of the profiles.

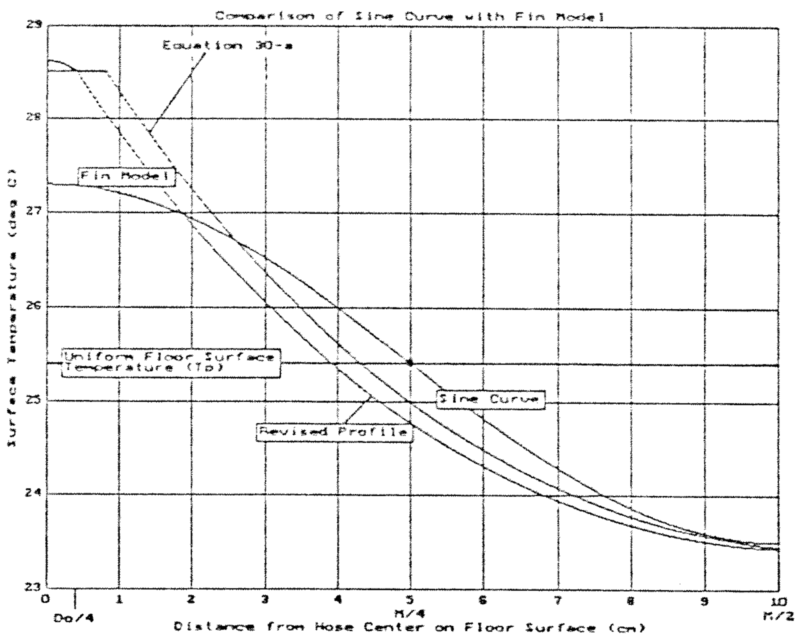

Figure. 2 Surface Temperature Profiles by the Composite Fin Model and the Sine Curve with an arbitrary boundary condition: $T_p'(M/2) = T_{min}$

Using T_{max} and taking the shortest path for the heat diffusion, the uniform outside surface temperature of the hose at design conditions can be approximated :

$$T_d = T_{max} + q_y \cdot [R_{co} + (L - D_o/2)/k_p + R] \qquad [33]$$

iermal resistance between the hose and slab material, R may be neglected. R_{co} is e thermal resistance of floor covers above the slab:

$$R_{co} = \sum_{j=1}^{n_k} (t_j / k_j) \tag{34}$$

ie required mean fluid temperature, T_w will be:

$$T_w = \frac{q_y \cdot M}{X \cdot \pi} \cdot \left[\frac{1}{\alpha \cdot D_i} + \frac{1}{2 \cdot k_h} \cdot \ln(D_o/D_i) \right] + T_d \tag{35}$$

ıere; $\alpha = 1056 \cdot [0.02 \cdot (T_w + 273) - 4.06] \cdot v_w^{0.8}/D_i^{0.2}$ [36]

r a turbulent flow.

4 Hydraulics

ie required fluid flow rate and the average velocity in the circuit serving the floor ınel at design conditions will be a function of the boiler load of the panel (Q_y/X), ·operties of the fluid and the design temperature drop ΔT_w in the circuit:

$$V = Q_y/(X \cdot \Omega \cdot C_p \cdot \Delta T_w) \tag{37}$$

$$v_w = 4 \cdot V/(\pi \cdot D_i^2) \tag{38}$$

or a turbulent flow, hose Reynolds number should be preferably above 8000, or at ast above 6000:

$$R_e = (v_w \cdot D_i)/\Phi \tag{39}$$

. radiant panel heating practice, R_e hardly exceeds $2 \cdot 10^4$. Therefore, the pressure ·op coefficient is strongly dependent on R_e (see Moody, 1944), and Colebrook's ıtural roughness function must be employed:

$$1/\sqrt{f} = 1.14 + 2 \cdot \log(D_i/\varepsilon) - 2 \cdot \log\left[1 + \frac{9.3}{R_e \cdot (\varepsilon/D_i) \cdot \sqrt{f}}\right] \tag{40}$$

ı spite of this fact, it is a general practice to calculate the friction factor by using ıuations which are valid only for $R_e \geq 10^6$. This is evident on typical pressure drop ıarts currently used by the radiant floor heating industry where the constant ·erage water velocity lines are orthogonal to the pressure drop lines. This ·ndition is mathematically possible if and only if the friction factor is assumed to be ıdependent from hose Reynolds number ($R_e \geq 10^6$). In an actual case where f is eynolds number dependent, these lines will not be orthogonal, and the angle of ıtersection angle depends on ε.

A typical pressure drop chart as developed in this study is shown in Figure 3.

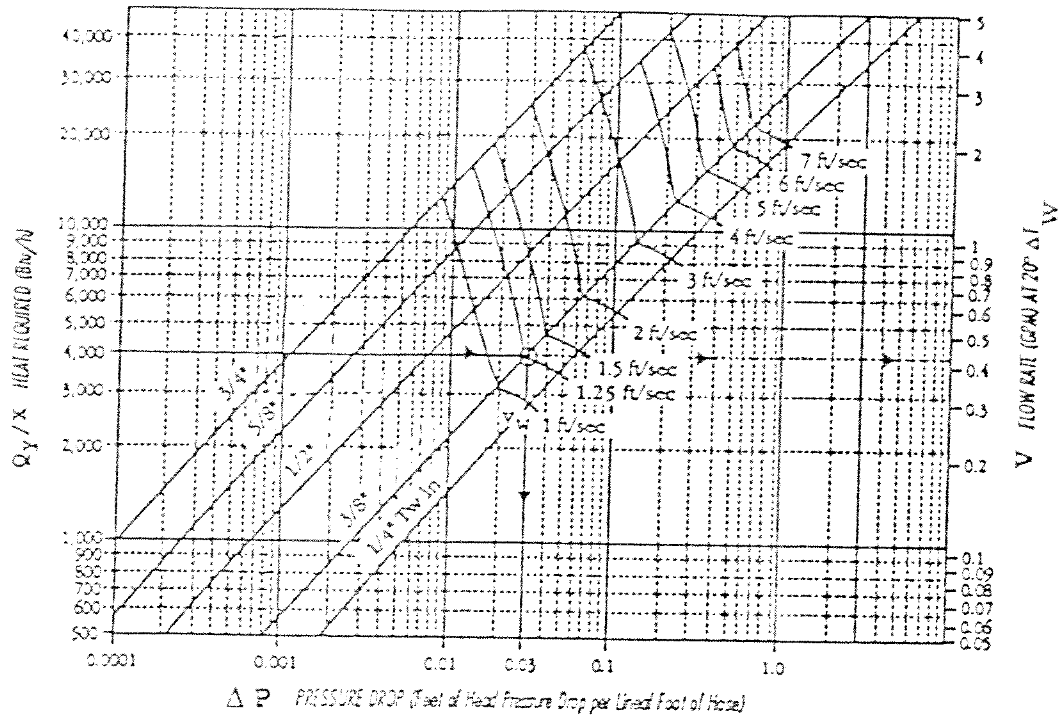

Figure 3. R_e Number Dependent Pressure Drop Chart

In a multiple, parallel circuit system with different hose lengths, the general practice is to multiply the longest circuit length by the unit pressure drop corresponding to the design flow rate in that circuit, in order to calculate the critical pressure drop. This misconception may result in oversized pumps. In fact, the pressure drop will be common in all parallel circuits, provided that pressure continuity is preserved by a proper manifold design and installation (Hansen, 1985), (Kilkis, 1992-c).

In this case, the flow rates in the circuits will change until equilibrium in pressure drop is established. i.e. : in a longer circuit the flow will be slower than the design value. In turn, the fluid temperature drop will increase in order to maintain the required heat output or vice versa. Formulating this equilibrium:

$$\Delta P_1 = \Delta P_2 \ldots = \Delta P_{np} \qquad [41]$$

Where the pressure drop in circuit j with hose length l_j will be:

$$\Delta P_j = f \cdot \Omega \cdot v_w^2/(2 \cdot D_i) \cdot l_j \qquad [42]$$

By accounting for singular pressure losses by introducing a factor like 1.1 to Equation 42 , and using Equation 38:

$$\Delta P_j = f \cdot \Omega \cdot [\frac{8 \cdot V^2}{\pi^2 \cdot D_i^5}] \cdot l_j \cdot 1.1 \qquad [43]$$

ning a pressure drop-equivalent hose length for a single circuit replacing the allel circuits on the manifold (Kilkis, 1992-c):

$$l_{eq} = [\sum_{j=1}^{n_p} (l_j)^{-1/2}]^{-2} , \qquad [44]$$

n, the flow rate in a given circuit (j) at pressure drop equilibrium:

$$V_j = [l_{eq}/l_j]^{1/2} \cdot \sum_{j=1}^{n_p} V_j \qquad [45]$$

second term is the total flow rate at the supply manifold at design conditions. ations 44 and 45 reveal that the actual flow rate in a given circuit will be tical to the design prediction if all circuit lengths are the same. Otherwise, the al heat output and the fluid temperature drop in every circuit will be somehow rent. They can be calculated using the following equations (Kilkis, 1992-c):

$$q_y = \frac{\int_{y=0}^{l_j} M \cdot a \cdot [T_w(y) - T_a'] \cdot d_y}{A_p} \qquad [46]$$

$$\frac{T_w(y) - T_a'}{T_{su} - T_a'} = e^{-\left|\frac{M \cdot a}{D_t \cdot v_w \cdot 1000}\right| \cdot y} \qquad [47]$$

a is the slope of the line for the given hose spacing M (see Figure 4) on the gn nomograph, and D_t is defined by :

$$D_t = \Omega \cdot C_p \cdot \pi \cdot D_i^2/4 \qquad [48]$$

Computer Program

computer program was developed in order to execute the new algorithm. This ram is sensitive to slab construction data, information about back and edge lation, temperature of the space or ground below, thermal properties of the , spacing of the hose, and depth from the surface as well as the altitude, room outdoor temperature, position of the room in the building, and floor covers. It mines the surface temperature distribution and calculates also the pressure and the required water flow rate as well as the required mean fluid erature for given design conditions. Heat loads are separately calculated in rdance with the provisions described in this article. Design constraints are:

(i) $T_p \leq 29°C$ (31°C in unoccupied peripheral zones)
(ii) $T_w \leq 70°C$ (special rubber hose); $T_w \leq 60°C$ (thermoplastic pipe)
(iii) $5°C \leq \Delta T_w \leq 15°C$
(iv) $A_p/M \cdot 1.1 \leq Z$
(v) $X \geq 0.80$
(vi) $Q_y/X \leq 5$ kW (for each circuit)
(vii) $q_y \leq 140$ W/m²
(viii) $0.5 \leq v_w \leq 1.5$ m s⁻¹
(ix) $M \geq 6 \cdot D_o$ (special rubber hose); $M \geq 10 \cdot D_o$(thermoplastic pipe)
(x) $\Delta P \leq 50$ kPa
(xi) $T_d \leq 65°C$
(xii) $T_{max} \leq 45°C$
(xiii) $0.025 \text{ m} \leq L \leq 0.060 \text{ m}$
(xiv) $R_e \geq 8000$

The computer program continuously checks the progress of the design with respect to these constraints. If the design is not in the technically feasible domain, it prompts the designer and declares possible corrective means. It is also capable of producing any design customized nomograph. A typical design nomograph is shown in Figure 4.
This nomograph provides the unique relationships among q_y, T_p, T_w and M for the given other design variables. It was prepared for a bare slab so that one may easily adjust the T_w readings for any $R_{co} > 0$ condition by the following approximation within + or - 2% accuracy:

$$T_w \approx T_w + q_y \cdot R_{co} \qquad [49]$$

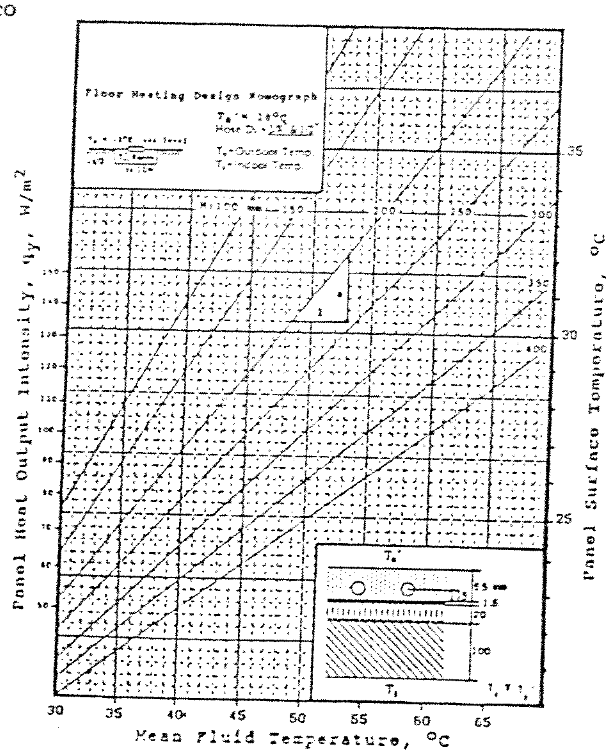

Figure 4. A Typical Design Nomograph

CASE STUDY

A two story residential home is going to be heated by a radiant floor heating ;tem. The building has a conventional heating load of 29 kW at a design outdoor aperature of -3°C. The standard indoor temperature is 22°C. The equivalent nfort indoor temperature is selected to be 18°C for radiant floor heating. The liant floor heating load was calculated to be 22.2 kW including the peak load aving factor. This is a 23% decrease from the standard load. The building has) m^2 total floor area and 275 m^2 is available for the floor panels. In a typical room h a heating load of 2.43 kW, two floor panels will be used with 15 m^2 surface area :h. This corresponds to a radiant heat load intensity of 81 W/m^2. Gross floor ıensions are 6 x 6 m. R_{co} is 0.085 m^2 K/W. Rubber hose with 0.0095 m I.D (3/8") d 0.016 m O.D. will be used. One side of the room is exposed to outdoors. Hose ıter line is 0.04 m below the surface of the slab. The necessary floor surface ıperature, T_p at design conditions is calculated to be 25.4°C. In this design case, V 4725 (see Equation 18) exactly agrees with this value. Computer calculations owing the composite fin model calculate the required mean water temperature to 42°C for an optimum hose spacing of 0.2 m on centers. DIN 4725 procedure :dicts the same temperature as 43.5°C. The difference is directly attributable to s procedure where it is assumed that heat flows along a straight line a-b (see ;ure 1), including the carpet. In fact, the carpet has a low thermal conductivity, :h that heat can not diffuse laterally at the same rate along the carpet. Therefore th a-b is a broken line which yields a smaller total thermal resistance. Using the nputer program result, the required water supply temperature, T_{su} will be : + 10/2) = 47°C, with a design temperature drop of 10°C. The temperature ›file on the floor surface is shown in Figure 2. Back and edge losses are 0.12 kW. is is about 6% of the heat output to the heated space. Taking this loss into :ount, the boiler load of this room will be 2.57 kW. The radiant heat output is % of the total.

e actual fluid temperature drop along the circuit (Equation 47) is shown in ;ure 5.

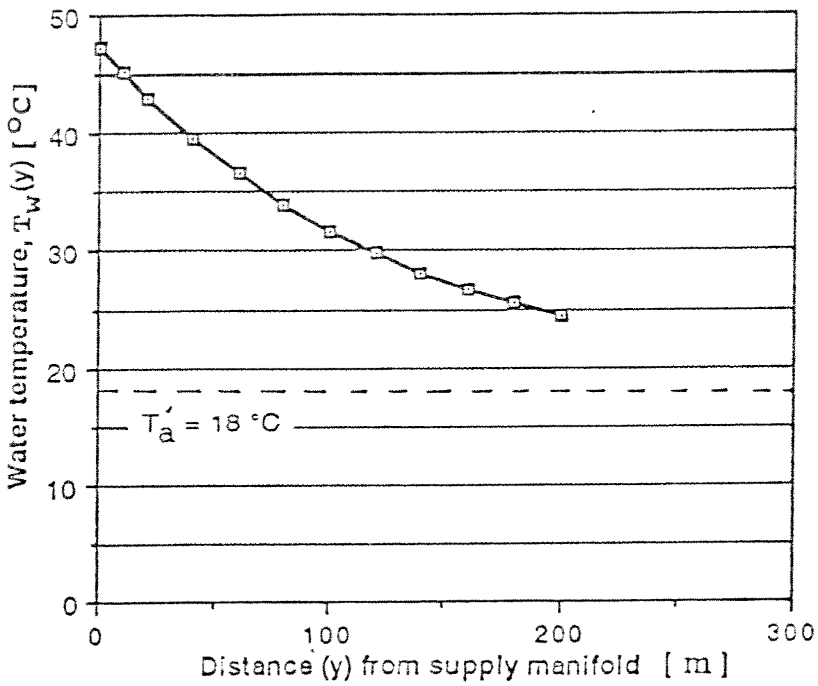

Figure 5. The Actual Mean Water Temperature Drop

A detailed finite element analysis indicated that the mean water temperature prediction with the simplified algorithm is within + or - 3% accuracy (Kilkis, 1993).

5. DISCUSSION OF RESULTS AND CONCLUSION

Today nearly one out of every three new single family homes built in industrialized nations are equipped with a radiant panel heating system. In Germany this ratio has already exceeded 50 % (1992-a). Due to its several attributes, radiant floor heating is becoming a key element in improving the technical and economical feasibility of waste heat and alternative energy utilization with or without the assistance of heat pumps . The performance of heat pumps are also improved. This is especially true for air to water type heat pumps where their feasibility are marginal especially in adverse climates. However in order to realize these benefits, a careful and detailed analysis of the indoor space coupled performance of the panel is essential along with a customized heat loss calculation. In this study, an analytical model was developed in order to predict the heat output of radiant floor panels as well as the steady state heat transfer in the panel. The interaction with the heated space was also modeled through a simplistic approach, which needs further refinement for a more accurate simulation. The comparison of the developed design algorithm with French (1990-a) and German standards (1989-a) has indicated that it is more comprehensive although the results proved to be close in the case design. However this close agreement may not be generalized. For example if the room had no outdoor exposure or more exposures, and the room size was too large, the results could be substantially different. In addition, DIN 4725 algorithm may over predict the water temperature if there is a heavy covering on the slab. Therefore, the new algorithm is especially useful for extreme design cases. It has also been observed that various nomographs that are in circulation may underestimate or overestimate the heat output intensity by up to 50 % partly due to the fact that all variables can not be incorporated into a single design nomograph. For this reason, a computer program was developed so that all the pertinent variables can be accounted for. The program is affordable by the designers and can be run on personal computers. It assists the designer in optimizing the design so that all the virtues of radiant floor heating system can in fact be practiced. Common misconceptions about the critical pressure drop and the friction factor calculations were also recognized and eliminated by introducing the appropriate solutions and the algorithm (Equations 45 through 48).

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\PPENDIX.A

Sensible thermal comfort is established when the sensible heat generated by the man body at a certain activity level and clothing, can be dissipated by the heat is from the body to the environment in the conditioned indoor space. Let the ensity of sensible heat generated per unit surface area of an adult body be W/1.6 m². Recognizing the radiant and convective heat loss intensities from the dy;

$$q_r = F_r \cdot {}_r \cdot \sigma \cdot (T_s - T_u) = 4.6 \cdot (T_s - T_u) \qquad [A\text{-}1]$$

$$q_c = \alpha_s \cdot (T_s - T_a) \qquad [A\text{-}2]$$

r the sensible thermal comfort:

$$80/1.6 = 4.6 \cdot (T_s - T_u) + \alpha_s \cdot (T_s - T_a) \qquad [A\text{-}3]$$

Here α_s is the surface convection coefficient, T_s is the average surface temperatu
of the adult.
With the boundary conditions for the following typical cases:

Radiant floor heating case:

T_s = 26.6°C ; AUST = T_a' + 3 ; $\alpha_s \sim 3$ W/m²K ;

Radiator heating case:

T_s = 26.2°C ; AUST = T_a - 2 ; $\alpha_s \sim 4$ W/m²K ;

Forced air heating case:

T_s = 25.8°C ; AUST = T_a - 2 ; $\alpha_s \sim 6$ W/m²K .

and using Equation A-3 the required indoor air temperatures:

Radiant floor heating case, T_a' : 18.2°C

Radiator(baseboard) heating case, T_a : 21.5°C

Forced air heating case, T_a : 21.9°C

AN ANALYSIS OF THE SOLAR ABSORPTION CYCLE WHEN COUPLED WITH IN-SLAB RADIANT COOLING PANELS

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Gazi University Middle East Tech. U.
Ankara Turkey

:T

ır provides the fundamentals and the ıorithm of radiant panel cooling systems ial emphasis given to the utilization of gy. A steady state heat transfer model loped in order to determine the heat n a cooled ceiling slab. More suitable tures of the radiant ceiling cooling th absorption type heat pumps and 'solar were identified.

ATURE

diant heat extraction term for the nposite fin, W/mK
face area of the cooling panel,m^2
al flat plate solar collector area, m^2
al ceiling area in the cooled space,m^2
al surface area of the cooled space closure (excluding the ceiling panels) ,m^2
a-weighted average design temperature the uncooled surfaces (excluding radiant ling),°C
nvective heat extraction term for the nposite fin, W/mK
ak load shaving factor,dimensionless
ating coefficient of performance, nensionless
oling coefficient of performance, nensionless
ecific heat, kJ/kg K
uivalent diameter of the floor in the oled space, m

D_i Hose inside diameter, m
D_o Hose outside diameter, m
e Surface emissivity of the ceiling panel, dimensionless
$ECOP_r$ Exergetic coefficient of performance in cooling, dimensionless
$ECOP_h$ Exergetic coefficient of performance in heating, dimensionless
F Collector heat removal factor
F_r Radiation interchange factor, dimensionless
h Altitude above sea level, m
k Thermal conductivity of the slab material, W/m K
k_e Eqv. thermal conductivity of the composite fin for lateral heat diffusion, W/m K
k_h Thermal conductivity of the hose material, W/m K
k_i Thermal conductivity of each ceiling element, W/m K
L Thickness of the slab below the hose center line (excluding ceiling elements), m
L_r Inside perimeter of the floor in the cooled space, m
M Hose spacing on centers, m
m Fin coefficient, m^{-1}
n_k Number of ceiling covers, dimensionless
Q Heat transfer rate, kW
Q_{ev} Cooling rate of the heat pump (evaporator heat transfer rate), kW
Q_a Heat gain of the cooled panel from its back, kW
Q_e Heat gain of the cooled panel from its edges, kW

Kilkis

Q_y' Sensible cooling load of the space when cooled by radiant panel, kW
q_y Radiant panel cooling intensity, W/m²
R Thermal resistance between the hose skin and the slab material, m²K/W
R_{co} Total thermal resistance of the panel elements, m²K/W
r Linearization factor for the radiation heat transfer term, K³
T_a' Indoor design temperature for a radiant panel cooled space,°C
T_b Outdoor design temperature,°C
T_d Skin temperature of the hose,°C
T_{dp} Dew point temperature,°C
T_i Indoor temperature of the space above °C
T_{max} Maximum panel surface temperature,°C
T_{min} Minimum panel surface temperature,°C
T_o Reference temperature,°C
T_p Uniform radiant panel surface temperature, °C
$T_p'(x)$ Local surface temperature of the ceiling panel,°C
T_{re} Return fluid temperature in the cooling circuit,°C
T_{su} Supply fluid temperature in the cooling circuit,°C
T_{total} Thickness of the composite fin (including ceiling coverings), m
T_w Mean fluid temperature in the circuit $(T_{su}+T_{re})/2$, °C
U_c Convection heat transfer coefficient on the ceiling panel, W/m²K
U_r Radiation heat transfer coefficient on the ceiling panel, W/m²K
v_w Average fluid velocity in the hose,m s⁻¹
W Half of the net spacing between adjacent hoses $(M-D_o)/2$,m
W_p Work rate into the pump(s),kW
X Panel cooling efficiency, dimensionless
x_i Thickness of each ceiling element,m
Z Available hose length, m

Greek symbols:

α Convection heat transfer coefficient between the fluid and the hose inner skin, W/m² K
ΔE_t Total exergy loss rate, kW
γ Density of the cooling fluid,kg/m³
ΔT_w Fluid temperature drop,°C
ΔP Fluid pressure drop, kPa
η Fin efficiency, dimensionless
σ Stefann-Boltzmann constant, 5.67×10^{-8} W/m²K⁴
η_{pl} Flat plate solar collector efficiency, dimensionless

Subscripts:

ab Absorber
c Convective
ev Evaporator
co Condenser
ge Generator
r Radiant

1. INTRODUCTION

The necessity of an absorption cycle with conventional cooling units creates certain design and operational constraints on active solar systems in space cooling. Panel cooling systems which demand moderate fluid temperatures establish a better tie-in with active solar energy systems and make them more feasible.

Main attributes of radiant panel cooling:
i. Sensible cooling load is decreased
Indoor air temperatures can be selected 2 to 3°C higher than the standard indoor design temperatures without any sacrifice of the desired human comfort (Kilkis,1990).
ii. Peak loads are shaved by cool storage
Mass of the building, which are partially cooled with radiation heat exchange with the cooling panel as well as the panel itself, moderate daily changes in the cooling load.
iii. Moderate fluid temperatures are required
Ceiling panels often require moderate chilled fluid (water or brine) temperatures compared to conventional cooling systems. As a prime feature of the radiant system, the mean fluid design temperature hardly gets below 14°C due to large radiant surfaces and reduced cooling loads. This also reduces the condensation risk on the supply and return circuits which are generally metallic pipes.
Although they are generally limited in scope, there are already existing theoretical and experimental studies on radiant panel heating systems either from the ceiling or floor. Unfortunately, lesser attention has been paid so far for radiant panel cooling systems especially in conjunction with active utilization of solar energy. Although radiant panel cooling is limited to sensible cooling, with

Kilkis

bove mentioned merits, and potential tages, it is a viable alternative cooling as reported by many authors very recently. s and Kosonen, 1992), (Kilkis, 1991,1993).

EORY

eat Extraction by a Radiant Ceiling Slab

cal in-slab type radiant ceiling cooling panel uction is shown in Figure 1. The heat ted from the conditioned space establishes nsible cooling effect. Therefore, the required load intensity bears a negative sign.

$$q_y = C \cdot (-Q_y')/A_p \cdot 1000 \quad (1)$$

load shaving factor, C depends on many eters like the climate, typical daily outdoor ature swings,and the cold storage capability building elements, in addition to the thermal ties and dimensions of the cooled slab. 1990) formulated an engineering rule for ining C .

g efficiency of the panel is:

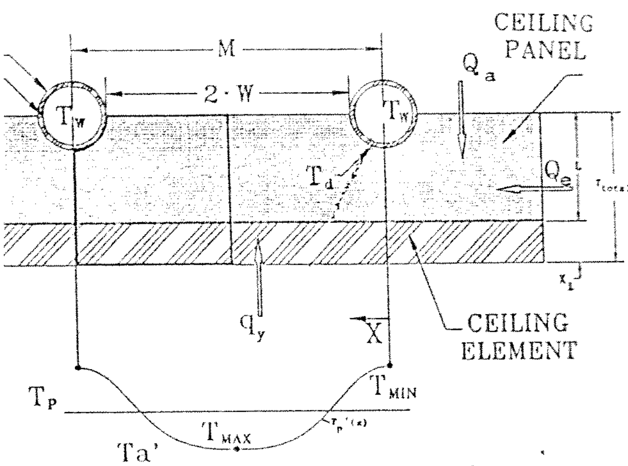
$$X = \frac{Q_y}{Q_y + Q_a + Q_e} \quad (2)$$

le cooling intensity is:

$$q_y = q_r + q_c = U_r \cdot (T_p - AUST) + U_c \cdot (T_p - T_a') \quad \{q_y \le 0 \text{ in cooling}\} \quad (3)$$

$$U_r = r \cdot F_r \cdot \sigma \quad (4)$$

$$U_c = (1\text{-}2.22 \times 10^{-5}h)^{2.627} \cdot (4.96/D_e)^{0.08} \cdot 2.67 \cdot |(T_p - T_a')|^{0.25} \quad (5)$$



1. Composite Ceiling Fin Model for Cooling

$$r = 4 \cdot ([T_p+273]/2+[AUST+273]/2)^3 \quad (6)$$

F_r is the simplified radiation interchange factor for $A_p/A_u \le 0.30$, (Raiss and Roedler,1969):

$$F_r \approx e \qquad \{A_p/A_u \le 0.30\} \quad (7)$$

and,

$$D_e = (4 \cdot A_r / L_r) \quad (8)$$

The area-weighted average design temperature of the uncooled indoor surfaces, AUST can be approximated by the following equation:

$$AUST = T_a' + c \cdot w \quad (9)$$

Here, c is the room position code. ie: c is one for an inner room, two for a room with one exposed wall, and three with two or more exposed walls.
w is a function of the outdoor design temperature:

$$w \approx 7/(45-T_b) \qquad \{26\ ^\circ C \le T_b \le 36\ ^\circ C\} \quad (10)$$

T_p can be solved for a required cooling intensity.

2.2 Steady State Composite Fin Model

If the peak load shaving feature is desired, the system must have a continuous circulation with temperature modulation. Furthermore, as the heat exchangers in absorption systems have long time constants, interrupted operation reduces the efficiency. Continuous operation also allows one to energize the generator at much lower temperatures, which increases the solar collector efficiency and the heat pump COP (Kreider and Kreith,1981). All these favor a steady state solution. The steady state composite fin model is shown in Figure 1 (Kilkis, 1993). Defining the fin efficiency:

$$\eta = \tanh(m \cdot W)/(m \cdot W) \quad (11\text{-a})$$

where,

$$m = \sqrt{\frac{U_r + U_c}{k_e \cdot T_{total}}} \quad (11\text{-b})$$

Here k_e is the equivalent thermal conductivity for lateral heat diffusion in the composite slab.

$$k_e = \frac{\left(\sum_{i=1}^{n_k} x_i k_i\right) + L \cdot k}{T_{total}} \qquad (12)$$

Superimposing the radiation and convection heat extraction intensities and solving for T_{min}:

$$T_{min} = \frac{q_y \cdot M + A \cdot AUST + B \cdot Ta'}{A+B} \qquad (13)$$

where A and B are the radiant and convective heat extraction rates for the composite fin having a surface area occupied by two adjacent hoses of unit length each:

$$A = [2 \cdot W \cdot \eta + D_o] \cdot U_r \qquad (14)$$

$$B = [2 \cdot W \cdot \eta + D_o] \cdot U_c \qquad (15)$$

Using T_{min} and taking the shortest path for heat diffusion :

$$T_d \cong T_{min} + q_y \cdot (R_{co} + (L - D_o/2)/k + R) \qquad (16)$$

R may be neglected.

$$T_w = \frac{q_y \cdot M}{X \cdot \pi} \cdot \left[\frac{1}{\alpha \cdot D_i} + \frac{1}{2 \cdot k_h}\right] \ln(D_o/D_i) + T_d \qquad (17)$$

Employing the turbulent flow model for water (Kreider and Kreith,1981):

$$\alpha = 1056 \cdot [0.02 \cdot \{T_w + 273\} - 4.06] \cdot v_w^{0.8}/D_i^{0.2} \qquad (18)$$

2.3 Solar Cooling

There are several methods for solar cooling. The cooling effect can be produced directly by an absorption cycle (Wilbur and Mitchell,1975), a regenerative desiccant process (Lunde,1976), (Roy and Gidaspow,1974), or a steam jet system. Absorption cooling systems require essentially only a heat source and has become one of the likely candidates for solar cooling. With recent developments in radiant panel cooling technology, which increase the collector efficiency and COP of the absorption heat pumps, the absorption cycle seems to become economically feasible. This system may be complemented by a conventional air handling system (Wilkins and Kosonen, 1992)

2.4 Absorption Heat Pumps

Water-ammonia systems require relatively high solar collector water outlet temperatures like 115°C for optimum operation (Shiran an et.al,1982). A water-lithium bromide system i simpler than the water-ammonia system an operates at a higher cooling ratio and require smaller heat exchanger surfaces, but may involv other design problems. (Karakas and et. al, 1990). For a minimum cost, the generator temperatur should be relatively high. (Alizadeh and et.al,1979) This may impose limitations on the utilization o flat plate collectors. At low solar insolatio levels, an auxiliary generator heater may b required. Due to low regenerating temperature required in a radiant system, silica gel-wate absorption systems as reported by Kluppel an Gurgel (1988) is another alternative. A water ammonia system is shown in Figure 2. Evaporato provides the necessary heat extraction through th radiant cooling panel circuit. Flat plate collector provide the necessary heat, for energizing th generator. A storage tank may also be used. Hea can be delivered on demand from the solar collecto tank.

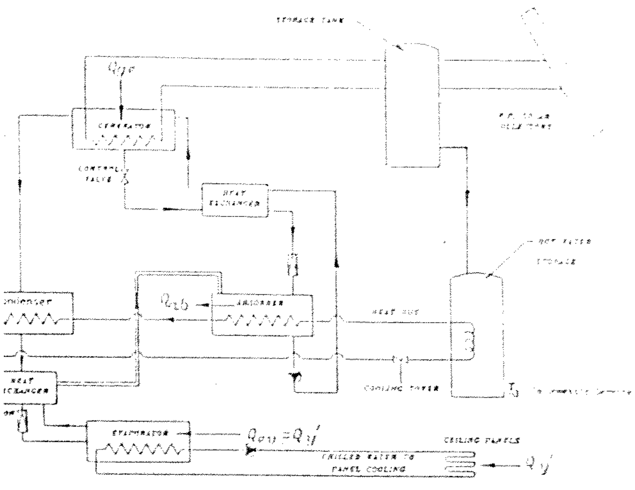
2.5 Performance Enhancement by Using Radia Panel Cooling

In design and optimization of the performance of a absorption cooling system, the most importa variable which has to be taken into account is th temperature of the generator, because the oth parameters of the system depend on the existi initial conditions and are fixed. The generat temperature affects the COP and ECOP of th system both for cooling and heating, as well as th flat plate solar collector efficiency, when oth design conditions are fixed. The required generat temperature however depends upon the temperatu of the chilled fluid to be produced for cooling. Th system, which requires very moderate chilled flu temperatures, allows reduction of the generat temperature at will.

$$COP_h = \frac{Q_{ab} + Q_{co}}{Q_{ge} + W_p} \qquad (19)$$

$$COP_r = \frac{Q_{ev}}{Q_{ge} + W_p} \qquad (20)$$

Kılıç, (4)



2. General Arrangement of Solar Absorption Cycle Radiant Cooling System

exergetic coefficient of performance for
ȝ and cooling are:

$$ECOP_h = \frac{Q_{co}(1-T_o/T_{co})+Q_{ab}(1-T_o/T_{ab})}{Q_{ge}(1-T_o/T_g)+W_p} \quad (21)$$

$$ECOP_r = \frac{Q_{ev}(1-T_o/T_{ev})}{Q_{ge}(1-T_o/T_g)+W_p} \quad (22)$$

ɔnship between $ECOP_h$ and T_{ge} is shown in 3 for a water-ammonia absorption heat (Ataer,1991). This figure reveals that there optimum generator temperature for maximum :ient of performance. This temperature is ɔn for COP and ECOP figures, and decreases ıe evaporator temperature. This feature also ates the need for an auxiliary generator The maximum $ECOP_h$ increases from 0.48 to 0.55 if T_{ev} increases to 10°C from 0°C. This se may be enhanced if the mean chilled water ·ature can be further increased to 15°C at an se of reducing the spacing between the hoses. 4 shows the variation of COP_r with T_{ev}. T_{ab}: 30 °C and T_{co}: 28 °C condition in this the optimum T_{ge} value drops from 75 °C to for T_{ev}: 12.5 °C. If absorber and condenser ·atures will be higher as listed in Figure 3, duced T_{ge} value will coincide with the 67 °C as predicted from Figure 3. For the latter case, COP_r is about 0.70, and indicates a noticeable increase of COP_r when compared with T_{ev}: 7.5 °C case ($COP_r \approx 0.65$). Generally, the temperature drop between the heat source and the generator is around 10°C (Alizadeh and et.al,1979). Taking this into consideration, the exit temperature from the solar collectors may be as low as about 80°C for a satisfactory operation. This increases the collector efficiency (Kreider and Kreith, 1981):

$$\eta_{pl} = F \cdot [X_1 - X_2((T_{ge}+10)-T_b)/I_{pl}] \quad (23)$$

X_1 and X_2 are the collector design variables.

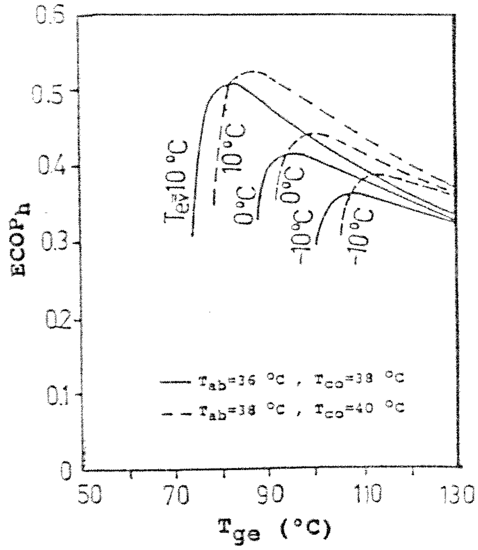


Figure 3. Change of $ECOP_h$ with the Generator and Evaporator Temperature

2.6 Computer Program

A computer program was developed for an in-slab radiant ceiling panel. The algorithm follows the theory given in this article. The dew point temperature at given indoor conditions is calculated and compared with the minimum ceiling surface temperature at design conditions. The following design constraints are recognized:

(i) $T_{min} > T_{dp}$	(ii) $T_{su} \geq 5°C$
(iii) $R_e > 6000$	(iv) $0.5 \leq v_w \leq 1.5\ ms^{-1}$
(v) $X \geq 0.80$	(vi) $3 \leq \Delta T_w \leq 7°C$
(vii) $\Delta P \leq 50$ kPa	(viii) $A_p/M \cdot 1.1 \leq Z$

Using the computer program, it is possible to generate specific design nomographs, too.

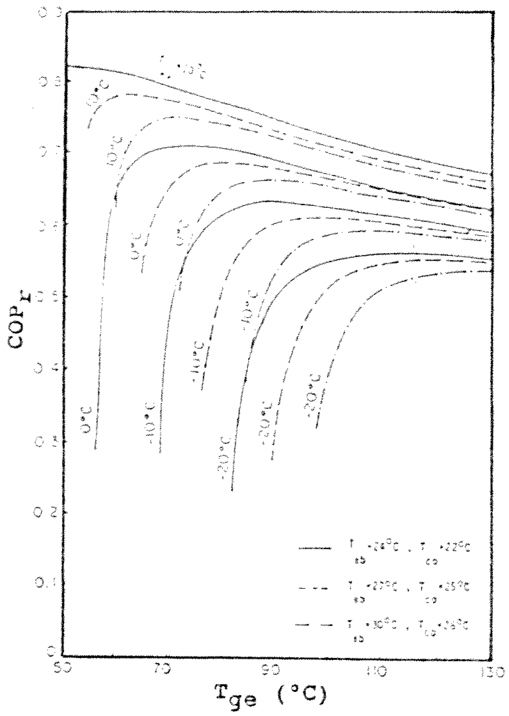


Figure 4. Change of COP$_r$ with the Generator and Evaporator Temperature

A case study for a solar house of 81 m^2 and a 6.5 kW sensible cooling load revealed that rubber hoses spaced at 0.15 m in a concrete slab can meet the sensible load at T_w= 19.8°C. Excess humidity is handled by a conventional unit. The load has dropped to 3.5 kW after applying the provisions for a radiant system and insulating the back of the slab. With this fluid temperature, the COP$_r$ of a water-ammonia absorption heat pump with 3.5 kW cooling capacity was calculated to be around 0.70 at an absorber temperature of 38 °C. With this figure and an 80% efficiency for solar energy storage system including the power consumption of the circulating pump of the heat pump, and 65% collector efficiency, the necessary collector area will be 30 m^2.
20 flat plate collectors with 1.5 m^2 absorption area each were selected. If a conventional cooling system was to be installed, the projected collector area would be 41m^2. COP$_r$ would drop to about 0.6 (Kilkis, 1993).

3. DISCUSSION OF RESULTS AND CONCLUSION

An analytical model for radiant ceiling cooling heat transfer under steady state conditions was developed. The computer program using this model revealed that the mean chilled fluid temperatures may be as high as 18 °C, especially if the slab material is concrete. This is a high temperature when compared with other conventional cooling systems. When coupled with the cooling load reducing and shaving features of radiant ceiling cooling, this moderate chilled fluid temperature requirement guarantees noticeable improvements in the COP of absorption type heat pumps and solar collector efficiencies. In the same manner the total exergy loss of the system decreases (Kilkis, 1993). Design calculations indicated that the performance of both the heat pump and solar collectors increase. These increase the competitiveness and feasibility of such alternative cooling systems.

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AN INVESTIGATION OF THE PARAMETERS EFFECTIVE UPON THE PERFORMANCE TESTING OF RADIANT PANELS

Prof.Dr.Birol KILKIS*
Heatway Radiant Floors and Snowmelting

ABSTRACT

Radiant panel heating and cooling is the most rapidly growing space conditioning system among other alternatives due to distinct attributes like energy efficiency and better tie-in features with alternative energy sources and heat pumps [1,2]. Although the fundamental principles of radiant panel systems are quite well established,the actual performance depends upon many factors that needs experimental verification both for design and rating purposes of panel products. Today,most of the engineering designs are carried out with certain assumptions and simplification which have not been fully justified with theory or experimental data. Although more comprehensive design tools are available [3], experimental rating of the panels seem to be still desirable in evaluating and sizing a radiant panel system. Such an experimental method needs to be able to simulate,manipulate and control all the pertinent factors. This presentation first identifies all the factors affecting the heat output or extraction capacity of a radiant panel under anticipated steady state operating conditions.

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Then, these factors will be classified and prioritized.
The second part of the presentation will mention an algorithm for interpreting the test data. A short survey of foreign testing standards will also be provided.

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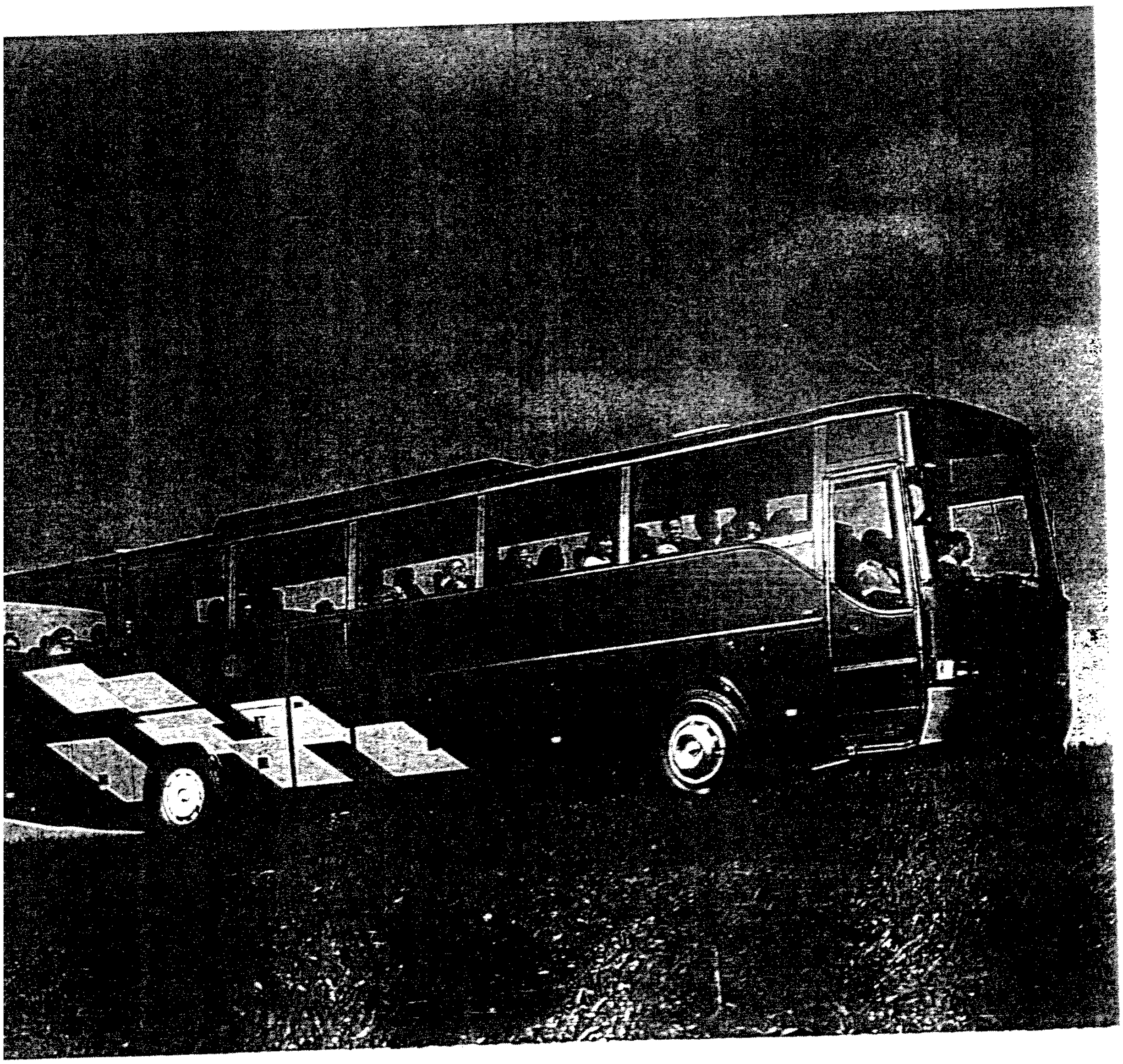


KATKILAR

Prof. Dr. Birol KILKIŞ

1949 Yılında Ankara'da doğdu. 1970 yılında ODTÜ Mak. Müh. Bölümünden mezun oldu. 1972 yılında Akışkanlar Mekaniği dalında Von Karman Enstitüsünden şeref diplomast aldı. 1981 yılı TÜBİTAK teşvik ödülünü kazanan Dr. Birol Kılkış halen ODTÜ'de görevli olup, aynı zamanda ABD ASHRAE kuruluşunda Panel Isıtma/soğutma konularında iki ayrı teknik kmomite üyeliğini sürdürmektedir. 166 adet bilimsel yayını mevcuttur. Evli ve iki çocuk sahibidir.

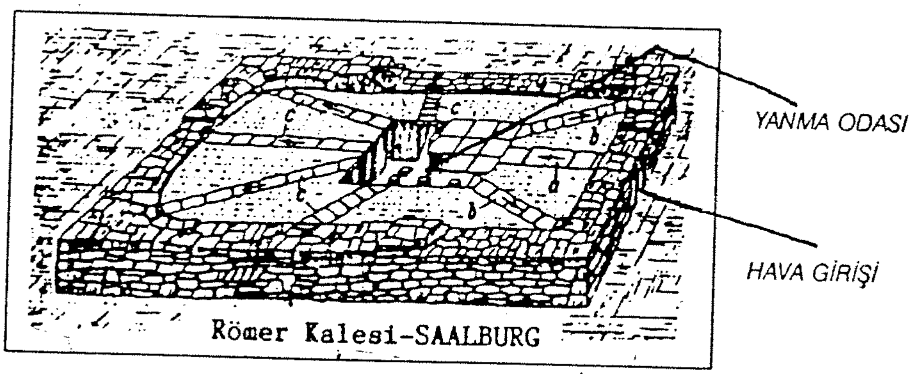
YERDEN ISITMA TEORİ ve UYGULAMA ESASLARI

GENEL TANITIM

Tarihçe

Döşeme, tavan veya duvar gibi yapı elemanı yüzeylerinden mekan ısıtılması uygulamaları çok eski çağlara kadar uzanmaktadır. İlk olarak M.Ö. 1200 yıllarında Güney Batı Anadolu'da, Kral Arzowa'nın Antalya yakınında Perge'de inşa ettirdiği sarayında bu sistemi kullandığı yapılan kazılardan anlaşılmaktadır. Yine aynı tarihlerde Çin'de ve Tibet'de, Ti-Kong ve Kao-Kang çağlarında benzer uygulamaların olduğu bilinmektedir. Pompei ve Saalburg'da yapılan kazılardan M.Ö.80 yıllarında Romalıların duvar içi ve döşeme altı kanallardan sıcak hava geçirerek yapıları ısıttıkları ortaya çıkmıştır (Şekil 1) Eskiden sıcak havanın duvar içlerinden geçirilmesi şeklinde yapılan ısıtma sistemleri günümüzde daha çok sıcak suyun beton içerisindeki borularda dolaştırılması şeklindeuygulanmaktadır.Buna ek olarak döşeme içindeki elektrik direnç kabloları ile mevzii ısıtma uygulamaları da yapılmaktadır.

Sıcak havalı sistemlere ise Doğu Anadolu'da, Kars dolaylarındaki 19. Yüzyıl taş yapılarında hala rastlanmaktadır. Çağdaş anlamda ilk uygu-



a- Taze Hava Menfeci, d- Döşeme Sıcak Hava Kanalı,

c- Duvardan Isıtma Kanal Geçişi (Duvardaki kanallar aynı zamanda baca görevi görür.

Şekil 1. Döşeme İçi Kanallar İle Isıtma (Eski Roma Çağı)

lama İngiltere'de A.H. Barker (1870-1954) tarafından yapılmış ve 1907 yılında 28477 numaralı Patenti "Panel Isıtması" ünvanı ile almıştır. Crittall firması ile birlikte Bolinbroke Hastanesi, Liverpool'da Adelphi Oteli, Aldwych'de Bush binası gibi değişik değişik tür yapılarda uygulamıştır. Aynı firma, Almanya, Finlandiya, Fransa, Hollanda, İtalya ve daha birçok ülkede Crittall sistemi adı ile uygulamalar yapmıştır. Kıta Avrupasında ilk firmalar ise 1929 yılında Hollanda'da ve 1930 yılında Almanya'da kurulmuştur. Yine aynı yıllarda, Amerika Birleşik Devletlerinde Indiana'da benzer bir yöntem, okul ısıtmasında uygulanmıştır. Almanya'da ise panel ısıtmasının esasları ve temel teorisi daha 1930 yılı başında ortaya konmuş ve kitap halinde yayınlanmıştır. Yüzey ısıtması özellikle kar ve buz eritmesi, soğutma, hastane ve spor salonu ısıtması gibi alanlarda öncelikle Amerika Birleşik Devletleri'nde ve Kanada'da uygulana gelmektedir. Ülkemizde ise münferit uygulamalar 1950'li yıllarda başlımış olup, bakır borular ile yapılan en belirgin uygulama Ankara'da inşa edilen Kocatepe Camiidir.

1960'lı yıllardan sonra Dünyada daha da hız kazanan yerden ve diğer yüzeylerden ısıtma, Ülkemizde de 1970'li yıllardan beri gelişen plastik teknolojisi ve hızlanan enerji tasarrufu ve dolaylı olarak hava kirliliğini önleme çabaları ile birlikte giderek artan bir hızla uygulamaya girmektedir. Şu anda Almanya'da yeni inşa edilen konutlarda panel ısıtma uygulama oranı %50'yi geçmiştir. Bugün birçok Avrupa ülkesinde olduğu gibi ülkemizde de yerden ısıtma tanınma ve denenme süresini çoktan geçirmiş ve tercih edilen, aranan bir sistem olarak günlük hayata yaygın bir şekilde girmiş bulunmaktadır.

Bununla birlikte hem tasarım, hem uygulama aşamasında, hatta terimlerde bir karmaşa hüküm sürmekte olup, 1993 yılı itibariyle uygulamaya geçecek olan iki Türk standardının hem tasarımcılara hem uygulayıcılara büyük ölçüde yardımcı olacağı muhakkaktır. Bu standardlar TÜBİTAK'ın desteklediği bir araştırma projesi çerçevesinde geliştirilmiş olup, özgün niteliğinden dolayı ISO'ya da teklif olarak sunulacaktır.

Düşemeden Isıtma Sistemi Nedir ?

Yerden Isıtma Sistemi, ülkemizde yeni, modern ve ekonomik bir ısıtma seçeneğini oluşturmaktadır. Herşeyden önce yerden ısıtma, insan anatomisine ve ısıtma tekniğine en uygun ve en verimli ısıtma sistemidir. Getirdiği enerji tasarrufu ile yerden ısıtma Ülke ekonomimize de katkıda bulunmaktadır. Enerji kaynaklarının giderek azalıp pahalılandığı Dünyamızda rakipsiz bir uygulama olarak dikkati çekmektedir.

Ayrıca Amerikan Underwriters Lab (UL) kalite belgesi denetiminde özel katmanlı lastik borular da başarı ile uygulanmaktadır. Klasik ısıtma sistemlerinin aksine daha düşük, örneğin 45°C - 50°C gibi su sıcaklıkları mekan ısıtmasında yeterli olduğu için, atık ısı, güneş enerjisi, jeotermal enerji gibi kaynakların ve ısı pompası gibi yeni ve alternatif sistemlerin en verimli bir şekilde kullanılmasına imkan tanımaktadır. Döşemeden, duvardan, tavandan ısıtma olabileceği gibi döşeme veya tavandan soğutma, hamam, havuz gibi özel mekanların ısıtması, sera ısıtması, toprakta ısı depolaması, köprü, havaalanı, yaya kaldırımı gibi yerlerde buz ve kar eritmesi gibi birçok değişik uygulamalarda vazgeçilmez bir sistemdir. Sıcak sulu sistemde ana ısıtma elemanı yapı malzemesi içersinden geçirilen yerden ısıtmaya uygun termoplastik veya özel lastik Polietilen Cross-link (PE-X), veya Polibüten (PB) termoplastik malzeme ile malzemeden mamul borudur. Polipropilen termoplastik malzeme ise genelde genelde iki türlü olabilmektedir. Birinci tip polipropilen malzeme Homopolimer (Tip 1) olarak tanımlanır ve özellikleri itibarı ile ve DIN 4726 standardına göre yerden ısıtmada kullanılamaz. İkinci tip polipropilen malzeme ise Ko-polimer (Tip 2) olarak tanımlanır. Random Ko-Polimer boruların kullanımı uygundur. Polietilen boruların ise cross-link işlemine tabi tutulmuş (ağdalanmış) olması gereklidir. Adı geçen standarda göre polietilen boru panel ısıtmada kullanılamaz. Ağdalanma işlemi ise üç yöntemle olabilmektedir. En uygunu engel yöntemidir.

Standardlara uygun olmayan boru kullanımı genellikle tamiri ve düzeltilmesi mümkün olmayan sorunlar yaratmaktadır. Mikrop ve toz sirkülasyonunun sakıncalı olduğu hastane, okul, kreş ve laboratuvar gibi mekanlarda radyatörlü ve konvektörlü sistemlerin yerine yerden ısıtma sisteminin kullanılması ile daha sağlıklı ve hijyenik bir ortam sağlanmaktadır. Sistemin diğer elemanları ise naylon branda, yüksek dansiteli strapor veya perlit ve bunun gibi yalıtım malzemeleri, boru tutturucu veya ek olarak çelik hasır, dağıtım kollektörü ve diğer yardımcı elemanlardır.

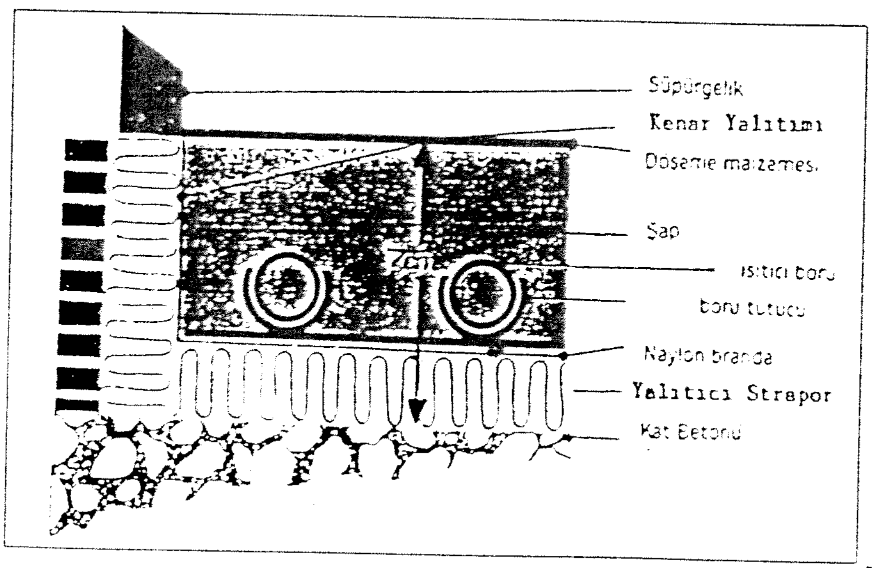
Döşeme Konstrüksiyonu

Yenden ısıtma sisteminde mermer, seramik, parke, halı gibi her türlü kaplama malzemesi kullanılabilir. Kullanılan özel şap, naylon branda ve strapor gibi yalıtım malzemeleri yapıda önemli ölçüde ek ısı ve ses yalıtımı sağlar. Şekil 2'de görülmektedir.

Bu sistemde oda ve kat sıcaklıklarının bağımsız ve merkezi kat kollektörlerinden kontrol imkanının bulunması, enerji tasarrufu ve daha iyi bir konfor sağladığı gibi toplu konutlarda işletme giderlerinin daha objektif olarak paylaştırımına imkan tanır. Isı Payölçer uygulaması çok daha kolaydır. Sistemini tabanda meydana getirdiği ek yükselti sadece 4 cm. kadardır.

Isıtma boruları tamamen döşeme içinde olduğundan mimaride yepyeni serbestilere, seçeneklere ve uygulamalara, iç mekan ferahlığına, konfora ve büyük ölçüde temizliğe imkan tanımaktadır. Radyatör ve benzeri bireysel ısıtıcılar olmadığından yerden tasarruf sağlar.

Bu tasarrufun yapı maliyetine oranı % 5 dolayındadır. Döşeme sıcaklığı aynı kişilerce sürekli kullanılan döşemelerde en çok 29°C alındığından canlılara hiçbir olumsuz etkisi olmadığı gibi aksine radyatör, soba gibi kızgın herhangi bir yüzey olmadığın-



Şekil 2. Tipi Döşeme Konstrüksiyon Kesiti

dan özellikle çocuklar ve hastalar için daha emniyetli bir yapı sunar. Aynı kişilerce üzerinde sürekli gezinilmeyen kenar bandı gibi döşemelerde bu sıcaklık sınırı DIN 4725'e göre en çok 35°C dir.

Maliyet

Döşemeden ısıtma sistemi klasik ısıtma sistemlerine göre ilk tesiste ucuz olduğu gibi, daha önemlisi sürekli rejimde uygulama sırasında % 20-30 hatta daha fazla enerji tasarrufu sağlar. Bu tasarrufun ana nedenleri yerden ısıtmanın termo-fiziksel özelliklerinden kaynaklanmaktadır.

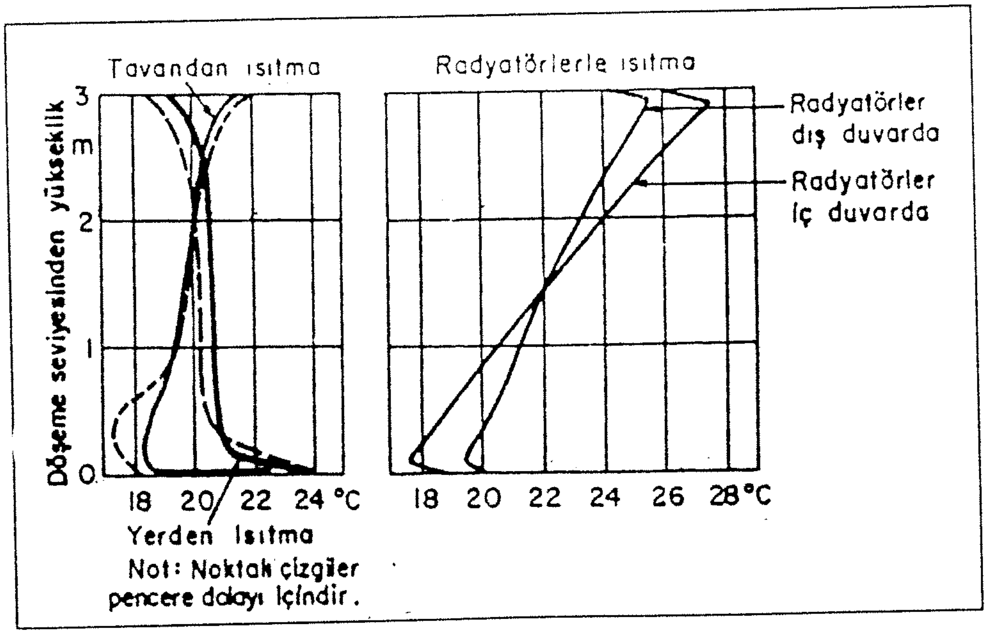
Sistemin Avantajları

Yerden ısıtma sistemine ilişkin avantajlar 11 ayrı grupta sınıflandırılabilir. Bunlar:

1. Enerji Tasurrufu:

Yerden ısıtılan bir yapıda oda içindeki sıcaklık dağılımları homojen ve odalar arasındaki sıcaklık farkları ise daha azdır. Yerden ısıtmada sıcak hava tavana yakın bölgelerde birikmez, bu nedenle 1.0, 1.5 m kadar yükseklikte insanların hissetmesi gereken konfor sıcaklığı standart değerlerden daha düşük, örneğin 18°C, hatta 16°C olarak kabul edilebilir. Odada hava dolaşımı ise daha yavaştır. Bu dolaşım yavaşlığı ve daha homojen sıcaklık dağılımı, infiltrasyon ısı kayıplarını büyük ölçüde azaltır. İç hesap sıcaklıklarının aynı konfor koşulları için daha düşük tutulması ile daha küçük ısıtma tesisi yeterli olduğu gibi işletme masrafları da büyük ölçüde azalır. Binanın ısıl ataletinden daha çok yararlanıldığından yani ısı bina yapı elemanlarında özellikle ısıtılan döşemede depolandığından bina konforu ani sıcaklık düşüşlerinden, aşırı soğuk günlerden fazla etkilenmez. Bu nedenle de bu gibi aşırı soğuk günler ısı yükü hesaplarında gözönünde tutulmaz.

Klasik bir ısıtma sistemi ile yerden ısıtma uygulaması yapılan bir odadaki sıcaklık dağılımı Şekil 3'de gösterilmiştir. İlk şekilde yerden ve tavandan ısıtma durumundaki sıcaklık dağılımı gösterilmiştir. Özellikle yerden ısıtmada odada çok düzgün bir sıcaklık dağılımının oluştuğu açıktır. Ayrıca döşemeden ısıtmada ısıl konfor yaklaşık % 60 ısıma % 40 ısı taşınımı ile sağlanır Bu oran insan vücudunun ısıl konfor için ihtiyaç duyduğu orantılar ile aynıdır. Bir radyatörlü sistemde ise bo oranlar sırası ile yaklaşık %20 ve



Şekil 3. Isıtılan Bir Mahalde Dikey Sıcaklık Dağılımı

%80'dir. Dolayısı ile döşemeden ısıtma, ısıl konfora ek bir sübjektif konfor boyutu daha eklemektedir.

ISO standardlarına göre insan boyunca sıcaklık farkı 3°C'de az olmalıdır. Görüldüğü üzere kaloriferli bir sistemde bu konfor koşulunun sağlanması oldukça zordur. Ayrıca yerden ısıtmada duvarlar da daha homojen ısınır ve sıcaklık farklılıkları daha azdır. Bu nedenle de konfor şartları daha kolay sağlandığından odayı aşırı ısıtmaya gerek kalmaz.

Duvar ve yer iç yüzey sıcaklıklarının karşılaştırması ise Şekil 4'de gösterilmiştir. Duvarlar daha sıcak olduğundan yerden ısıtmada konfor koşulu daha kolay elde edilmiş olur.

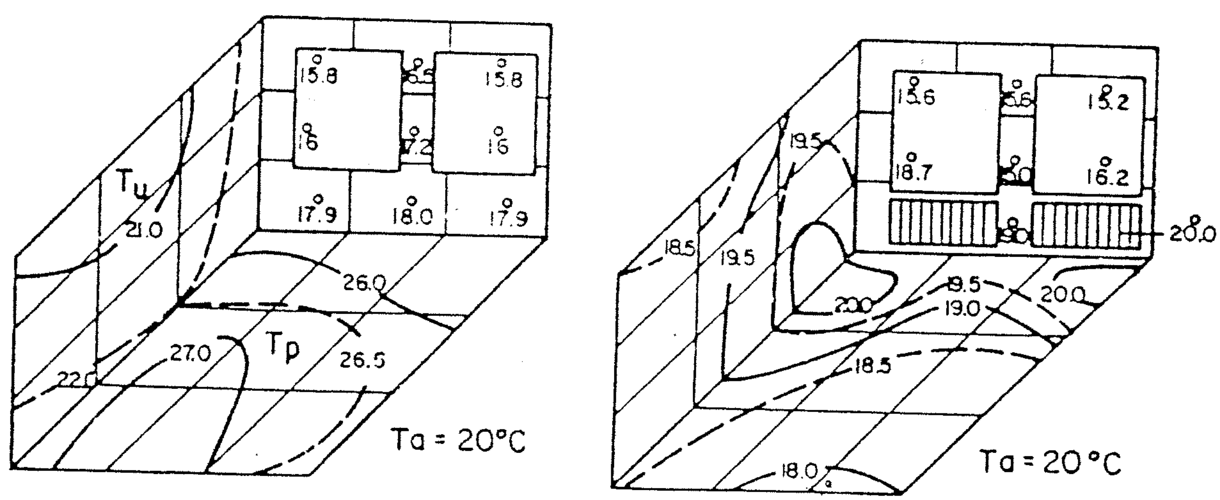
Şekilden görüldüğü üzere duvar iç sıcaklıkları radyatörlü sisteme oranla yaklaşık + 2°C daha fazla olabilmektedir. Bu yolla hem insanların ısınma duyularına en fazla hitap eden ışıma (radyasyon) ile ısınma oranı artmakta hem de binanın ısıl ataleti daha iyi kullanılmış olmaktadır. Buna ek olarak genellikle dış duvar ve pencere altlarına konan radyatörlere yakın dış duvarlarda olduğu gibi bir aşırı ısınma söz konusu olmadığından ısı kaçaklarıda azalmaktadır.

Yerden ısıtma "kendi kendini ayarlayabilen" bir sistemdir. Kısacası ısıtılan bir döşemenin ısı arzı iç sıcaklık ve duvar sıcaklığına göra kendi kendine azalır veya çoğalır. Bu nedenle pahalı kontrol ve ayar cihazlarına gerek duymadan, hatta insan müdahalesine gerek bırakmadan dış hava sıcaklığına, dolayısı ile ısıtma ihtiyacına göre kendini döşeme yüzeyi sıcaklığı aracılığı ile belirli bir ölçüde kontrol eder. Eğer ısı yükü azalmış ise sistemdeki ısı mekana dağılmadan ya ısı yükü daha fazla odalara gider veya kazana dönmüş olur (madde 9).

Işıma yüzey alanları fazla olmasının yanısıra, yatay bir ısıtıcı plaka konumundaki döşeme yüzeyi en ideal ısı taşınım katsayısına sahiptir. Yukarıda açıklanan tüm konuların ışığı altında yerden ısıtmanın en ekonomik ve bilinçli ısıtma yöntemi olduğu ortaya çıkmaktadır.

2. Isıl Konfor ve Sağlık:

İnsan bedeni genellikle 1-1.5 m yükseklikteki hava sıcaklığından etkilendiği gibi en çok ışıma ile ısınma ihtiyaçları-



a - Yerden Isıtma.

b - Radyotörlü Sistem

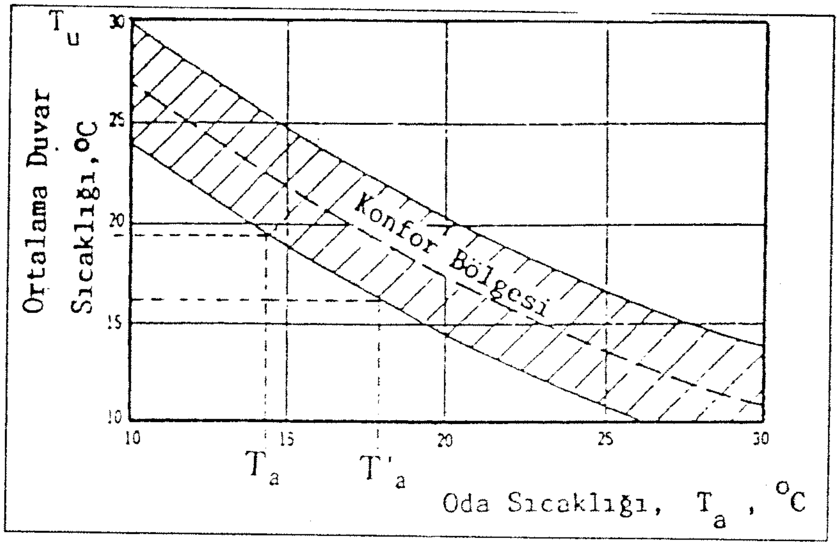
Şekil 4. Yerden Isıtmada ve Radyotörlü Sistemde Duvar ve Döşeme Sıcaklıkları nı giderirler. Yerden ısıtmáda beden seviyesinde daha sıcak bir havanın bulunması, insanı çevreleyen duvarların iç yüzeylerinin daha sıcak olması nedeni ile ışıma ısı kazançlarının daha yüksek olması, ayrıca soğuk hava cereyanlarının ve de pencere ve dış kapılardan sızan havanın daha az olması nedeni ile, yerden ısıtmada insanların konfor gereksinimi daha iyi karşılanır.

Bina içinde soğuk ve cereyanlı koridor ve mahallerin olmaması da ayrı bir konfor faktörüdür. Yerden ısıtmada yaşanılan hacim içersinde hareketli hiçbir mekanik ısıtma aksamı yoktur. Bu yüzden üflemeli konvektörler ve sıcak hava apareylerinde olduğu gibi gürültü ve titreşim de söz konusu değildir. Ayrıca yerden ısıtmada döşemeye serilen yalıtıcı starapor köpük ve naylon branda ideal bir ek ısı yalıtımı olduğu kadar ses yalıtımı da sağlar. Bunun sonucunda katlar arasında gürültü iletimi azaldığından, daha sessiz ve sakin bir ortam oluşur. Radyatör, konvektör gibi çok sıcak ve sert yüzeyler olmadığı için özellikle yaşlıları ve çocukları ilgilendiren cilt yanması ve çarpıp yaralanma tehlikeleri de tamamen bertaraf edilmiştir. Odaların temizlenmesi ise daha kolay yapılabilmektedir. Ayrıca kuru ve soğuk zeminlerde mikroorganizmaların üremesi sorunu yoktur. Bu nedenle de yerden ısıtma, hastanelerde özellikle tavsiye edilmektedir. Ayrıca yüksek sıcaklıktaki radyatör yüzeyleri arasından geçerken ısınan hava nemini kaydeder ve kurur. Giderek kuruyan bir hava ortamında çeşitli rahatsızlık belirtileri ve havayı nemlendirme gereği ortaya çıkar. Bu amaçla ek nemlendirme cihazları gerekebilmekte veya odayı havalandırma yolu seçilmektedir. Bu ise insanları ya kuru bir ortamda oturmaya ya da geçiçi de olsa pencereleri açarak üşüme veya ısı kayıplarını arttırma ikilemi ile başbaşa bırakmaktadır. Yerden ısıtma sisteminde ise hacimlerin nemlendirilme ihtiyacı minimum düzeydedir. Sıcak yüzeylerde havanın pişmesi (baking-effect) nedeni ile koku oluşması gibi bir sorun döşemeden ısıtmada sözkonusu değildir. Diğer sistemlerde ise bu sorun sık sık odayı havalandırma gereğini doğurur. Klasik sistemlerde hava dolaşımının daha hızlı olması nedeni ile radyatör yüzeylerinden kalkan tozlar bütün odaya yayılmaktadır. Yerden ısıtmada ise hava hareketlerinin daha zayıf olması nedeni ile ısıtılan mahaller içersinde toz ve mikrop dolaşım tehlikesi minimum düzeydedir. Sırf bu nedenledir ki sağlık bakımından daha önemli yapılarda örneğin hastaneler, sanatoryumlar, koğuşlar, düşkünler yurdu, kreşler ve bakımevlerinde öncelikle yerden ısıtma tavsiye edilmektedir. Bu gerçek ise günlük hayatta karşılaşılan "yerden ısıtmada toz daha çok kalkar" kuşkusunu bertaraf etmektedir.

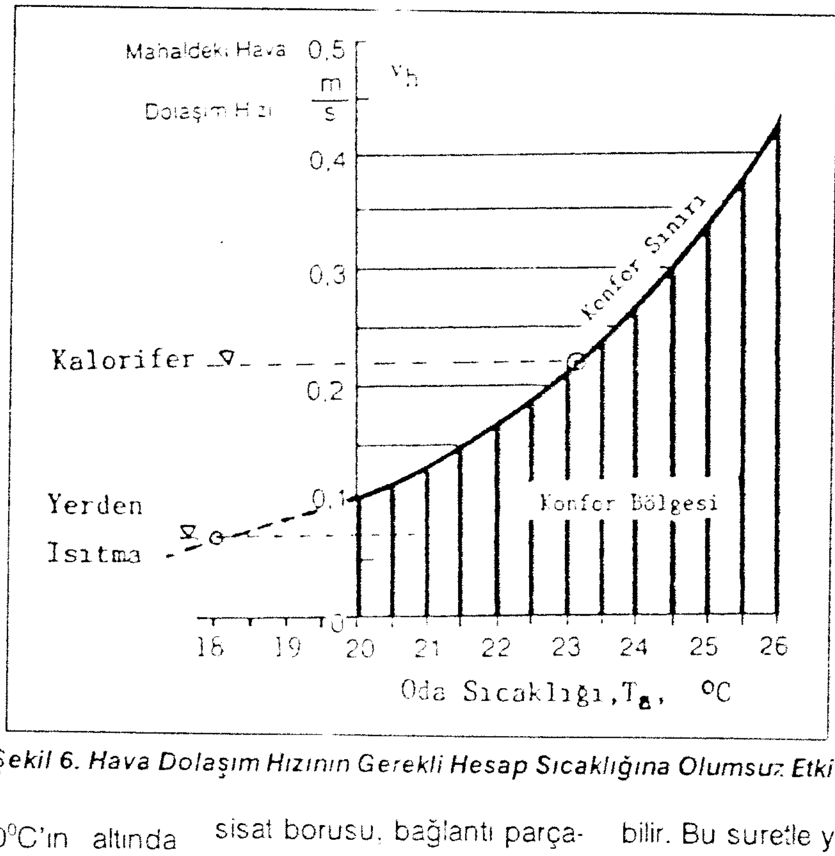
Her dairenin kendine ait bir veya birkaç kollektör dolabının bulunması nedeni ile ve her oda modülasyonu (yerden ısıtma birimi) ayrı vana ile kontrol edilebildiğinden konfor gereksinimine göre bağımsız ve tamamen o dairede yaşayan sakinlerin isteğine göre kolaylıkla ayar yapma imkanı da yerden ısıtmanın ayrı bir sübjektif konfor faktörüdür. Şekil 5'de konforlu bir odada gerekli iç sıcaklığın duvarların ortama iç yüzey sıcaklığına olan bağımlılığı görülmektedir.

Görüldüğü üzere yerden ısıtmada duvar sıcaklıkları daha yüksek olduğu için oda hesap sıcaklığı (T_a) daha az alınabilir. Buna karşın kaloriferli bir sistemde aynı hesap sıcaklığı en az 18°C olmalıdır. Şekil 6'da ise hava dolaşım hızının (v_h) konfor için gerekli oda sıcaklığına olan etkisi görülmektedir. Hava dolaşımı hızlandıkça konfor koşulları daha yüksek hesap sıcaklıkları gerektirmektedir.

Yerden ısıtma sistemlerinde genellikle 0.1 m/s. dolayında bir hava dolaşımı söz konusudur. Bu nedenle de kon-



Şekil 5. İç Duvar-Oda Sıcaklığı-Konfor İlişkisi



Şekil 6. Hava Dolaşım Hızının Gerekli Hesap Sıcaklığına Olumsuz Etkisi

for sıcaklığı 20°C'ın altında olabilmektedir.

3. Mimari ve Estetik Avantajlar:

Yerden ısıtmada göze görünür hiçbir ısıtıcı eleman, tesisat borusu, bağlantı parçası yoktur. Ana kontrol kollektörü ise bir dolap içinde kolaylıkla gizlenebilir. Isıtıcı aparey ve tesisat borusu olmadığı için mekanın her yeri istenen her hangi bir amaçla kullanılabilir. Bu suretle yerden ısıtma yer ve hacim tasarrufu da sağlar. Mimar ve tesisat mühendislerine ısıtıcıların konumlanması sorunundan uzak, yeni serbestlikler ve boyutlar kazandırılmakta, estetik yönünden yeni imkanlar tanınmaktadır.

4. Uygulama ve Bakım Kolaylığı

Uygulaması kolaydır. 2 kişilik bir ekip bir günde 150 m^2 lik ısıtma yüzeyini rahatlıkla döşeyebilir. Daha az kolon borusu kullanıldığından kolonlar daha çabuk çekilir. Özel PE-X borular ve özel lastik hortumlar normal işletme koşullarında çok uzun ömürlüdürler. Malzeme termoplastik veya lastik olduğundan birbirine benzemeyen metalik malzemelerde görülen katodik korozyona rastlanmaz. Daha az boru bağlantısı olduğu için sızdırma, paslanma ve arıza olasılığı daha azdır. Ayrıca iç yüzeyleri çok kaygan olan borular zamanla tıkanmaz ve çürümez. İyi bir işçilik ile uygulanan bir ile sistem pratik olarak bakım ve onarım gerektirmez. İçersinde unutulsa dahi radyatörlere kıyasla ısıtılmayan zamanlarda donma tehlikesi yer döşemesi boruları için daha azdır. Sistemin hava yapma olasıġı daha az olup, kollektörden hava alınması kolay olduğu gibi, bu otomatik olarak da yapılabilir. Aynı kollektöre çok uzun ve çok kısa boru parkurları bağlansa bile basınç dengelenmesi kendiliğinden oluşur.

Yapı duvarları iç yüzeyleri radyatörlü sistemlerde olduğu gibi islenmez, kir tutmaz. Bu nedenle oldukça sık ve pahalı badana ve boya giderleri söz konusu değildir. Temiz bir ortam gerektiren bilgisayar, televizyon gibi pahalı elektronik aygıtların devrelerinde zamanla kir ve toz birikmez.

5. İşletme Kolaylığı:

Sistem belirli ölçüde kendini ayar edebildiği için aslında pahalı bir kontrol tesisatına gerek yoktur. Nominal yükte ve sürekli rejimde çalışan bir kazana 4 yollu bir motorlu vana ile su geri dönüşünü sağlamak yeterli olmaktadır. Borularda dolaşan sıcak su 50°C dolayında olduğu için aynı kazan ve vanadan sıcak su boylerini beslemek, gerektiğinde ise (çok soğuk dış hava koşullarında), boylerde depolanmış sudan ısıtmada yararlanmak mümkündür. Yakıt pay ölçerin daha basitce uygulanabilir olması nedeni ile apartmanlarda işletme giderlerinin daha kolay ve adil bir şekilde payedilmesi mümkün olmaktadır.

6. Mali Avantajlar:

Uygulamada en az % 20 enerji tasarrufu nedeni ile işletme giderlerinde ekonomi sağlar. Ayrıca su sıcaklığının 50°C civarında olması nedeni ile atık enerji kullanımı, ısı pompalı sistemlerin uygulanması gibi daha çok enerji tasarruf edici yöntemlerin daha etkin kullanıma ortam hazırlar. Daha küçük bir kazan tesisatı gerektiğinden, ilk yatırım da ekonomi sağlanması söz konusudur.

7. Çevre Sorunlarının Azaltılmasına Katkı:

Yerden ısıtmada klasik kazanlar kullanılsa dahi, enerji tasarrufu nedeni ile çevreye atılan parçacık ve zararlı gazlarda teorik olarak en az% 20 azalma söz konusudur. Ayrıca çevreyi kirletmeyen güneş enerjisi, toprak ısısı, atık enerji, jeotermal enerji, kuyu ve deniz suyu (ısı pompası ile) gibi ısı kaynaklarından etkin olarak faydalanmaya imkan ve ortam sağlar.

8. Yeni Uygulama Ufukları ve Isı Kaynağı Seçenekleri:

Yol, köprü, sera, hemzemin geçitlerde, kar ve buz eritme, cami, havuz ve hamam gibi yerlerde daha iyi bir konforun sağlanması, hacim soğutması gibi uygulamalarda aynı sistem, aynı beceri ve prensiplerle uygulanabilmektedir. Bu geniş yelpaze başka hiç bir ısıtma sisteminde mevcut değildir. Örneğin aynı ısıtıcı borulardan, yazın soğutma yapmak, boru ve bağlantılarında hiç bir düzeltme ve ayarlama yapmadan mümkündür. Sadece sıcak su yerine serin su geçirilmesi yeterli olmaktadır. Serin su sıcaklığının 7°-8°C yerine 12°-14°C olması, kışın da sıcak su sıcaklığının 90°-70°C yerine 45°-°C dolayında olması alternatif enerji kaynaklarının verimli ve daha sürekli kullanımına ve daha ucuz enerji depolama yöntemlerinin kullanılmasına imkan verdiği gibi teçhizatın daha küçük ve ucuz olmasında önemli rol oynar.

9. Kendi Kendini Kontrol ve Dengeleme İmkanı (Self Regulating System):

Yerden ısıtma sisteminde, olarak ısı arzı ısı yüküne göre kendi kendine değişir. Bunun nedeni döşeme yüzey sıcaklığının ısı arzında en büyük etken olmasıdır. Isıtma soğutma kapasitesi döşeme yüzey sıcaklığı ile ortam ve duvar sıcaklığı ile orantılıdır. Bu bağımlılık ışıma ısı transferinde daha da belirgindir. Döşeme yüzey sıcaklığı (26^0-29^{0}C), duvar iç yüzey sıcaklığı (16^0-18^{0}C) ve iç hava sıcaklığı (16^0-20^{0}C) arasındaki farklar radyatörlü sisteme (yüzey sıcaklığı > 70^{0}C) oranla daha azdır. Bu nedenle de sistem sıcaklık değişikliklerine daha duyarlıdır. Örneğin dış hava veya ortam sıcaklığındaki ufak bir artış döşeme sıcaklığı ile ortam sıcaklığı azaltacak bu da ısı transferini aksi yönde etkileyecektir.

10. Sıcak ve Soğuk Su Tesisatında Kullanılabilirlik:

İlgili standardlar çerçevesinde belirli termoplastik borular doğrudan sıcak ve soğuk su tesisatında kullanılabilir. Bu yöntemle büyük ölçüde işçilik ve yatırım tasarrufu sağlanır. Korozyon ve tıkanma problemi ortadan kalkar. Bu tür tesisatta kullanılan termoplastik boru et kalınlığı dolayısı ile basınç dayanımı daha fazla seçilir.

11. Hafiflik ve Doğal Enerji Kaynaklarını Daha Az Tüketme

Termoplastik borular eskiden yerden ısıtmada kullanılan bakır borulara oranla çok daha hafif ve ucuz olup döşenmesi daha kolaydır. Boru yüzeyleri daha düzgün olup bakır boruların lehimleme dezavantajını da giderir. En önemlisi termoplastik boruların yapımında çok daha az birincil enerji tüketilmektedir.

Metalik malzemede fosil yakıtlar üretim için gerekli enerji girdisinde büyük miktarlarda kullanılırken, termoplastik boruların hammaddesi olarak doğrudan doğruya fosil yakıtyan ürünleri kullanıldığı halde çok daha az birincil enerji kaynağı tüketilmiş olmaktadır.

Soğutmaya ilişkin özel avantajlar ise 3 ayrı grupta toplanabilir. Bunlar:

1. Güneş Enerjisi ile Soğutma İmkanı:

Güneş enerjisi ile tahrikli absorbsiyonlu bir ısı pompası kolaylıkla 12^{0}C-14^{0}C arasında su serinletmesini yapabilir. Bu ise tavandan soğutmaya ayrılabilen geniş yüzey alanları aracılığı ile etkin bir soğutmaya genellikle yeterlidir.

2. İnsan Anatomisine Uygunluk:

İnsanlar en çok ışıma ile ve başlarında serinlik duyusuna haizdirler. Geniş ışıma yüzeyi ve yüksek ışıma transfer oranı ile tavan yüzeyinden soğutma ideal bir soğutma yöntemi olmaktadır.

3. Güneş Enerjisine Bağımlı Olmadan Çalışabilme:

Yazın güneş enerjisi olmasa bile sıcak servis suyu özellikle otel ve tatil beldelerinde gereklidir. Örneğin sıcak su hazırlayan bir ısı pompası kullanıldığında buharlaştırıcı tarafında soğutma gerçekleştirilebilir. Tavandan soğutmada su sıcaklığı klasik soğutma düzenine göre daha yüksek olabildiği için ısı pompasının perfonmansı ve verimi daha yüksek olur. Dolayısı ile bir ısı pompası ile verimli bir şekilde sıcak su hazırlanırken mekan soğutması da gerçekleştirilmiş olur. Özetle soğutulan mekandan çekilen ısı ile sıcak su elde edilir.

Dezavantajlar:

Yerden ısıtma veya tavandan soğutmanın bunca avantajına karşın dezavantajı veya alınması gereken önlemler var mıdır?

Mutlaka her sistemde olduğu gibi, dezavantaj denilmeşe de bazı kısıtlar, ve alınması gerekli önlem ve dikkat edilmesi gerekli hususlar vardır. Bunlar şu şekilde özetle-mı?

İyi bir tasarım ve işçilikle yapılan ve uygun boru kullanılan bir uygulamada söz konusu değildir.

4. 30 yıl sonunda sistem ne olacak?

30 yılın hemen ardından borular hemen eskiyecek anlamı çıkarılmamalıdır. Aslında beklenen ömür 50 yıl dolayındadır. Bu ise bir çok betonarme binanın ömründen de uzundur.

5. Tabana konan halı, parke gibi kaplamalar sıcaktan etkilenir mi?

Taban sıcaklığı bu gibi malzemelerin fiziksel olarak yıpranmasına etki etmeyecek kadar düşüktür. Bununla birlikte boydan boya sentetik halılarda sıcakta genleşme payı gözönüne alınarak halılar buna göre kesilmeli, ayrıca özel fırınlanmış parke kullanımına itina edilmelidir. Ülkemizde yerden ısıtma için daha uygun ısıl ve fiziksel özellikte parke yapımına bu yıl başlanmıştır.

6. Borular kireç tutar mı?

Düşük sıcaklık, yüzey özelliği ve sistemde buharlaşan suyun çok az olması nedenleri ile kireç oluşmaz, sistemin hava yapma olasılığı da daha azdır. Ancak bunun için sistemde dolaşan su hızı uygun seçilmeli, gerekirse katkı maddesi kullanılmalıdır. Korozyonu önleyen katkı maddeleri de olmakla birlikte bunlar çok dikkatli ve bilgi sahibi kişilerce sisteme eklenmelidir. Aksi taktirde yarardan çok zarar getirebilirler.

7. Halıdan veya yerden daha çok toz kalkmaz mı?

Hava dolaşımının az, oda neminin normal düzeyde kalması nedenleri ile bilakis mikroorganizma üremesi olasılığı daha azdır. Halı diplerine toz ve kirin kaçması da önlendiği için yüzeyden bu tozların süpürülmesi daha da kolaydır. Ancak uzun zaman temizlenmeyen halılardan, çok ufak (mikroskopik) tozların 40-50 cm kadar yukarılara zaman zaman kalkması söz konusu olabilir.

8. Normal plastik veya lastik boru kullanamaz mıyız?

Normal veya standart dışı plastik boru kullanıldığında yerden ısıtmanın tüm avantajları kısa sürede yok olur ve sistem iş göremez hale gelebilir.

9. Yerden döşeme pahalı ve lüks bir uygulama mıdır?

Yerden ısıtma her ne kadar çok daha konforlu ve temiz bir yaşam ortamı sunuyorsa da esas avantajı enerji tasarrufu nedeni ile işletme giderlerinin azlığı, bakım giderlerinin az oluşu ve hatta ilk yatırım maliyetlerinin diğer sistemlere oranla düşük olabilmesidir.

10. Yerden ısıtma maliyeti m^2 üzerinden verilebilir mi?

Böyle bir fiyatlandırma çok yanıltıcı olur. Zira ısı yükü ve yerden ısıtma verimleri şehirden şehire, binadan binaya değişir. Her yeni proje ve uygulama için ciddi bir analiz gerekmektedir. Bu amaçla TÜBİTAK desteğinde bir standard tasrım algoritmesi geliştirilmiş ve aynı amaçla da bir bilgisayar programı yapılmıştır.

11. Sisteme plastik borulardan oksijen sızıp tesiste korozyona neden olmaz mı?

Termoplastik ve lastik boruların her türü, özellikle düşük yoğunluktaki borular (Low Density) oksijen geçirgendirler. Dolayısı ile sistem suyunda oksijen olması doğaldır. Ancak bu oksijen miktarının normal çelik aksamlı klisak sistemlerdeki bağlantılardan oluşun oksijen sızıntısından çok daha fazla olduğu söylenemez. Bu fazlalık genellikle %30 dolayındadır. Bununla birlikte her sistemde olduğu gibi yerden ısıtma sistemlerinde de oksijen giderici önlemlerin alınması tavsiye olunur. Bu önlemler: Boruların oksijene geçirimsiz olması oksijen absorplayıcı katkı ilavesi, kazan ve ısıtma yüzeyleri arasındaki parkura ısı değiştirgeci (eşanjör) konulması şeklinde olabilir.

Ayrıca sistem düşük sıcaknebilir:

1. Tesis tamamlandıktan sonra, boruların ve çoğu zamanda kollektör bağlantılarının değiştirilmesi veya zorluğu hemen heken imkansızdır. Sistem performansı döşeme veya tavanda beton dökülmesinden ve kaba işçiliğinden başlayarak tüm evrelerde çok iyi bir işçilik ve bilgi düzeyi ve birikimi gerektirir.

2. m_2 başına ısı yükü 100 kcal/h'yı geçen durumlarda ek yalıtım veya ek ısıtıcı gerektirmesi. Aslında bu bir kısıt veya dezavantaj değildir. Bilakis binanın iyi yalıtılmadığını gösterir.

Zira normal yalıtımlı binalarda, çok küçük çatı veya köşe odaları hariç böyle bir durumda Türkiye koşullarında karşılaşılmaz.

3. Binaya yerleşim sırasında yere delik açılması, dübel takılması, tavana askı çakılması gibi uygulamalarda dikkatli davranması gerekir.

Yerden ısıtmada binanın ısıl ataleti devrede olduğu için çok ani sıcaklık ayarları yapmak zordur. Ayrıca çok seyrek ısıtılan yapılarda özel bir rejime sokma süresi gereklidir.

4. Isıtmada su gidiş/dönüş sıcaklık farkı 10°C alındığı için su debisi daha fazla seçilir. Bu ise dolaşım pompası kapasitesini ayrıca ana kolon boru çaplarını etkiler. Buna karşılık kolon ve vana sayısı daha azdır.

5. Klasik sisteme göre inşası bitirilmiş ve kullanılmakta olan binalarda uygulama yapılması durumunda kapı ve eşiklerin yeniden düzenlenmesi sorunu vardır.

6. Termoplastik borular morötesi ışınlara duyarlıdırlar. Bu nedenle imalat sonrası özel ambalaj içersinde pazarlanır. Özel ambalajından çıkartılmış ve açık havada uzun süre bekletilmiş borular yerden ısıtmada kullanılmamalıdır. Özel laastik hortmlarda ise bu sorun yoktur.

7. Sürekli ısıtma rejiminde termoplastik borular genelde 60°C'ın üzerinde çalıştırılmamalıdır. Özel ağdalanmış borularda bu sınır 75°C'a çıkabilmektedir. Bu sınır özel lastik hortumlar için de geçerlidir.

8. Termoplastik borular bazı yağ ve bitümlü malzemeye duyarlı olabilir. Dolayısı ile beton katkı maddelerinin seçiminde DIN 8078 Standardı Ek 1'deki "katkılara uyumluluk" listesi gözönünde tutulmalıdır.

9. Piyasada döşemeye uygun osun ya da olmasın bir çok türde ve markada boru mevcuttur. Bu boruların gözle tanımı ve ayırdedilmesi genelde zordur.

10. Bu tür termoplastik boruların seçimi sırasında içerisinden sıcak su dolaştırılması gerekli olmaktadır.

11. Oksijen bariyersiz boru kullanıldığı zaman sistemin metal kısımlarında ve kazan yüzeylerinde pompa aksamında aşırı korozyona rastlanabilir. Bunun için ya oksijen bariyerli boru kullanılmalı, ya da kazan devresi bir eşanjörle ayrılmalıdır.

Genellikle Sorulan Sorular ve Cevapları

1. Yerden ısıtma sağlığa, özellikle çoçuklara zararlı değil midir?

Hayır, döşeme yüzey sıcaklığı hiç bir zaman 29°C'a dan yukarı seçilmediği gibi gerektiğinde 25°C'a kadar düşük seçilmektedir. Dünyadaki çeşitli Standartlar 26°C ile 29°C arasındaki döşeme sıcaklığında hiç bir olumsuz etkinin tesbit edilmediğini vurgulamaktadır.

2. Döşemeden ısıtmada geç ısınılmaz mı?

Binanın ısıl ataleti nedeni ile bu bir dereceye kadar doğru olmakla birlikte aslında bu önemli bir avantajdır. Herşeyden önce ısıtma mevsimine girerken kazan belki 2-3 gün erken yakılmaya başlanacak fakat İlkbaharda aynı şekilde 2-3 gün önce kazan söndürülecektir. Ayrıca enerji depolaması nedeniyle pik ısıtma yüklerinde azalma sözkonusudur.

3. Yere döşenen borular sık sık delinmez, tıkanmaz

lıklarda çalıştığı için bakteriyolojik korozyon sözkonusúdur. Bu durumlarda su özelliğine bağlı olarak katkı maddeleri kullanılmalıdır.

12. Tavandan soğutmada tavanda nem birikmez mi?

Uygun bir analiz ve tasarımla bu sorun genellikle söz konusu değildir. Çok nemli ortamlarda nem alıcı paket cihazlar tavsiye olunur.

Özel Uygulamalar

Yerden ısıtma prensibinin avantajları ve özellikleri, hem yeni ısı (soğuk) kaynaklarını hem de bazı yeni diğer uygulamaları da gündeme getirmiştir. Bunlar özetle:

1- Yeni ısı (soğuk) kaynakları:

- Güneş enerjisi
- Toprak ısısı - Deniz veya göl suyu
- Yeraltı suyu
- Jeotermal enerji
- Her türlü atık enerji

Isı pompaları döşemeden ısıtmada daha verimli olmaktadır.

2- Döşemeden ısıtma boruları ile diğer uygulamalar:

- Duvardan ısıtma,
- Tavandan soğutma,
- Zemin ve yol ısıtması, kar ve buz eritmesi,
- Toprakta ısı depolaması,
- Sıcak su soğuk su tesisatı,
- Sera ısıtması,
- Soğuk hava deposu zemin toprağının donmadan korunması,
- Radyatörlü sistemlerde gidiş/dönüş bağlantı boruları,
- Pipe-line borularının belirli sıcaklıkta tutulması, - Çim sahaların korunması,
- Açı sahaların ısıtılması.

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TÜBİTAK

DEVELOPMENT OF A NEW DESIGN ALGORITHM AND NOMOGRAPH FOR HEATING AND COOLING PANEL SYSTEMS

by

Prof.Dr.İ.B.KILKIŞ

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:FACE

ıel heating and cooling systems have a wide range of attributes ıging from energy economy to human comfort. This feature ,ever,does not single out the absolute necessity of a thorough ierstanding of the concept,and a comprehensive design for its :cess. Some say that the system is very forgiving and the industry ıms to rely on some "proven" rules of thumb. In the contrary,panel ıting and cooling are one of the most complex space conditioning ;tems to understand and design. In spite of this,technical :erature is relatively few,although some nations have already ındardized the design procedure.

ıpter 6 in ASHRAE Handbook:Systems and Equipment is the only :erence in America and probably in many other overseas countries. .s wide and international circulation probably puts this chapter in ıuch more vital position than the other handbook chapters. First of .,the information must be up to date,and in a sense,state of the :. Then,it must be complete and provide sufficient design :ormation for the systems that it covers.
.s chapter was reviewed from a designer's birds eye,also keeping in ıd that a handbook should be very concise and simple,yet accurate. : fundamental theory and the content of Chapter 6 was carefully :iewed by spending more than 700 single man-hours. Among many ıgestions,a unique nomograph and an analytical algorithm is hereby ıposed for the next issue of Chapter 6. This tiresome but very :asant task was performed in part by a grant from TUBITAK and :hnical support from Middle East Technical University,and US .ustry which l also believe that they make my participipitation in :RAE T.C. 6.5 mutually beneficial and successful. This report fully :uments and substantiates all the suggestions made for the next ue of Chapter 6 which is due 1996. l believe that in its new mat,it will continue to be a preferred guide and a reference for engineers and HVAC industry in facing the challenges and coping h the developments taking place all over the World at the verge of next century.
ope you will enjoy reading this report and like the suggested new pter 6 for the 1996 Handbook.

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.T.C.6.5,ASHRAE
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NTENT

DEVELOPMENT OF A NEW DESIGN ALGORITHM AND NOMOGRAPH FOR HEATING AND COOLING PANEL SYSTEMS

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TRACT

ed on existing literature and data,guidelines for panel heating cooling systems given in Chapter 6,ASHRAE Handbook,1992 were luated. Results indicated the need for new design nomographs and er fundamental revisions. In order to accomplish this task without rcomplicating the guidelines,a simple analytical algorithm was eloped. It enables to manipulate all design variables so that an imum solution can be searched with sufficient engineering uracy. A unique nomograph was also prepared. This single nomograph be used either for floor heating,ceiling heating,or ceiling ling panels of various types. Any desired indoor temperature and a ge of AUST values and circuit spacing can be entered. Due to its ular nature,this combined nomograph can be easily edited to erate alternative nomographs with different formats and content. e studies and comparisons revealed that the new algorithm and ign nomograph are expected to provide more comprehension, xibility,and convenience to the designer.

1. Basic Theory

Operation of radiant panels is based on controlling the effective surface temperature to meet the sensible space heating and cooling loads. For a proper design the following steps are essential:

a- Defining or calculating the indoor design conditions like surface temperatures and the air circulation.

b- Calculating the sensible load,the corresponding heat load intensity and the effective temperature at the panel surface.

c- Calculating the heat diffusion in the panel and deriving an optimum (optimal) design.

Main design inputs are the magnitude and type (heating or cooling) of the space conditioning load,desired indoor temperature t_a,air dynamics,dimensions and the elements of the conditioned space.

Area averaged temperature of uncontrolled surfaces(AUST) is a derived input.

Main design parameters are; type,size,dimensions and structure of the panel(s),circuit spacing,mean water (brine) temperature, material and the dimensions of the hydronic tube (or electric cable),arrangement,distribution and layout of the circuit.

1.1. Panel Surface Heat Transfer.

The heat transfer at the panel surface takes place primarily by radiation and is complemented by natural convection. At the presence of mechanical ventilation and/or some extreme temperatures of uncontrolled surfaces,an enhancement of the natural convection may also take place (Harris,Everett,Sartain,1959,ASHRAE,1992,Drake,1993).

.1. Radiation heat transfer intensity

ardless of the panel location (i.e:floor,ceiling or wall) iation heat transfer intensity is (ASHRAE,1992):

$$q_r = 0.15 \cdot 10^{-8} \cdot [(t_p+460)^4 - (AUST+460)^4] \quad [1]$$

hough ASHRAE Handbook,Chap.6 already provides certain rules,it is her difficult to determine AUST accurately. It may be quite ferent than the indoor air temperature at design conditions lkis,1993-a,Drake,1993). Drake shows that AUST primarily depends the amount of outdoor exposure and fenestration. His calculations ealed temperature differences up to 9°F in panel heating.

kis(1993-a) developed a simple rule for AUST in panel heated or led spaces:

$$AUST \approx t_a - d \cdot z \quad [2]$$

e d is the room exposure code. It is 0.5,1,2 or 3 respectively for ms with no outdoor exposure,one side exposed (with fenestration up 5%),one side exposed (with a higher fenestration ratio),and two or e exposed sides.

s the outdoor temperature adjustment factor:

$$z \approx \frac{50}{13 + t_b} \qquad \{\text{panel heating, } t_b \geq -5°F\} \quad [3\text{-A}]$$

$$z \approx \frac{23}{t_b - 113} \qquad \{\text{panel cooling, } 80 \leq t_b \leq 100°F\} \quad [3\text{-B}]$$

e t_b is the outdoor design temperature.

se equations approximate and simplify the calculation process,if ired,by reducing the number of variables to two,namely t_b,and the ition of the room in the building.

le 1 compares this rule for panel heating wherever possible,with

Drake's findings (1993). It reveals a relatively close agreement and confirms that there may be substantial differences between AUST and t_a. Harris,Everett and Sartain (1959) also reported similar observations.

Table 1. Calculation of AUST in Panel Heating

t_b °F	**room position**	**degree of of fenestration**	**U value of windows**	**(AUST - t_a)[°F] Eqn.3-A**	**Drake**
0	inner room (no exposure) (d=0.5)	---	---	1.92	<1
0	exposed room (one side) (d=1)	5%	~1	3.85	3.1
0	exposed room (one side) (d=2)	10%	~1	7.69	7.6
0	corner room (two sides exposed) (d=3)	20%	~1	11.53	9.9

1.1.2. Convective heat transfer intensity

Determination of the convective heat transfer intensity is more controversial: Harris,Everett and Sartain (1959) had experimentally shown that the presence of cold indoor surfaces may enhance the natural convection from a heated floor. ASHRAE Handbook (1992) states that this enhancement can not be generalized for an engineering ign. Although in principle,this holds true,there might be tances where the enhancement can not be ignored: experiments has wn that air at a colder temperature sweeps floor panels ake,1993). In fact this is the film temperature which governs the ural convection heat transfer (ASHRAE,1992). This "sweeping" by a m of colder air is directly related to AUST. Therefore,a sonable and compromising solution is to equate the adjacent air m temperature to AUST in convection heat transfer calculations for loor heating panel. Although this argument can also hold true for ling cooling,there is no supportive evidence for wall panels and ling heating panels.

and et.al (1956) carried out panel heating experiments in a 12 ft 24½ ft test room (D_e:16.1 ft). They obtained correlations which sider the effects of the size (D_e) of the space,and the height (H) the wall panel (see ASHRAE Handbook,1992,Chap.6,Eqns. 6,7 and 8):

$= 0.041 \cdot (t_p-t_a)^{1.25}/D_e^{0.25}$ ceiling heating **[4]**

$= 0.39 \cdot (t_p-t_a)^{1.31}/D_e^{0.08}$ floor heating or ceiling cooling **[5]**

$= 0.29 \cdot (t_p-ta)^{1.32}/H^{0.05}$ wall heating or cooling **[6]**

effects of D_e and H may be ignored at experimental conditions,by igning D_e:16.1 ft and H: 8.8 ft from their test room. sequently the following equations were obtained (see ASHRAE dbook,1992, Chap.6,Equations 8,9 and 10):

$q_c = 0.020 \cdot (t_p-t_a)^{1.25}$ ceiling heating **[7]**

$q_c = 0.31 \cdot (t_p-t_a)^{1.31}$ floor heating or ceiling cooling **[8]**

$q_c = 0.26 \cdot (t_p-ta)^{1.32}$ wall heating or cooling **[9]**

must be noted that a proper rounding off operation to two nificant digits in deriving Equations 7 and 8 is the reason for a or difference between their counterparts in ASHRAE Handbook.

For large indoor spaces like hangars or exhibition halls, the size effect becomes important in floor and ceiling panels. For example in a 200 by 400 ft floor heated space,natural convection heat transfer decreases by about 30%. In ceiling heating, size effect is even greater. Therefore,for large indoor spaces,the former pair of equations for floor and ceiling panels should be used.

The biggest controversy is about convection from a heated ceiling: The downward natural convection process in ceiling panel heating is a difficult one,and is very sensitive to the existing boundary conditions. Therefore,depending upon test conditions maintained,the agreement among many empirical equations is indeed very weak. Keeping this in mind,one has to pick the correlation which suits the expected indoor design conditions most: In practice,with the objective of implementing and maintaining a stronger downward convection,cold (unheated) strips (sections) between ceiling panels are left.

In addition,the indoor conditions will most likely provide initiatives like the temperature gradient between the ceiling and an adjacent cold window surface,or air circulation due to draft, infiltration or mechanical ventilation. Any correlation should acknowledge these factors. Kilkis (1992-a) suggested to use the following equation after Kollmar and Liese (1957):

$$q_c = 0.13 \cdot (t_p - t_a)^{1.25} \qquad \text{ceiling heating} \qquad \textbf{[10]}$$

Under practical circumstances,this will yield more realistic heat output intensities.

According to the experimental data by Harris,Everett and Sartain (1959) the constant for the above equation is about 0.22 instead of 0.13,when averaged over the ceiling panel area. This shows that Equation 10 is valid and indeed conservative.

L.3. Combined surface heat transfer intensity

nbined heat transfer intensity q at the panel surface is the sum of and q_c:

$$q = q_r + q_c \qquad [11]$$

nust meet or exceed the sensible heating (or cooling) load per unit iel area. In panel heating t_p is greater than AUST and t_a. erefore q_r, q_c and consequently q will be positive (see Equations ',8,and 9). Therefore any heat transfer **from** the panel surface to conditioned space is regarded positive. Consequently,they will be jative in panel cooling where heat is extracted (removed) from the iditioned space and transferred **into** the panel. In floor heating l ceiling heating ASHRAE Handbook represents q by symbols q_u and q_d spectively. No symbol is mentioned for the heat extraction moval) intensity in ceiling cooling. It will be appropriate to use l within the scope of the sign convention used. For wall panels ere is not any symbol assigned either. It is suggested to use the ibol q_s (sidewise heat transfer from/to the wall),following the licit terminology where **up** is for floor heat transfer direction l **down** for ceiling heat transfer direction in heating. Table 1 umarizes this terminology and the sign convention:

le 1. Terminology for Combined Heat Transfer Intensity

anel	Combined Panel Surface Heat Transfer Intensity,**q:**		
inction	**Floor**	**Ceiling**	**Wall**
ating	q_u	q_d	q_s
oling	N/A	$-q_d$	$-q_s$

1.1.4. Panel heat losses and gains

Inevitably some heat is lost from the back and edges of a heated panel to the colder environment. q_b is the heat loss per unit panel area(Btu/h·ft²). Chapter 6 provides sufficient information to calculate back and edge losses in heating for certain panel types and positions. In cooling,heat is transferred from warmer environment to the panel. Simple heat transfer calculations for these heat gains are expected to yield sufficiently accurate information. Following the sign convention established,intensity of the heat gain in a cooling panel is also negative ($-q_b$). Panel heat losses (or gains) should only be considered in sizing the boiler (chiller).

1.2. Heat Diffusion in the Panel.

Either electric or hydronic,the panel surface temperature control is accomplished with heated (cooled) circuits placed at a finite spacing,M on their centers. These circuits may be embedded into the panel or attached to it by a suitable means of bonding,clamping or stapling (see Figure 3).

1.2.1 Thermal modeling of the panel

Depending upon circuit spacing and all other material properties,the mean temperature of the circuit has to be calculated in order to meet a specified load at given design indoor conditions. Spacing of the circuit effects the following:

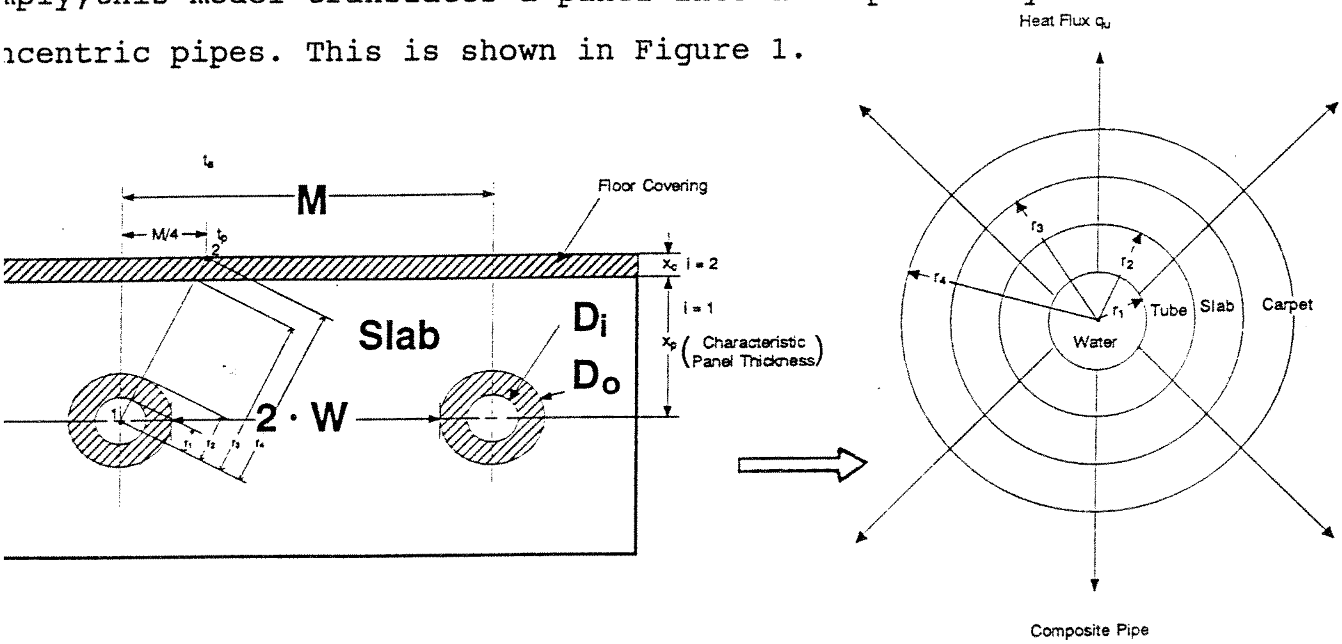
i- Heat output (removal) per square foot of panel area,

ii- Panel surface temperature distribution,

iii-Heat diffusion in the panel,

iv- Temperature drop across the tubing.

The first three parameters have a primary effect on the required mean ter (brine) temperature. The latter has a secondary influence. finite element solution is quite readily available by private or mmercial computer codes. However, this does not seem practical yet r everyday design activities. At this point one has to make a oice between available nomographs with limited scope and accuracy ee Appendix 6 for sample nomographs) and a reliable analytical tool r a better control and manipulation of the design process. There e few models available. One such a model (Krinninger,1989) assumes at every panel layer surrounds the embedded circuit concentrically. mply,this model translates a panel into a composite system of ncentric pipes. This is shown in Figure 1.



gure.1. Polar Representation of a Cartesian Slab (Krinninger,1989)

order to introduce the effect of circuit spacing **M**,panel thermal sistance,r_u is calculated along line 1-2 instead of a vertical line ght above the tube (electric cable). Point 2 is at a distance **M/4** om the centerline on the panel surface. Therefore,the length of ne 1-2,consequently r_u will be related to the circuit spacing **M**. s relationship can only account for parameters iii and iv ntioned above. The primary disadvantage of this model is quite obvious: the actual heat flux from the tube is not radially symmetric. Depending upon the specific configuration and insulation, it is quite distorted. Figure 2 (Kilkis and Sager,1993) shows a finite element solution for an in-slab floor panel with insulation, below the tubing. Therefore,radial symmetry assumption will not hold unless the upward and downward heat transfer rates are equal.

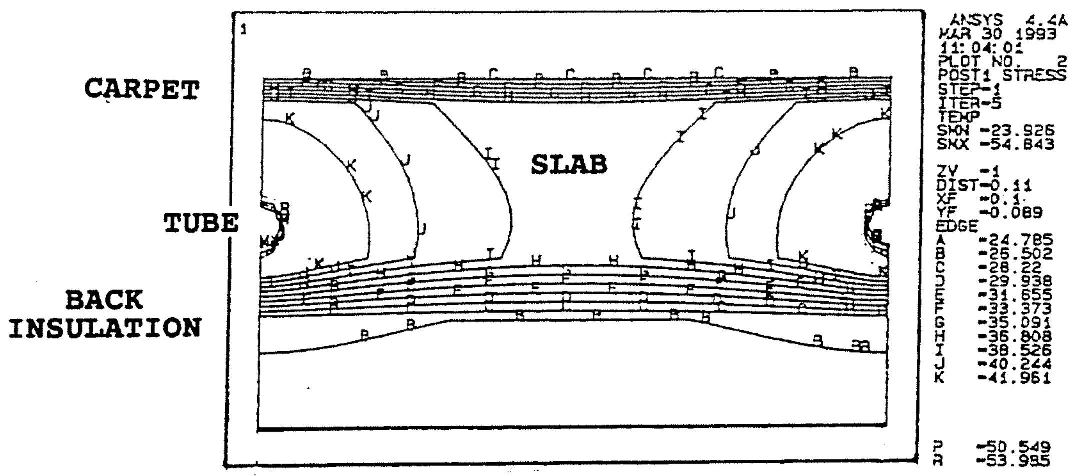


Figure 2. A Typical Finite Element Solution (Kilkis and Sager,1993)

Even this does not help much,because floor covers are at the top floor surface and not at the ceiling of the space below !. In other words,a carpet on the floor has only a limited thermal exposure to the tube perimeter and does not cover the heated (cooled) tube all around, 360°(see Figure 1). This makes the model inaccurate especially for heavy floor coverings:usually floor coverings like a carpet are relatively thin and have low thermal conductivity. Therefore,the lateral diffusion in floor coverings is quite negligible regardless of the tube spacing. This is obvious from the temperature lines in the carpet layer. Also a very high temperature

›p in the vertical direction should be noted. This indicates that floor coverings with low thermal conductivity and/or relatively ıll thickness,the significance of M is negligible. In contrary, .s method directly proportions the temperature drop to the spacing floor coverings too. Thus, for close spacings like 4" or 6" the ın water temperature is underestimated. Opposite is true for large ıcings like 15" or 18". In between,the results are quite :isfactory (see Example 1 in Appendix 2). In addition,the polar łel is not applicable to other panel types,in particular when the :cuit is not embedded but attached or bonded to the panel surface :e in a sub-floor application or metal ceiling panels.

!.2. Panel thermal resistance

› key design parameter is the panel thermal resistance,r_u. One ;t recognize and accurately calculate the four components of r_u:

$$r_u = r_s+r_t+r_p+r_c \quad [12]$$

›re;

r_s = thermal contact resistance between the tubing (electric cable) and the panel

r_t = thermal resistance of the pipe in a hydronic system (it is not applicable to electric resistance heating)

r_p = thermal resistance of the panel body

r_c = thermal resistance of the panel coverings

· an in-slab application or other panel types where a good and :ure attachment or bonding exists,r_s may be neglected. ASHRAE .delines (1992) relate r_u to tube (electric wire) spacing,M. This a limited role in honoring the effects of M on design (see :tion 1.2.1). At the absence of any information about the details this relationship (see Tables 2,3 and 4 in Chap.6),it is very .ely that a heat diffusion model similar to Figure 1 was used.

There are two practical disadvantages of using $r_u(M)$:

a- Effects of spacing,M can not be fully accounted.

b- It requires to calculate r_u or refer to the tables given in the ASHRAE Handbook every time M is changed in a design.

In order to avoid these difficulties,simple yet more comprehensive definitions for each element of the panel thermal resistance was introduced. **Appendix 1** gives these new definitions.

Being independent from the spacing M,these definitions will enable to assign a unique r_u value to any given panel. Instead,several effects of spacing will be more thoroughly encountered by Equations 13,14 and 16 given in the next section. Consequently,above mentioned disadvantages will be eliminated. It must be noted that an r_u value calculated from the equations given in Appendix 1 for any given panel and its coverings will not correspond to $r_u(M)$ values listed in tables or shown on the nomographs for various spacings like 3",6" and 9" in the ASHRAE Handbook,Chapter 6. In fact these tables cover only a limited range of M and may seriously restrict the design.

1.2.3. Simplified analytical algorithm

An analytical model for the heat diffusion in the panel was developed with the following attributes under the TUBITAK, MISAG-12 grant:

1-Comprehensive and realistic,

2-Simple,yet sufficiently accurate,

3-Recognize and enable to manipulate all pertinent design variables for an optimization search,

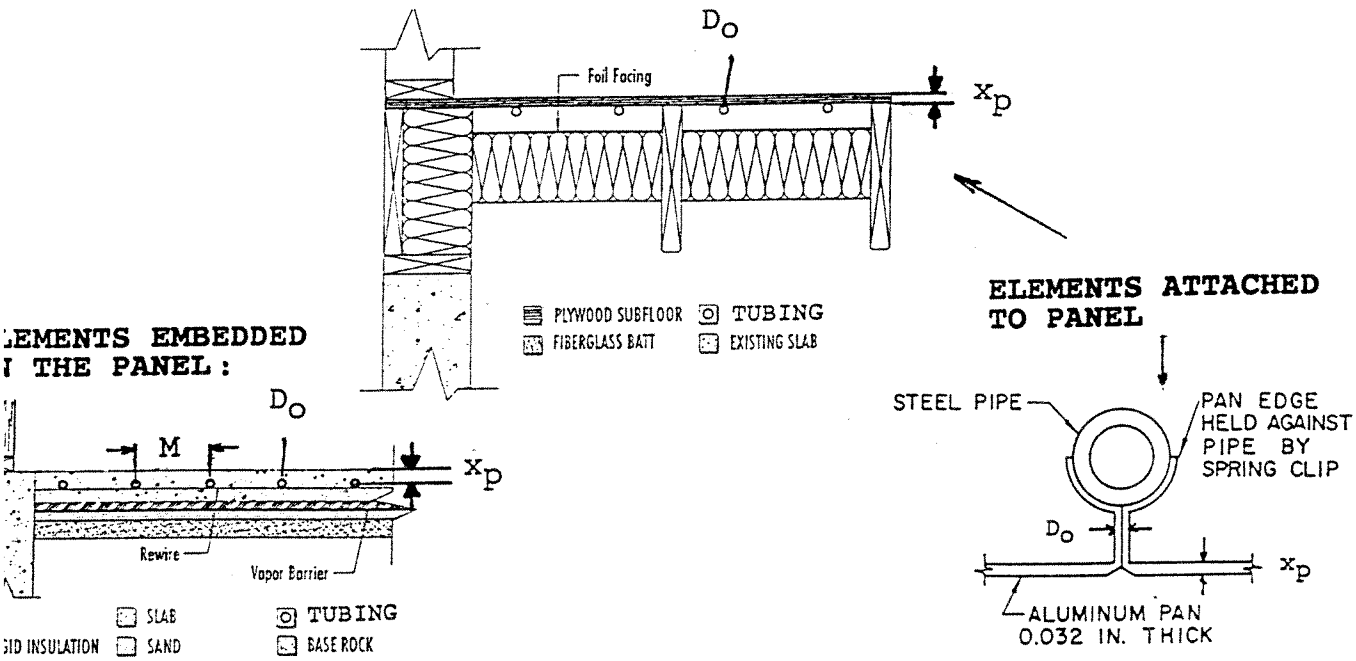
4-Applicable to many panel types,including but not limited to slabs,

5-Applicable to any panel position including walls,

6-Applicable to heating and cooling,

7-Applicable to electric and hydronic systems, ıccommodate any material property and configuration.

ɜ details of this algorithm was given by Kilkis (1993-a). Typical solutions confirmed it(Kilkis and Sager,1993). In order to :ilitate and simplify its everyday use,some simplification were le without sacrificing the accuracy (see **Appendix 2)**. It can easily ıdle virtually all types of panels. See Figure.2 for typical types.



in-slab b- sub-floor c- metal ceiling

ure 3. Typical Panel Types That Can Be Solved by the Algorithm.

ˈ a complete design manipulation only four equations are needed:

$$t_d \approx t_a + \underbrace{\frac{(t_p - t_a)\cdot \mathbf{M}}{(2\cdot W\cdot \cap + D_o)}}_{A} + \underbrace{q \cdot (r_p + r_c + r_s)}_{B} \qquad \textbf{[13]}$$

part A spacing **M** is significant in two ways: a- **M** is explicit in numerator,b- **M** is implicit in ∩ (see Equation 14). r_p and r_c also ect ∩ (see Appendix 2). Usually r_p is more significant than r_c. ever,M is not significant in part B. This agrees with the finite element solutions:the temperature drop in the floor coverings is only a function of its thermal resistance,r_c and the heat transferred,q:

q : combined surface heat transfer intensity ($q_d, q_u, q_s, -q_u$ or $-q_s$) Btu/h·ft²

$\mathbf{M}$: tube (cable) spacing on centers,ft

D_o : Outer diameter of the tube (cable),ft or characteristic contact width of the tube with the panel (see Table B-1 in Appendix 2)

$2 \cdot W$: net spacing ($\mathbf{M}-D_o$),ft

η : $\tanh(m \cdot \mathbf{W})/(m \cdot \mathbf{W})$,dimensionless **[14]**

$\approx 1/(m \cdot \mathbf{W}) \quad \{\text{if } m \cdot W > 2\}$

m = fin coefficient,ft^{-1}

$$m : \left[\frac{q}{(t_p - t_a) \cdot 2 \cdot \Sigma_{i} \, (k_i \cdot x_i)_{\, i:1 \text{ to } n}} \right]^{1/2} \qquad \{t_p \neq t_a\} \qquad \textbf{[15]}$$

k_i and x_i are the thermal conductivity (Btu/h·ft·°F) and characteristic thickness (ft) of each layer,"i" (up to n layers) including the body of the panel (see Figures 1 and 3). For a hydronic system the required mean water (brine) temperature will be:

$$t_w = (q + q_b) \cdot \mathbf{M} \cdot r_t + t_d \qquad \textbf{[16]}$$

This algorithm is equally applicable for cooling or heating by wall,floor or ceiling panels,provided that appropriate equations for q_r and q_c are used. With this algorithm,several design problems were solved. Solutions revealed that all the objectives were achieved:

1-Accuracy is expected to be better than 4% in terms of required mean circuit temperature,based on a finite element solution.

2-All design variables are explicit and can be manipulated.

hanks to the sign convention given in Table 1,and the way the quations were derived and arranged,the same algorithm is pplicable for panel cooling without any change.

t is applicable to many panel types and configurations,provided hat their dimensions and thermal properties are known.

he same algorithm is applicable for ceiling,floor and wall anels,by using the appropriate surface heat transfer equations.

ith only 4 equations,it can be easily adopted to any programmable ocket calculator,if desired.

r different design problems were solved. They are given in **endix 2.** Example 1 is about an in-slab floor heating. Example 2 is ut a plaster ceiling slab for cooling. It should be noted that hough the new algorithm seems to apply only for in-slab panels,it indeed applicable to any type by using their characteristic panel ckness,x_p: Example 3 gives the solution for an under sub-floor ting application. Example 4 gives the solution for a metal ceiling el heating application.

n the ASHRAE guidelines were challenged with the same design blems,it was evident that most of the panel types and other design ameters like size and the material of the tube,spacing,panel struction and panel function are not covered effectively. example,design and sizing of a heating system for a joisted -floor can not be made at all. To a large extent the same is true ceiling cooling systems and wall panels for heating or cooling. t section gives a detailed account of these shortcomings.

Development of New Design Nomographs

. Design Nomographs:General.

definition,a design nomograph is a dedicated graphical compilation of technical data or theory which enables to perform certain design calculations in a convenient and visual environment. A well designed and organized nomograph may be useful for instruction and demonstration. However a nomograph can not substitute available data or theory. It is just a utility tool and unless accompanied by the underlying equations,it does not appeal much to the engineers and designers. The desire to have access to them is quite understandable with the advent of hand held computers,because at the absence of the equations used,it is usually very difficult to transform a nomograph into a computer program. There are nomographs or tables available in the literature or company catalogs for floor heating. Appendix 6 provides some of these,including Drake's suggested nomograph with AUST compensation(Drake,1993). ASHRAE guidelines provide two charts. One is for floor heating,the other is ceiling heating (see Figures 7 and 8 in Appendix 4). To the best of the Author's knowledge there is not any widely available and general purpose nomograph for panel cooling,(except Kilkis,1993-a). It is also known that a DIN Standard for panel cooling is on the way.

The success of a nomograph depends upon the following:

1- Should be based upon a sound and complete model and/or reliable data. Underlying equations and data should be available.
2- Should acknowledge and accommodate all design variables and treat them to be true variables.
3- Allow an explicit manipulation of all important design variables.
4- Should state the expected margin of error in terms of the primary design variable(s).
5- Cover the appropriate ranges of each design variable.
6- Permit multiple ways of use and several ways of data entry with

equal ease. (i.e: enter q_d,obtain t_w and t_p at specified r_u,t_a,AUST and circuit spacing,or: enter t_p,determine q_d and required t_w at specified r_u,AUST,circuit spacing etc..).

Should be accurate and consistent with the theory/data used.

Multi-purpose (i.e: should be able to handle both cooling and heating problems,and different panel types).

Should be modular and flexible so that nomographs with varying contents and levels of sophistication may be easily derived.

Eliminate any need to repeat calculations or refer to other tables or charts.

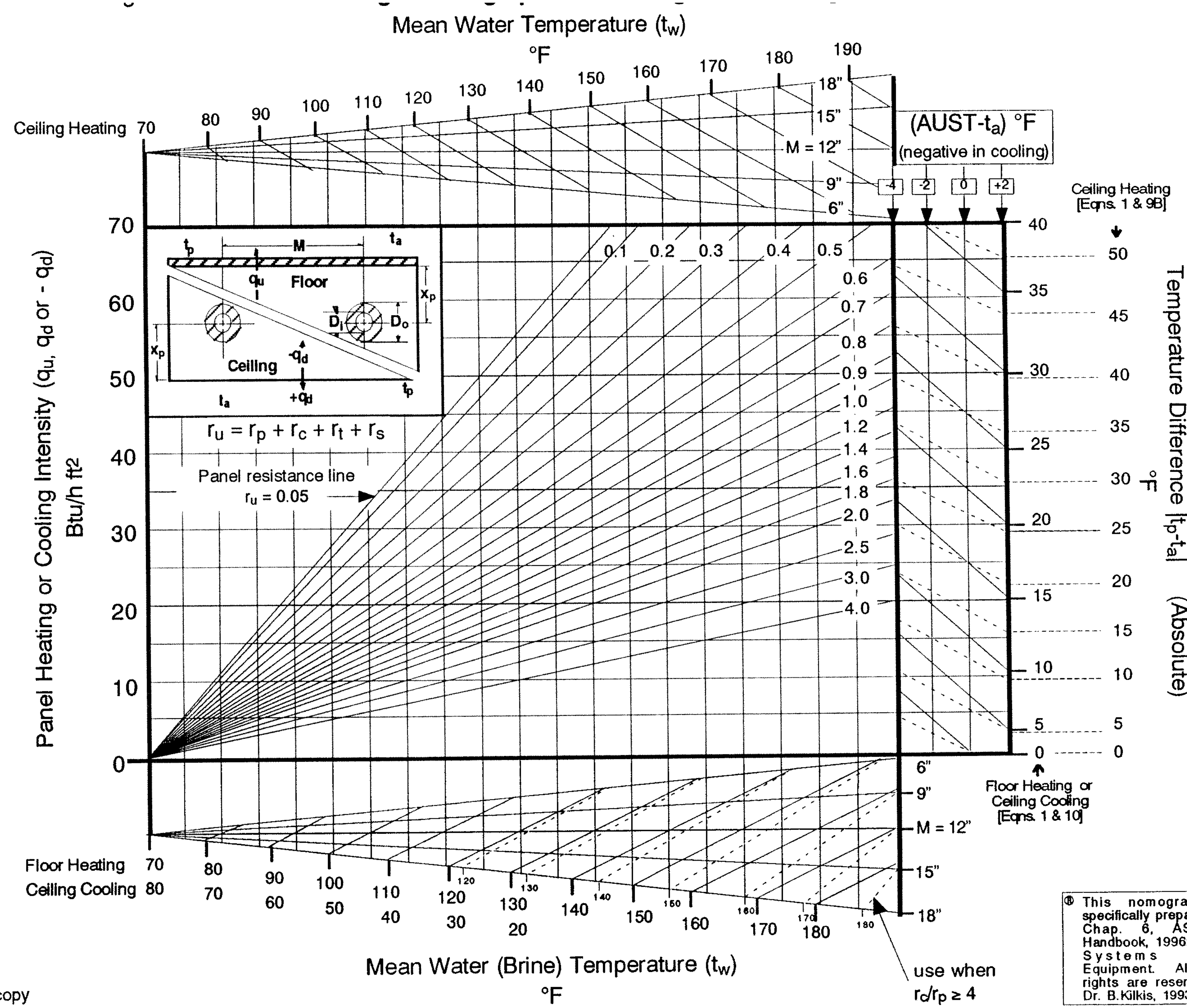
tion 3 will concentrate on several shortcomings of the design ographs and the referred tables in Chap.6,ASHRAE Handbook,1992. endix 6 also reveals that nomographs available in public domain not meet all of the above criteria. While the new algorithm vides a sufficient and capable analytical design tool,nomographs still be desirable for everyday hand calculations about panel ting and cooling systems. Such a nomograph may also enhance the semination of the theory and have an instructional value.

. New Universal Design Nomograph

h the desired features in mind and using the new algorithm,a que "Combo" nomograph[R] was specifically developed by the Author Chap.6. This new nomograph is shown in Figure 4. While the core mat of Figures 7 and 8 in Chap.6 was retained for the convenience designers who are accustomed to those figures,new nomograph resents a completely different alternative. Appendix 3 gives the ails about its construction. Its features are:

an be used for both panel heating and cooling. The left hand ertical axis can read q_u,q_d,or $-q_d$ (see Table 1 for terminology).

an be used both for ceiling and floor panels: t_w axis at the



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bottom has scales both for floor heating and ceiling cooling. The counterpart at the top is for ceiling heating. The right hand vertical axis has two separate scales in order to accommodate both floor heating,ceiling cooling and ceiling heating.

3-Can be used virtually for any kind of panel,provided that new r_u definitions are used (see Appendix 1 for r_u and Appendix 2 for examples of different panel designs).

4-Permits direct manipulation of the following parameters:

. Available circuit spacing,M : 6",9",12",15",18" (interpolation is possible and easy). This entry is either from the top (ceiling heating) or bottom horizontal axis, which replaced the t_a axis in Figures 7 and 8,in Chap.6.

Current ASHRAE charts does not explicitly acknowledge M. They refer to tables 2,3 and 4 to adjust r_u for a limited range of M. This necessitates to calculate r_u every time M is changed.

. Panel thermal resistance,r_u:0.05 to 4.0 (0.05 was added,because metal cooling panels may even have less thermal resistance. r_u:5 line was removed because of its impracticality even with heavy floor coverings). r_u lines are linear and makes interpolation or extrapolation easier. (see also Drake,1993). It must be noted that the range of q was reduced to 70 from 100,because 100 Btu/h·ft² roughly corresponds to about 110°F in floor heating which is not permissible. (ASHRAE Standard 55-1992 limits t_p to 84°F on frequently occupied floor surfaces). In this range r_u lines are quite linear,which is also obvious from Figures 7 and 8,in Chap.6 (see Also Drake,1993). This linearity facilitates interpolation and reading too.

Indoor air temperature, t_a: The nomograph was so designed that there is no need to spare a separate axis for t_a like in Figures 7 and 8. Instead, after an accurate linearization process, a new vertical axis was added on the right hand side where the variable group (t_p-t_a) enables to enter virtually any t_a value (see Appendix 3). Note that this group is automatically (t_a-t_p) in cooling. There are two sets of lines: The solid lines represent floor heating and ceiling cooling. The broken lines are for ceiling heating. Their scales are different and are shown individually on the nomograph. There are four vertical lines each representing a different AUST relative to the indoor air temperature. These are namely t_a**+2**, t_a**+0**, t_a**-2**, t_a**-4** in heating and t_a**-2**, t_a**-0**, t_a**+2** and t_a**+4** in cooling. An interpolation between these lines is possible. i.e: a line for t_a**+1** can be imaginarily drawn half way between t_a**+0** and t_a**+2**.

While mean water temperature lines focus at an air temperature of 70°F, this scale can also be made generic by converting this scale into a (t_a+10), (t_a+20)..format, instead of 80°F, 90°F.. type of scale marks. In this case, 70°F node will be replaced by t_a. In ASHRAE charts, t_a alternatives are only 70, 72, 74 and 76°F. Drake(1993) replaced this axis and t_a with a wider range of AUST. This chart is however only for floor heating. (see also comments by Kilkis(1993-e) and Appendix 7).

AUST: In the new nomograph there is no need to limit the choices of AUST. Here, AUST is related to t_a in the right hand vertical axis. AUST can be t_a+2, t_a, t_a-2 and t_a-4. While t_a can be any temperature, AUST does so too.

Effective panel temperature, t_p: This relationship is not represented by a curve in the core chart any more: t_p can be directly read on the right hand vertical axis for any given t_a. This axis registers absolute value of $(t_p - t_a)$. This makes the readings in cooling consistent with the sign convention used.

- Mean water (brine) temperature, t_w: It is possible to read t_w for floor heating/ceiling cooling and ceiling heating systems respectively from two separate axes, one at the bottom and the other at the top, respectively.

  Any t_w reading will correspond to the skin temperature of the electric cable, t_d. This temperature is important in determining the actual resistance of the electric cable at operating temperatures which may be substantially different than the one measured at room temperature. This effect was not addressed in any of the previous nomographs or models. (except TSE, 1993).

  M lines for 6, 9, 12, 15 and 18" adjust t_w readings in a combined function of q and r_u. In floor heating the significance of M decreases when part B in Equation 13 is dominant (i.e: when fin efficiency is high like in a concrete slab and low heat output) while $r_c >> r_p$. This phenomena is represented by the broken lines on M axis. In ceiling cooling and ceiling heating, normally $r_p >> r_c$ condition is not a practical situation.

Consequently, 6 primary design variables can be directly manipulated. In Figures 7 and 8, Chap.6 this number is only 4.

5-Permits indirect manipulation of the following parameters:

- Tube dimensions D_o/D_i, thermal conductivity k_h of the tube material, panel characteristic thickness x_p, panel thermal conductivity k_p, and panel type.

Consequently 5 primary design variables can be indirectly but numerically manipulated through r_u.

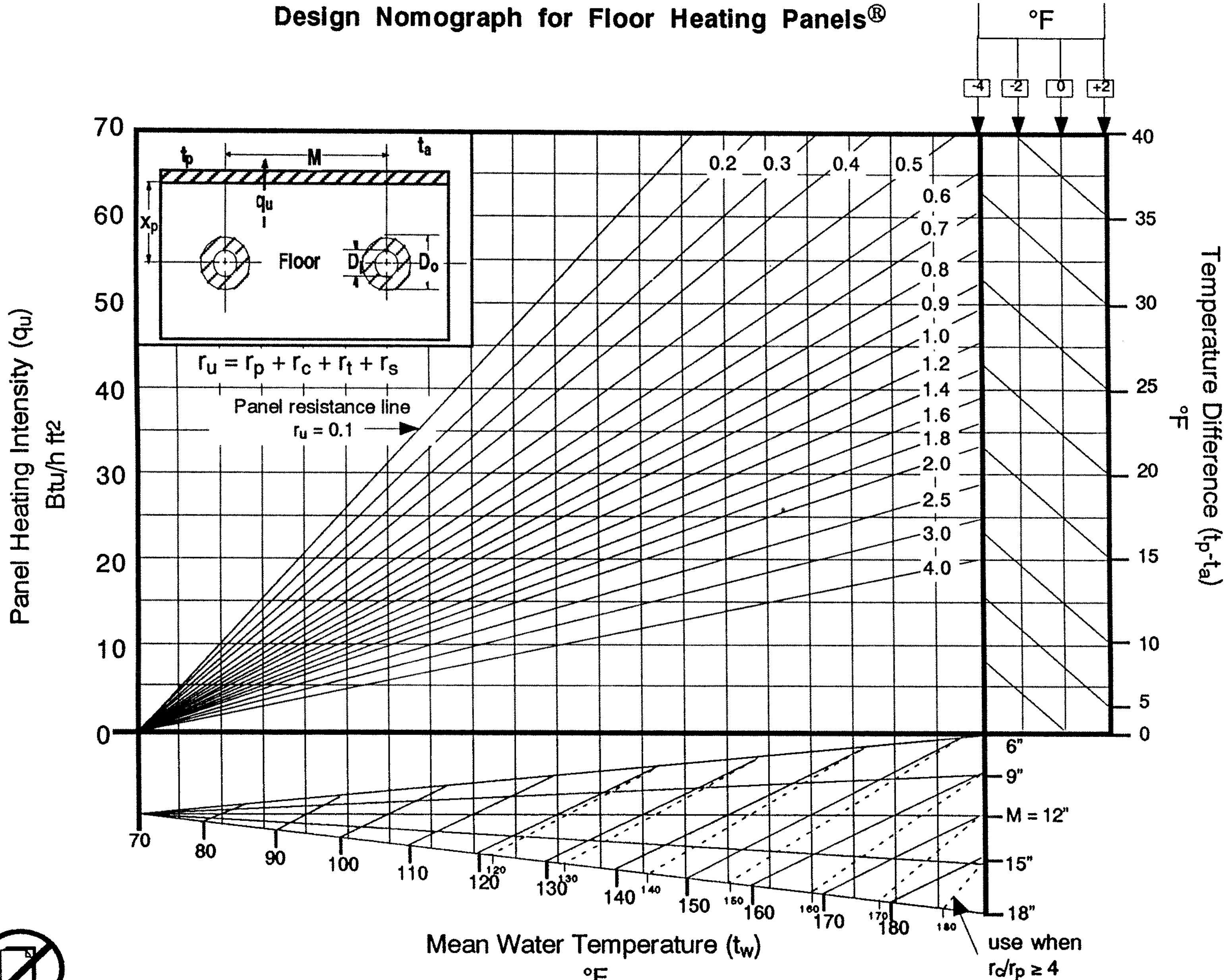

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Figure 5. Floor Heating Panel Nomograph

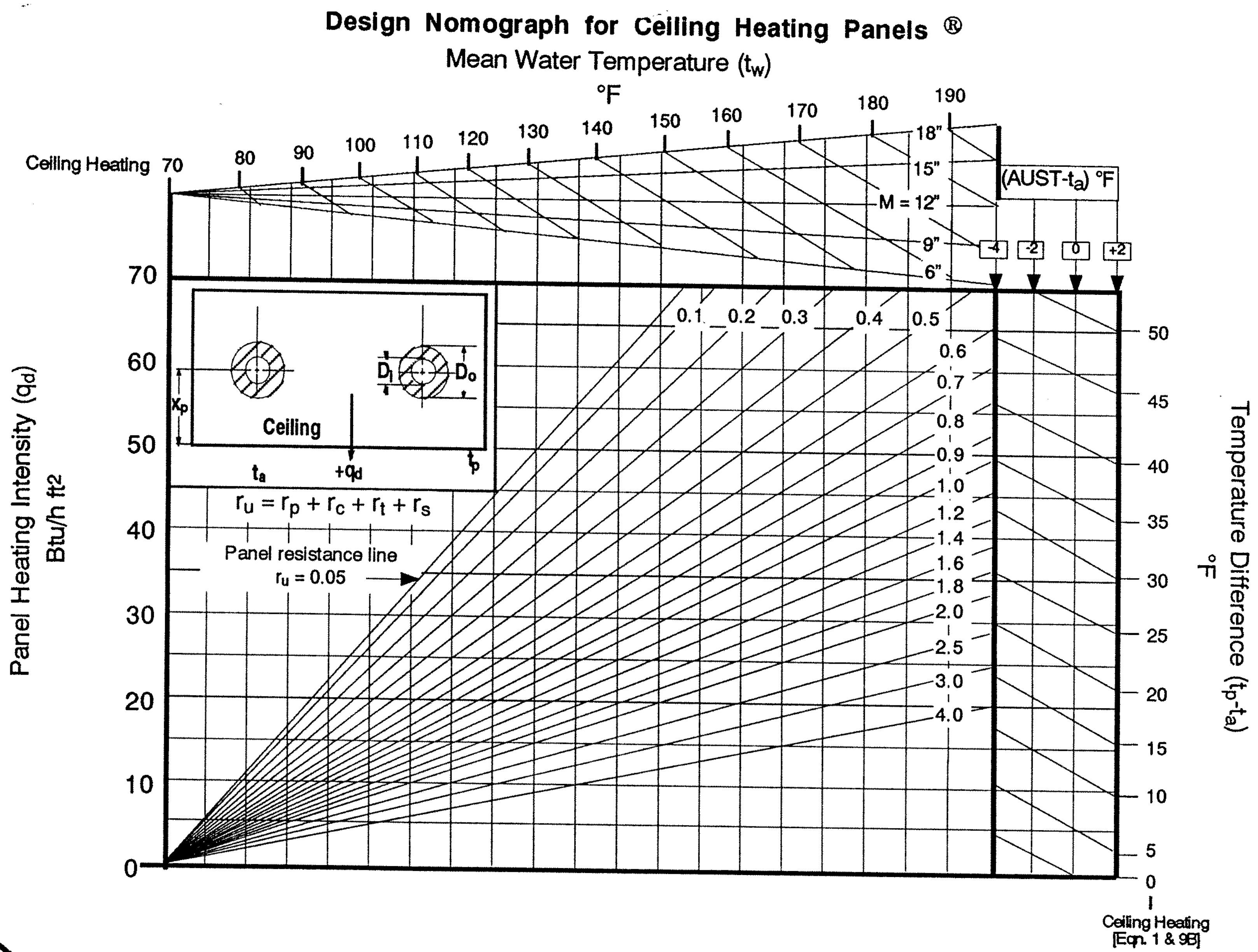

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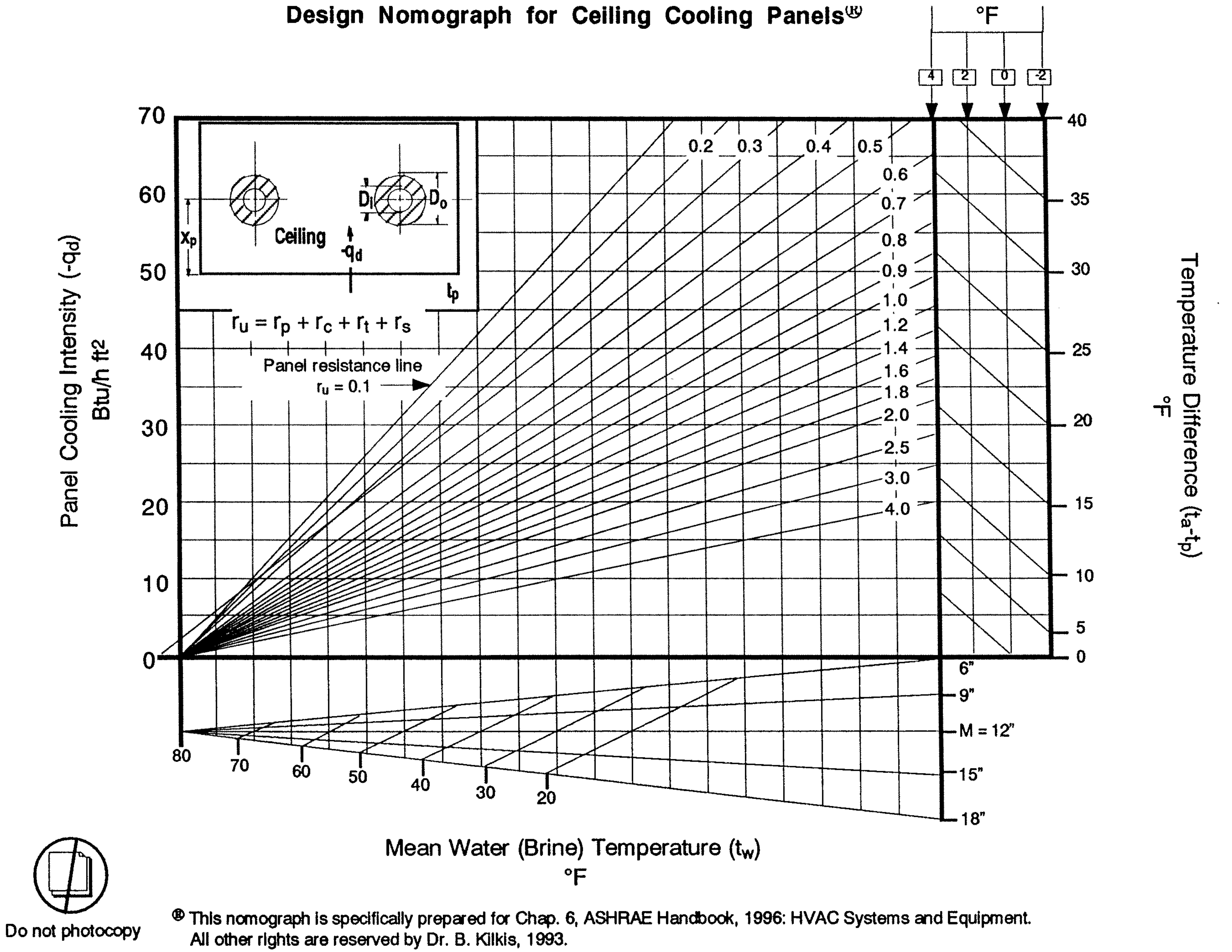
Figure 6. Ceiling Heating Panel Nomograph

Design Nomograph for Ceiling Cooling Panels®



® This nomograph is specifically prepared for Chap. 6, ASHRAE Handbook, 1996: HVAC Systems and Equipment. All other rights are reserved by Dr. B. Kilkis, 1993.

Figure 7. Ceiling Cooling Panel Nomograph

In Figures 7 and 8 this number is only one. In conclusion,new nomograph recognizes 6 more design variables in a single chart.

6-Modular and flexible. Packing this much information in a single nomograph was not easy at all. This was achieved by re-grouping the design parameters and assigning various common scales and multi scales around the periphery of the core chart. Core chart now includes only r_u,because t_p was removed. This makes the core chart independent of panel location and function. Consequently,removing or altering the perimeter axes will generate other alternative nomographs,while the core remains the same. Figures 5,6 and 7 show derived nomographs for floor heating,ceiling heating and ceiling cooling respectively. In a similar sense,it would not be difficult to generate a wall panel nomograph for heating and cooling.

7-Multiple input and output ports:
One may easily and directly enter either from q axis,t_w axis, (t_p-t_a) axis or M axis. This feature makes the nomograph also versatile for the analysis and evaluation of existing systems.

8-Accurate and consistent: The new nomograph's accuracy and consistency with the theory is shown in the following table. This table compares the results for t_p and t_w, for four different design problems which are explained in the next section.

2.3. How to use ? Examples

Four examples were solved by the nomograph. The details of these problems and their analytical solutions are given in Appendix 2,in the same order. Only a brief description be given here. All the manipulations on the nomograph are shown in Figure 8.

mple.1 Floor heating (see Example 1 in Appendix 2)

ign inputs:

30 Btu/h·ft²

1.34; (r_c/r_p:4.73 > 4)

9" O.C.

70°F

T:68°F (AUST-t_a:-2)

er the nomograph from left hand axis with 30 Btu/h·ft² and ectly proceed to the right hand vertical axis. At (AUST-t_a) tical line equal to -2,one reads about 16°F on the floor ting/ceiling cooling scale. This is (t_p-t_a). Knowing that t_a is F then t_p can be solved: t_p: 70+16:86°F.

staying on the same horizontal line at 30 Btu/h·ft²,intersect h r_u:1.34 (This is pretty close to 1.4 line). Then,go down until broken M:9" line is intersected. The broken line is used because this example r_c/r_p is greater than 4. This gives t_w: 131°F.

mple.2 Cooling with a plaster ceiling

ign inputs:

-20 Btu/h·ft² (note that q axis accommodates -q_d values directly)

0.489 (see Example 2 in Appendix 2)

6" O.C.

80°F

T:81°F (AUST-t_a:+1) (note that in cooling,sign is reversed. See the nomograph)

er the nomograph with 20 Btu/h·ft². Extend this line up to the ht hand axis until the interpolated (AUST-t_a):+1 vertical line is ched. At this point,using the floor heating/ceiling cooling scale resented by the solid lines,(t_a-t_p) is 12°F.

Therefore, t_p: 80 - 12°F : 68°F. Then,from r_u line:0.489,go down to mean water temperature axis. At 6" O.C line,t_w is about 56°F.

Example.3 Floor heating in a sub-floor construction

Design inputs:

q_u:30 Btu/h·ft²

r_u:1.86 (r_c/r_p:1.6<4,see Example 3 in Appendix 2)

M:6" O.C.

t_a:70°F, AUST:68°F (AUST-t_a:-2)

Then,from the nomograph,read (t_p-t_a) first on the right hand axis for floor heating/ceiling cooling scale. It is about 16°F.

Therefore,t_p : 70 + 16°F : 86°F.

In this problem r_c/r_p is less than 4. Therefore,solid lines should be used. This gives t_w: 140°F on the lower t_w axis at M:6" line.

Example.4 Ceiling heating with metal panels

Design inputs:

q_d:30 Btu/h·ft²

r_u:0.2 (see Example 4 in Appendix 2)

M :12" O.C.

t_a:74°F,AUST:70°F (AUST-t_a:-4)

Using the broken lines and the corresponding scale for ceiling heating on the (t_p-t_a) axis at right:

(t_p-t_a): 20°F at (AUST-t_a):-4°F vertical line (see Figure 8).

Therefore,t_p: 74°F + 20°F : 94°F. Intersecting q_d:30 Btu/h·ft² and r_u:0.2 lines and then this time going vertically up for t_w scale for ceiling cooling,t_w is about 105°F on 12" O.C line.

Table 2 compares the result with the analytical algorithm.

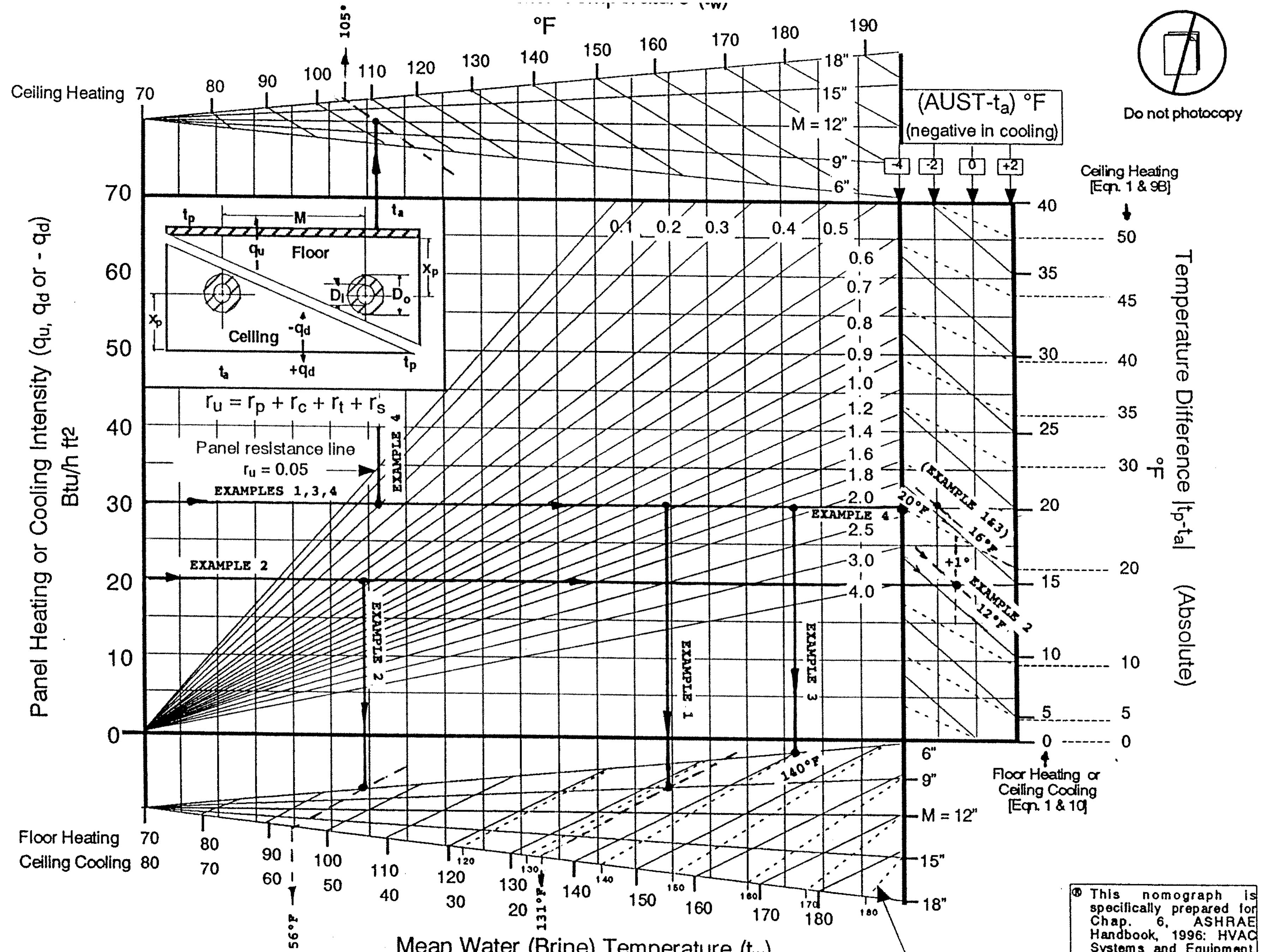


Figure 8. Nomograph Worksheet for Examples 1,2,3, and 4 °F

Table. 2 Comparison of Nomograph Solutions with Theory

Problem	Temperature Solution °F			
	Nomograph		Theory	
	t_p	t_w	t_p	t_w
Example 1	86	131	86	128
Example 2	68	56	68.5	56.5
Example 3	86	140	86	138
Example 4	94	105	94	103

In these examples the nomograph is consistent with the theory used. This consistency is further analyzed in Appendix 3.

Chapter 6 in ASHRAE Handbook was challenged with the same problems. Examples 2 and 3 could not be solved for t_w. The other two problems could be solved,but the results were different from DIN and the new algorithm (see Appendix 2). The design capabilities of ASHRAE guidelines are reviewed in Table 3,next section.

3. A Brief Review of the ASHRAE Handbook

In general,a handbook is rated with its technical content,relevance and conciseness. ASHRAE Handbooks are no exception and indeed keep displaying very good examples for many decades. ASHRAE Handbook Reviser's guide(1990) also emphasizes this feature.

In quote:

> "..ASHRAE Handbook volumes are the recognized authority in engineering procedures and current practices in the field of heating,..".

..application..oriented chapters cover current engineering
ractices.."

kis has already reported his suggestions and the results of his iew (Kilkis 1992-a,Kilkis,1992-b and Kilkis,1993-c),within the pe of the Reviser's Guide. The following section gives a condensed ormation about the evaluations of the Author:

EXT:

ucture:

enerally accepted order of content for a handbook chapter is; roduction,definition of basic concepts and terms,applications, ory,charts,tables,fundamentals of design,examples(if required), erences and appendices(if required). After receiving many itions and modifications on the same text over the years,current RAE Chapter 6 seems to need a re-organization of its structure, tent and order of presentation:

current text has some information which are repeated many es(i.e:information about plaster ceiling temperature limit is eated twice,ceiling panels are mentioned in two different tions, Radiant Energy Transfer section precedes the General luation section and there is another section on radiant heat nsfer,later).

s may make it rather difficult for the reader to grasp the whole cept.

hnical Language:

e of the terms and symbols are not consistent. For example,in ation 4 subscript f is used for the emittance of a fictitious face. Right below this equation,subscript r is used for the same iable. Same is also true for F_e and F_c symbols which are used for same radiation interchange factor in Equations 3 and 4.

Some symbols are not defined or explained: D in Figure 4 is unknown. It should be D_e; equivalent diameter.

t_p and r_u are not clearly defined. The reader does not really know how r_u is calculated or derived. t_p is the effective surface temperature. It is in fact a defined temperature: It is the surface temperature of a uniformly heated (cooled) plate which has the same heat transfer intensity with the actual panel. This is not explained and the reader is not informed about the temperature nonuniformity in actual panels.

Some variables have no symbol (like combined heat transfer intensity in cooling and wall heating).

These examples indicate that the text was not entirely overviewed for the uniformity of the language and terminology while individual corrections and additions were made over the years.

Technical language needs some cosmetic but fundamentally important changes. Some examples:

-**Radiation removed** .. by a cooling panel (should be appropriate to say heat removed by radiation.. by a cooling panel)

-**Heat transfers...** (should be: Heat is transferred..)

-**Radiant energy exchanges..** (here heat and energy is confused. Should be: Radiant heat is exchanged...)

-..**source of energy,creating rays** (should be: ..source of heat, transmitted at a wavelength...

-..**Radiation transfer..**(should be: .. heat transfer by radiation)

-..**wavelength produced by temperature of..**(should be: ..at a wavelength corresponding to the temperature of..)

-..**heat transfer is lost..**:(could be:heat transfer rate diminishes..)

e convection coefficient q_c..(here q_c is not a coefficient it is e convection heat transfer intensity in Btu/h·ft²)

issivity: Chapter 3 (see page 8) in Fundamentals Handbook very early states that for "real surfaces" the terminology adopted by HRAE is **emittance**,reflectance and absorptance.. Emisssivity, flectivity and absorptivity are the words used for a "polished" rface. Therefore the current text is not consistent with ASHRAE rminology.

ures and tables:

Figure 4,lines representing ASHRAE Laboratory data is checked th the original paper: these lines do not coincide with the iginal data.

me figures are not consistent with convection equations in terms the sign. For example,in Figure 9 heat removal is positive,but uation 10 gives a (-)ve convection heat transfer intensity in oling because $t_p<t_a$. This confusion is a consequence of the fact at the current Chapter has not identified any appropriate sign nvention for the heat transfer,which is a must in a chapter where ating and cooling are handled together.

gures 5 and 6 are helpful only in panel heating to determine AUST. ere is no any data regarding AUST in cooling.

gures 7 and 8 were criticized earlier (Kilkis 1993-b and Kilkis 93-e). Table 3 given in the next section shows that these figures n only handle 3 out of 12 panel types which are mentioned in the isting Chapter. These charts are not documented in terms of the eory and model used.

e reader can not understand how these figures were drawn and does t get any clue about its accuracy or applicable limitations. There not any reference cited either.

-Tables 2,3,and 4 seriously restrict the design,because spacing and tube material menu are quite limited. Also panel construction menu is very limited. In addition,the difference between any r_u value with respect to ferrous or nonferrous tube options is only about 0.04. Even the most thermally conductive plastic pipe with least thickness has an r value of about 0.2. (see Table B-2 and Equation B-4 in Appendix 1). This indicates that tables are not consistent with general theory of heat transfer in pipes (DIN,1990).

-Although some figures equally apply for panel cooling and wall panels in addition to floor heating panels,they are only labeled for floor heating.

-Temperature units are mixed: some are written in deg.F,some has just F,few has the correct symbol °F. Some does not have a temperature symbol at all.

-In some figures,the title and the figure itself use different symbols or terms. For example,titles of Figures 22 and 23 use the term pipe,but figures use the term tube. Figure 22 is not a correct engineering drawing either.

-Figure 9 in IP edition is not consistent with Equations 5 and 10. Sensecall(1992) pointed out this problem earlier. Kilkis (1992-b) explained that Figure 9 employs Wilkes and Peterson Data from Figure 4,instead of ASHRAE data for natural convection.

-Proper sizing and location of the expansion tank is very important in hydronic panel systems. Figure 10 does not mention these critical elements,except the choke valve and the water pump.

The text of several Figures were corrected. These corrections in draft form are given in Appendix 7 in order to help the reader to distinguish the typed revisions mentioned in the next section.

ıations:

ıation 2 is wrong. There should be no 100 terms in the ıominators,because this is already included in Stefan-Boltzman ıstant. F_e symbol is not consistent with F_e. The superscript 2 ›ve q_r is confusing.

ıation 5B takes care of the condition t_p<AUST in cooling. However .s is not distinguished in the convection equations. Therefore one :s positive q_r but negative q_c in cooling. However Figure 9 "adds" ·m together.

ıations 9 and 10 has small rounding errors. New constants after :erring to original Min and et.at's data is 0.20 and 0.31 ;pectively.

: above equations are for the heat transfer at the panel surface. ·re is not any other equation in the entire text to complement the ;ign. The design is solely based on Figures 7 and 8. Therefore ›ling,wall heating,some floor heating systems like sub-floor ›lications and heated air panels can not be designed. In SI version the same chapter there are many basic unit conversion errors too. ·se were already pointed out and an errata sheet was released in 2 (Kilkis,1992-a).

ıpleteness:

: design material in the chapter is not complete.Table 3 gives a ·vey about the design capabilities of the current Chapter 6. :hough about 13 panel systems are described in detail with ustrations,only three of them can actually be designed completely. other cases the required mean water temperature needs to be ;ssed at best for a given spacing. Examples to the last category is ling cooling,wall heating or cooling,heated floor,and sub floor ting. AUST cannot be calculated for cooling too,because Figures 5

Table 3

Investigation of Actual Design Capabilities of Chapter 6 ASHRAE Handbook, for the Applications Mentioned in the Same Chapter

Panel Information			Design Capabilities							
			Outputs Given		Design Parameters Handled:					
Function	Location	Type	Tp	Tw	M	Ta	AUST	Tubing	Coverings	Comments
Heating	Floor	Slab	Yes	Yes	1/2	1(3)	0	1/2	1	Design is possible
		Tube in/under subfloor	Yes	No	0	0	0	0	0	Design not possible
		Thin Slab on Subfloor	Yes	No	0	0	0	0	0	Design not possible
		Air Heating	Yes	No (t_{air})	0	0	0	0	0	Design not possible
	Ceiling	Plaster	Yes	Yes	1/2	1	0	0	N/A	Design is possible (only for types given in Table 4)
		Metal	Yes	Yes	1/2	1	0	N/A	N/A	Design is possible (only for types given in Table 1)
		Embedded	Yes	No	0	0	0	0	N/A	Design not possible
	Wall	Any type	Yes	No	0	0	0	0	0	Design not possible
Cooling	Ceiling	Embedded	Yes	No	0	0	0	0	N/A	Design not possible
		Plaster	Yes	No	1/2	0	0	1/2	N/A	Design not possible
		Metal	Yes	No	1/2	0	0	1/2	N/A	Design not possible
	Wall	Any type	Yes	No	0	0	0	0	0	Design not possible

1) Available ta in Figures 7 and 8 are only 70, 77, 74 and 76ºF
2) Tubing is only classified as metal/non-metal. Material can not be specified.
3) **Legend: 1 = Fully handled; 1/2 = Partly handled; 0 = Not handled**

6 are only for heating. There is not any analytical algorithm or ograph to determine the tube spacing and mean chilled water ine) temperature (see Example 2 in Appendix. 2).

floor heating r_c and r_d are mentioned but the way they are related r_u are unknown to a novice designer.

sistency:

onsistencies within the text, references and the data cited are tioned in different sections of the report.

erences:

erences are not updated. There is not any reference for panel ling. Citations to ASHRAE standards and publications need to be ated too. i.e: The maximum permissible floor temperature is given be 85°F,referring to ASHRAE Standard 55-1981. This is now 84°F ording to ASHRAE Standard 55-1992.

advantages of panel heating and cooling systems are referred to p.3 in ASHRAE Fundamentals Handbook. This chapter mentions high perature radiant system's probable disadvantages,and these have no evance to Chap.6

UNDAMENTAL THEORY

rent Chapter 6 bases the fundamentals of panel design on Figures and four tables for r_u. If r_u is not available in these tables n the design can not be completed.

validity of the figures and the tables were criticized earlier in s report. Appendix 1 discusses the importance of the split of r_u. other words, just to figure out r_u may not be sufficient in many es especially in floor heating.

Let us take two cases with the same tube spacing,M:

Case 1:

A concrete heated slab with heavy carpeting

r_u:2.2 (r_p: 0.3, r_c: 1.7, r_t: 0.2, r_s is negligible).

Case 2:

A plywood heated subfloor

r_u:2.2 (r_p: 0.8, r_c: 1.0, r_t: 0.2, r_s:0.2).

Both cases have the same r_u and according to Figure 8,the mean water temperature will be the same for a given heat output intensity !
It is very clear that this is not true,because in the first case the concrete slab will have a better heat diffusion and temperature distribution than a plywood subfloor. This will definitely effect the solution, especially if the required heat output intensity is high. Although both cases have the same total thermal resistance r_u,the latter will require a higher mean water temperature. It should be also pointed out that there is no any other design tool in the chapter in order to correct the design at such instances. The new analytical algorithm however can model the r_u split. The reader is urged to solve the above two cases using the new algorithm given in Appendix 2.
The new nomograph also recognizes that the split of r_u may be very important,and that is why broken lines were incorporated at the lower t_w axis on the right hand section. These lines display that r_c/r_p ratio is effective in design. If the new nomograph would be used,then the mean water temperature should be read from the broken lines for case 1 (r_c/r_p:5.66 and >4). For the latter case,solid lines should be used (r_c/r_p:1.25 and <4).Except tables 1,2,3 and 4 where r_u is given in terms of a limited hose spacing M, the effect of hose spacing on t_w is not defined. Also Figures 7 and 8 do not recognize AUST.

Revised Chapter 6 for ASHRAE 1996 Handbook

General Comments

p.6 of ASHRAE Handbook,1992 was extensively revised and updated lowing the general principles given in Reviser's Guide (ASHRAE, 0). Although the original philosophy and text was retained to a ge extent,the format and the content was substantially rganized. This reorganization also helped to include more ormation while the length of the text was kept the same. It should noted that the length of the text of the suggested Chapter 6 will shorter in ASHRAE Handbook when the fonts will be reduced to RAE style. The reader is also advised to compare the suggested pter 6 with Panel Heating and Cooling Chapter in 1984 Handbook: t handbook included some more design data at an expense of many es and yet the coverage was limited.

s compaction of information was made possible primarily by the new lytical algorithm which applies virtually to any kind of panel tem,provided that the characteristic panel thickness is properly ined. This algorithm therefore made it possible to eliminate the d to give separate design information for each panel system tioned in the text: Drake (1993) already pointed out that current p.6 needs to cover more data about tube spacing etc. Indeed the rent design information is sufficient only for about 3 panel types hough 12 panel types or so are described in the text (see le 3). However,with the current ASHRAE definition for the panel istance (a function of hose spacing) it would be necessary to add more table in order to cover the design information for every itional panel system. The new algorithm also eliminated Tables 2,3 4. Thus the analytical algorithm and the new nomograph enabled to cover more information in a rather simple manner. It must be noted that the new nomograph can be disintegrated into a set of simpler nomographs if required. (see Figures 5,6,and 7 in Section 3). Due to extensive nature of revisions,the entire chapter was typed on a word processor for the convenience of ASHRAE. This suggested Chap.6 for ASHRAE Handbook:1992.

4.2 The Suggested Chapter 6

This Chapter is enclosed. Highlights of this Chapter are:

1- Consistency is established.

2- 14 more references were added in order to reflect the recent developments,including panel cooling.

3- A sign convention was established for heating and cooling panels. This eliminated the confusion and probable mistakes.

4- r_u is defined independent of M. This enabled to eliminate the need of tables for every panel type and dimension. In fact this definition is more sound in the sense that the split is recognized.

5- A single nomograph replaces Figures 7 and 8 yet can be used also for cooling. AUST,M and r_c/r_p are recognized.

6- Analytical algorithm can handle all panel types and configurations.

7- Figures were revised and corrected.

8- Terminology and the technical language were improved,including the labels of the figures.

9- Consistency between the figures and their cited data was established by redrawing them whenever necessary.

10-Equations were revised,corrected and a new analytical algorithm was included.

Definitions of terms and symbols were reviewed and the missing ones were generated.

A new table for the thermal conductivity for various non metallic tubes were added (see Table 2 in the new text).

R table for floor coverings was expanded (see Table 3 in the new text).

A new convection equation for ceiling heating was added.

Equation 5-b was removed because it became obsolete after the new sign convention.

Design of air heated floor was introduced.

The change in the electrical resistance with the skin temperature of the cable was introduced by an additional equation.

The information in Distribution and Layout section were individually transferred to more appropriate sections and this section was removed.

CHAPTER 6

PANEL HEATING AND COOLING

ANEL heating and cooling systems use controlled emperature surfaces in the floor, walls or ceiling; the erature is maintained by circulating water, air, or electricity ıgh a circuit embedded in or attached to the panel. They be combined with a central station air system of one-zone, tant temperature, constant volume design, or with dual- , reheat, multi-zone or variable volume systems.

heating and cooling, room thermal conditions are ıtained primarily by radiation heat transfer, rather than by 'ection. A controlled temperature surface is referred to as a ınt panel if 50% or more of the heat transfer at the surface is ıdiation. This chapter is concerned with surfaces which are ›rimary sources of sensible heating and cooling within the litioned space, and whose temperatures are controlled.

gh-temperature surface radiant panels over 160°F gized by gas, electricity, or high-temperature water are ıssed in Chapter 15.

PANEL HEATING AND COOLING SYSTEMS

most common forms of panels are as follows:

Metal ceiling panels
Embedded tubing or attached piping in ceilings, walls, or floors
Electric ceiling panels
Air-heated floors or ceilings
Electrically heated ceilings or floors

:ating applications usually consist of tube, pipe coils, or ric resistance elements embedded in masonry floors or er ceilings. This construction is suitable where loads are e and solar effects are minimized by building design. 'ever, in buildings where glass areas are large and load ıges occur faster, the slow response, lag, and override effect asonry panels are unsatisfactory. Light metal panel systems ɔnd quickly to load changes (Berglund *et al.*, 1982). These ›ften located in the ceiling because it is exposed to all other ıces and objects in the room. It is not likely to be covered, e the floors, and higher surface temperatures can be used.

arm air and electric heating elements are two design :epts used in systems influenced by local factors. The warm ʏstem has a special cavity construction where air is ›lied to a cavity behind the panel surface. The air leaves the y through a normal diffuser arrangement and is supplied to oom. Generally, these systems are used as floor radiant :ls in schools and in floors subject to extreme cold, such as ı overhang. Cold outdoor temperatures and heating medium temperatures must be analyzed with regard to potential damage to the building construction. Electric heating elements embedded in the floor or ceiling construction and unitized electric ceiling panels are used in various applications to provide both full heating and spot heating of the space.

Thermal comfort, as defined in ASHRAE *Standard* 55-1992, is "that condition of mind which expresses satisfaction with the thermal environment." No system is completely satisfactory unless the three main factors controlling heat transfer from the human body (radiation, convection, and evaporation) result in thermal neutrality.

Designers sometimes think that panel heating and cooling systems are desirable only for certain types of buildings and only in some climates. However, maintaining correct conditions for human comfort primarily by radiation is possible for even the most severe climatic conditions (Buckley, 1989).

Panel heating and cooling systems provide an acceptable thermal environment by controlling surface temperature within an occupied space, thus primarily affecting the radiant heat transfer. With a properly designed system, a person should not be aware that the environment is being heated or cooled. The mean radiant temperature (MRT) has a strong influence on body comfort. When the temperature of the surfaces comprising the building deviates excessively from the ambient temperature (particularly outside walls with large amounts of glass), convective systems sometimes have difficulty in counteracting the discomfort caused by cold or hot surfaces. Heating and cooling panels neutralize these deficiencies and stabilize radiation losses or gains by the body.

Most building materials have surfaces with relatively high emittance factors and, therefore, absorb, re-radiate, and reflect radiant heat from active panels. Warm radiant panels are effective because heat transferred by radiation is absorbed and reflected by the irradiated surfaces, and not transmitted through the construction. Glass is opaque to the wavelengths emitted by active panels and, therefore, transmits little of the long-wave radiation to the outside. This is significant because all surfaces within the room tend to assume temperatures that result in an acceptable thermal comfort condition.

GENERAL EVALUATION

The principal advantages of radiant panel systems are:

- Comfort levels can be better than those of other conditioning systems, because radiant loads are treated directly, and air motion in the space is at normal ventilation levels.
- Space conditioning equipment is not needed at the outside

walls—simplifying the wall, floor, and structural systems.

- Almost all mechanical equipment may be centrally located —simplifying maintenance and operation.
- No space is required within the room for space conditioning equipment. This feature is especially valuable in hotels, hospital patient rooms, and other applications where space is at a premium, where maximum cleanliness is essential, or where dictated by legal requirements.
- Draperies and curtains can be installed at the outside wall without interfering with the space conditioning system.
- Cooling and heating can be simultaneous, without central zoning or seasonal changeover, when four-pipe systems are used.
- Supply air quantities usually do not exceed those required for ventilation and dehumidification.
- A 100% outdoor air system may be installed with less severe penalties in terms of refrigeration load because of reduced air quantities.
- A common central air system can serve both the interior and perimeter zones.
- The modular panel option provides flexibility to meet changes in partitioning.
- Wet surface cooling coils are eliminated from the occupied space, reducing the potential for septic contamination.
- The panel system can use the automatic sprinkler system piping (see NFPA 13, Chapter 5, Sections 5-6). The maximum water temperature must not fuse the heads.
- Space conditioning by panel heating and cooling systems and minimum supply air quantities provide a draft-free environment.
- Noise associated with fan coil or induction units is eliminated.
- Peak loads are leveled as the result of thermal energy storage in the panel, walls and partitions exposed to it.
- The systems are energy-efficient and give the opportunity to reduce conditioning costs (Brunk, 1993). It is possible to use water at low temperatures—available from solar collector systems, waste heat sources, condenser water heat reclaim systems, or heat pumps—by selecting the tube spacing accordingly. Solar absorption systems may be effectively employed for cooling (Kilkis, 1993-a).
- Can be safely used where high-temperature heating units are prohibited by the presence of hazardous or toxic materials.

HEAT TRANSFER AT PANEL SURFACES

A heated or cooled panel transfers heat to or from a room by radiation and natural convection. Some forced convection may also be present if air velocity is high.

Radiation Heat Transfer Intensity

Heat: (1) is transferred by radiation at the speed of light; (2) travels in straight lines and can be reflected; and (3) elevates the temperature of solid objects by absorption, but does not heat the gaseous medium (air) through which it travels.

Heat is exchanged continuously between all bodies in a building environment by radiation. The rate of radiation heat transfer depends on the following factors:

- Temperature (of the emitting surface and receiver)
- Emittance (of the radiating surface)
- Reflectance, absorptance, and transmittance (of the receiver)
- The view factor between the emitting surface and rec (viewing angle of the occupant to the radiant source)

A critical factor is the structure of the body surface. In general, rough surfaces have low reflectance and high emittance/absorptance characteristics. Conversely, smoot polished surfaces have high reflectivity and low absorpti emissivity characteristics.

One example of radiant heating is the feeling of warmt when standing in the sun's rays on a cool, sunny day. Sor the rays come directly from the sun and include the entire electromagnetic spectrum. Other rays from the sun impin surrounding objects and are absorbed or reflected. This generates a secondary source of radiant heat—at a wavel corresponding to a combination of the temperatures of th objects and the reflected rays. If a cloud passes in front of sun, there is an instant sensation of cold. This sensation is caused by the decrease in radiant heat received from the although there is little, if any, change in the surrounding ambient air temperature.

The basic equation for a multi-surface enclosure with g diffuse isothermal surfaces is derived by radiosity formula methods (Chapter 3 of the 1993 ASHRAE *Handbook—Fundamentals*). This equation may be written as:

$$q_r = J_p - \sum_{j=1}^{n} F_{pj} J_j \qquad (1)$$

where

q_r = net radiation heat transfer intensity on the pan surface, Btu/h · ft^2
J_p = radiosity of the panel surface, Btu/h · ft^2
J_j = radiosity of all other surfaces in room, Btu/h · f
F_{pj} = radiation angle factor between panel surface a another surface in the room (dimensionless)
n = number of surfaces other than the panel

Equation (1) is applicable to simple and complex enclo with varying surface temperatures and emittances. The ne ation transferred by the panels can be found by determini unknown J_j if the number of surfaces is small. More comp enclosures require computer calculations.

Radiation angle factors can be evaluated using the char Chapter 3 of the 1993 ASHRAE *Handbook—Fundamenta* Fanger (1972) shows room-related angle factors, or they r developed from algorithms in ASHRAE *Energy Calculatio* (1976).

Several methods have been developed to simplify Equation (1) by reducing a multi-surface enclosure to a tw surface approximation. In the MRT method, the radiant in change in a room is modeled by assuming that each surfa radiates to a fictitious surface that has a given area, emitta and temperature—simulating the same heat transfer from surfaces in the real multi-surface case (Walton, 1980). In lition, angle factors do not need to be determined in the luation of a two-surface enclosure. The MRT equation may written as:

$$q_r = \sigma F_e \left[T_p^{\,4} - T_r^{\,4}\right] \qquad (2)$$

ere

q_r = net radiation heat transfer at the panel surface from/to room surfaces, Btu/h · ft²

T_p = effective surface temperature of heated (cooled) panel, °R

T_r = temperature of the fictitious surface (unheated or uncooled), °R

σ = Stefan-Boltzman constant, 0.1714 x 10^{-8}·Btu/h · ft² ·°R

ıe temperature of the fictitious surface is given by an tance weighted average of all surfaces other than the panel.

$$T_r = \sum_{j\neq p}^{n} A_j \epsilon_j T_j \Big/ \sum_{j\neq p}^{n} A_j \epsilon_j \qquad (3)$$

re

= area of surfaces other than panel

= surface emittances other than panel (dimensionless)

j = effective surface temperature of unheated (uncooled) surfaces, °R

ıen the emittances of an enclosure are closely equal and ces exposed to the panel are marginally unheated ɔoled), then Equation (3) becomes the area-weighted .ge temperature (AUST) of unheated (or uncooled) sur- exposed to the panels.

ɜ radiation interchange factor or Hottel equation for two- ɔe radiation heat exchange is given by:

$$e = \frac{1}{1/F_{p\text{-}r} + \left[(1/\epsilon_p) - 1\right] + A_p/A_r\left[(1/\epsilon_r) - 1\right]} \qquad (4)$$

ɜ

e = radiation interchange factor (dimensionless)

·r = radiation angle factor from panel to fictitious surface (1.0 for flat panel)

. = area of panel surface and fictitious surface, respectively

. = emittance of panel surface and fictitious surface, respectively (dimensionless)

ctice, the emittance of nonmetallic or painted metal non- ing surfaces is about 0.9. When this emittance is used in on (4), the combined factor is about 0.87 for most rooms. stituting this value in Equation (2), the constant becomes 0.15. Min *et al.* (1956) showed that this constant was n their test room. The radiation equation for heating or g becomes:

$$q_r = 0.15 \cdot 10^{-8}\left[(t_p + 460)^4 - (AUST + 460)^4\right] \qquad (5)$$

where

t_p = effective panel surface temperature, °F

AUST = area-weighted average temperature of uncontrolled surfaces in the room, °F

According to Equation (5), q_r is positive in heating and negative in cooling. This also establishes the general sign convention for this chapter, which states that any heat flux from the panel is regarded positive. Otherwise, it is negative.

The actual transfer by radiation in a room may be somewhat different from that given by Equation (5) because of non-uniform temperatures, irregular room surfaces, variations in surface emittance of materials, etc. However, the equation is accurate to within 10% when used in conventional heating and cooling calculations. Stemman (1989) found that the accuracy of Equations (2) and (5) was improved by adding back geometric angle factors to these equations. These factors allow the accuracy of the MRT method to approach that of Equation (1) and still maintain a first order temperature solution.

Radiation heat transfer calculated from Equation (5) is given in Figure 1. The values apply to ceiling, floor, or wall panel heat output. Heat removed by a cooling panel for a range of practical temperatures is given in Figure 2.

In many specific instances where normal multi-story commercial construction and fluorescent lighting are used, the room temperature at the 5-ft. level will closely approach the average uncooled surface temperatures (AUST). In structures where the main heat gain is through the walls or where incandescent lighting is used, the wall surface temperatures tend to rise considerably above the room air temperature.

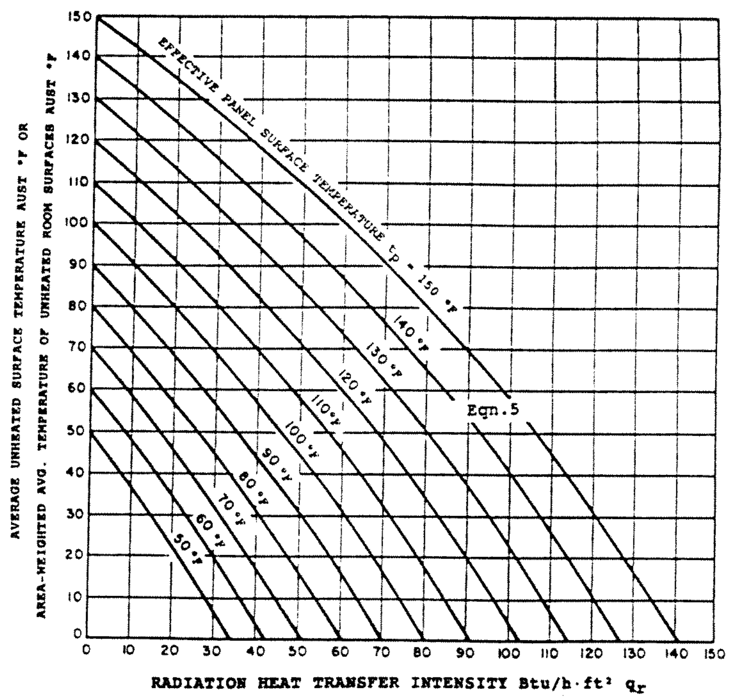


Fig. 1 Radiation Heat Transfer Intensity at Heated Ceiling, Floor, or Wall Panel Surface

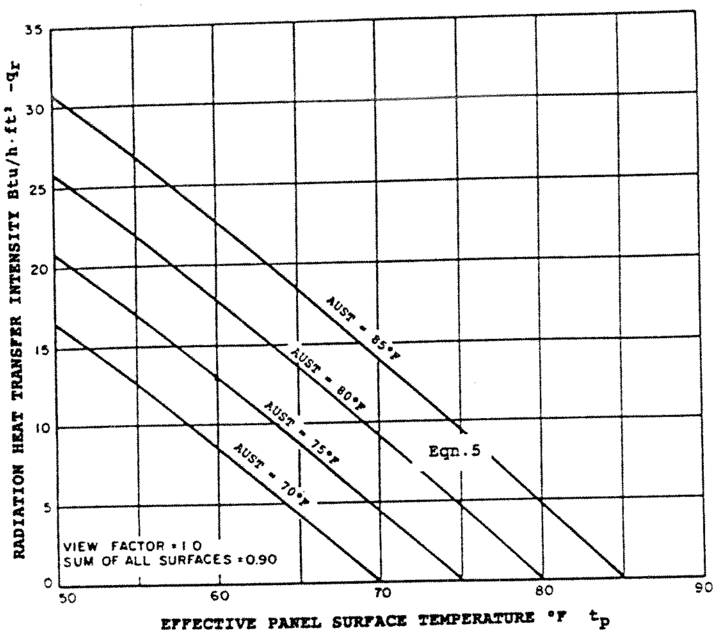


Fig. 2 Heat Removed by Radiation at Cooled Ceiling or Wall Panel Surface

Convection Heat Transfer Intensity

The convection heat transfer intensity q_c is defined as the heat transferred by convection in Btu/h ·ft² · °F difference between air and effective panel surface temperatures. Convection heat transfer values are not easily established. Convection in panel systems is usually natural; that is, air motion is generated by the warming or cooling of the boundary layer of air. In practice, however, many factors such as a room's configuration interfere with or affect natural convection. Infiltration, the movement of occupants, and mechanical ventilating systems can introduce some forced convection that will disturb the natural process.

Parmelee and Huebscher (1947) included the effect of forced convection on heat transfer from panels as an increment to be added to the natural convection coefficient. Harris and Sartain (1959) proposed a similar approach after observing rather strong downwash of air near cold window surfaces. However, increased heat transfer from forced convection should not be systematically used because the increments are unpredictable in pattern and performance, and forced convection does not significantly increase the total capacity of the panel system—except in rooms with large glass areas. Convection in a panel system is proportioned to the difference between the effective panel surface temperature and the temperature of the airstream layer directly wetting the panel. The most consistent measurements are obtained when the air layer temperature is measured close to the region where the fully developed stream begins, usually 2 to 2.5 in. away from the panel surface.

Min *et al.* (1956) determined natural convection coefficients 5 ft. above the floor in the center of a 12 ft. by 24 1/2 ft. room.

Natural convection from an "all-heated" ceiling,

$$q_c = 0.041\,(t_p - t_a)^{1.25}/D_e^{0.25} \quad (6$$

Natural convection from a heated floor or cooled ceiling

$$q_c = 0.39\,(t_p - t_a)^{1.31}/D_e^{0.08} \quad (7$$

Natural convection from a heated or cooled wall panel,

$$q_c = 0.29\,(t_p - t_a)^{1.32}/H^{0.05} \quad (8$$

where

q_c = heat transfer intensity by natural convection, Btu/h · ft²
t_a = temperature of airstream layer, °F
D_e = equivalent diameter of panel (4 area/perimete
H = height of wall panel, ft

Schutrum and Humphreys (1954) measured panel performance in furnished test rooms that did not have un temperature surfaces and found no variations large enou be significant in heating practice. Schutrum and Vouris (1 established that the effect of room size was also usually insignificant. Very large spaces like hangars and warehou are exceptions. In this case, for the natural convection he transfer intensity q_c, Equations (6) and (7), should be use Otherwise, convection equations can be simplified to:

Natural convection from an all-heated ceiling,

$$q_c = 0.020\,(t_p - t_a)^{1.25} \quad (9A)$$

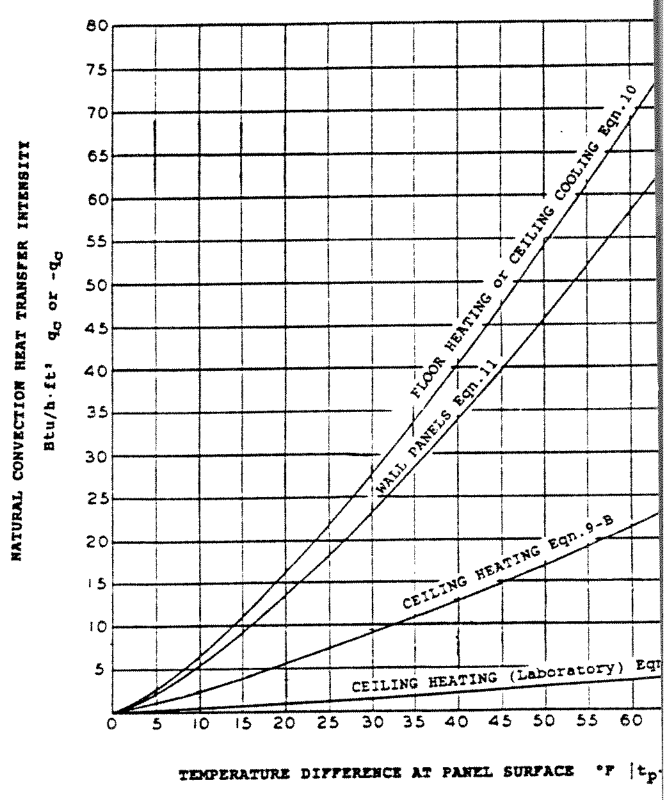
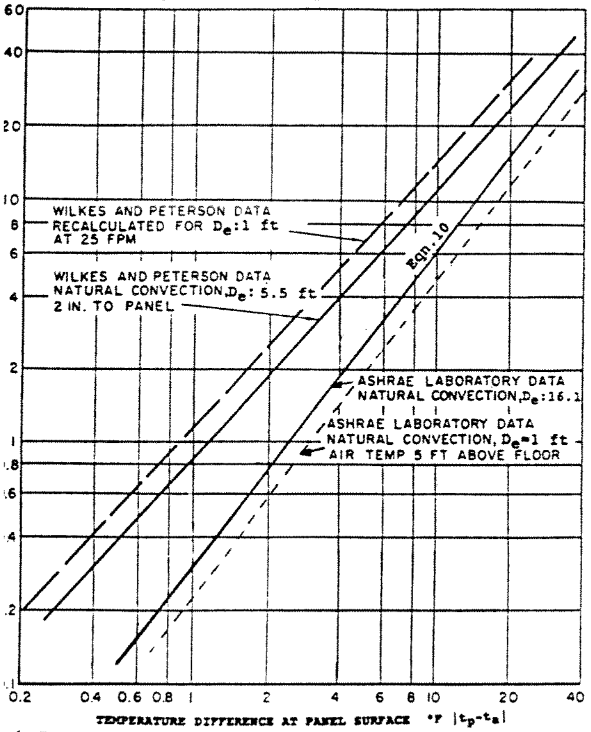


Fig. 3 Natural Convection Heat Transfer Intensity Floor, Ceiling, and Wall Panel Surfaces [Equations (9) (10) and (11)]



4 Heat Removal by Ceiling Cooling Panels with Natural Convection [Equation (10)]

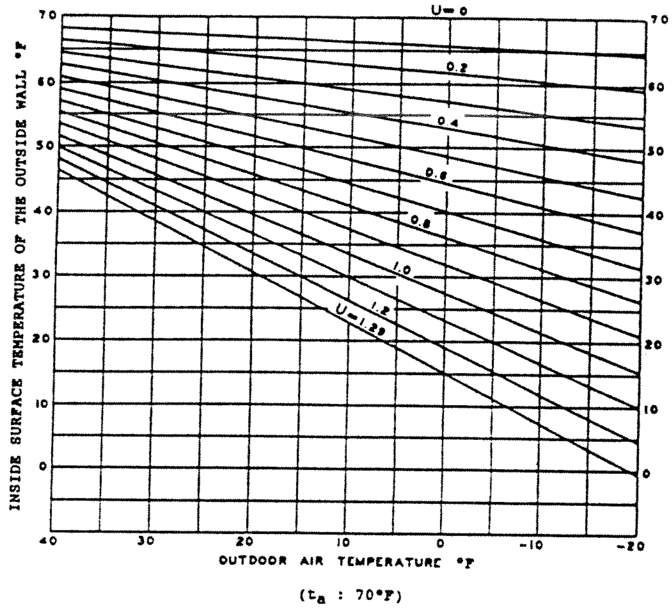


Fig. 5 Relation of Inside Surface Temperature to Overall Coefficient of Heat Transfer

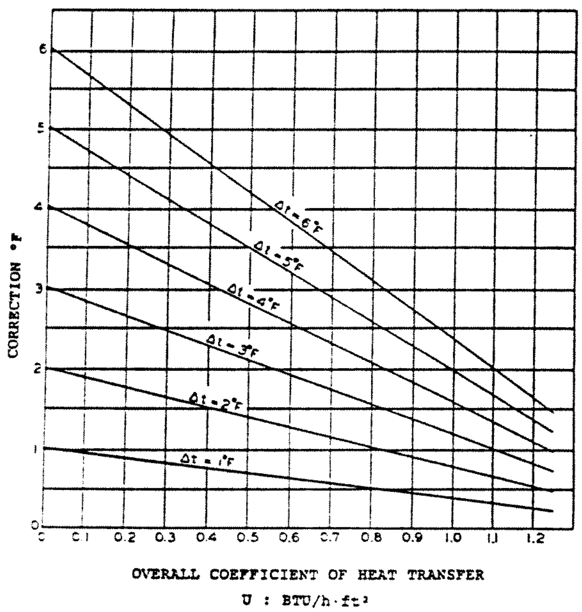


Fig. 6 Inside Wall Surface Temperature Correction for Air Temperatures Other than 70°F

ıtural convection from a heated ceiling may be enhanced :aving "cold strips" (unheated ceiling sections) between the ıg panels. These strips help in initiating the natural convec- (Raiss and Roedler, 1958). In this case, Equation (9A) may :placed by the following equation (Kollmar and Liese,):

$$q_c = 0.13\left(t_p - t_a\right)^{1.25} \quad (9B)$$

ırge spaces, this equation should be adjusted with the icient: $(16.1/D_e)^{0.25}$

:al convection from a heated floor or cooled ceiling,

$$q_c = 0.31\left(t_p - t_a\right)^{1.31} \quad (10)$$

:al convection from a heated or cooled wall panel,

$$q_c = 0.26\left(t_p - t_a\right)^{1.32} \quad (11)$$

r normal conditions t_a can be taken as the indoor design :erature. In spaces with large glass areas, t_a may be taken to AUST (Drake, 1993).

:ooling t_p is less than t_a, therefore, q_c is also negative.

: 3 shows heat transfer by natural convection at floor, ınd ceiling heating panels as calculated from ions (9A) (9B) (10) and (11).

ıre 4 compares heat removal by natural convection at ɟ ceiling panel surfaces, as calculated by Equation (10) ata from Wilkes and Peterson (1938) for specific panel An additional curve illustrates the effect of forced convec- n the latter data. Similar adjustment of the ASHRAE data is ropriate, but the effects would be much the same.

bined Heat Transfer Intensity

combined heat transfer intensity on the panel surface can ermined by adding the radiant heat transfer from Figure 1 or 2 to the convective heat transfer from Figure 3 or 4, respectively. Alternatively, equations for q_r and q_c can be employed, so that:

$$q = q_r + q_c \quad (12)$$

Following is the terminology for q:

- = q_u in floor heating (heat output)
- = q_d in ceiling heating (heat output)
- = $-q_d$ in ceiling cooling (heat removal)
- = q_s in wall heating (heat output)
- = $-q_s$ in wall cooling (heat removal)

calculating the AUST in the room. In calculating the AUST, the surface temperature of the interior walls is assumed to be the same as the room air temperature. The inside surface temperatures of outside walls and exposed floors or ceilings can be obtained from Figure 5 for a 70°F room temperature. Figure 6 gives corrections for other temperatures.

Tests by Schutrum *et al.* (1953a, 1953b), and simulation by Kalisperis (1985) based on a CAD program developed by Kalisperis and Summers (1985), show that the temperatures are almost equal. Steinman *et al.* (1989) noted that these temperatures may not be appropriate for enclosures with large glass areas or a high percentage of outside wall and/or ceiling surface area. These surfaces decrease (increase in cooling) AUST, which increases the heat transfer by radiation

Figure 7 shows the combined radiation and convection heat removal intensity for cooling, as given in Equations (5) and (10). The data in Figure 7 do not include heat gains from sun, lights, people, or equipment: manufacturer's data includes these heat gains.

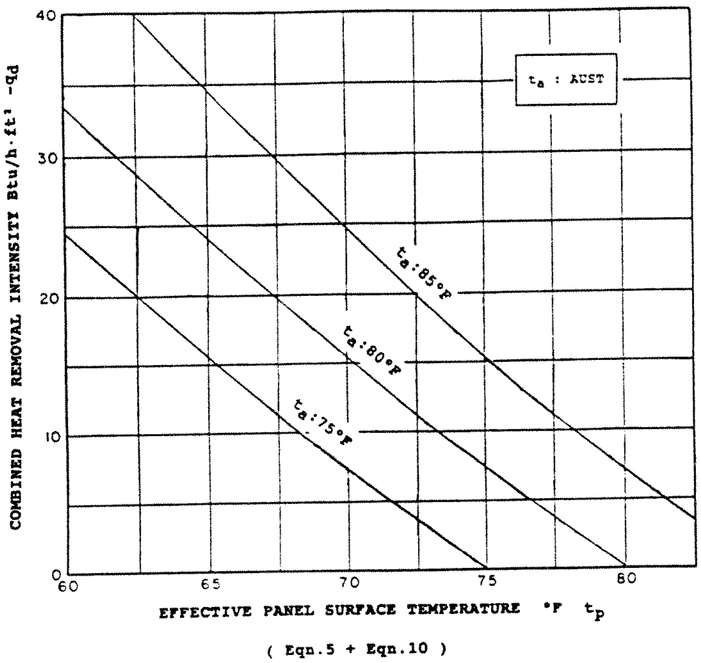


Fig. 7 Cooled Ceiling Panel Performance in Uniform Environment with No Infiltration and No Internal Heat Sources

Panel Thermal Resistance

Heat transfer in the panel to the surface is an important consideration. Any hindrance in between will affect the efficiency of the system. The thermal resistance to heat flow may vary considerably among panel systems, depending on the type of bond between the tubing (or wiring) and the panel material. Corrosion or adhesion defects between lightly touching surfaces, the method of maintaining contact, and other factors may change the bond with time. The actual thermal resistance of any proposed system should be verified by testing whenever practical. Specific resistance and performance data be obtained from the manufacturer, whenever available Basically, there are four types of thermal resistance:

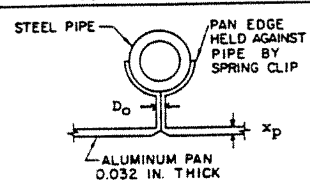
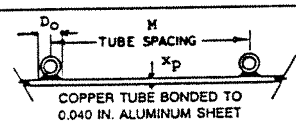
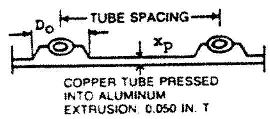
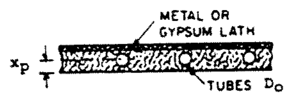
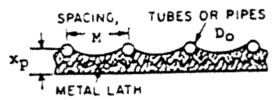
- Thermal resistance of the tube wall in a hydronic sy
- Thermal resistance between the tube (electric wire) the panel material, r_s
- Thermal resistance of the panel material, r_p
- Thermal resistance of the panel covers, r_c

The sum of these thermal resistances is the panel therma resistance, r_u:

$$r_u = r_t + r_s + r_p + r_c \quad (13$$

Generally, if the tubes or electric cables are embedded i slab, r_s may be neglected. However, if the tubes are atta the panel, r_s may be significant depending upon the qua bonding. Table 1 gives typical r_s values for various ceilin panels.

Table 1. Thermal Resistance of Ceiling Panels

Type of Panel	Thermal Resistance ft²·°F·h/Bt r_p	r_s
	x_p/k_p	0.20
	x_p/k_p	0.05
	x_p/k_p	0.05
	$(x_p-D_o/2)/k_p$	≈0
	$(x_p-D_o/2)/k_p$	≤0.05

r_p may be calculated if the characteristic panel thickne (ft) and the thermal conductivity of the panel material, k_p (Btu/h · ft· °F) are known.
If the tubes (electric wires) are embedded in the panel:

$$r_p = \frac{\left(x_p - D_o/2\right)}{k_p} \quad (14A$$

Here, D_o is the outer diameter of the tube (electric cable Hydronic floor heating by a heated slab or gypsum-plast ceiling heating are typical examples.

he tubes are attached to the panel:

$$r_p = \frac{x_p}{t_p} \quad (14B)$$

tal ceiling panels (see Table 1) and tubes under subfloor (see ure 20) are typical examples.

The following expression is a simplified version of the rmal resistance equation for a tube with circular cross tion and a thermal conductivity of k_t (Btu/h · ft · °F).

$$r_t = \frac{0.116 \cdot D_o/D_i - 0.107}{k_t} \quad (1.1 \le D_o/D_i \le 2) \quad (15)$$

ere D_i is the inside diameter of the tube, in ft. In metallic es, r_t may be ignored.

ently, the use of nonmetallic tubes has increased. alues of different tubing material is given in Table 2. Long gths are available up to 600 ft, which simplify installation. ne manufacturers have also developed splicing or fusion iniques for shorter tube lengths.

le 2. Thermal Conductivity of Typical Tube Material

erial	k_t (Btu/h · ft · °F)
l	48.50
per	17.00
nless Steel	9.20
density polyethylene (LDPE)	0.18
n density polyethylene (HDPE)	0.24
s-linked polyethylene (VPE)	0.22
ile reinforced rubber heat transfer hose (HTRH)	0.17
propylene block co-polymer (PP-C)	0.13
propylene random co-polymer (PP-RC)	0.14
butylene (PB)	0.13

nel coverings like carpets and pads on the floor can have a ounced effect on the performance of the panel systems. added thermal resistance r_c reduces the panel surface heat fer. In order to re-establish the required performance, the erature of the water must be increased (decreased in ng). Thermal resistance of a panel covering is:

$$r_c = \frac{x_c}{k_c} \quad (16)$$

re, x_c and k_c is the thickness (ft) and thermal conductivity h · ft · °F) of each panel covering. If there is more than one ing, individual r_c values should be added. Table 3 gives l r_c values for floor coverings.

el Heat Losses or Gains

at transfer from or to (in cooling) the back and edges of s is considered a panel loss (gain). Panel heat losses s) are part of the building load if the heat transfer exceeds uilding. Then these losses should be considered in siz- e boiler (chiller). If the heat transfer is limited to another tioned space, the panel loss (gain) is a source of heat for that space instead. In either case, the magnitude of panel loss should be determined. Panel loss (gain) per square ft. of panel area is the panel heat loss (gain) intensity, q_b. Following the sign convention adopted, it is negative in cooling.

Table 3. Thermal Resistance of Floor Coverings

Description	Thermal resistance, r_c ft^2 · °F · h/Btu
Bare concrete, no covering	0.00
Asphalt tile	0.05
Rubber tile	0.05
Light carpet	0.60
Light carpet with rubber pad	1.00
Light carpet with light pad	1.40
Light carpet with heavy pad	1.70
Heavy carpet	0.80
Heavy carpet with rubber pad	1.20
Heavy carpet with light pad	1.60
Heavy carpet with heavy pad	1.90
3/8" hardwood	0.54
5/8" wood floor (oak)	0.57
1/2" oak parquet and pad	0.68
Linoleum	0.12
Marble floor and mudset	0.18
Rubber pad	0.62
Prime urethane underlayment, 3/8"	1.61
48-oz. waffled sponge rubber	0.78
Bonded urethane, 1/2"	2.09

Panel heat loss (gains) should be kept to a reasonable amount by insulation. For example, a floor panel may overheat the basement below, and a ceiling panel may cause the temperature of a floor surface above it to be too high for comfort unless it is properly insulated.

In suspended ceiling panel systems, heat is transferred from the ceiling panel to the floor slab above (heating) and vice versa (cooling). This decreases the desired performance. For example, the ceiling panel surface temperature is affected because of heat transfer to or from the panel and the slab by radiation and, to a much smaller extent, by convection. The radiation component can be approximated using Figure 1. The convection component can be estimated from Figure 3 or 4. In this case, the temperature difference is that between the top of the ceiling panel and the mid-space of the ceiling. The temperature of the ceiling space should be determined by testing, since it varies with different panel systems. However, much of this heat transfer is nullified when insulation is placed over the ceiling panel, which, for perforated metal panels, also provides acoustical control.

If lighting fixtures are recessed into the suspended ceiling space, radiation from the top of the fixtures raises the overhead slab temperature and transfers heat to the space by convection. This energy is absorbed at the top of the cooled ceiling panels by radiation, as in Figure 2, and by convection, generally in accordance with Equation (9A). The amount of heat the top of the panel absorbs depends on the system. Most manufacturers have information available. Similarly, panels installed under a roof absorb additional heat, depending on configuration and insulation.

The heat loss from most panels can be calculated by using the coefficients given in Chapter 22 of the 1993 ASHRAE *Handbook—Fundamentals*. These coefficients should not be used to determine the downward heat loss from panels built on grade because the heat flow from them is not uniform (Sartain and Harris, 1956; ASHAE 1956, 1957). The heat loss from panels built on grade can be estimated from Figure 8 or from data in Chapter 25 of the 1993 ASHRAE *Handbook—Fundamentals*.

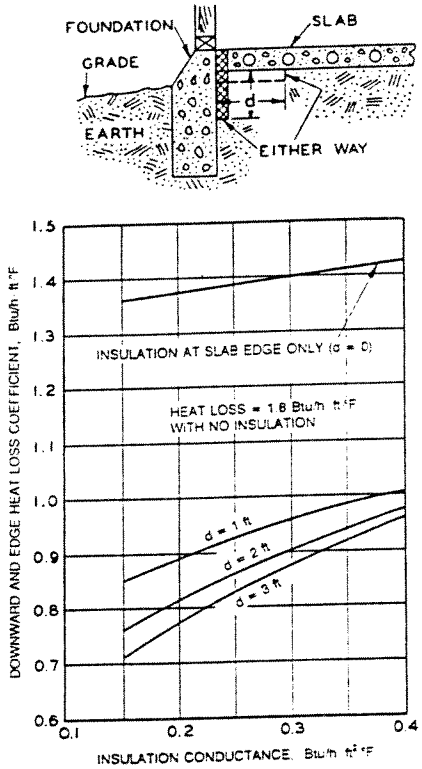


Fig. 8 Downward and Edgewise Heat Loss Coefficient for Concrete Floor Slabs on Grade

DESIGN OF RADIANT PANELS

The control of the panel surface temperature is accomplished by either hydronic or electric circuits. The required effective surface temperature, t_p, can be calculated by using applicable heat transfer equations for q_r and q_c. At a given AUST and t_a, it can be directly obtained from Figure 9, on its right hand vertical axis. Either way, AUST must be determined or assumed first. In a hydronic radiant panel, the mean water (brine) temperature necessary to maintain a required heat transfer intensity primarily depends upon the required effective surface temperature, the tube spacing and the panel resistance. Mean water (brine) temperature (t_w) readings in Figure 9 will correspond to skin temperature of the cable in electric resistance heating. After t_p is determined, the following simplified algorithm (Kilkis and Sager, 1993, TS 10464, 1993), which was used to generate Figure 9, may be directly used:

$$t_d \cong t_a + \frac{(t_p - t_a)\cdot M}{(2\cdot W\cdot \eta + D_o)} + q\cdot(r_p + r_c + r_s) \quad (17)$$

where

t_d = the mean skin temperature of the tubing (electric cable), °F

q = combined heat transfer intensity on the panel surface, Btu/h · ft²

t_a = indoor design temperature, °F

D_o = outer diameter of the tube, in ft, or characteris contact width of the tube with the panel (see Table 1).

M = spacing of the circuit on centers, ft

$2\cdot W$ = net spacing between tubing (electric cables), $(M - D_o)$, ft

η = fin efficiency, dimensionless

$$\eta = \frac{\tanh(f\cdot W)}{(f\cdot W)} \quad (18)$$

$$\eta \cong 1/(f\cdot W) \qquad \textit{if } (f\cdot W) > 2$$

f : fin coefficient, ft⁻¹

$$f \cong \left[\frac{q}{(t_p - t_a)\cdot 2\cdot \sum_{i:1}^{n} k_i \cdot x_i}\right]^{1/2} \qquad t_p \neq t_a \quad (19)$$

Here, n is the number of layers with different materials, including the panel itself. k_i and x_i are the thermal condu (Btu/h · ft · °F) and characteristic thickness (ft) of each lay

For a hydronic system, the required mean water (brine temperature is:

$$t_w = (q + q_b)\cdot M\cdot r_t + t_d \quad (20)$$

Here, q_b is the intensity of the back and edge heat losses heated panel (positive) or gains (negative) in a cooled pa These equations may be easily adapted for a small compu program.

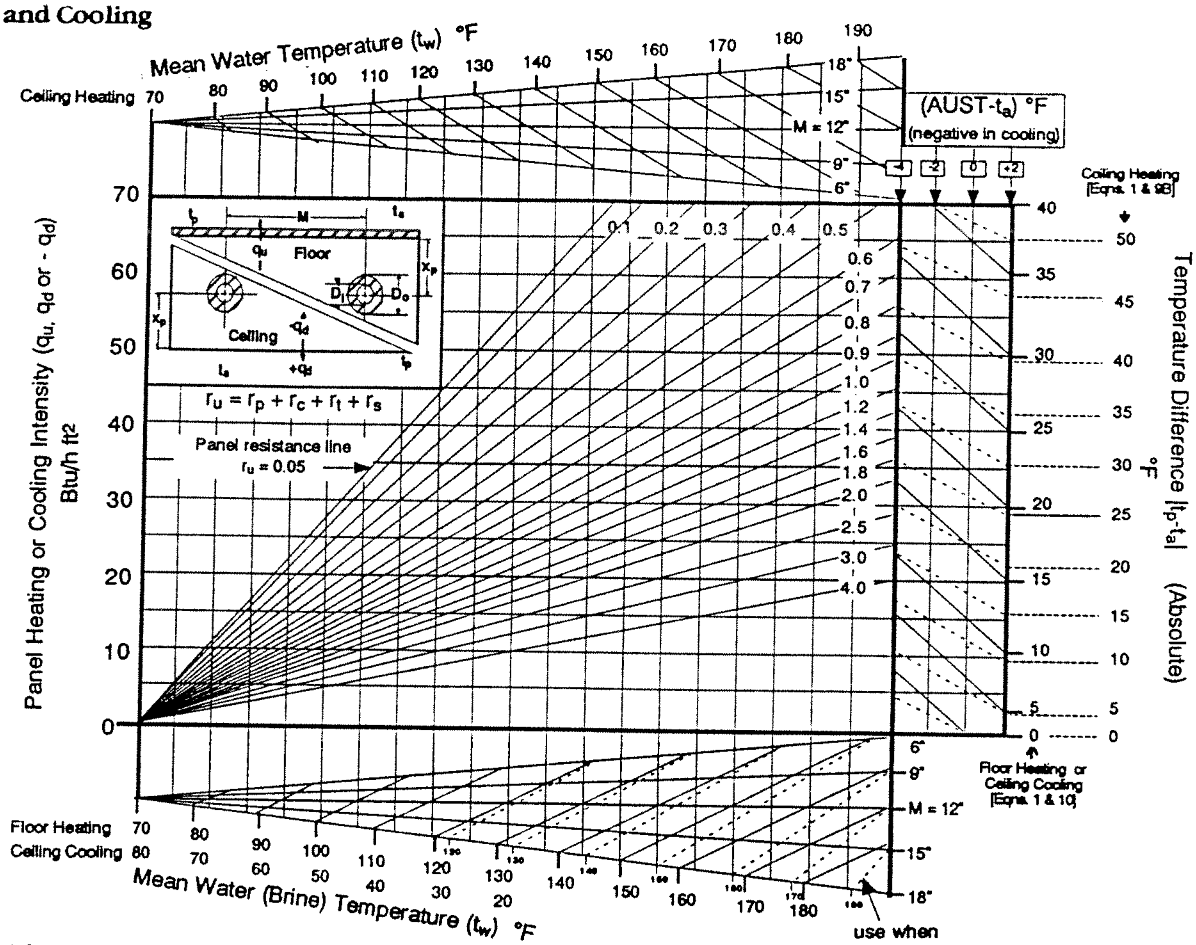
GENERAL DESIGN CONSIDERATIONS

The sensible heating and cooling loads are calculated i similar way for conventional systems. However, it should noted that:

- Indoor design air temperature may be chosen lower heating) or higher (in cooling) for an equivalent hum thermal comfort. See Chapter 51 in ASHRAE *Handbo* 1995 *Applications*, and Buckley, *et. al.*, (1987).
- Natural infiltration rate may be lower due to a more uniform air temperature, pressure, and velocity field maintained in the conditioned space.
- Peak loads are leveled (shaved) to the extent of ther energy stored in the panel, walls, and partitions expo to it.

Because the mean radiant temperature (MRT) within a heated space increases as the heating load increases, the controlled air temperature during this increase may be lo without affecting comfort. In ordinary structures with nor infiltration loads, the required reduction in air temperatur small—enabling a conventional room thermostat to be us

In panel heating systems, lowered night temperature ca duce less satisfactory results with heavy panels such as concrete floors. These panels cannot respond to a quick increase or decrease in heating demand within the relativ



9 Combined Nomograph For Cooling and Heating Panel Design

t time required, resulting in a very slow reduction of the
e temperature at night and a correspondingly slow pickup
e morning. Light panels, such as plaster or metal ceilings
walls, may respond to changes in demand quickly enough
atisfactory results from lowered night temperatures. Tests
metal ceiling panel demonstrated the speed of response to
omparable to that of convection systems. However, very
fuel savings can be expected—even with light panels—
ss the lowered temperature is maintained for long periods.
luced non-occupancy temperatures are employed, some
is of providing a higher-than-normal rate of heat input for
warm-up is necessary; for example, fast-acting radiant
ig panels (Berglund, 1982).
tal radiant heating panels, hydronic and electric, are
ed to building perimeter spaces for heating in much the
way as finned-tube convectors. Metal panels are usually
led in the ceiling and are integrated into the ceiling design.
ayout and arrangement of panels usually considers
ectural design.
titions may be erected to the face of hydronic panels, but
the active heating portion of electric panels because of
ble element overheating and burnout. Electric panels are
sized to fit the building module with a small removable
or dummy panel at the window mullion to accommodate
partitions. Hydronic panels can run continuously.
cutting and fitting, field modification of hydronic panels is
ble; however, modification should be kept to a minimum
p installation costs down. Electric panels cannot be
ied in the field.

Thermal expansion or contraction of panels, and other material in or adjacent to it, should be anticipated. Mean water temperature in hydronic systems or the cable temperature in electrically-heated systems must be selected according to the temperature limits of the materials in contact with the panel.

Oxygen-induced or bacteriological corrosion in hydronic systems may deteriorate the performance. Suitable additives should be considered.

Placing the thermostat on a side wall where it can see the outside wall and the panel should be considered. The normal thermostat cover reacts to the panel, and the radiant effect of the panel on the cover tends to alter the control point so that the thermostat controls 2 to 3°F lower when the outdoor temperature is at a minimum and the panel temperature is at a maximum in heating. Experience indicates that rooms heated with radiant panels are more comfortable under these conditions than when the thermostat is located on a back wall.

Floor panels are limited to surface temperatures of up to 84°F for comfort reasons in frequently occupied surfaces. (ASHRAE *Standard* 52-1992). For surfaces with little or no occupation, like perimeter bands not extending more than 3 ft. into the room, this limit is 95°F (DIN 4725, 1990).

In applications with normal ceiling heights, heating panels that exceed 160°F should not be located over the occupied area. Refer to Figure 15 for selecting effective ceiling surface temperatures.

Other factors to consider when using panel cooling systems:

- Evaluate the panel system to take full advantage of optimizing the physical building design.

- Select recessed lighting fixtures, air diffusers, hung ceilings, and other ceiling devices to provide the maximum ceiling area possible for use as ceiling panels.
- The air-side design must be able to maintain humidity levels at or below design conditions at all times to eliminate any possibilities of condensation on the panels. This becomes more critical if space dry- and wet-bulb temperatures are allowed to drift as an energy conservation measure, or if duty cycling of the fans is used.
- Do not place cooling panels in or adjacent to high humidity areas.

HYDRONIC PANEL SYSTEMS

Design Considerations

Hydronic radiant panel systems are similar to other air-water systems in the arrangement of the system components.

Chapter 3 and other chapters in this handbook covering hydronic systems apply to radiant panels. Layout and design of radiant panels for heating and cooling begins early in the job. The type of panel location type and construction chosen influences the design and, conversely, thermal considerations may dictate what type to be used.

Hydronic radiant panels can be used with two- and four-pipe distribution systems. Figure 10 shows the arrangement of a typical system. It is common to design for a 20°F temperature drop for heating across a given grid and a 5°F rise for cooling, but other temperature differentials may be used, if applicable.

Panel design requires determination of the panel area, panel type, supply water temperature, water flow rate, pressure drop, boiler (chiller) load, and the panel arrangement. Panel performance is directly related to room conditions. Air-side design must also be established. Heating and cooling loads may be calculated by procedures covered in Chapters 22 through 27 in the 1993 ASHRAE *Handbook—Fundamentals*. Provisions as listed in General Design Considerations, if applicable, may be applied. The general design procedure is as follows.

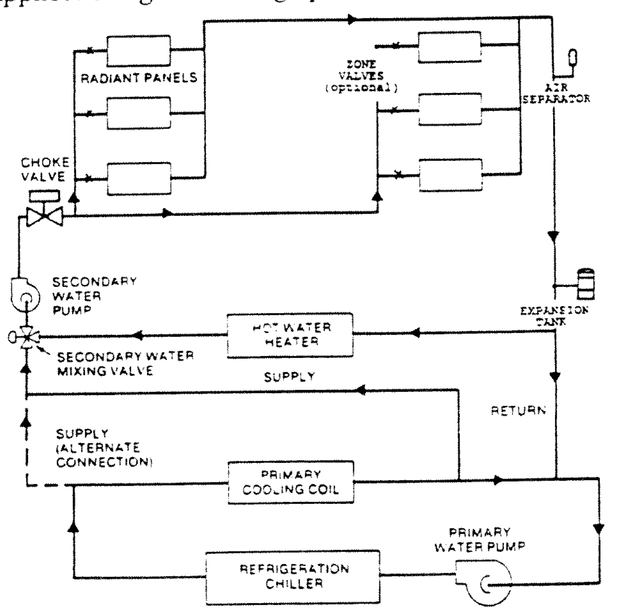


Fig. 10 Primary/Secondary Water Distribution System with Mixing Control

For cooling:

1. Determine room design dry-bulb temperature, rel[...] humidity, and dew point. Determine surface temp[...]tures of unheated surfaces.
2. Calculate room sensible and latent heat gains.
3. Establish minimum supply air quantity.
4. Calculate latent cooling available from the air.
5. Calculate sensible cooling available from the air.
6. Determine panel cooling load.
7. Determine panel area for cooling.
8. Determine required effective surface temperature o[...] panel. Refer to Figure 9, or use the equations.
9. Select the spacing of cooling circuits.
10. Determine mean water (brine) temperature for coo[...]

For heating:

1. Designate room design dry-bulb temperature for h[...]
2. Calculate room heat loss.
3. Determine panel area for heating.
4. Determine surface temperatures of unheated surfa[...] Refer to Figures 5 and 6 to find surface temperature[...] exterior walls and exposed floors and ceilings. Inte[...] walls are assumed to have surface temperatures eq[...] the room air temperature. Calculate AUST.
5. Design the panel arrangement.
6. Select the spacing of heating circuits.
7. Determine the mean water temperature.
8. Thermal comfort requirements should be checked [...] the following steps (see Chapter 8 of the 1993 ASH[...] *Handbook—Fundamentals*, and NRB, 1981).
   a. Determine occupant's clothing value (clo value) [...] metabolic rate (MET). (See Tables 1D and 4A, Ch[...] of the 1993 ASHRAE *Handbook—Fundamentals*[...]
   b. Determine the optimum operative temperature at[...] coldest point in the room (see Figure 15, Chapter [...] the 1993 ASHRAE *Handbook—Fundamentals*, o[...] Figure 2.3.22 for other values).
   c. Determine the mean radiant temperature (MRT) a[...] coldest point in the room (see Fanger, 1972).
   d. From the definition of operative temperature, est[...] the optimum room design temperature at the col[...] point in the room. If the optimum room design te[...]ature varies greatly from the designated room des[...] temperature, designate a new temperature.
   e. Determine the MRT at the hottest point in the roo[...]
   f. Calculate the operative temperature at the hottest [...] in the room.
   g. Compare the operative temperatures at the hottes[...] coldest points in the room. For light activity and n[...] clothing, the acceptable operative temperature ra[...] 68 to 75°F (see NRB, 1981, for other ranges). If th[...] range is not acceptable, the heating system must [...] modified.
   h. Calculate radiant temperature asymmetry (NRB, 1[...] Acceptable ranges are <18 °F for windows and <[...] for warm ceilings.
9. Determine water flow rate and pressure drop. Refe[...] manufacturer's specifications for the pressure drop characteristics of specific products. Pressure drop i[...]

parallel circuits of uneven lengths tend to balance (Kilkis, 1993-b; Hansen, 1985). Pump sizing should be made accordingly.

he application, design, and installation of hydronic panel ems have certain requirements and techniques such as the owing:

.. As with any hydronic system, look closely at the tubing or piping system design. It should be designed to ensure that water of the proper temperature, and in sufficient quantity, is available to every grid or coil at all times. Reverse-return systems may be considered to minimize balancing problems.

. Individual panels can be connected for parallel flow using headers, or for sinuous or serpentine flow. To avoid flow irregularities within a header-type grid, the water channel or lateral length should be greater than the header length. If the laterals in a header grid are forced to run in a short direction, this problem can be solved by using a combination series parallel arrangement.

Serpentine flow will ensure a more even panel surface temperature throughout the heating or cooling zone.

Noises from entrained air, high-velocity, or high-pressure drop devices, or from pump and pipe vibrations must be avoided. Water velocities should be high enough to secure a turbulent flow and to prevent separated air from accumulating and causing air binding. Recommended water velocity range is between 3 ft/sec. and 7 ft/sec.

Where possible, avoid automatic air-venting devices over ceilings of occupied spaces.

The expansion tank must be adequately sized. An undersized expansion tank may lead to leakages at manifolds and enhance oxygen-induced corrosion of metal surfaces (Metzner, 1986).

Design piping systems to accept thermal expansion adequately. Do not allow forces from piping expansion to be transmitted to panels. Thermal expansion of the ceiling panels must be considered.

In circulating water systems, rubber, plastic tube, steel, and copper pipe are widely used in ceiling, wall, or floor panel construction. Where coils are embedded in concrete or plaster, no threaded joints should be used for either pipe coils or mains. Steel pipe should be the all-welded type. Copper tubing should be soft-drawn coils. Fittings and connections should be minimized. Changes in direction should be made by bending. Solder-joint fittings for copper tube should be used with a medium emperature solder of 95% tin, 5% antimony, or capillary orazing alloys. All piping should be subjected to a hydrostatic test of at least three times the working pressure. Maintain adequate pressure in embedded piping while oouring concrete.

f throttling valve control is used, either the end of the nain should have a fixed bypass, or the last one or two ooms on the mains should have a bypass valve to maintain water flow in the main. Thus, when a throttling valve nodulates, there will be a rapid response. In hospital pplications, valves should be located in the corridor, outside patient rooms.

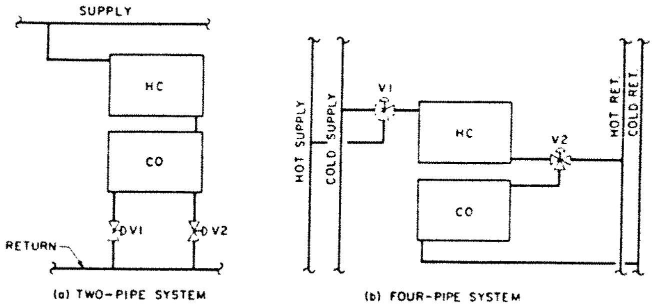


Fig. 11 Split Panel Piping Arrangement for Two-Pipe and Four-Pipe Systems

8. When the panel chilled water system is started, the circulating water temperature should be maintained at room temperature until the air system is completely balanced, the dehumidification equipment is operating properly, and building humidity is at design value.
9. When the panel area for cooling is greater than the area required for heating, a two-panel arrangement (Figure 11) can be used. Panel HC (heating and cooling) is supplied with hot or chilled water year-round. When chilled water is used, the controls function activates panel CO (cooling only), and both panels are used for cooling.
10. To prevent condensation on the room side of cooling panels, the panel water supply temperature should be maintained at least 1°F above the room design dew-point temperature. This minimum difference is recommended to allow for the normal drift of temperature controls for the water and air systems, and also to provide a factor of safety for temporary increase in space humidity.
11. Selection of summer design room dew point below 50°F is generally not economical.
12. The most frequently applied method of dehumidification utilizes cooling coils. If the main cooling coil is six rows or more, the dew point of the air leaving will approach the temperature of the water leaving. The cooling water leaving the dehumidifier can then be used for the panel water circuit.
13. Several chemical dehumidification methods are available to control latent and sensible loads separately. In one application, cooling tower water is used to remove heat from the chemical drying process, and additional sensible cooling is necessary to cool the dehumidified air to the required system supply air temperature.
14. When chemical dehumidification is used, hygroscopic chemical-type dew-point controllers are required at the central apparatus and at various zones to monitor dehumidification.

15. When cooled ceiling panels are used with a variable air volume (VAV) system, the air supply rate should be near maximum volume to assure adequate dehumidification before the cooling ceiling panels are activated.
16. Design operable windows to discourage unauthorized opening.
17. Pump and tube sizing must ensure a turbulent flow in the coils.

Hydronic Ceiling Panels

Metal ceiling panels. Metal ceiling panels can be integrated into a system that heats and cools. In such a system, a source of dehumidified ventilation air is required in summer, so the system is classed as an air-water (hybrid) system (Wilkins and Kosonen, 1992). Also, various amounts of forced air are supplied year-round. When metal panels are applied for heating only, a ventilation system may be required, depending on local codes.

The ceiling panel systems are an outgrowth of the perforated metal, suspended acoustical ceilings. These radiant ceiling systems are usually designed into buildings where the suspended acoustical ceiling can be combined with panel heating and building module, which provides extensive flexibility for zoning and control; or the panels can be arranged as large continuous areas for maximum economy. Some ceiling installations require active panels to cover only a portion of the room, and compatible, matching acoustical panels for the remaining ceiling area.

Three types of metal ceiling systems are available. The first consists of light aluminum panels, usually 12 in. by 24 in., attached in the field to 0.5-in. galvanized pipe coils. Figure 12 illustrates a metal ceiling panel system that uses 0.5-in. pipe laterals, on either 6-, 12-, or 24-in. centers, hydraulically connected in a sinuous- or parallel-flow welded system. Aluminum ceiling panels are clipped to these pipe laterals and act as a heating panel when warm water is flowing, or as a cooling panel when chilled water is flowing.

The second type of panel consists of a copper coil secured to the aluminum face sheet to form a modular panel. Modular panels are available in sizes up to about 36 in. by 60 in., and are

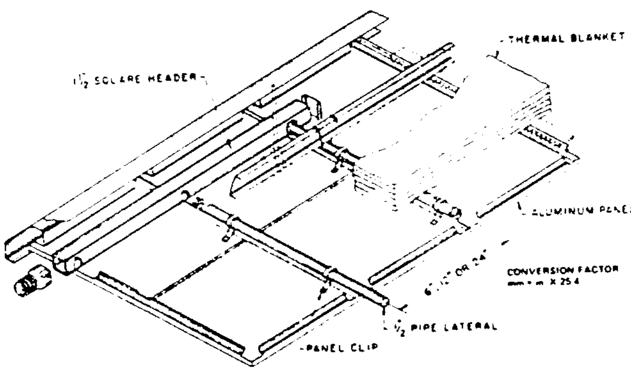


Fig. 12 Metal Ceiling Panels Attached to Pipe Laterals

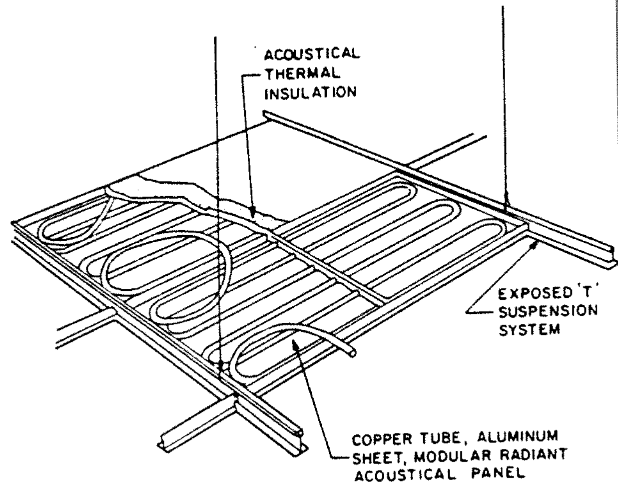


Fig. 13 Metal Ceiling Panels Bonded to Copper Tubin

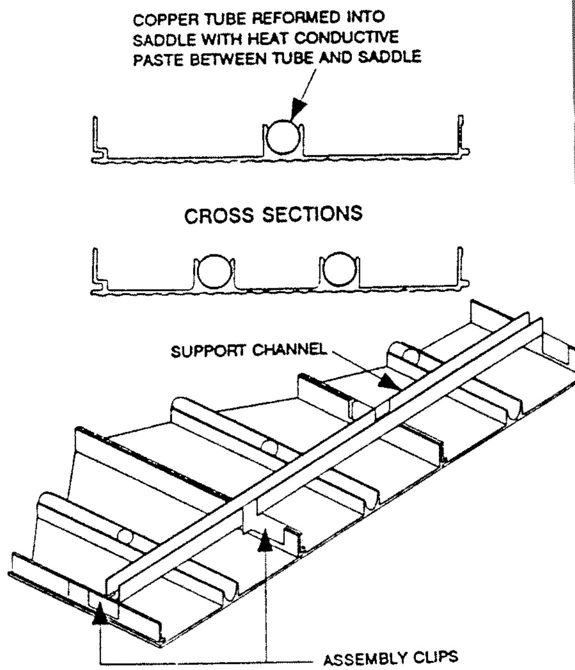


Fig. 14 Extruded Aluminum Panels with Integral Co[p]
Tube

held in position by various types of ceiling suspension systems—most typically, a standard suspended T-bar 24 in. in. exposed grid system. Figure 13 illustrates a metal panel copper tubing bonded to an aluminum panel. Metal ceiling panels can be perforated so that the ceiling becomes sound absorbent when acoustical material is installed on the back panels. The acoustical blanket is also required for thermal r sons, so that the reverse loss or upward flow of heat from th metal ceiling panels is minimized.

The third type of panel is an aluminum extrusion face sh with a copper tube mechanically fastened into a channel h on the back of the face sheet. Extruded panels can be manu tured in almost any shape and size. Extruded aluminum pa often used as long, narrow panels at the outside wall, and independent of the ceiling system. Panels 15 or 20 in. wide ıally satisfy the heating requirements of a typical office build- . Lengths up to 20 ft. are available. Figure 14 illustrates metal ıels using a copper tube pressed into an aluminum rusion, although other methods of securing the copper tube 'e proven equally effective.

'erformance data for extruded aluminum panels vary with rusion surface configuration, copper tube/aluminum con- , and test procedures used. Hydronic ceiling panels have a thermal resistance and respond quickly to changes in space ıditions. Table 1 gives the thermal resistance values of ous ceiling panels.

Ietal radiant ceiling panels can be used with any of the all-air ling systems described in Chapter 2. Double glazing and vy insulation in outside walls have reduced transmission t losses. As a result, infiltration and reheat have become a ıter concern.

litional design considerations are as follows:

Locate ceiling panels adjacent to the outside wall and as close as possible to the areas of maximum load. The panel area within 3 ft. of the outside wall should have a heating capacity equal to or greater than 50% of the wall transmission load.

Ceiling system designs based on passing return air through perforated modular panels into the plenum space above the ceiling are not recommended, because much of the panel heat will be lost to the return air.

When selecting heating design temperatures for a ceiling panel surface temperature, the design parameters are:

a. Excessively high temperatures over the occupied zone will cause the occupant to experience a "hot head effect".
b. With normal ceiling heights of 8 to 9 ft., panels less than 3 ft. wide at the outside wall can be designed for 235°F surface temperature. If panels extend beyond 3 ft.. into the room, the panel surface temperature should be limited to the values as given in Figure 15. The surface temperature of concrete or plaster panels is limited by construction.

Allow sufficient space above the ceiling for installation and connection of the piping that forms the radiant panel ceiling.

Any allowance of unheated ceiling sections, "cold strips", will enhance the heat output by convection. [See Equation (9B) versus (9A).]

tal radiant acoustic panels provide heating, cooling, sound ption, insulation, and unrestricted access to the plenum :. They are easily maintained, can be repainted to look and have a life expectancy in excess of 30 years. The n is a basic air-and-water system—easy to control, and nds quickly. First costs are competitive with other sys- and a life cycle cost analysis often shows that the long life : equipment makes it the lowest cost in the long run. The n has been used in hospitals, schools, office buildings, ts, and exposition facilities.

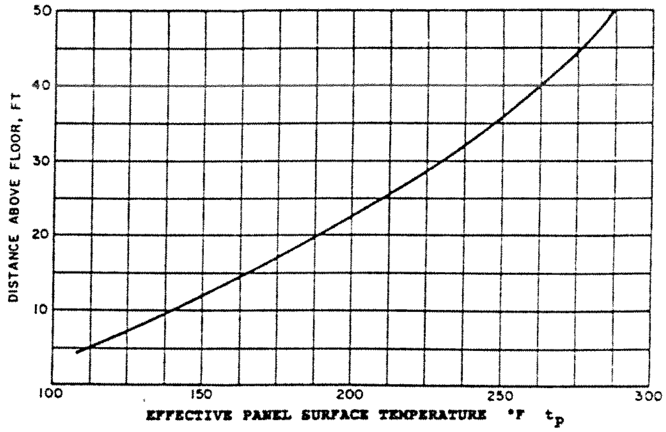


Fig. 15 Suggested Design Ceiling Surface Temperatures at Various Ceiling Heights

Metal radiant panels can also be integrated into the ceiling design to provide a narrow band of radiant heating around the perimeter of the building. The radiant system offers advantages over baseboard or overhead air in appearance, comfort, operating efficiency, cost, maintenance, and product life.

Piping or Tubing Embedded in Ceilings. One of the following types of construction is generally used:

1. Pipe or tube is embedded in the lower portion of a concrete slab, generally within 1 in. of its lower surface. If plaster is to be applied to the concrete, the piping may be placed directly on the wood forms. If the slab is to be used without plaster finish, the piping should be installed not less than 0.75 in. above the undersurface of the slab. Figure 16 shows this method of construction. The minimum coverage must comply with local building code requirements.
2. Pipe or tube is embedded in a metal lath and plaster ceiling. If the lath is suspended to form a hung ceiling, the lath and heating coils are securely wired to the supporting members so that the lath is below, but in good contact with, the coils. Plaster is then applied to the metal lath, carefully embedding the coil as shown in Figure 17.
3. Smaller diameter copper or plastic tube is attached to the underside of wire lath or gypsum lath. Plaster is then applied to the lath to embed the tube, as shown in Figure 18. See also Table 1 for plaster ceiling panels.
4. Other forms of ceiling construction are composition board, wood paneling, etc., with warm water piping, tube, or channels built into the panel sections.

Coils are usually the sinuous type, although some header grid-type coils have been used in ceilings. Coils may be plastic, rubber tube, ferrous, or nonferrous pipe spaced from 4.5 to 9 in. on centers—depending on the required output, pipe or tube size, and other factors.

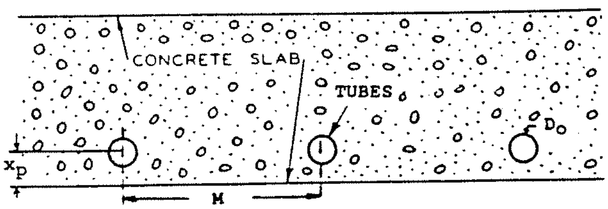


Fig. 16 Tubes in Structural Concrete Slab

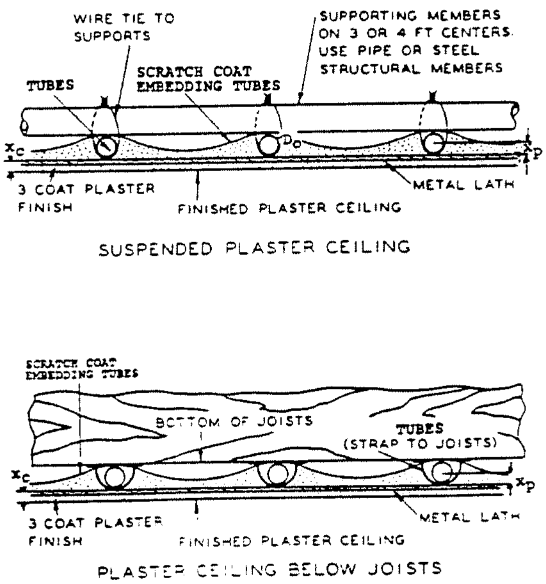


Fig. 17 Tubes in Plaster Above Lath

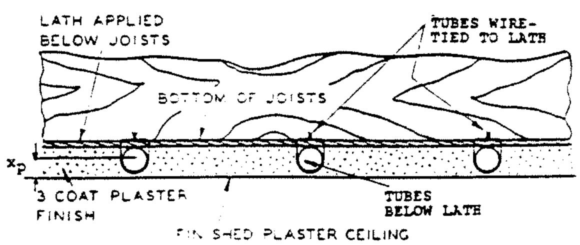


Fig. 18 Tubes in Plaster Below Lath

Where plastering is applied to coils, a standard three-coat gypsum plastering specification is followed—with a minimum of 0.38 in. of cover below the tubes or pipes when they are installed below the lath. Generally, the surface temperature of plaster panels should not exceed 120°F. This can be accomplished by limiting the water temperature in the pipes or tubes in contact with the plaster to a maximum temperature of Insulation should be placed above the coils to reduce *re loss*, the difference between heat supplied to the coil and useful output to the heated room.

To protect the plaster installation and to ensure proper drying, heat must not be applied to the panels for two w after all plastering work has been completed. When the s is started for the first time, the water supplied to the pane should not be higher than 20°F above the prevailing roo temperature at that time, and not in excess of 90°F. Wate should be circulated at this temperature for about two da then increased at a rate of about 5°F per day to 140°F.

During the air drying and preliminary warm-up period should be adequate ventilation to carry moisture from th panels. Paint or paper should not be applied to the pane before these periods have been completed, or while the are being operated. After paint and paper have been app an additional, shorter warm-up period (similar to first-tim starting) is also recommended.

Hydronic Floor Panels

Interest has increased in radiant floor heating with the duction of non-metallic tubing and new design, applicati and control techniques. Whichever method is used for o floor output and comfort, it is important that the heat be distributed throughout the floor. Spacing is generally 4- t on center for the coils. Wide spacing under tile or bare fl can cause uneven surface temperatures.

Embedded Piping or Tubing in Concrete Slab. Pl rubber tube, ferrous, and nonferrous pipe are used in flo slabs, too. The coils are constructed as sinuous-continuo coils, or arranged as header coils with the pipes spaced f 6 to 18 in. on centers. The coils are generally installed wi to 4 in. of cover above them. Insulation is recommended reduce the perimeter and back losses. Figure 19 shows th application of pipe coils in slabs resting on grade. Coils s be embedded completely, and should not rest on an inte Any supports used for positioning the heating coils shoul non-absorbent and inorganic. Reinforcing steel, angle iro pieces of pipe or stone, or concrete mounds can be used wood, brick, concrete block, or similar materials should s coils. A waterproofing layer is desirable to protect insulat and piping.

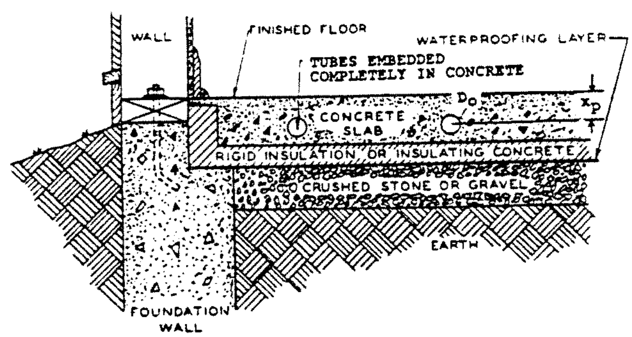


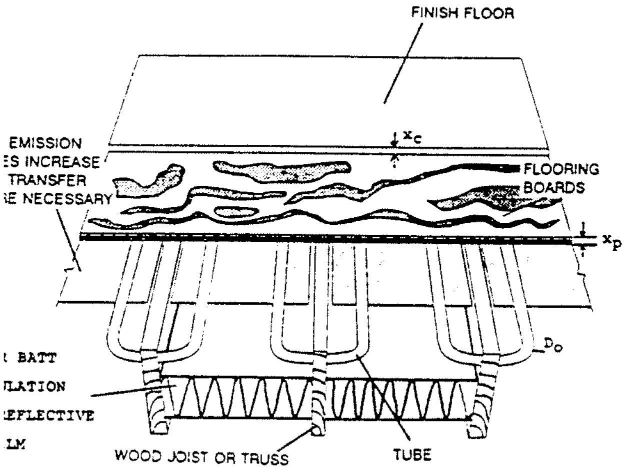
Fig. 19 Tubes in Floor Slab on Grade

Where coils are embedded in structural load-supporting slabs ›ve grade, construction codes may affect their position. ıerwise, the coil piping is installed as described for slabs ting on grade.

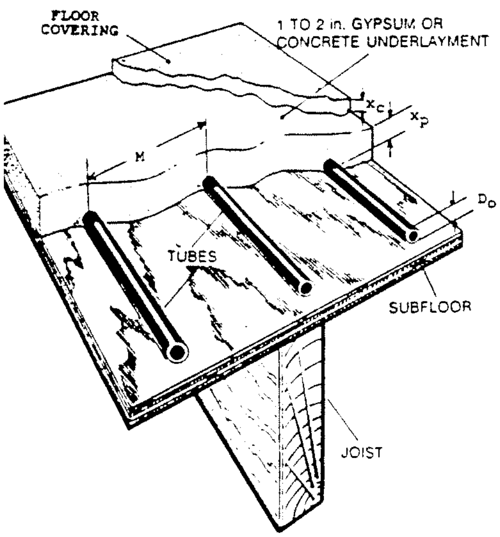
The warm-up and start-up periods for concrete panels are ilar to those outlined for plaster panels.

mbedded systems may fail at some point during their life- e. Adequate valves and properly labeled drawings will help solate the point of failure.

uspended Floor Piping or Tubing. Piping or tubing may ıpplied on or under suspended wood floors using several hods of construction. Coils may be in a layer of concrete or sum on the floor, or mounted in or below the subfloor, or ched directly to the underside of the subfloor. Metal plates att insulation with a reflective layer will improve heat ısion in the subfloor. Both alternatives are illustrated in ıre 20. In design calculations, fin coefficient may be taken lar to the case with metal plates if a batt insulation with a ective layer is used..



20 Tube Under Subfloor



1 Embedded Tube in Thin Slab

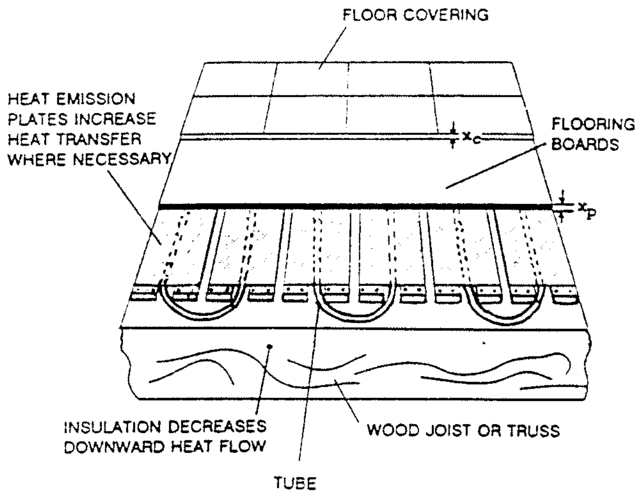


Fig. 22 Tube In Subfloor

Figure 21 illustrates construction with tubing embedded in concrete or gypsum. The thickness of the embedding material is generally 1 to 2 in. when applied to a wood subfloor.

Gypsum products specifically designed for floor heating can generally be installed 1 to 1.5 in. thick because they are more flexible and crack resistant than concrete. When concrete is used, it should be of structural quality to reduce cracking due to movement of the wood frame or shrinkage. Embedding material must provide a hard, flat, smooth surface that can accommodate a variety of floor coverings. The extra structure load due to embedding material must be considered in the design and layout of the subfloor.

Figure 22 illustrates that tubing may be installed in the subfloor. The tubing is installed on top of the rafters between the subflooring members. Heat diffusion and surface temperature uniformity can be improved by the addition of metal heat transfer plates, which spread the heat beneath the finished flooring.

Hydronic Wall Panels

Although piping embedded in walls is not as widely used as floor and ceiling panels, it can be constructed by any of the methods outlined for ceilings or floors. Its design is similar to other hydronic panels (see Equations 17 to 20). Heat transfer at the surface of wall panels is given by Equations (5) and (11).

AIR-HEATED OR COOLED PANELS

Several methods have been devised to warm or cool interior room surfaces by circulating air through passages in the floor, walls, or even the ceiling. In some heating cases, the heated air is recirculated in a closed system. In others, part or all of the air is passed through the room, on its way back to the furnace, to provide supplementary heating and ventilation. Figure 23 illustrates one common type of construction. Compliance with applicable building codes is important.

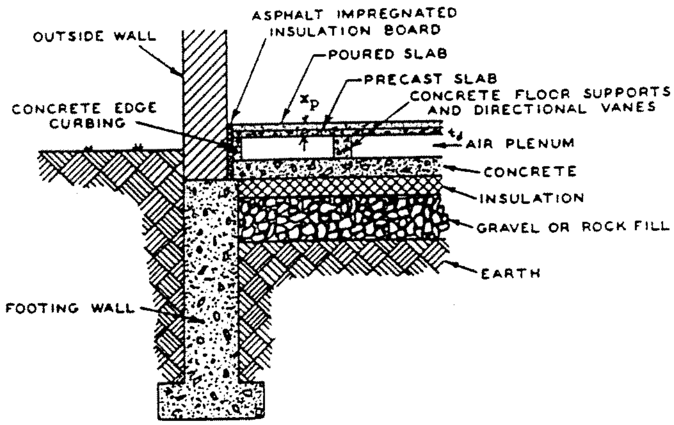


Fig. 23 Air Heated Floor Panel Construction

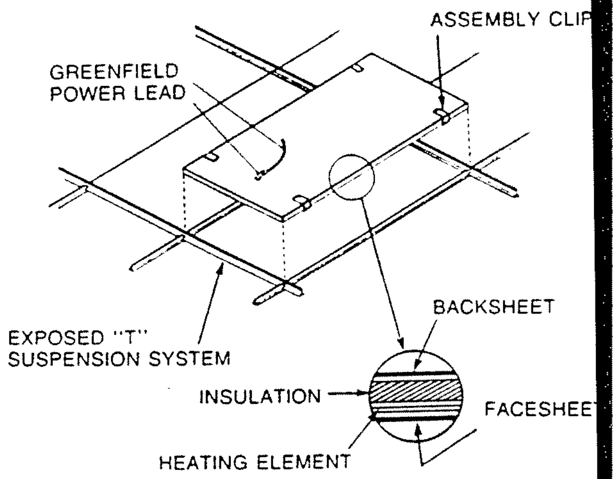


Fig. 24 Electric Heating Panel

In principle, all the heat transfer equations for the panel surface and the design algorithm, explained in the section "Design of Radiant Panels", are applicable. In these systems, however, the fluid (air) moving in the duct has a virtually continuous contact with the panel. Therefore, η is about 1, D_o is 0, and M is unity. Equation (17) gives the required skin temperature of the plenum, t_d. The design of air side can be carried out by following the principles given in Chapters 23 and 32 of the 1993 ASHRAE *Handbook—Fundamentals*.

ELECTRICALLY HEATED SYSTEMS

All design calculations are similar to the hydronic heating systems, except the cable layout. Several different forms of electric resistance units are available for heating interior room surfaces. These include: (1) electric heating cables that may be embedded in concrete or plaster or laminated in drywall ceiling construction; (2) prefabricated electric heating panels to be attached to room surfaces or floor joist space; and (3) electrically heated fabrics or other materials for application to, or incorporation into, finished room surfaces.

Electrically Heated Ceilings

Prefabricated electric ceiling panels. A variety of prefabricated electric heating panels are available for either supplemental or full-room heating. These panels are available in sizes from 2 ft. by 4 ft. to 6 ft. by 12 ft. They are constructed from a variety of materials such as gypsum board, glass, steel fiberglass, or vinyl. Different panels have rated inputs varying from 10 to 95 W/ft^2 for 120, 208, 240, 277, and 347V service. Maximum operating temperatures vary from about 100 to about 300°F, depending on watt density. National and local codes should be followed when placing partitions, lights, and air grilles adjacent to or near electric panels.

Panel heating elements may be embedded conductors, laminated conductive coatings, or printed circuits. Non-heating (cold) leads are connected and furnished as part of the panel.

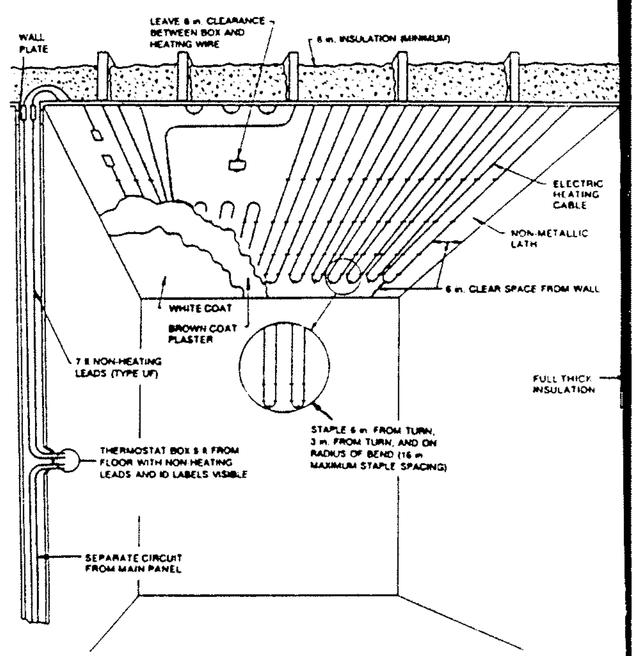


Fig. 25 Electric Heating Panel for Wet Plaster Ceili

Some panels can be cut to fit available space; others m installed as received. Panels may be either flush or surfac mounted and, in some cases, are finished as part of the c Rigid panels that are about 1 in. thick, and weigh from 6 l about 25 lb for a 2ft. by 4 ft. panel, are usually made for l ceilings (see Figure 24). Panels may also be (1) surface-mounted on gypsum board and wood ceilings, or (2) rec between ceiling joists. Panels range in size from 4 ft. wid ft. long. The maximum output is 95 W/ft^2.

Electrical cables embedded in ceilings. Electric he cables for embedded or laminated ceiling panels are fact assembled units furnished in standard lengths of 75 to 18 These cable lengths cannot be altered in the field. The ca

;emblies are normally rated at 2.75 W per linear foot and are ɔplied in capacities from 200 to 5000 W in roughly 200-W rements. Standard cable assemblies are available for 120, 3, 240, and 347V. Each cable unit is supplied with 7-ft. nonıting leads for connection at the thermostat or junction box. :lectric cables for panel heating have electrically-insulated ʼerings resistant to medium temperature, water absorption, ng effects, and chemical action with plaster, cement, or ing lath material. This insulation is normally a polyvinylchlo- : (PVC) covering which may have a nylon jacket. The out- : diameter of the insulation covering is usually about 0.12 in. or plastered ceiling panels, the heating cable may be stapled ;ypsum board, plaster lath, or similar fire-resistant materials ı rust-resistant staples (Figure 24). With metal lath or other ducting surfaces, a coat of plaster (brown or scratch coat) is lied to completely cover the metal lath or conducting ace before the cable is attached. After fastening on the lath applying the first plaster coat, each cable is tested for tinuity of circuit and for insulation resistance of at least 000 ohms measured to ground.

ıe entire ceiling surface is finished with a covering of mally non-insulating sand plaster about 0.50 to 0.75 in. <, or other approved non-insulating material applied ırding to manufacturer's specifications. The plaster is ied parallel to the heating cable rather than across the runs. le new plaster is drying, the system should not be ener- d, and the range and rate of temperature change should be low by other heat sources or by ventilation until the plaster ɔroughly cured. Vermiculite or other insulating plaster es cables to overheat and is contrary to code provisions.

r laminated drywall ceiling panels, the heating cable is :d between two layers of gypsum board, plasterboard, or thermally non-insulating fire-resistant ceiling lath. The : is stapled directly to the first (or upper) lath, and the two s are held apart by the thickness of the heating cable. It is ıtial that the space between the two layers of lath be *letely* filled with a non-insulating plaster or similar materi- iis fill holds the cable firmly in place and improves heat 'er between the cable and the finished ceiling. Failure to : space completely between the two layers of plasterboard illow the cable to overheat in the resulting voids and cable failure. The plaster fill should be applied according nufacturer's specifications.

:tric heating cables are ordinarily installed with a 6-in. ıeating border around the periphery of the ceiling.

n. clearance must be provided between heating cables ıe edges of the outlet or junction boxes used for surface- ted lighting fixtures. A 2-in. clearance must be provided 'ecessed lighting fixtures, trim, and ventilating or other ngs in the ceiling.

ting cables or panels must be installed only in ceiling :hat are not covered by partitions, cabinets, or other ctions. However, it is permissible for a single run of d embedded cable to pass over a partition.

National Electric Code requires that all general power ;ht wiring be run above the thermal insulation, or at least ɔove the heated ceiling surface, or that the wiring be d.

In drywall ceiling construction, the heating cable is always installed with the cable runs parallel to the joist. A 2.5-in. clearance between adjacent cable runs must be left-centered under each joist for nailing. Cable runs that cross over the joist must be kept to a minimum. Where possible, these crossings should be in a straight line at one end of the room.

For cable having a watt density of 2.75 W/ft, the minimum permissible spacing is 1.5 in. between adjacent runs. Some manufacturers recommend a minimum spacing of 2 in. for drywall construction.

The spacing between adjacent heating cables can be determined by using Equation (21):

$$M = A_n/C \tag{21}$$

where

M = cable spacing, ft.
A_n = net panel area, ft^2
C = length of cable, ft

A_n in Equation (21) is the net panel area available after deducting the area covered by the non-heating border, lighting fixtures in the ceiling, cabinets, and other obstructions. For simplicity, Equation (21) contains a slight safety factor, and small lighting fixtures are usually ignored in determining net ceiling area.

Resistance of the electric cable must be adjusted according to its temperature at design conditions:

$$R' = R \cdot \frac{(1 + \alpha_e \cdot t_d)}{(1 + \alpha_o \cdot t_d)} \tag{22}$$

Here,

R' = electrical resistance per ft of electric cable at standard temperature, ohm/ft..
α_e = thermal coefficient for the material resistivity, °F^{-1}
α_o = thermal expansion coefficient, °F^{-1}
t_d = skin temperature of the electric cable at operating conditions, °F. [See Equation (17).]

The 2.5-in. clearance required under each joist for nailing in drywall applications occupies one-fourth of the ceiling area, if the joists are 16 in. on center. Therefore, for drywall construction, the net area A_n must be multiplied by 0.75. Many installations have a spacing of 1.5 in. for the first 2 ft. from the cold wall. Remaining cable is then spread over the balance of the ceiling.

Electrically Heated Wall Panels

Cable embedded in walls similar to ceiling construction is occasionally found in Europe. Because of possible damage from nails driven for hanging pictures or from building alterations, most codes in the United States prohibit such panels. Some of the prefabricated panels described in the preceding section are also used for wall panel heating.

Electrically Heated Floors

Electric heating cable assemblies, such as those used for ceiling panels, are sometimes used for concrete floor heating systems. Since the possibility of cable damage during installation is greater for concrete floor slabs than for ceiling panels, these assemblies must be carefully installed. After the cable has been placed, all unnecessary traffic should be eliminated until the concrete covering has been placed and hardened.

Preformed mats are sometimes used for electric floor slab heating systems. These mats usually consist of PVC-insulated heating cable woven in, or attached to, metallic or glass fiber mesh. Such mats are available as prefabricated assemblies in many sizes from 2 to 100 ft^2, and with various watt densities ranging from 15 to 25 W/ft^2. When used with a thermally treated cavity beneath the floor, a heat storage system is provided, which may be controlled for off-peak heating.

Mineral-insulated (MI) heating cable is another effective method of slab heating. MI cable is a small-diameter, highly durable, flexible heating cable composed of solid electric-resistance heating wire or wires surrounded by tightly compressed magnesium oxide electrical insulation and enclosed by a metal sheath. MI cable is available in stock assemblies in a variety of standard voltages, watt densities, and lengths. A cable assembly consists of the specified length of heating cable, waterproof hot-cold junctions, 7-ft. cold sections, UL-approved end fittings, and connection leads. Several standard MI cable constructions are available, such as single conductor, twin conductor, and double cable. Custom-designed MI heating cable assemblies can be ordered for specific installations.

Other outer-covering materials that are sometimes specified for electric floor heating cable include (1) silicone rubber, (2) lead, and (3) tetrafluoroethylene (Teflon).

For a given floor heating cable assembly, the required cable spacing is determined from Equation (21). In general, cable watt density and spacing should be such that floor panel watt density is not greater than 15 W/ft^2. Higher watt densities (up to 25 W/ft^2) are often specified for the 2-ft border next to cold walls. Check with the latest issue of the *National Electric Code*, and other applicable codes, to obtain information on maximum panel watt density and other required criteria and parameters.

Floor heating cable installation. When PVC-jacketed electric heating cable is used for floor heating, the concrete slab is laid in two pourings. The first pour should be at least 3 in. thick and, where practical, should be insulating concrete to reduce downward heat loss. For a proper bond between the layers, the finish slab should be placed within 24 hours of the first pour, with a bonding grout applied. The finish layer should be at least 1.5 in. and not more than 2 in. thick. This top layer must not be insulating concrete. At least 1 in. of perimeter insulation should be installed as shown in Figure 26.

Preformed mats can be embedded in the concrete in a continuous pour. The mats are positioned in the area between expansion and/or construction joints and electrically connected to a junction box. The slab is poured to within 1.5 to 2 in. of the finished level. The surface is rough screeded and the mats placed in position. The final cap is applied immediately. Since the first pour has not set, there is no adhesion problem between the first and second pour, and a monolithic slab results. A variety of contours can be developed by using heater wire attached to glass fiber mats. Allow for circumvention of obstructions in the slab.

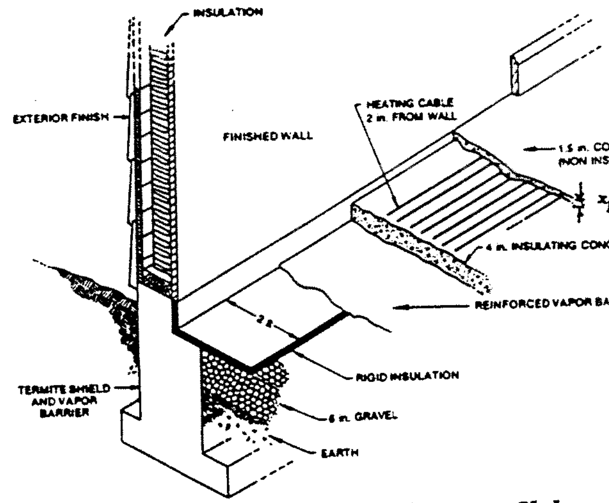


Fig. 26 Electric Heating Cable in Concrete Slab

The cable is installed on top of the first pour of concret closer than 2 in. from adjoining walls and partitions. Meth fastening the cable to the concrete include:

1. Staple the cable to wood nailing strips fixed in th surface of the rough slab. The predetermined cab spacing is maintained by daubs of cement, plaste paris, or tape.
2. In light or uncured concrete, staple the cable dire the slab using hand-operated or powered staplin machines.
3. Nail special anchor devices to the first slab to hol cable in position while the top layer is being pou

M1 electric heating cable can be installed in concrete sl using either one or two pours. For single-pour applicatio cable is fastened to the top of the reinforcing steel before pour is started. For two-layer applications, the cable is lai top of the bottom structural slab and embedded in the fin layer. Proper spacing between adjacent cable runs is mai tained by using prepunched copper spacer strips nailed t lower slab.

CONTROLS

Automatic controls for panel systems may differ from th for other space conditioning systems because of the therm inertial characteristics of the panel and the increase in the radiant temperature within the space under increasing loa However, lightweight systems using thin metal panels or underlay with low thermal heat capacity may be successf controlled with conventional control technology using in sensors. Many of the control principles for hydronic syste described in Chapter 12 and Chapter 14 also apply to pan systems. Because radiant panels do not depend on air-sid equipment to distribute energy, many control methods ha been used successfully; however, a control interface betw heating and cooling should be installed to prevent simult ous heating and cooling.

High mass panels, such as concrete radiant slabs, requi control approach different from low mass panels. Becaus thermal inertia, significant time is required to bring such ıssive panels from one operating point to another, say from ›cation setback to standard operating conditions. This will ›ult in long periods of discomfort from low temperature, then ssibly periods of uncomfortable and wasteful overshoot. reful economic analysis may reveal that a nighttime setback ıtegy is not warranted.

)nce a slab is at operating conditions, the control strategy ›uld endeavor to supply the slab with heat at the rate that ıt is being lost from the space (MacCluer *et al.,* 1989). For ›ronic slabs with constant circulator flow rate, this equates to dulating the difference between the outgoing water and the ırning water temperatures, accomplished via mixing valves, l modulation, or, for constant thermal power sources, via se-width modulation ("bang-bang" or "on-off" control). Slabs h embedded electric resistance cable can be controlled by se-width modulators, such as the common round thermostat ı anticipator or its solid state equivalent.

related approach, outdoor reset control, has enjoyed wide ›ptance. An outdoor reset control measures the outdoor air perature, calculates the supply water temperature required steady operation, and operates a mixing valve or boiler to ieve that supply water temperature. If the heating load of controlled space is primarily a function of the outdoor air perature, or indoor temperature measurement of the con- ed space is impractical, then outdoor reset control alone is cceptable control strategy. When other factors such as solar ıternal gains are also significant, then indoor temperature back should be added to the outdoor reset.

all radiant panel applications, precautions must be taken to ent excessive temperatures. A manual boiler bypass or r means of reducing the water temperature may be ssary to prevent new panels, and covering materials like d, from drying out too rapidly.

›ling Controls

›ntrolling the panel water circuit temperature by mixing, exchange, or using the water leaving the dehumidifier is al. Other considerations are listed in the General Design litions section. It is imperative to dry out the building space ›e starting the panel water system—particularly after ıded down periods, such as weekends. Such delayed ıg action can be controlled manually or by a device.

ıel cooling systems require the following basic areas of erature control: (1) exterior zones, (2) areas under sed roofs to compensate for transmission and solar loads, 3) control of each typical interior zone to compensate for ıal loads. For optimum results, each exterior corner zone imilarly loaded face zone should be treated as a separate ›ne. Panel cooling systems may also be zoned to control ›rature in individual exterior offices, particularly in ›ations where there is a high lighting load, or for corner s with large glass areas on both walls.

: temperature control of the interior air and panel water y should not be functions of the outdoor weather. The ıl thermostat drift is usually adequate compensation for ghtly lower temperatures desirable during winter er. This drift should be limited to result in a room ›rature change of not more than 1.5°F. Control of the interior zones is best accomplished by devices that reflect the actual presence of the internal load elements. Frequently, time clocks and current-sensing devices are used on lighting feeders.

Because air quantities are generally small, constant volume supply air systems should be used. With the apparatus arranged to supply air at an appropriate dew point at all times, comfortable indoor conditions can be maintained throughout the year with a panel cooling system. As with all systems, to prevent condensation on window surfaces, the supply air dew point should be reduced during extremely cold weather according to the type of glazing installed.

Electric Heating Slab Controls

For comfort heating applications, the effective surface temperature of the floor slab is held to a maximum of 84°F in occupied surfaces. Therefore, when the slab is the primary heating system, thermostatic controls sensing air temperature should not be used to control temperature; instead, the heating system should be wired in series with a slab-sensing thermostat. The remote sensing thermostat in the slab acts as a limit switch to control maximum surface temperatures allowed on the slab. The ambient sensing thermostat controls the comfort level. For supplementary slab heating, as in kindergarten floors, a remote sensing thermostat in the slab is commonly used to tune in the desired comfort level. Indoor-outdoor thermostats are used to vary the floor temperature inversely with the outdoor temperature. If the heat loss of the building is calculated for 70 to 0°F and the floor temperature range is held from 70 to 84°F with a remote sensing thermostat, the ratio of outdoor temperature to slab temperature is 70:15, or approximately 5:1. This means that a 5°F drop in outdoor temperature requires a 1°F increase in the slab temperature. An ambient sensing thermostat is used to vary the ratio between outdoor and slab temperatures.

A time clock is used to control each heating zone if off-peak slab heating is desirable.

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Conclusions

apter 6 of ASHRAE Handbook on panel heating and cooling was viewed. In this process suggestions and comments made by others re also considered. With the help of an international project,an alytical algorithm and set of nomographs were developed and tested. ese were adapted for the ASHRAE Handbook and a new text was epared. Case studies and comparisons revealed that the new text cludes more information with substantially increased design pability,and coverage,although the text was virtually kept in its iginal length.

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LIST OF APPENDICES

Appendix 1. Calculation of Panel Resistance, r_u

Appendix.1 Calculation of the Panel Resistance, r_u

There are four elements of panel thermal resistance, r_u:

$$r_u = r_s + r_t + r_p + r_c \quad \textbf{[B-1]}$$

Here;

r_s = thermal contact resistance between the tubing (electric cable) and the panel,

r_t = thermal resistance of the pipe in a hydronic system, (not applicable to electric resistance cable)

r_p = thermal resistance of the panel body,

r_c = thermal resistance of the panel coverings.

In an accurate design it is also important to know the split of r_u, because particularly in floor heating it may significantly affect the design although r_u remains the same (see equations in Appendix 2).

In an in-slab application or other panel types where a good and secure attachment or bonding exists, r_s may be neglected.

Thermal resistance of the panel body in the direction of heat transfer between the conditioned space and the circuit may have two forms:

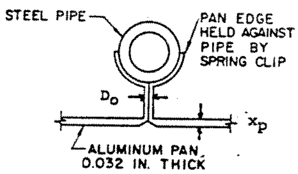
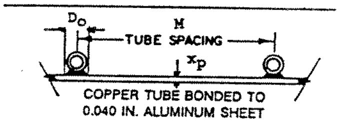
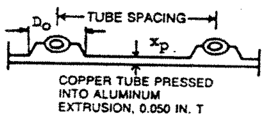
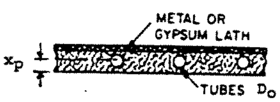
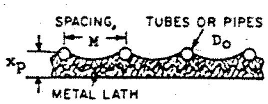
$r_p \approx x_p/k_p$ for panels with attached or bonded circuits (see Figures 3-b and 3-c, in Section 1.2.3) **[B-2]**

or;

$r_p \approx (x_p - D_o/2)/\cdot k_p$ for panels with embedded circuits (see Figure 3-a, in Section 1.2.3) **[B-3]**

Table B-1 gives information about typical r_s and r_p values for metal ceiling panels.

able B-1. Thermal Resistance for Metal Ceiling Panels

Type of Panel	Thermal Resistance ($h \cdot ft^2 \cdot {}^\circ F/Btu$) r_s	r_p
	0.20	x_p/k_p
	0.05	x_p/k_p
	0.05	x_p/k_p
	≈ 0	$(x_p - D_o/2)/k_p$
	≈ 0	$(x_p - D_o/2)/k_p$

r thermal resistances can be calculated by the following essions:

$\approx (0.116 \cdot D_o/D_i - 0.107)/k_t \quad \{1.1 \le D_o/D_i \le 2.0\}$ **[B-4]**

k_t is the thermal conductivity of the tube material.

Table B-2 gives typical values for several tube materials.

Table B-2. Thermal Conductivity of Different Tube Materials

Material	k_t (Btu/h·ft·°F)
Aluminum,hard drawn	135
Steel	48.5
Copper	170
Stainless steel	9.2
Low density polyethylene (LDPE)	0.18
High density polyethylene (HDPE)	0.24
Cross-linked polyethylene (VPE)	0.22
Textile reinforced rubber heat transfer hose (HTRH)	0.17
Polypropylene block co-polymer (PP-C)	0.13
Polypropylene random co-polymer (PP-RC)	0.14
Polybutylene (PB)	0.13

Thermal resistance of any panel covering can be obtained from; manufacturer's catalog,Table 5 in Chapt.6 in ASHRAE Handbook,or from the following equation if the effective thickness,x_c and the thermal conductivity,k_c of the material is known:

$$r_c = x_c/k_c \qquad \textbf{[B-5]}$$

If there are more than one cover,each individual thermal resistance should be added.

It must be noted that above definitions of thermal resistance exclude the effect of spacing,M.

Appendix 2. The New Analytical Algorithm and Examples

Appendix. 2 Description of the Analytical Algorithm and Examples

The analytical algorithm is based upon the composite fin model (Kilkis,1993-a). The original model acknowledges that surface heat transfer components namely radiation and natural convection may observe different temperature gradients, because AUST is not usually equal to t_a. This may be an important factor especially if this difference is too big. However even in these extreme conditions,its impact on the mean water temperature solution may be within a few degrees F,if the tube spacing is not greater than 12". In order to free the designer from some calculations within an acceptable engineering accuracy,the following simplified equation was adapted:

$$t_d \approx t_a + \frac{(t_p - t_a)\cdot \mathbf{M}}{(2\cdot W\cdot \cap + D_o)} + q \cdot (r_p + r_c + r_s) \qquad \textbf{[A-1]}$$

here,

$\mathbf{t_d}$ = the required skin temperature of the tubing or electric cable,°F

$\mathbf{t_a}$ = indoor design temperature,°F

$\mathbf{q}$ = combined surface heat transfer intensity (q_d,q_u,q_s or $-q_u$ or $-q_s$),Btu/h·ft²

M = circuit spacing on centers,ft

$\mathbf{D_o}$ = outer diameter of the tube(cable),ft or characteristic contact width of the tube with the panel (see Table B-1 in Appendix 1)

2·W = net spacing (**M** - D_o),ft

η = fin efficiency, dimensionless

$: \tanh(m \cdot \mathbf{W})/(m \cdot \mathbf{W})$ **[A-2]**

$\approx 1/(m \cdot \mathbf{W}) \quad \{\text{if } m \cdot W > 2\}$

m = fin coefficient, ft^{-1}

$$m : \left[\frac{q}{(t_p - t_a) \cdot 2 \cdot \Sigma_i \, (k_i \cdot x_i)_{i:1 \text{ to } n}} \right]^{1/2} \qquad \{t_p \neq t_a\} \qquad \textbf{[A-3]}$$

and x_i are the thermal conductivity (Btu/h·ft·°F) and
ıracteristic thickness (ft) of each layer (up to n) including the
ıel body itself (see Figure 3 in Section 1.2.2).
: exact solution for the temperature drop across the tube wall in a
lronic system is (Kilkis,1993-d and Krinninger,1989):

$$ta\ t_w = \frac{(q + q_b) \cdot M}{\pi} \cdot \left[\frac{1}{\alpha \cdot D_i} + \frac{1}{2 \cdot k_t} \cdot \ln\,(Do/Di) \right] \qquad \textbf{[A-4]}$$

s the convection heat transfer coefficient on the water(brine)
.e. In a turbulent regime it is around 400 Btu/h·ft²·°F. Therefore
typical $1/(\alpha \cdot D_i)$ term is in the order of 0.05. When compared to
second term in the parenthesis (typical values in the range
o 1.5) it may be neglected. Back and edge loss (gain) intensity
is in the order of 10% of q with proper insulation. $\ln(D_o/D_i)$ can
linearized within the practical range of standard pipe dimensions.
s is shown in Figure A-1. After these simplification and allowing

a nominal value for the $1/(\alpha \cdot D_i)$ term, the thermal resistance of the tube wall will be approximated by the following equation:

$$r_t \approx (0.116 \cdot D_o/D_i - 0.107)/k_t \quad \{1.1 \le D_o/D_i \le 2.0\} \tag{A-5}$$

Using this thermal resistance for a hydronic system, the required mean temperature will be:

$$t_w = (q + q_b) \cdot \mathbf{M} \cdot r_t + t_b \tag{A-6}$$

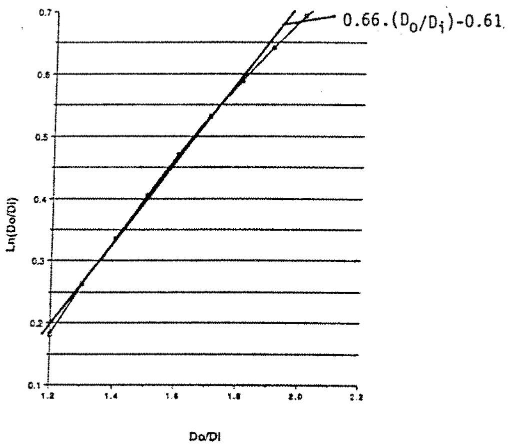


Figure A-1

Figure A-2 shows the variation of fin efficiency with $(m \cdot W)$. Following equation accurately fits this function:

$$\eta = 0.843 \cdot (m \cdot W)^{-0.9085} \tag{A-7}$$

It should also be noted that $1/(m \cdot W)$ can also approximate this function if $(m \cdot W) \ge 2$.

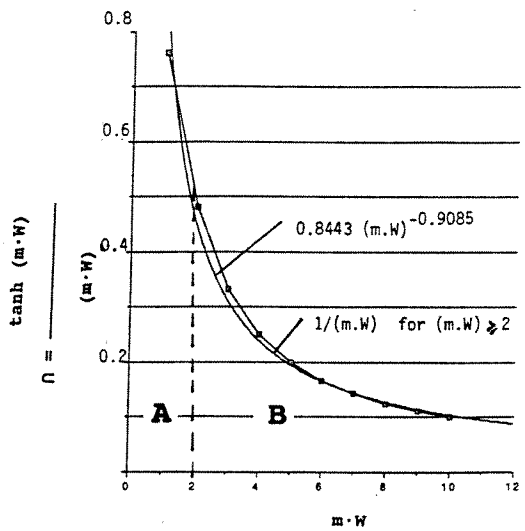


Figure A-2

mple 1. In-slab floor heating

ign inputs:

door design temperature, t_b: 10°F

room type : One outdoor exposed side, fenestration about 5%

t_a : 70°F

q_u : 30 Btu/h·ft² (required)

q_b : 5 Btu/h·ft² (estimated)

floor panel consists of the following;

crete slab :

our, tubes placed at mid-thickness (2" cover)

k_1: k_p: 0.81 Btu/h·ft·°F

x_1: x_p: 2"/12 ft (characteristic panel thickness)

transfer tubing:

material : rubber, k_h: 0.17 (see Table B-2 in Appendix 1)

D_o : 3/4" : 0.0625'

D_i : 1/2" : 0.0416'

D_o/D_i : 1.5

refore, r_t : 0.39 (see Equation A-5)

and, r_p : (2/12-0.0625/2)'/0.81 = 0.167

(see Equation B-3 in Appendix 1)

spacing, M: 9" O.C (9/12')

or covering: Carpet

k_2: k_c : 0.04

x_2: x_c : 3/8" (0.03125')

refore, r_c : 0.78 (see Equation B-5 in Appendix 1)

ecting r_s, r_u will be:

r_u : 0.167 + 0.39 + 0.78 = 1.34 h·ft²·°F/Btu

Solution:

First determine or approximate AUST. From Equations 2 and 3-A in Section 1.1.1, with t_b=10°F and d=1:

$$z : 2.17°F$$

$$AUST : 70-2.17 \approx 68°F$$

Then determine the required effective surface temperature, t_p using Equations 1 and 8 in Sections 1.1.1 and 1.1.2:

$$t_p \approx 86\ °F$$

(Note: in Equation 8, air temperature adjacent to floor panel is equated to AUST)

It should be noted that the new nomograph given in Section 3 readily gives this solution for any q and t_a for a variety of AUST.

This nomograph also reads t_p 86°F.

Step.1. Determine fin characteristics:

$$W : (9/12-0.0625)/2 = 0.34375'$$

$$2\cdot\sum_{i=1}^{2}(k_i\cdot x_i) : 2\cdot(0.81\cdot 2'' + 0.04 \cdot 3/8'')/12 = 0.2725$$

$$m : \left[\frac{30}{(86-70)\cdot 0.2725}\right]^{1/2} = 2.62\ ft^{-1}$$

$$\eta : \tanh(2.62\cdot 0.34375)/(2.62\cdot 0.34375) = 0.795$$

Step.2. From Eq.13 in Sec.1.2.3 determine the tube skin temperature.

$$t_d : 70 + \frac{(86-70)\cdot 9/12'}{(2\cdot 0.34375\cdot 0.795 + 0.0625)} + 30\cdot(0.167+0.78)$$

: 70 + 19.7 + 28.4 = 118°F

p.3 This is a hydronic system,therefore determine the mean water temperature.

m Equation A-6;

$$t_w : 118 + (30 + 5)\cdot 9/12\cdot 0.39 = 128°F$$

new nomograph reads t_w 131°F.

following table shows a list of required mean water temperatures other tube spacings.

le A-1 Variation of t_w with M

ube Spacing O.C. (inches)	t_w (°F) Calculated	New Nomograph
6"	123	126
9"	128	131
12"	134	137
15"	142	142

s table shows that tube spacing has a considerable effect on t_w n for a concrete slab panel.

IRAE Solution:

the same problem would be solved by the existing ASHRAE guidelines

9" on centers:

m Table 2 in Chap.6,for 4" concrete,heat flow ratio about 10 and ferrous tube;thermal resistance of bare concrete floor ≈ 0.42

ng the same thermal resistance for carpet;

r_u : 0.42 + 0.78 = 1.2

Then from Figure 8 in Chap.6 (see Appendix 4);

t_w is read approximately 120°F

Table 2 in Chapter 6 (see Appendix 4) gives only one more option for tube spacing namely,12". Using this spacing,

r_u : 0.51 + 0.78 = 1.29

and again using Figure 8;

t_w : 124°F

Results obtained for the same problem by the approach underlying DIN Standard for floor heating (Krinninger,1989),current ASHRAE guidelines and the new algorithm are compared in Table A-2.

Table A-2 Comparison of Different Methods

Tube Spacing	t_w		
M (inches O.C)	New Algorithm	Current ASHRAE	DIN Standard
6	123	cannot solve	112
9	128	120	125
12	134	124	140
15	142	cannot solve	151

It has already been explained that DIN method underestimates mean water temperature at close spacings like 4" and 6". The opposite is true for wide spacings like 15" and 18" (see Section 1.2.1). In spite of this fact,there is a better resemblance between the results of the new algorithm and DIN when compared to current ASHRAE results.

least they confirm each other that current ASHRAE guidelines seem underestimate t_w at any tube spacing. Furthermore,current ASHRAE delines cover only a limited range of tube spacing.

mple 2. Plaster ceiling cooling slab

ign inputs:

door design temperature, t_b:95°F

room type : One outdoor exposed side,fenestration about 5%

t_a : 80°F

q_u : -20 Btu/h·ft² (required)

q_b : -3 Btu/h·ft² (estimated)

ling panel consists of the following;

ee coat gypsum plaster slab, 2" thick, below gypsum lath:

" Pe-X tubes are tied directly to gypsum lath at 6" on centers.

re is no panel covering.

k_1 : k_p: 0.42 Btu/h·ft·°F (given)

x_1 : x_p: 2"/12 ft (characteristic panel thickness)

: transfer tubing:

material : PE-X, k_h : 0.22 (see Table B-2 in Appendix 1)

D_o : 1/2" : 0.04166'

D_i : 3/8" : 0.03125'

D_o/D_i : 1.333

efore, r_t : 0.216 (see Equation A-5)

and, r_p : (2/12-0.04166/2)'/0.42 = 0.347 (see Eqn.B-3 in App.1)

spacing,M: 6" O.C (6/12')

l covering: none

efore, r_c : 0

Neglecting r_s, r_u will be:

$$r_u : 0.347 + 0.216 = 0.563 \ h\cdot ft^2\cdot{}^\circ F/Btu$$

Solution:

From Equations 2 and 3-B in Section 1.1.1 with t_b=90°F and d=1:

$$z : -1^\circ F$$

$$AUST : 80 + 1 \approx 81^\circ F$$

The exact value for the required effective surface temperature, t_p using Equations 1 and 8 in sections 1.1.1 and 1.1.2:

$$t_p = 68.5^\circ F$$

Note: in Equation 8, air temperature adjacent to ceiling panel is equated to AUST.

The new nomograph reads t_p 68°F for (AUST-t_a):1°F

ASHRAE Solution:

When Equations 5B and 10 in Chap.6 are used with t_a:80°F and AUST:81°F, t_p is 68°F. However Figure 9 in Chap.6, reads t_p:70°F. The reason of this difference is not only the fact that Figure 9 equates t_a to AUST. The cause of this discrepancy between the equations and Figure in IP edition of Chap.6 was already explained in Kilkis's letter (1992-b)to ASHRAE, in a response to D.B.Sensecall's comments (1992). Appendix 5 shows these correspondences.

Shortly, Figure 9 uses Wilkes and Peterson data from Figure 4 instead of Equation 10 in Chap.6. In fact, this discrepancy among the equations and figures in the existing ASHRAE guidelines is a typical cause for confusion and inaccuracies in design.

In Chap.6 and Table 4 for plaster ceiling cooling panels for r_u(M), there is not any equation or chart to solve the rest of the problem. It must be noted that Figure 7 is only for ceiling ting,therefore by using the information available in the existing RAE guidelines the mean chilled water (brine) temperature can not calculated for any type of ceiling cooling panel and tube spacing.

tinuing the design by the new algorithm;

p.1.Determine fin characteristics:

$$W : (6/12-0.04166)/2 = 0.2292'$$

$$\cdot \sum_{i=1}^{1}(k_i \cdot x_i) : 2\cdot(0.42 \cdot 2")/12 = 0.14$$

$$m : \left[\frac{-20}{(68.5-80)\cdot 0.14}\right]^{1/2} = 3.52 \text{ ft}^{-1}$$

$$\eta : \tanh(3.52\cdot 0.2292)/(3.52\cdot 0.2292) = 0.83$$

).2. From Equation 13,Section 1.2.3 the tube skin temperature:

$$t_d : 80 + \frac{(68.5-80)\cdot 6/12'}{(2\cdot 0.2292\cdot 0.83+ 0.04166)} - 20\cdot(0.347)$$

$$: 80 - 13.6 - 6.94 \approx 59°F$$

.3. From Eq.A-6;t_w : $59 + (-20 - 3)\cdot 6/12\cdot 0.216 = 56.5°F$.

new nomograph reads this 56°F.

Example 3. Under sub-floor heating

Design inputs:

Outdoor design temperature, t_b: 10°F

room type : One outdoor exposed side, fenestration about 5%

t_a : 70°F

AUST : 68 °F (see Example 1)

q_u : 30 Btu/h·ft² (required)

q_b : 7 Btu/h·ft² (estimated)

Floor panel consists of the following:

0.5" plywood subfloor with aluminium plates 0.032"thick.

k_1: k_p: 0.085 Btu/h·ft·°F

x_1: x_p: 0.5"/12 ft (characteristic panel thickness)

k_2: 135

x_2: 0.032/12'; r_2:2.10^{-5} (negligible)

Heat transfer tubing:

Hoses are stapled beneath the plywood sub-floor.

material : rubber, k_h : 0.17

D_o : 3/4" : 0.0625'

D_i : 1/2" : 0.0416'

D_o/D_i : 1.5

therefore, r_t : 0.39 (see Equation B-4 in Appendix 1)

and, r_p : (0.5/12)'/0.085 = 0.49 (see Eqn.B-2 in App.1)

tube spacing, M: 6" O.C (6/12')

Floor covering: Carpet

k_2: k_c : 0.04

x_2: x_c : 3/8" (0.03125')

therefore, r_c : 0.78 (see Equation B-5 in Appendix 1) is assumed to be 0.2. Then r_u will be:

$$r_u : 0.49 + 0.39 + 0.78 + 0.2 = 1.86\ h\cdot ft^2\cdot °F/Btu$$

ution:

$$t_p \approx 86\ °F \qquad \text{(see Example 1)}$$

new nomograph also reads 86°F (see Section 2).

p.1.Determine fin characteristics:

$$W :(6/12-0.0625)/2 = 0.21875'$$

$$2\cdot\sum_{i=1}^{3}(k_i\cdot x_i) : 2\cdot(0.085\cdot0.5" + 0.04\cdot3/8" + 135\cdot0.032)/12 = 0.73$$

$$m : \left[\frac{30}{(86-70)\cdot\ 0.73}\right]^{1/2} = 1.6\ ft^{-1}$$

$$\eta : \tanh(1.6\cdot0.21875)/(1.6\cdot0.21875) = 0.96$$

).2. Determine tube skin temperature from Eq.13 in Section 1.2.3;

$$t_d : 70 + \frac{(86-70)\cdot6/12'}{(2\cdot0.21875\cdot0.96+ 0.0625)} + 30\cdot(0.2+0.49+0.78)$$

$$: 70 + 16.5 + 44.1 = 131°F$$

.3 Determine the mean water temperature:

Equation A-6;

$$t_w : 131 + (30 + 7)\cdot6/12\cdot0.39 = 138°F$$

new nomograph reads this temperature 140°F.

AE Solution:

ent ASHRAE guidelines give a comprehensive explanation about

: sub-floor hydronic panel heating system,including construction details. (see Figures 21,22,and 23 in Chap.6). However not any equation,table or chart are available for the completion of the design for such a panel heating system,with the only exception that t_p can be determined using Figure 8. This figure reads t_p 85°F and agrees reasonably with the calculated value of 86°F.

Example 4. Ceiling heating with metal panels

Design inputs:

Outdoor design temperature,t_b:0°F

room type : two outdoor exposed sides

"Cold strips" of 1 foot exists between 6′ ceiling metal panels

t_a : 74°F

q_u : 30 Btu/h·ft² (required)

q_b : 5 Btu/h·ft² (estimated)

Ceiling panel consists of an aluminum pan with steel pipe attached by spring clip to the pan edge. This is shown in Table B-1,first figure. There is no panel covering.

k_1 : k_p:135 Btu/h·ft·°F (from Table B-2 in Appendix 1)

x_1 : x_p:0.032/12 ft (characteristic panel thickness)

Heat transfer tubing:

material : steel, k_h : 48.5 (see Table B-2 in Appendix 1)

D_o : 1" : 0.0833′

D_i : 3/4" : 0.0625′

D_o/D_i : 1.33

therefore, r_t : 10^{-3} (it is negligible)

and, r_p : $(0.032/12)/135 \approx 2\cdot10^{-5}$ (it is negligible)

(see Equation B-2 in App.1)

e spacing,M: 12" O.C (1')

el covering: none

refore, r_c : 0

m Table B-1 r_s is 0.20

refore, r_u : 0.20 h·ft²·°F/Btu

ution:

n Equations 2 and 3-B in Section 1.1.1 with t_b=0°F and d=3:

$$z : 3.85 \approx 4°F$$

$$AUST : 74 - 4 \approx 70°F$$

ng Equations 1 and 10 in sections 1.1.1 and 1.1.2:

$$t_p = 94°F$$

new nomograph also reads t_p 94°F for (AUST-t_a) : -4°F

).1.Determine fin characteristics:

$$W :(12/12 - 2\cdot 0.032/12)/2 \approx 0.5'$$

: that in this equation D_o is replaced by the root thickness of pan edge (2·0.032") which establishes the contact width between pipe and the panel surface.

$$\sum_{i=1}^{1}(k_i \cdot x_i) : 2\cdot(135\cdot\ 0.032")/12 = 0.72$$

$$m : \left[\frac{30}{(94-74)\cdot 0.72}\right]^{1/2} = 1.44\ ft^{-1}$$

$$\eta : \tanh(1.40\cdot 0.5)/(1.40\cdot 0.5) \approx 0.86$$

.2. Determine tube skin temperature:

$$t_d : 74 + \frac{(94-74)\cdot 12/12'}{(2\cdot 0.5\cdot 0.86+ 0.0053)} + 30\cdot(0.20)$$

In this case the skin temperature of the steel pipe will approximate the mean water temperature.

$$t_w \approx t_d : 74 + 23.1 + 6 \approx 103°F$$

New nomograph given in Section 2 reads this temperature 105°F

ASHRAE Solution:

Using Table 1 in ASHRAE Handbook,Chap.6 for 12" spacing $r_u(M)$ is 0.61

From Figure 7, mean water temperature is predicted 119°F

Same figure reads the effective panel surface temperature,t_p 102°F.

The discrepancy between these results and the new algorithm is mainly due to two reasons: First,Figure 7 in Chap.6 is plotted with convection heat output intensity given from Equation 7 in Chap.6. This underestimates the heat output and consequently calls for a higher t_p. Second,the heat diffusion model is a very approximate one which was explained earlier in Section 1. These two effects are combined when tube spacing is greater than about 9". Otherwise second cause decreases the amount of discrepancy.

Appendix 3. Construction of the New Universal Nomograph

Appendix. 3 Construction of the New Universal Nomograph[R]

3.1. General

There are six major design parameters in a typical panel design. These are:

q : Heat transfer intensity at the panel surface

t_a : Air temperature

AUST: Area averaged temperature of uncontrolled surfaces

t_p :Effective panel surface temperature

r_u :Panel resistance

M :Tube spacing

Other variables which effect the design are:

Pipe size and its thermal conductance,slab material and position of pipes in the slab,back and edge insulation,floor coverings,elements of r_u (r_p,r_t,r_s,r_c).

Provided that r_u is properly calculated in terms of its elements,the only problem about representing all these elements on a nomograph by a single r_u is the fact that mean water temperature actually depends on the split between the elements of r_u. This is significant especially when r_c and r_p are dissimilar (see Equation 13). Provided that mean water temperature is adjusted with r_c/r_p, r_u will adequately represent the pipe,slab overpour (panel) thickness,floor coverings etc.

3.2. q versus t_p

t_p is a direct function of q if t_a and AUST are fixed. This primary relationship can be illustrated independent of panel type, construction,and other variables regarding the heat diffusion in the panel. In a typical nomograph t_a is usually an input variable. This necessitates to spare an axis for t_p in terms of various t_a values. Another alternative is to plot individual nomographs,each with a ed t_a value. This alternative however needs to prepare many ographs. The main difficulty here is the radiation heat transfer m which has t_a and AUST with powers of four. Similarly the vection term is not exactly linear. i.e: in floor heating it is portional to $(t_p-t_a)^{1.31}$. However this gives us the clue that if radiation term can be linearized, at least we will obtain a ear (t_p-t_a) term. In fact,the radiation term can be accurately earized in a given range of operation. By standards,floor perature is well defined. In ceiling cooling t_p is limited by t_a $<t_a$) and the dew point temperature corresponding to the maximum oor relative humidity which is suggested to be less than 50% by RAE Chapter 6 for a satisfactory operation. Ceiling panel perature has a broader range.It is only limited by the head fort (see Figure 16 in Chap.6).

ng that AUST is already related to t_a (Kilkis,1993-a):

$$\text{AUST} : t_a \pm f \qquad \textbf{[C-1]}$$

total heat transfer intensity in a generic form will be :

$$q : A\cdot(t_p-t_a\pm f) + B\cdot(t_p-t_a)^c \qquad \textbf{[C-2]}$$

aking the upper and lower practical limits of t_p into ideration,the combined heat transfer equation can be linearized a set of f:

$$q \approx D\cdot(t_p-t_a)^e \qquad \{\ f\text{:fixed}\ \} \qquad \textbf{[C-3]}$$

also eliminates the need to dedicate a separate axis for t_p in s of t_a,because t_p is now directly and linearly related to t_a. enables to set up a new scale with the variable group (t_p-t_a) in ing. In order to be consistent with cooling,this must be in lute terms.

he new nomograph,the right hand vertical axis establishes this approach, where AUST is permitted to vary between ±2 and ±4. Constant (t_p-t_a) lines are linear which makes it easy to interpolate for AUST. This approach also eliminates the t_p line and the t_a axis at the bottom in Figures 7 and 8 in the current Chapter 6 in ASHRAE Handbook. Consequently, a single nomograph can handle any t_a value on the right hand vertical axis, which eliminates the need for a specific t_a axis.

Equation C-3 was linearized at the following pivots in the range of $0 \le q \le 70$ Btu/h·ft²:

In floor heating: t_p:80°F for $(t_p-t_a) \le 25$°F

t_p:85°F for $25<(t_p-t_a) \le 35$°F

(t_p-t_a) varies between 0 and 40°F.

Equations used: Equation 1 + Equation 5.

Ceiling heating and floor heating use the same equations for the combined heat transfer intensity at the panel surface. Therefore, the only difference between floor heating and ceiling cooling would be difference of the pivot point for linearization. However the difference was found to be within the reading accuracy.

In ceiling heating:

t_p:95°F for $(t_p-t_a) \le 25$°F

t_p:100°F for $25<(t_p-t_a) \le 35$°F

Equations used: Equation 1 + Equation 10 (with "cold strips" on the ceiling).

Two sets of lines of constant (t_p-t_a) were drawn in this range with increments of 5°F. Broken lines are for ceiling heating.

Four vertical lines represent f values of +2,0,-2 and -4.

With this set up the lines were constructed with the following method:

-Select (t_p-t_a): 0,5,10,15 etc..

: t_p at pivot

$$t_a : t_p-(t_p-t_a)$$

: AUST;

$$AUST : t_a \pm f \qquad f:+2,0,-2,-4\ °F$$

heat transfer equation to find the corresponding q and plot this the axis for the given f value.

he convection equation the adjacent air film temperature is taken l to AUST (Drake,1993. See also Section 1.1.2 of this report) following table shows the accuracy of this linearization process floor heating for different t_a and t_p values. Figure 8 in Chapter HRAE Handbook was also used to find the corresponding q values. are listed in the last column.

e C-1. Comparison of the linearization process

AUST:t_a-1

a	t_p °F	q Btu/h·ft² Equations 1 + 5	New Nomograph	Fig.8 Chap.6 ASHRAE
0	90	36.0	36.0 (+0%)	41 (+14%)
6	90	24.5	25.0 (+2%)	28 (+14%)
0	80	17.2	17.5 (+1%)	18 (+4%)
6	80	7.3	7.5 (-2%)	7.5(+3%)
0	100	57.9	57.0 (-2%)	74 (+28%)
0	105	69.2	67.0 (-3%)	89 (+30%)

This table also gives the % error compared to the first column representing the equations for surface heat transfer intensity. Results indicate that the linearization process in the new nomograph is within 3% when compared to the calculated values. Figure 8 however exhibit errors up to 30% especially at high heat transfer rates. This indicates two possibilities:

The t_p line on Figure 8 in Chap.6 is wrong,or in contrary to to the text in Chapter 6,AUST was selected different than t_a-1.

In order to investigate the latter possibility the calculations were repeated for AUST: t_a-2. The results are shown in Table 2. This table now also includes the predictions given by by DIN (1990).

Table C-2. Comparison of the linearization process with DIN and Fig.8

AUST:t_a-2

t_a °F	t_p °F	q Btu/h·ft² Eqns 1 + 5	New Nomograph	DIN	Fig.8 Chap.6 ASHRAE
70	90	38.5	38.0 (-1%)	39.6	41 (+6%)
76	90	27.0	27.5 (+2%)	27.1	28 (+4%)
70	80	19.0	19.0 (+0%)	18.5	18 (+5%)
76	80	8.3	8.5 (+2%)	6.7	7.5(-10%)
70	100	60.0	58.0 (-3%)	61.0	74 (+23%)
70	105	71.4	69.0 (-3%)	74.0	89 (+24%)

This table shows that the linearization of the new nomograph is even

er than DIN linearization when compared to the equations.
re 8 in Chap.6 overpredicts the heat transfer intensity at high
alues. However its overall agreement is somehow better when
ared with its previous figures in Table 1. This indicates that
ably Figure 8 was drawn with AUST:ta-2 assumption.

Mean Water Temperature t_w:

relationship between q, r_u, M and t_w is very complicated for a
graph. In floor heating where there is a possibility of having
y floor coverings, the split of r_u also becomes significant. This
licated relationship was solved in two steps:
1: Mean water temperature requirement in ceiling heating is
arily different than floor heating, because $(t_p - t_a)$ relationship
ifferent for q. This necessitated two different axis. The top one
or ceiling heating. It was put on the top because this matches
the position of the ceiling (physically, ceiling is at the top in
om). The bottom axis is for floor heating. This axis was also
for ceiling cooling where the heat transfer mechanism is the
except the sign change and the temperature drops are subtracted
ead of addition, starting from the panel surface. The pivot air
erature is 70°F for heating and 80°F for cooling.
.2:The significance of tube spacing depends both on r_u, its split
q. This significance is reflected in the nomograph by oblique t_w
s. The pivot for these lines is M:12'. These lines were drawn
that when one follows any given r_u line but inctreases q, then
solution is exact. When one keeps q fixed but changes r_u, then the
p ratio becomes significant. However this complication only
ies for floor heating. This is represented by the dashed lines
r_c/r_p ratio starts to become significant on t_w axis.

Appendix 4. Current Edition of Chap.6,ASHRAE Handbook,1992
Systems and Equipment,IP Edition

I-P

CHAPTER 6

PANEL HEATING AND COOLING

RADIANT panel systems use controlled temperature surfaces in the floor, walls or ceiling; the temperature is maintained by circulating water, air, or electric resistance. Radiant panel systems may be combined with a central station air system of one-zone, constant temperature, constant volume design, or with dual-duct, reheat, multizone or variable volume systems. A controlled temperature surface is referred to as a radiant panel if 50% or more of the heat transfer is by radiation to other surfaces seen by the panel. This chapter is concerned with surfaces whose temperatures are controlled and are the primary source of heating and cooling within the conditioned space.

High-temperature surface radiant panels over about 300°F energized by gas, electricity, or high-temperature water are discussed in Chapter 15.

RADIANT ENERGY TRANSFER PRINCIPLES

Radiant energy (1) is transmitted at the speed of light; (2) travels in straight lines and can be reflected; and (3) elevates the temperature of solid objects by absorption, but does not heat the air through which it travels.

Radiant energy exchanges continuously between all bodies in a building environment. The rate at which radiant energy is transferred depends on the following factors:

- Temperature (of the emitting surface and receiver)
- Emissivity (of the radiating surface)
- Reflectivity, absorptivity, and transmissivity (of the receiver)
- The view factor between the emitting surface and receiver (viewing angle of the occupant to the radiant source)

A critical factor is the structure of the body surface. In general, rough surfaces have low reflectivity and high emissivity/absorptivity characteristics. Conversely, smooth or polished surfaces have high reflectivity and low absorptivity/emissivity characteristics.

One example of radiant heating is the feeling of warmth when standing in the sun's rays on a cool, sunny day. Some of the rays come directly from the sun and include the entire electromagnetic spectrum. Other rays from the sun impinge on surrounding objects and are absorbed or reflected. This generates a secondary source of energy, creating rays that are a combination of the wavelength produced by the temperature of the objects and the wavelength of the reflected rays. If a cloud passes in front of the sun, there is an instant sensation of cold. This sensation is caused by the decrease in radiant energy received from the sun, although there is little, if any, change in the surrounding ambient air temperature.

Thermal comfort, as defined in ANSI/ASHRAE *Standard* 55-1981, is "that condition of mind which expresses satisfaction with the thermal environment." No system is completely satisfactory unless the three main factors controlling heat transfer from the human body (radiation, convection, and evaporation) result in thermal neutrality.

Designers sometimes think that a radiant heat transfer system is desirable only for certain types of buildings and only in some climates. However, maintaining correct conditions for human comfort by radiant heat transfer is possible for even the most severe climatic conditions (Buckley 1989).

Panel heating and cooling systems provide an acceptable thermal environment by controlling surface temperature within an occupied space, thus affecting the radiant heat transfer. With a properly designed system, a person should not be aware that the environment is being heated or cooled. The mean radiant temperature (MRT) has a strong influence on body comfort. When the temperature of the surfaces comprising the building deviates excessively from the ambient temperature (particularly outside walls with large amounts of glass), convective systems sometimes have difficulty in counteracting the discomfort caused by cold or hot surfaces. Heating and cooling panels neutralize these deficiencies and minimize radiation losses or gains by the body.

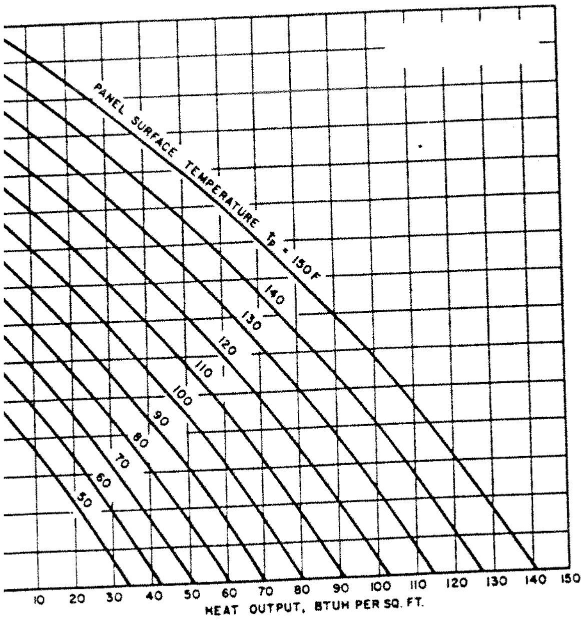
Most building materials have surfaces with relatively high emissivity factors and, therefore, absorb, reradiate, and reflect radiant heat from active panels. Warm ceiling panels are effective because their radiant heat is absorbed and reflected by the irradiated surfaces and not transmitted through the construction. Glass is also opaque to the wavelengths emitted by active panels, and therefore, transmits little of the long-wave radiant heat to the outside. This is significant because all surfaces within the room tend to assume temperatures that result in an acceptable thermal comfort condition within the space.

GENERAL EVALUATION

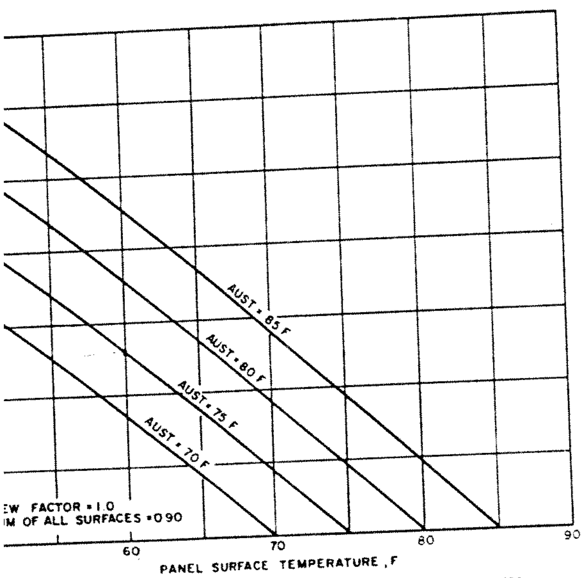
Principal advantages of radiant panel systems are:

- Comfort levels can be better than those of other conditioning systems because radiant loads are treated directly and air motion in the space is at normal ventilation levels.
- Mechanical equipment is not needed at the outside walls, simplifying the wall, floor, and structural systems.
- All pumps, fans, filters, and so forth are centrally located, simplifying maintenance and operation.
- No space is required within the air-conditioned room for mechanical equipment. This feature is especially valuable in hospital patient rooms and other applications where space is at a premium, where maximum cleanliness is essential, or where dictated by legal requirements.
- Cooling and heating can be simultaneous, without central zoning or seasonal changeover, when four-pipe systems are used.
- Supply air quantities usually do not exceed those required for ventilation and dehumidification.

The preparation of this chapter is assigned to TC 6.5, Radiant Space Heating and Cooling.



. 1 **Radiation Heat Transfer from Heated Ceiling, Floor, or Wall Panel**



:g. 2 **Heat Removed by Radiation to Cooled Ceiling or Wall Panel**

:rature at the 5-ft level will closely approach the average ›led surface temperatures (AUST). In structures where the heat gain is through the walls or where incandescent light- used, the wall surface temperatures tend to rise considera- ›ove the room air temperature.

/ection Transfer

e convection coefficient q_c is defined as the heat transferred nvection in Btu/h·ft²·°F difference between air and panel eratures. Heat transfer convection values are not easily estab- 1. Convection in panel systems is usually natural; that is, air on is generated by the warming or cooling of the boundary of air. In practice, however, many factors such as a room's guration interfere with or affect natural convection. Infiltra- the movement of persons, and mechanical ventilating sys- can introduce some forced convection that will disturb the ral process.

Parmelee and Huebscher (1947) included the effect of forced convection on heat transfer from panels as an increment to be added to the natural convection coefficient. However, increased heat transfer from forced convection should not be used, because the increments are unpredictable in pattern and performance, and forced convection does not significantly increase the total capacity of the panel system.

Convection in a panel system is a function of the panel surface temperature and the temperature of the airstream layer directly below the panel. The most consistent measurements are obtained when the air layer temperature is measured close to the region where the fully developed stream begins, usually 2 to 2.5 in. below the panels.

Min *et al.* (1956) determined natural convection coefficients 5 ft above the floor in the center of a 12 ft by 24 ft room. Equations (6) to (11), derived from this research, can be used to calculate heat transfer from panels by natural convection.

Natural convection from a heated ceiling

$$q_c = 0.041\,(t_p - t_a)^{1.25}/D_e^{0.25} \tag{6}$$

Natural convection from a heated floor or cooled ceiling

$$q_c = 0.39\,(t_p - t_a)^{1.31}/D_e^{0.08} \tag{7}$$

Natural convection from a heated or cooled wall panel

$$q_c = 0.29\,(t_p - t_a)^{1.32}/H^{0.05} \tag{8}$$

where

q_c = heat transfer by natural convection, Btu/h·ft²
t_p = temperature of panel surface, °F
t_a = temperature of air, °F
D_e = equivalent diameter of panel (4 area/perimeter), ft
H = height of wall panel, ft

Schutrum and Humphreys (1954) measured panel performance in furnished test rooms that did not have uniform temperature surfaces and found no variations large enough to be significant in heating practice. Schutrum and Vouris (1954) established that the effect of room size was also usually insignificant. The convection equations can therefore be simplified to:

Natural convection from a heated ceiling

$$q_c = 0.021\,(t_p - t_a)^{1.25} \tag{9}$$

Natural convection from a heated floor or cooled ceiling

$$q_c = 0.32\,(t_p - t_a)^{1.31} \tag{10}$$

Natural convection from a heated or cooled wall panel

$$q_c = 0.26\,(t_p - t_a)^{1.32} \tag{11}$$

Figure 3 shows heat output by natural convection from floor, wall, and ceiling heating panels as calculated from Equations (9) and (10).

Figure 4 shows heat removed by natural convection by cooled ceiling panels as calculated by Equation (10) and data from Wilkes and Peterson (1938) for specific panel sizes. An additional curve illustrates the effect of forced convection on the latter data. Similar adjustment of the ASHRAE data is inappropriate, but the effects would be much the same. For preliminary design, use 1 Btu/h·ft²·°F between room design and panel temperature.

Combined Heat Transfer (Radiation and Convection)

The combined heat transfer from a panel to a room can be determined by adding the radiant heat transfer from Figure 1 or 2 to the convective heat transfer from Figure 3 or 4, respectively. Figures 1 and 2 require calculating the AUST in the room. In calculating the AUST, the surface temperature of the interior walls is assumed to be the same as the room air temperature. The inside surface temperatures of outside walls and exposed floors or ceil-

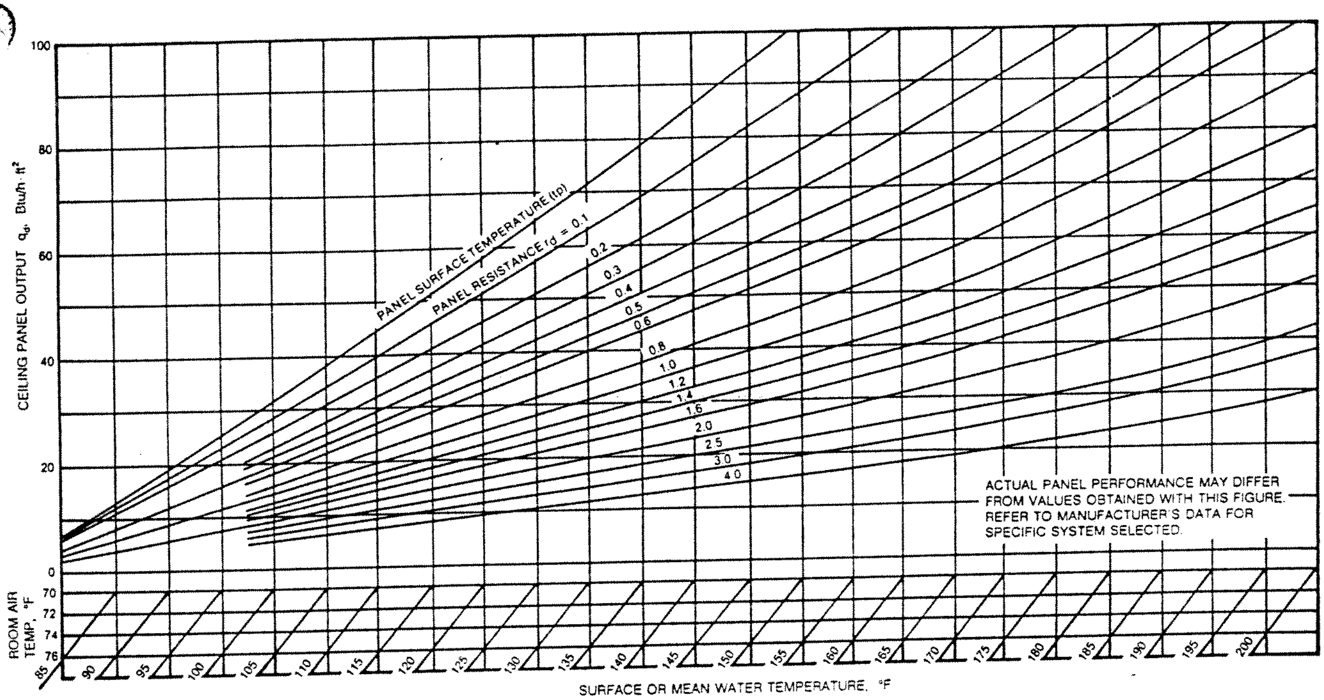


Fig. 7 **Ceiling Panel Design Graph Showing Panel Surface Temperature and Mean Water Temperature versus Output Downward**

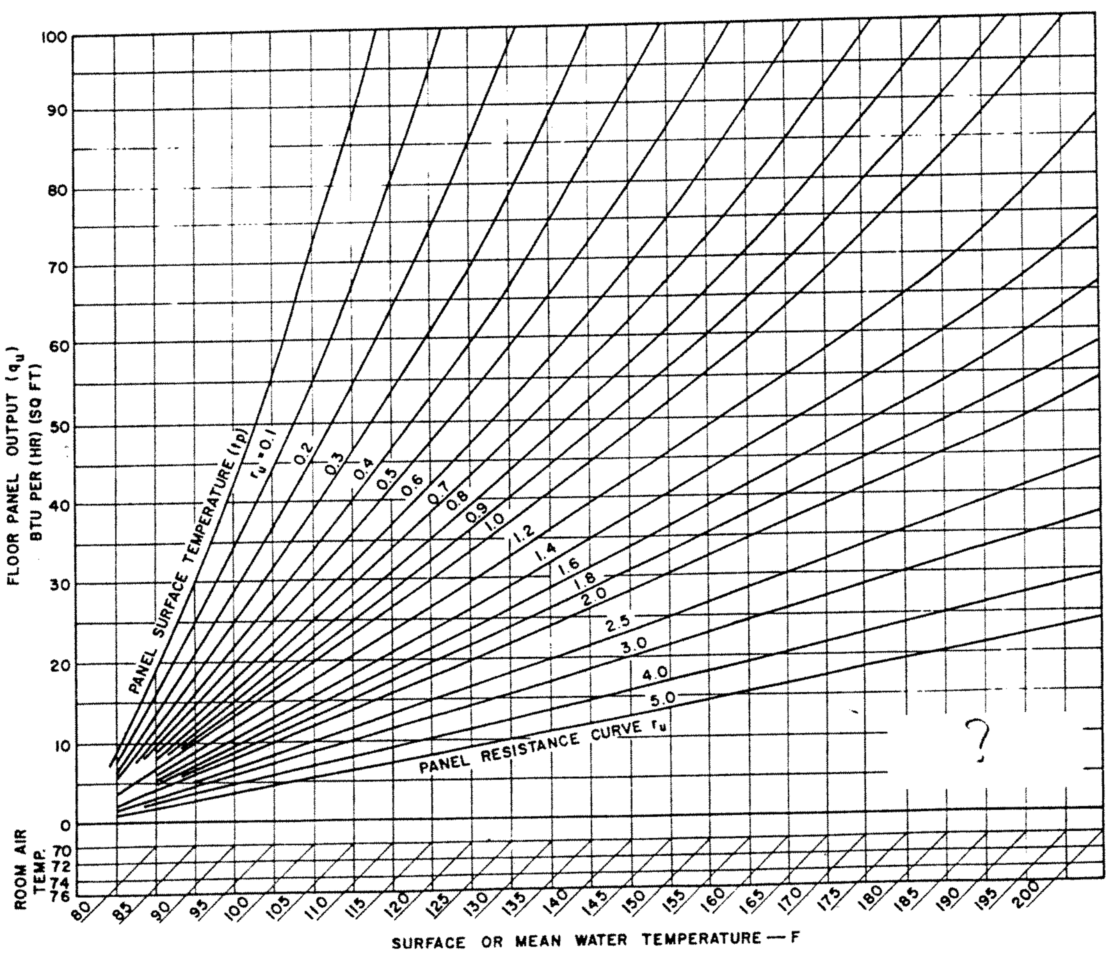


Fig. 8 **Floor Panel Design Graph Showing Panel Surface Temperature and Mean Water Temperature versus Output Upward**

2 Thermal Resistance of Bare Concrete Floor Panels (Heating)

ıction	Spacing, in.	Heat Flow Ratio, q_u/q_d or q_u/q_{de} 1 Up	1 Down	3 Up	3 Down	5 Up	5 Down	10 Up	10 Down
		Panel Thermal Resistance, $ft^2 \cdot °F \cdot h/Btu$							
ıncrete Slab with 2-in. Cover									
	9	0.57	0.52	0.46	0.84	0.43	1.17	0.42	1.97
rous tube	12	0.73	0.68	0.58	1.16	0.54	1.65	0.51	2.86
errous	9	0.49	0.42	0.41	0.66	0.39	0.90	0.38	1.80
in. rous tube	12	0.63	0.55	0.50	0.93	0.48	1.30	0.46	2.35
ıncrete Slab with 2-in. Cover									
	9	0.59	0.70	0.47	1.05	0.45	1.39	0.43	2.25
rous tube	12	0.78	0.90	0.60	1.40	0.56	1.97	0.54	3.21
	9	0.51	0.61	0.43	0.87	0.41	1.13	0.40	1.78
rous pipe	12	0.68	0.78	0.54	1.23	0.51	1.63	0.49	2.61
	9	0.47	0.55	0.40	0.77	0.39	0.98	0.38	1.50
s pipe	12	0.63	0.71	0.50	1.07	0.48	1.44	0.46	2.36
nferrous tube	12	0.59	0.66	0.48	0.98	0.46	1.30	0.45	2.11
ous pipe	15	0.73	0.83	0.57	1.21	0.54	1.73	0.51	2.74

ameters are nominal.
ward heat flow from panel
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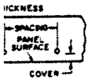
le 3 Thermal Resistance of Concrete Ceiling Panels (Heating)



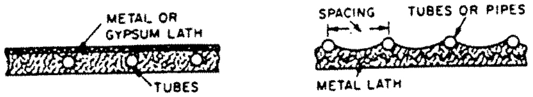
Construction[a]	Spacing, in.	Heat Flow Ratio,[b] q_u/q_d 0 Up	0 Down	0.5 Up	0.5 Down	1.0 Up	1.0 Down
		Panel Thermal Resistance, $ft^2 \cdot °F \cdot h/Btu$					
ıncrete Slab with 1-in. Cover							
	9	3.6	0.30	0.9	0.35	0.7	0.45
rous tube	12	5.1	0.35	1.1	0.45	0.9	0.55
ferrous or	9	2.6	0.25	0.7	0.30	0.6	0.35
ı. non- s tube	12	4.0	0.30	0.9	0.40	0.8	0.50
ferrous pipe	9	2.1	0.20	0.6	0.25	0.6	0.30
5-in. non-	12	3.3	0.30	0.8	0.35	0.7	0.40
s tube	15	4.5	0.35	1.0	0.45	0.8	0.55
rrous pipe	9	1.6	0.20	0.5	0.25	0.5	0.25
	12	2.6	0.25	0.7	0.30	0.9	0.40
	15	3.6	0.30	0.9	0.40	0.7	0.45
ıncrete Slab with 1-in. Cover							
nonferrous	9	3.6	0.30	1.0	0.35	0.8	0.40
	12	5.2	0.35	1.2	0.45	1.0	0.55
ferrous pipe	9	2.9	0.25	0.9	0.30	0.8	0.35
5-in. non- s tube	12	4.0	0.30	1.1	0.40	0.9	0.45
ferruous	9	2.2	0.20	0.8	0.30	0.7	0.30
r 1-in.	12	3.3	0.30	1.0	0.35	0.8	0.40
rrous tube	15	4.3	0.35	1.1	0.40	0.9	0.50
rrous pipe	9	1.7	0.20	0.7	0.25	0.7	0.25
	12	2.7	0.25	0.9	0.30	0.8	0.35
	15	3.7	0.30	1.0	0.40	0.9	0.45

ıward heat flow from panel
ıwnward heat flow from panel
ameters are nominal.
ling panel also acts as a floor panel to the extent of its upward heat flow.
ward heat flow is high and the space above is occupied, check floor surface
ure for possible foot discomfort. Also check effect on heating requirements
ace above. Do not supply the major portion of the upper room's heating
ıents by the upward heat flow of a ceiling panel below.

Table 4 Thermal Resistance of Plaster Ceiling Panels (Heating or Cooling)

Panel Construction

Standard gypsum plaster (three coats) with 3/8-in. nominal nonferrous tube or 1/2-in. nominal ferrous pipe above metal lath tied at 8-in. intervals with good tube imbedment, or 3/8-in. nominal nonferrous tube below metal or gypsum lath.



Spacing, in.	Thermal Resistance to Downward Heat Flow r_d, $ft^2 \cdot °F \cdot h/Btu$ — Heat Flow Ratio q_u/q_d: 0	0.2	0.4	0.6	0.8	1.0
4.5	0.30	0.34	0.38	0.42	0.46	0.50
6	0.45	0.51	0.57	0.63	0.69	0.75
9	0.75	0.85	0.95	1.05	1.15	1.25
16	1.15	1.29	1.43	1.57	1.71	1.85

Notes:
1. Any ceiling panel also acts as a floor panel to the extent of its upward heat flow. If the upward heat flow is high and the space above is occupied, check floor surface temperature for possible foot discomfort (see Schutrum *et al.* 1953). Also check effect on heating requirements of the space above. Do not supply the major portion of the upper room's heating requirements by the upward heat flow of a ceiling panel below.
2. Recommended maximum inlet water temperature t_{max} = 140°F.

Table 5 Thermal Resistance of Floor Coverings

Description	Resistance r_{uc}, $ft^2 \cdot °F \cdot h/Btu$
Bare concrete, no covering	0.00
Asphalt tile	0.05
Rubber tile	0.05
Light carpet	0.6
Light carpet with rubber pad	1.0
Light carpet with light pad	1.4
Light carpet with heavy pad	1.7
Heavy carpet	0.8
Heavy carpet with rubber pad	1.2
Heavy carpet with light pad	1.6
Heavy carpet with heavy pad	1.9

Effect of Floor Coverings

Floor coverings can have a pronounced effect on the performance of a floor heating panel system. The added thermal resistance of the floor covering reduces upward heat flow and increases the heat flow to the underside of the panel. To maintain a given upward heat flow after a floor covering has been added, the temperature of the heating medium must be increased. Data on the thermal resistance of common floor coverings are given in Table 5.

Where covered and bare floor panels exist in the same system, it may be possible to maintain a high enough water temperature to satisfy the covered panels and balance the system by throttling the flow to the bare slabs. In some instances, however, the increased water temperature required when carpeting is applied over floor panels makes it impossible to balance floor panel systems in which only some rooms have carpeting, unless the piping is arranged to permit zoning using more than one water temperature.

Panel Heat Losses

Heat transferred from the upper surface of ceiling panels, the back surface of wall panels, the underside of floor panels or the

fer to manufacturer's data for panel surface temperatures higher than those given in Figure 7.
5. Design the panel arrangement.
17. Thermal comfort requirements should be checked in the following steps (see Chapter 8 of the 1989 ASHRAE *Handbook—Fundamentals* and NRB 1981).
    a. Determine occupant's clothing value (clo value) and metabolic rate (MET) (see Tables 1D and 4A, Chapter 8 of the 1989 ASHRAE *Handbook—Fundamentals*).
    b. Determine the optimum operative temperature at the coldest point in the room (see Figure 15, Chapter 8 of the 1989 ASHRAE *Handbook—Fundamentals* or NRB Figure 2.3.22 for other values).
    c. Determine the mean radiant temperature (MRT) at the coldest point in the room (see Fanger 1972).
    d. From the definition of operative temperature, establish the optimum room design temperature at the coldest point in the room. If the optimum room design temperature varies greatly from the designated room design temperature, designate a new temperature.
    e. Determine the MRT at the hottest point in the room.
    f. Calculate the operative temperature at the hottest point in the room.
    g. Compare the operative temperatures at the hottest and coldest points in the room. For light activity and normal clothing, the acceptable operative temperature range is 68 to 75 °F (see NRB 1981 for other ranges). If the range is not acceptable, the heating system must be modified.
    h. Calculate radiant temperature asymmetry (NRB 1981). Acceptable ranges are < 18 °F for windows and < 9 °F for warm ceilings.
18. Determine water flow rate and pressure drop.

The application, design, and installation of panel systems have certain requirements and techniques such as the following:

1. As with any hydronic system, look closely at the piping system design. Piping should be designed to ensure that water of the proper temperature and in sufficient quantity is available to every grid or coil at all times. Reverse-return systems should be considered to minimize balancing problems.
2. Individual panels can be connected for parallel flow using headers, or for sinuous or serpentine flow. To avoid flow irregularities within a header-type grid, the water channel or lateral length should be greater than the header length. If the laterals in a header grid are forced to run in a short direction, this problem can be solved by using a combination series-parallel arrangement. Serpentine flow will ensure a more even panel surface temperature throughout the heating or cooling zone.
3. Noises from entrained air, high-velocity, or high-pressure drop devices or from pump and pipe vibrations must be avoided. Water velocities should be high enough to prevent separated air from accumulating and causing air binding. Where possible, avoid automatic air venting devices over ceilings of occupied spaces.
4. Design piping systems to accept thermal expansion adequately. Do not allow forces from piping expansion to be transmitted to panels. Thermal expansion of the ceiling panels must be considered.
5. In circulating water systems, plastic, steel, and copper pipe or tube are used widely in ceiling, wall, or floor panel construction. Where coils are embedded in concrete or plaster, no threaded joints should be used for either pipe coils or mains. Steel pipe should be the all-welded type. Copper tubing should be soft-drawn coils. Fittings and connections should be minimized. Changes in direction should be made by bending. Solder-joint fittings for copper tube should be used with a medium temperature solder of 95% tin, 5% antimony, or capillary brazing alloys. All piping should be subjected to a hydrostatic test of at least three times the working pressure. Maintain adequate pressure in embedded piping while pouring concrete.
6. Placing the thermostat on a side wall where it can see the outside wall and the warm panel should be considered. The normal thermostat cover reacts to the warm panel, and the radiant effect of the panel on the cover tends to alter the control point so that the thermostat controls 2 to 3 °F lower when the outdoor temperature is a minimum and the panel temperature is a maximum. Experience indicates that radiantly heated rooms are more comfortable under these conditions than when the thermostat is located on a back wall.
7. If throttling valve control is used, either the end of the main should have a fixed bypass, or the last one or two rooms on the mains should have a bypass valve to maintain water flow in the main. Thus, when a throttling valve modulates, there will be a rapid response.
8. When selecting heating design temperatures for a ceiling panel surface, the design parameters are:
    a. Excessively high temperatures over the occupied zone will cause the occupant to experience a "hot head effect."
    b. Temperatures that are too low can result in an oversized, uneconomical panel and a feeling of coolness at the outside wall.
    c. Locate ceiling panels adjacent to perimeter walls and/or areas of maximum load.
    d. With normal ceiling heights of 8 to 9 ft, panels less than 3 ft wide at the outside wall can be designed for 235 °F surface temperature. If panels extend beyond 3 ft into the room, the panel surface temperature should be limited to the values as given in Figure 16. The surface temperature of concrete or plaster panels is limited by construction.
9. Floor panels are limited to surface temperatures of less than 85 °F for comfort reasons.
10. When the panel chilled water system is started, the circulating water temperature should be maintained at room temperature until the air system is completely balanced, the dehumidification equipment is operating properly, and building humidity is at design value.

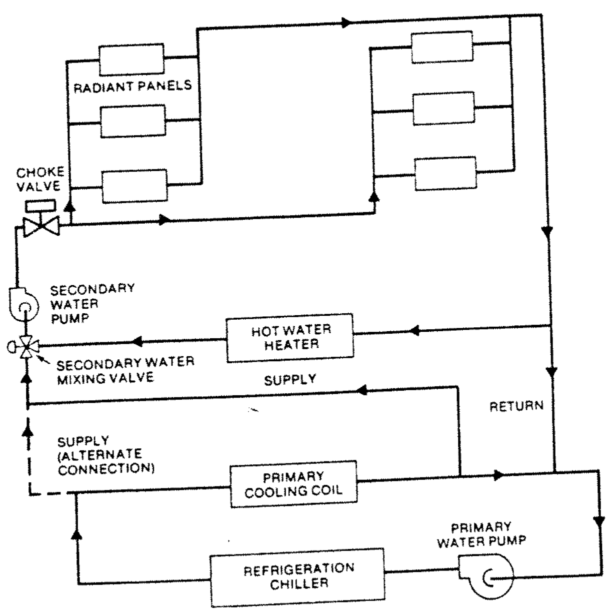
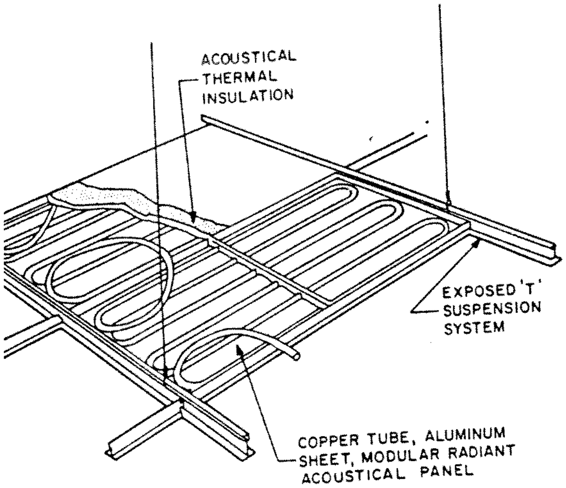


Fig. 11 Primary/Secondary Water Distribution System with Mixing Control



14 **Metal Ceiling Panels Bonded to Copper Tubing**

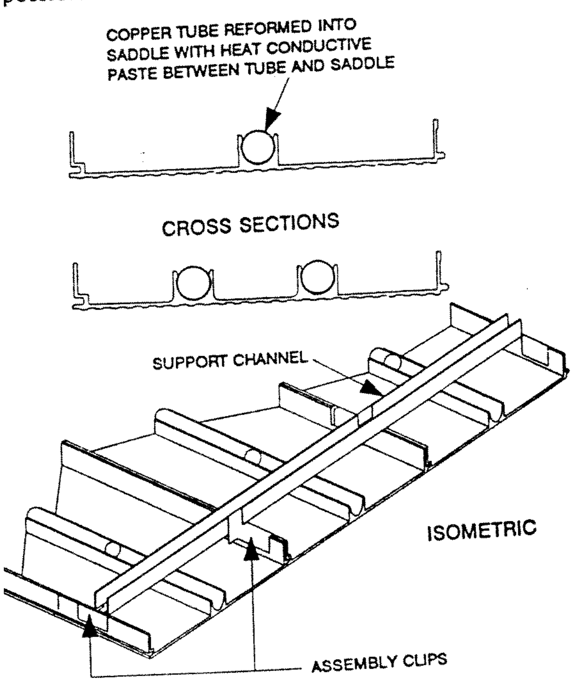
stance and respond quickly to changes in space conditions. shows thermal resistance values for various ceiling conns.

l radiant ceiling panels can be used with any of the all-air ; systems described in Chapter 2. Chapters 23 through 25 1989 ASHRAE *Handbook—Fundamentals* describe how ulate heating loads. Double glazing and heavy insulation ide walls has reduced transmission heat losses. As a result, tion and reheat have become of greater concern. Additional considerations are as follows:

imeter radiant heating panels, not extending more than 3 nto the room, may operate at higher temperatures, as cribed under "Design Considerations" in the Heating and oling Systems section.

dronic panels operate efficiently at low temperature and are table for condenser water heat reclaim systems.

cate ceiling panels adjacent to the outside wall and as close possible to the areas of maximum load. The panel area



15 **Extruded Aluminum Panels with Integral Copper Tube**

within 3 ft of the outside wall should have a heating capacity equal to or greater than 50% of the wall transmission load.

4. Ceiling system designs based on passing return air through perforated modular panels into the plenum space above the ceiling are not recommended, because much of the panel heat transfer is lost to the return air system.
5. When selecting heating design temperatures for a ceiling panel surface or mean water temperature, the design parameters are:
   a. Excessively high temperatures over the occupied zone will cause the occupant to experience a "hot head effect."
   b. Temperatures that are too low can result in an oversized, uneconomical panel and a feeling of coolness at the outside wall.
   c. Give the technique in item 3 priority.
   d. With normal ceiling heights of 8 to 9 ft, panels less than 3 ft wide at the outside wall can be designed for 235°F surface temperature. If panels extend beyond 3 ft into the room, the panel surface temperature should be limited to the values as given in Figure 16.
6. Allow sufficient space above the ceiling for installation and connection of the piping that forms the radiant panel ceiling.

Metal radiant acoustic panels provide heating, cooling, sound absorption, insulation, and unrestricted access to the plenum space. They are easily maintained, can be repainted to look new, and have a life expectancy in excess of 30 years. The system is quiet, comfortable, draft-free, easy to control, and responds quickly. The system is a basic air-and-water system. First costs are competitive with other systems, and a life cycle cost analysis often shows that the long life of the equipment makes it the lowest cost in the long run. The system has been used in hospitals, schools, office buildings, colleges, airports, and exposition facilities.

Metal radiant panels can also be integrated into the ceiling design to provide a narrow band of radiant heating around the perimeter of the building. The radiant system offers advantages over baseboard or overhead air in appearance, comfort, operating efficiency and cost, maintenance, and product life.

DISTRIBUTION AND LAYOUT

Chapter 3 and chapters covering hydronic systems apply to radiant panels. Layout and design of metal radiant ceiling panels for heating and cooling begins early in the job. The type of ceiling chosen influences the radiant design and conversely, thermal considerations may dictate what ceiling type to be used. Heating panels should be located adjacent to the outside wall. Cooling panels may be positioned to suit other elements in the ceiling. In applications with normal ceiling heights, heating panels that

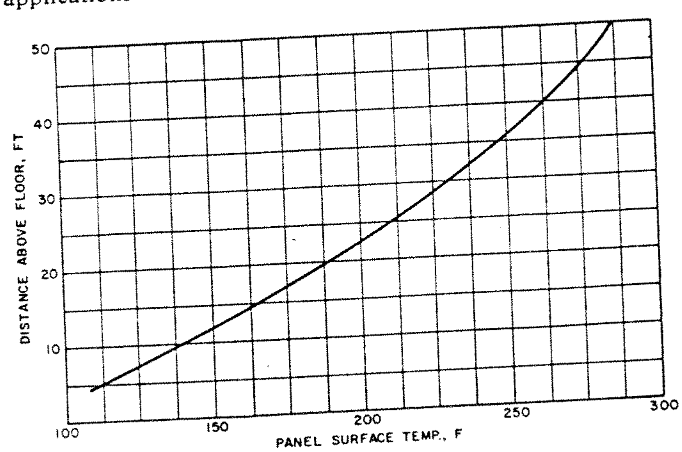


Fig. 16 Suggested Design Ceiling Surface Temperatures at Various Ceiling Heights

stone, or concrete mounds can be used. No wood, brick, concrete block, or similar materials should support coils. A waterproofing layer is desirable to protect insulation and piping.

Where coils are embedded in structural load-supporting slabs above grade, construction codes may affect their position. Otherwise, the coil piping is installed as described for slabs resting on grade.

The warm-up and start-up period for concrete panels are similar to those outlined for plaster panels.

Embedded systems may fail sometime during their life. Adequate valves and properly labeled drawings will help to isolate the point of failure.

Suspended Floor Piping

Piping may be applied on or under suspended wood floors using several methods of construction. Piping may be attached to the surface of the floor and be embedded in a layer of concrete or gypsum, mounted in or below the subfloor, or attached directly to the underside of the subfloor using metal panels to improve heat transfer from the piping. Whichever method is used for optimum floor output and comfort, it is important that the heat be evenly distributed throughout the floor. Pipes are generally spaced on 4- to 12-in. spacing. Wide spacing under tile or bare floors can cause uneven surface temperatures.

Figure 21 illustrates construction with piping embedded in concrete or gypsum. The thickness of the embedding material is generally 1 to 2 in. when applied to a wood subfloor. Gypsum products specifically designed for floor heating can generally be installed 1 to 1.5 in. thick because they are more flexible and crack resistant than concrete. When concrete is used, it should be of structural quality to reduce cracking due to movement of the wood frame or shrinkage. Embedding material must provide a hard, flat, smooth surface that can accommodate a variety of floor coverings.

As illustrated in Figure 22, tubing may also be installed in the subfloor. The tubing is installed on top of the rafters between the subflooring members. Heat transfer can be improved by the addition of metal heat transfer plates which spread the heat beneath the finished flooring. This construction is illustrated in the figure.

A third construction option is to attach the tube to the underside of the subfloor with or without metal heat transfer plates. The construction is illustrated in Figure 23.

Transfer from the hot-water tube to the surface of the floor is the important consideration in all cases. The floor surface temperature affects the actual heat transfer to the space. Any hindrance between the heated water tube and the floor surface will affect the efficiency of the system. As pointed out in the Panel Thermal Resistance section, the method that most effectively transfers and spreads the heat evenly through the subfloor with the least resistance produces the best results.

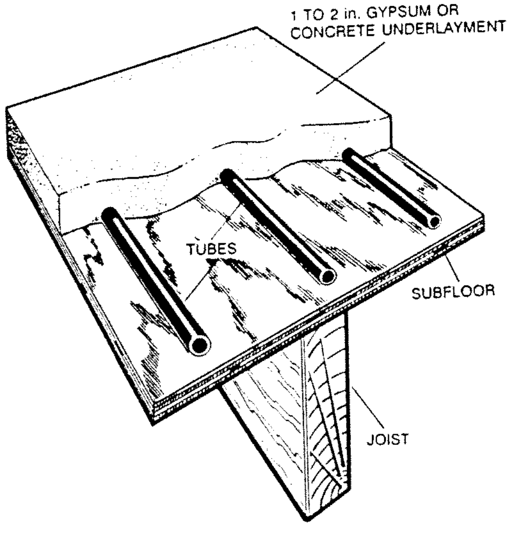


Fig. 21 Embedded Pipe

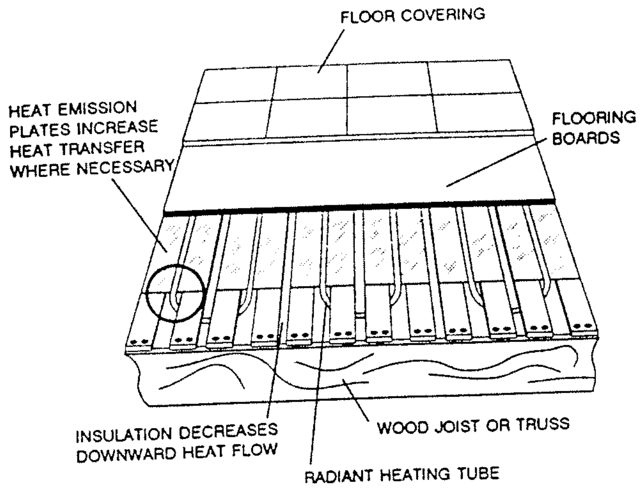


Fig. 22 Pipe in Subfloor

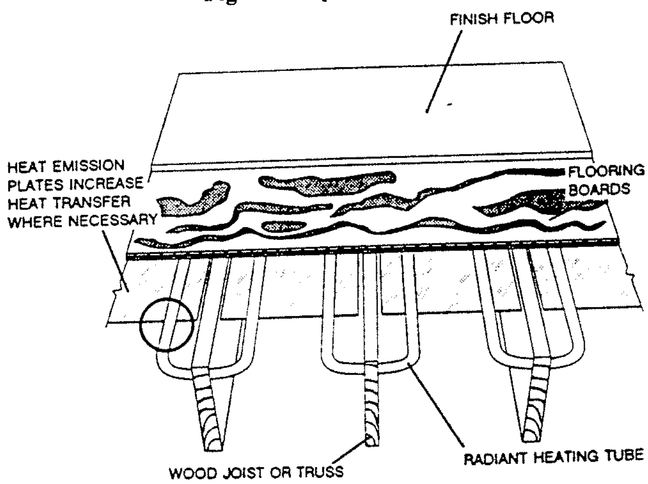


Fig. 23 Pipe under Subfloor

ELECTRICALLY HEATED SYSTEMS

Several different forms of electric resistance units are available for heating interior room surfaces. These include: (1) electric heating cables that may be embedded in concrete or plaster or laminated in drywall ceiling construction; (2) prefabricated electric heating panels to be attached to room surfaces; and (3) electrically heated fabrics or other materials for application to, or incorporation into, finished room surfaces.

Electrically Heated Ceilings

Prefabricated electric ceiling panels. A variety of prefabricated electric heating panels are available for either supplemental or full-room heating. These panels are available in sizes from 2 ft by 4 ft to 6 ft by 12 ft. They are constructed from a variety of materials such as gypsum board, glass, steel fiberglass, or vinyl. Different panels have rated inputs varying from 10 to 95 W/ft^2 for 120, 208, 240, and 277 V service. Maximum operating temperatures vary from about 100 to about 300°F, depending on watt density. National emperature change should be kept low by other heat or by ventilation until the plaster is thoroughly cured. Ver- or other insulating plaster causes cables to overheat and ary to code provisions.

minated drywall ceiling panels, the heating cable is placed two layers of gypsum board, plasterboard, or other ther- noninsulating fire-resistant ceiling lath. The cable is stapled to the first (or upper) lath, and the two layers are held apart ickness of the heating cable. It is essential that the space the two layers of lath be *completely* filled with a nonin- plaster or similar material. This fill holds the cable firmly and improves heat transfer between the cable and the l ceiling. Failure to fill the space completely between the ers of plasterboard may allow the cable to overheat in the g voids and cause cable failure. The plaster fill should be according to manufacturer's specifications.

ric heating cables are ordinarily installed with a 6-in. non- border around the periphery of the ceiling. An 8-in. clear- ust be provided between heating cables and the edges of let or junction boxes used for surface-mounted lighting fix- A 2-in. clearance must be provided from recessed lighting s, trim, and ventilating or other openings in the ceiling.

ing cables or panels must be installed only in ceiling areas not covered by partitions, cabinets, or other obstructions. er, it is permissible for a single run of isolated embedded o pass over a partition.

National Electric Code requires that all general power and iring be run above the thermal insulation or at least 2 in. the heated ceiling surface, or that the wiring be derated. rywall ceiling construction, the heating cable is always ed with the cable runs parallel to the joist. A 2.5-in. clear- etween adjacent cable runs must be left centered under each or nailing. Cable runs that cross over the joist must be kept inimum. Where possible, these crossings should be in a it line at one end of the room.

cable having a watt density of 2.75 W/ft, the minimum per- le spacing is 1.5 in. between adjacent runs. Some manufac- recommend a minimum spacing of 2 in. for drywall uction.

spacing between adjacent runs of heating cable can be nined using Equation (12):

$$s = 12\,A_n/C \tag{12}$$

,

cable spacing, in.
net panel heated area, ft^2
length of cable, ft

t panel area A_n in Equation (12) is the net ceiling area avail- after deducting the area covered by the nonheating border, ng fixtures, cabinets, and other ceiling obstructions. For sim- y, Equation (12) contains a slight safety factor, and small ng fixtures are usually ignored in determining net ceiling area. e 2.5-in. clearance required under each joist for nailing in all applications occupies one-fourth of the ceiling area, if the are 16 in. on center. Therefore, for drywall construction, the rea A_n must be multiplied by 0.75. Many installations have cing of 1.5 in. for the first 2 ft from the cold wall. Remain- able is then spread over the balance of the ceiling.

trically Heated Wall Panels

ble embedded in walls similar to ceiling construction is occa- ally found in Europe. Because of possible damage from nails n for hanging pictures or from building alterations, most s in the United States prohibit such panels. Some of the abricated panels described in the preceding section are also for wall panel heating.

Electrically Heated Floors

Electric heating cable assemblies, such as those used for ceiling panels, are sometimes used for concrete floor heating systems. Since the possibility of cable damage during installation is greater for concrete floor slabs than for ceiling panels, these assemblies must be carefully installed. After the cable has been placed, all unnecessary traffic should be eliminated until the concrete covering has been placed and hardened.

Preformed mats are sometimes used for electric floor slab heating systems. These mats usually consist of PVC-insulated heating cable woven in, or attached to, metallic or glass fiber mesh. Such mats are available as prefabricated assemblies in many sizes from 2 to 100 ft^2 and with various watt densities ranging from 15 to 25 W/ft^2. When used with a thermally treated cavity beneath the floor, a heat storage system is provided, which may be controlled for off-peak heating.

Mineral-insulated (MI) heating cable is another effective method of slab heating. MI cable is a small-diameter, highly durable, flexible heating cable composed of solid electric-resistance heating wire or wires surrounded by tightly compressed magnesium oxide electrical insulation and enclosed by a metal sheath. MI cable is available in stock assemblies in a variety of standard voltages, watt densities, and lengths. A cable assembly consists of the specified length of heating cable, waterproof hot-cold junctions, 7-ft cold sections, UL-approved end fittings, and connection leads. Several standard MI cable constructions are available, such as single conductor, twin conductor, and double cable. Custom-designed MI heating cable assemblies can be ordered for specific installations.

Other outer-covering materials that are sometimes specified for electric floor heating cable include (1) silicone rubber, (2) lead, and (3) tetrafluoroethylene (Teflon).

For a given floor heating cable assembly, the required cable spacing is determined from Equation (12). In general, cable watt density and spacing should be such that floor panel watt density is not greater than 15 W/ft^2. Higher watt densities (up to 25 W/ft^2) are often specified for the 2-ft border next to cold walls. Check with the latest issue of the *National Electric Code* and other applicable codes to obtain information on maximum panel watt density and other required criteria and parameters.

Floor heating cable installation. When PVC-jacketed electric heating cable is used for floor heating, the concrete slab is laid in two pourings. The first pour should be at least 3 in. thick and, where practical, should be insulating concrete to reduce downward heat loss. For a proper bond between the layers, the finish slab should be placed within 24 h of the first pour, with a bonding grout applied. The finish layer should be at least 1.5 in. and not more than 2 in. thick. This top layer must not be insulating concrete. At least 1 in. of perimeter insulation should be installed as shown in Figure 26.

The cable is installed on top of the first pour of concrete not closer than 2 in. from adjoining walls and partitions. Methods of fastening the cable to the concrete include:

1. Staple the cable to wood nailing strips fixed in the surface of the rough slab. The predetermined cable spacing is maintained by daubs of cement, plaster of paris, or tape.
2. In light or uncured concrete, staple the cable directly to the slab using hand-operated or powered stapling machines.
3. Nail special anchor devices to the first slab to hold the cable in position while the top layer is being poured.

Preformed mats can be embedded in the concrete in a continuous pour. The mats are positioned in the area between expansion and/or construction joints and electrically connected to a junction box. The slab is poured to within 1.5 to 2 in. of the finished level. The surface is rough screeded and the mats placed in position. The final cap is applied immediately. Since the first pour re than 1.5 °F. Control of the interior zones is best accomplished by devices that reflect the actual presence of the internal load elements. Frequently, time clocks and current-sensing devices are used on lighting feeders.

Because air quantities are generally small, constant volume supply air systems should be used. With the apparatus arranged to supply air at an appropriate dew point at all times, comfortable indoor conditions can be maintained throughout the year with a panel cooling system. As with all systems, to prevent condensation on window surfaces, the supply air dew point should be reduced during extremely cold weather according to the type of glazing installed.

Electric Heating Slab Controls

For comfort heating applications, the surface temperature of a floor slab is held to a maximum of 80 to 85 °F. Therefore, when the slab is the primary heating system, thermostatic controls sensing air temperature should not be used to control temperature; instead, the heating system should be wired in series with a slab-sensing thermostat. The remote sensing thermostat in the slab acts as a limit switch to control maximum surface temperatures allowed on the slab. The ambient sensing thermostat controls the comfort level. For supplementary slab heating, as in kindergarten floors, a remote sensing thermostat in the slab is commonly used to tune in the desired comfort level. Indoor-outdoor thermostats are used to vary the floor temperature inversely with the outdoor temperature. If the heat loss of the building is calculated for 70 to 0 °F and the floor temperature range is held from 70 to 85 °F with a remote sensing thermostat, the ratio of outdoor temperature to slab temperature is 70:15, or approximately 5:1. This means that a 5 °F drop in outdoor temperature requires a 1 °F increase in the slab temperature. An ambient sensing thermostat is used to vary the ratio between outdoor and slab temperatures. A time clock is used to control each heating zone if off-peak slab heating is desirable.

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Appendix 5. Critical Review of Figure 9. ASHRAE Handbook,Chap.6

HEATWAY
Radiant Floors & Snowmelting

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Springfield, MO 6
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800/255/1996 (USA and Ca

Dear Mr. Buckley, 8/7/1992

Thank you for your kind letter dated 30 July,1992 and its enclosures. l have checked Figure 9 ,the related equations and the text in Chapter 6 of 1992 Systems and Equipment Handbook,in response to Mr.Parsons' and Sensecall's letters. As will be explained below,the discrepancy between Figure 9 and Equations 5B + 10 is not due to any additional correction factors applied to Figure 9. In fact, Figure 9 is an arithmetic sum of convective and radiative heat transfer expressions. However, Equation 10 is based on Min and et.al.'s data for natural convection,where Figure 9 seems to be based on natural convection data by Wilkes and Peterson. As both data can be seen on Figure 4, the latter yields higher heat transfer rates than Min and et.al's (ASHRAE Lab.)data. So my conclusion is that the inconsistency is due to the use of different natural convection data in Equation 10 and in Figure 9. Therefore ,when compared to Fig.9,one gets smaller total heat transfer rates through the equations . This also answers the question raised by Mr. Sensecall. Please find below my detailed explanations,suggestions and recommendations at below:

- The text already mentions that the convective heat transfer equation (Eq.10) is after Min and et.al data (1956,then ASHAE LAB). After ignoring room size effect,original equation (Eq.7) simplifies to Equation(10). Although l have reservations for such a simplification,as l already mentioned in my report submitted to

'C 6.5 in Baltimore,at the moment lets take Eq.10 as granted: ...sing Equation 10, one obtains q_c:11.1 Btu/h.ft^2 as noted in Mr.Parsons' letter. Exactly the same value can be read from Figure 4 using ASHAE,1956 data line (first line from right).On the same figure,original Wilkes and Peterson Data reads around 16 Btu/h.ft^2 ...hen one adds this convective value instead of 11.1 to the radiant ...art given either by Equation 5B or Figure 2,then it becomes ...ossible to read same values as given in Figure 9.i.e:13.2+16=29.2. ...his value now perfectly matches with Figure 9 reading for the ...iven example. This is a sound indication that Wilkes and Peterson ...ata was employed in drawing Figure 9 instead of ASHAE data. ...therwise one gets systematically lower values through Equation ...0. 1 checked few more points .Please see the copy enclosed for IP ...ersion.

...I have noted that there is no any discrepancy between Eq.5B (in its ...evised format in SI) and Figure2.

...igure 9 is drawn for the condition of Tair=AUST. As already ...entioned in the text,this condition usually does not hold and the ...oom air temperature lines in Figure 9 does not intercept panel ...emperature axis at their value.i.e: 24^{o}C line does not necessarily ...intercept 24^{o}C point on the ceiling panel temperature axis.

...In order to clarify this matter,Figure 9 needs to have a label ...learly indicating this condition i.e: "for T_a=AUST".

...I have also noted thet there are some rounding offs on the scales ...of SI version of Figure 9. This partly explains the differences in ...eadings from IP and SI versions of this Figure.

...My conclusion:

③

For 1996 Handbook:

It is my suggestion that Figure 9 should be re-drawn using ASHAE data or any updated data that l would like to introduce and thus set the consistency between Equation 10 and Figure 9 (l can draw them),

Possibly replace Equation 10 with an updated model,

Keep Figure 4 in its present form or add more data on it for giving an in depth information to the reader about different models,but make it very clear in the text about which line is preferred and used in Figure 9.

As l have submitted to you earlier,there are other major points that seem to need to be updated and modified (like Figure 7).l will re-explain them on proper ASHRAE forms soon.

In the short run:

l would like to suggest the following;

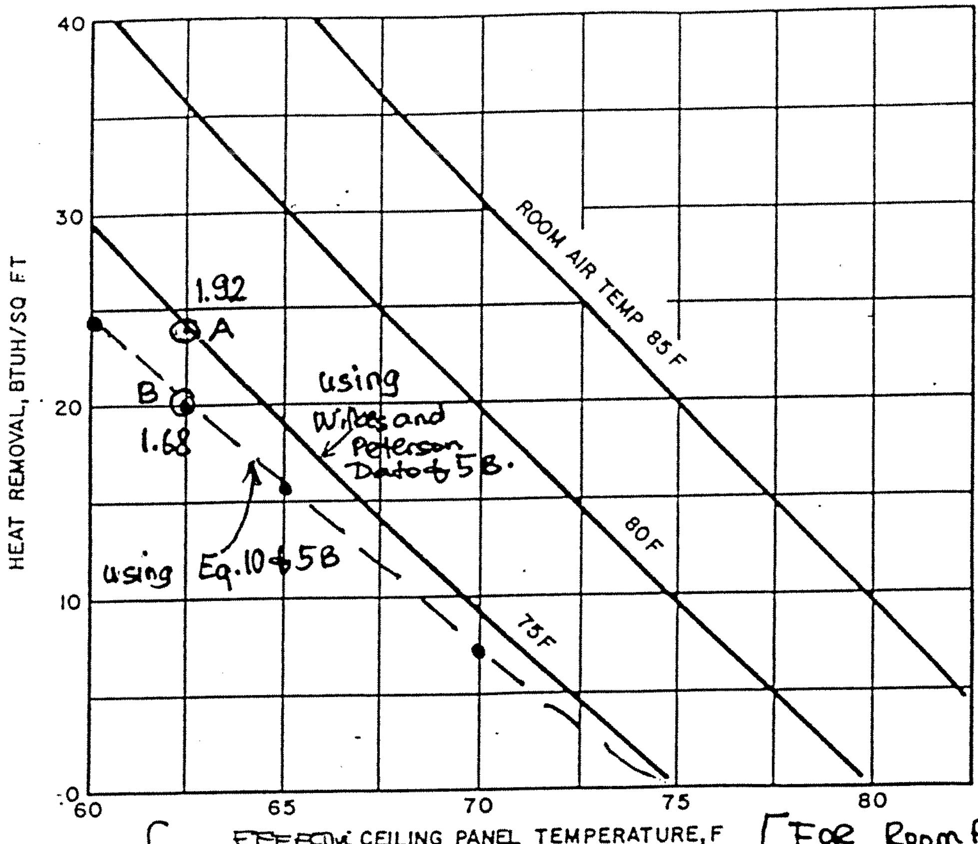
-Figure 9 should bear additional sub-titles to clarify the matters. See attached figures.

Mr Sensecall points out in his letter that Figure 9 now yields outputs in a range lower than the original IP version. In original IP version, Wilkes and Peterson data actually yields intensities around 2 Btu/sq.ft.h.deg.F (See point A yielding 1.96 in IP version of attached Figure). Using Equation 10,however this value is less,like 1.68 as Point B shows on the same figure. l am sure that as long as Equation 10 is honored,the latter number will be common in practice (i.e.; 1.7 Btu/sq.ft.h.deg.F or 9 to 10 W/sq.m.deg.C. instead of 2 and 11-11.5 respectively.

I do hope that the above information will be lpful,and looking forward to meet you in Chicago.

Prof.Dr.Birol KILKIS

:Mr.R.A.Parsons.



[For Room Air Temp = . AUST]

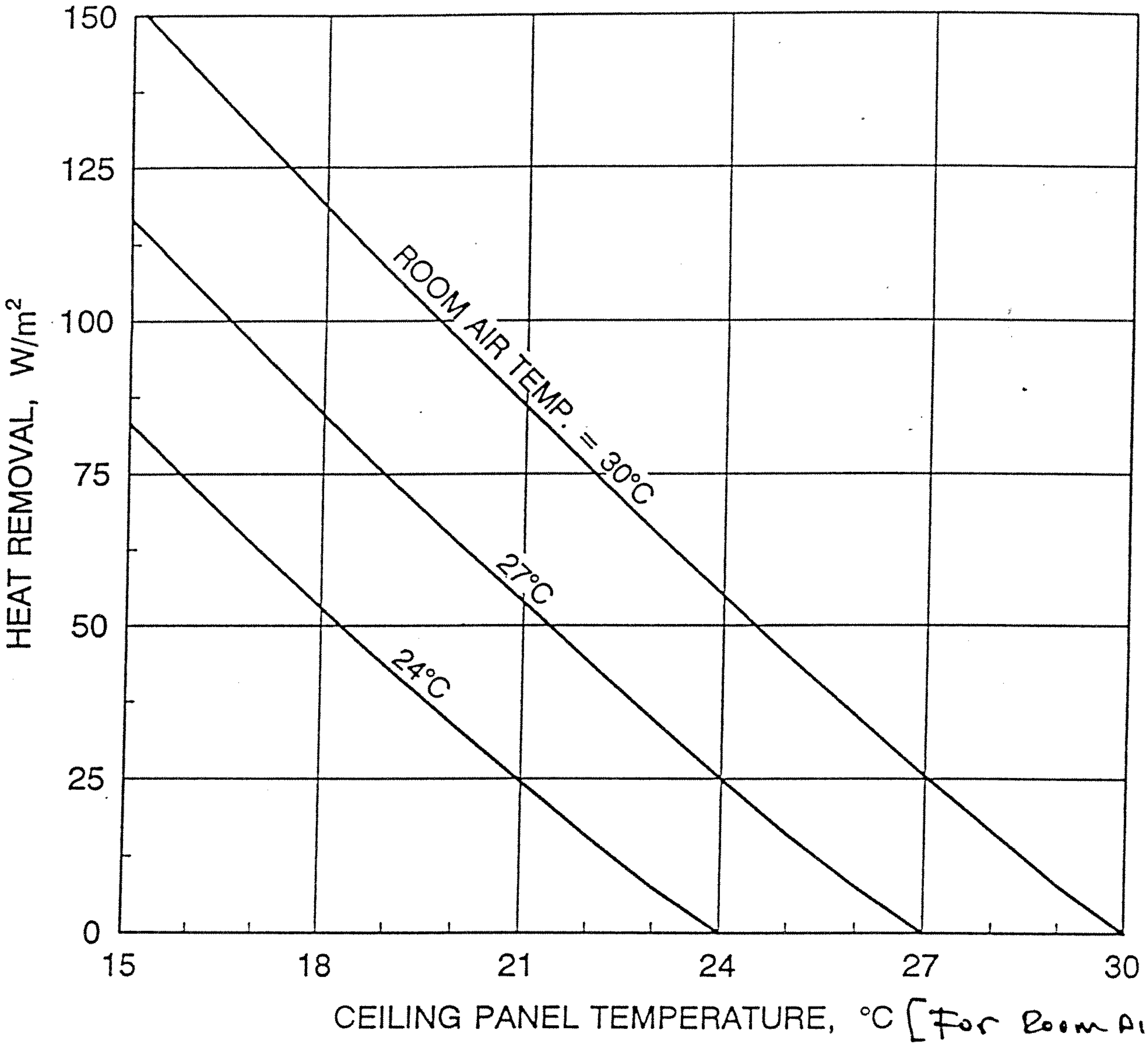
...se are my ...ggestions for the ...rt run. *

(After Wilkes and Peterson Data for heat removal by natural convection with D = 5½, 2 in to panel)

An International Organization

50%

6



[For Room Air Temp.= AUST]

* → (After Wilkes and Peterson Data for heat removal by natural convection with D= 1.68 m, 50mm to panel)

F9 SI
Ch 6
92 S

ASHRAE American Society of Heating, Refrigerating and Air-Conditioning Engineer
1791 Tullie Circle, NE • Atlanta, Georgia 30329 ☎404-636-8400 • Tlx 705343 ASHRAE • Fax 404-321-547

Robert A. Parsons
Editor
ASHRAE Handbook

30 July 1992

Norman Buckley
Buckley Associates
7700 Briarwood Drive
Crestwood, KY 40014

Dear Norm:

Here is a letter from Mr. Sensecall that questions the accuracy of Figure 9 in the panel heatin and cooling chapter 6 in the SI edition of the 1992 Handbook. I simply added Equations (5B) and (10) to generate the graph. According to the text, these equations were used to generate Figures 2 and 4 for natural convection.

So I checked the I-P version of Figure 9. I calculated equations (5B) and (10) for t_p = 60°F ar t_a = AUST = 75°F (Δt = 15°F). I got q_r = 13.2 Btu/h·ft^2 and q_c = 11.1 Btu/h·ft^2. These numbers seem to agree with the values shown in Figures 2 and 4. But the combined heat transfer value of 13.2 + 11.1 = 24.3 does not agree with the value of about 29 shown in Figur 9. In other words, is the statement in the next to the last paragraph on page 6.4 correct? It states, "Figure 9 shows the combined radiation and convection transfer for cooling, as given in Figures 2 and 4." Or does figure 9 include some additional correction factors?

I look forward to your clarification. Also thanks again for the SI corrections from Dr. Kilkis. I sent him a copy of this letter in case he would like to check Figure 9 as well.

Sincerely,

Robert A. Parsons

Robert A. Parsons

cc: B. Kilkis

P.S. I did use the correct coeff (4.93) for Eqn in the SI figure

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FAX MESSAGE

To: A.S.H.R.A.E.

Fax No: 0101 404 321 5478

From: Mr Douglas B Sensecall

Membership No: 2000621

Date: 17 July 1992

Re: A.S.H.R.A.E. HANDBOOK 1992 - S.I. EDITION - TECHNICAL QUERY

Have recently received my copy - many thanks for excellent reading.

I do observe however, that one subject area of specific interest to me has changed for the worse.

On page 6.6 figure 9, the graph is drawn for a lower output of about 9 or 10 watts/m² per degree Celcius - whereas all previous issues were drawn for about 11-11.5 watts/m² per degree Celcius (2 btu/ft² per degree Farenheit).

Since I do not have sight of your U.S. Imperial Units version, I would be most pleased if you could fax a copy of the same graph from the 1992 non S.I. volume.

All previous issues on my shelf are of course in Imperial units and will thus be able to make a direct comparison.

Regards

Pub Sales JUL 17 1992

ASHRAE American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

1791 Tullie Circle, NE • Atlanta, Georgia 30329-2305 ☎ 404-636-8400 • Fax 404-321-5478

June 2, 1992

FAX TO: Mr. Douglas B. Sensecall

FROM: Robert A. Parsons, Handbook Editor Robert A. Parsons

Here is the I-P version of Figure 9 on page 6.6 in the 1992 ASHRAE Handbook. After reviewing the two versions, I am concerned that the SI figure may be incorrect. I will ask the technical committee that prepared that chapter to check.

Thanks for bringing the problem to my attention.

Appendix 6. Typical Examples of Design Charts from Literature and Manufacturers

AUTHOR'S NOTE:

The next two pages are from a design manual for a specific product and panel type. This manual refers to ASHRAE Handbook and gives some figures for the floor R values. These R values are used to approximate the supply water temperature depending upon the required floor heat output. It is interesting that neither R-Value Table nor the Output Chart considers the effect of hose spacing at all. Instead, recommended spacings are given in zone size chart. No adjustments for indoor air temperatures different than 70°F and AUST are available.

ʼstem Sizing

Heat Loss Calculations

STEP **1** **(If designing for floor warming only, proceed to Step 3.)** Use any one of the several industry standard heat-loss calculation guides (e.g. I = B = R Heat-Loss Calculation Guide). A sample heat loss can be found on page 14. Pay special attention to rooms with high heat-loss potentials. When properly insulated, heat loss at the edge of the slab, or downward loss is not significant in slab-on-grade applications. In general, good insulating techniques are a major factor contributing to the success of an Infloor Heating System.

Required Floor Output

STEP **2** The heat lost from a room must be replaced by heat from the floor. The total heat loss divided by the *adjusted gross floor area* (do not include cabinet areas or other areas that do not generate heat) will produce the required floor output in BTUs/ square foot.

Floor Coverings

STEP **3** Floor coverings have a pronounced affect on the performance of radiant floors due to their insulative qualities. Add the R-value of the desired floor covering to that of the carpet pad (if used). The lower the R-value, the better suited for radiant floors. Floor coverings will determine the actual floor output.

ɹe Table

Of Various Floor Coverings

ue	Floor Covering	Tufts/ Sq. In.	Depth (Inches)
20	Bare floor	—	—
21	Linoleum or vinyl sheet goods	—	—
22	Ceramic tile	—	—
54	Hardwood	—	3/8
55	Nylon level loop	86	1/8
55	Nylon level loop	48	1/8
57	Nylon level loop	67	3/16
58	Nylon level loop	80	1/8
78	Acrylic level loop	80	3/16
93	Hardwood	—	3/4
95	Polyester plush	54	1/4
03	Acrylic level loop with foam back	80	1/4
2	Nylon plush	88	1/4
13	Nylon high low tip sheared	55	varies
51	Nylon shag	28	1
56	Polyester high low tip sheared	54	varies
71	Acrylic plush	44	1/2
13	Nylon plush	80	7/8
10	Acrylic plush	58	11/16
16	Nylon saxony	29	9/16
9	Wool plush	45	1/2
6	Nylon shag	22	1 1/4

\+

R-Value	Carpet Pad Underlayments	Depth (Inches)
0.31	Infloor Carpet Cushion	1/4
0.62	Slab rubber	—
0.78	Waffled sponge rubber 48 oz.	—
1.61	Prime urethane 2.2 lbs. density	3/8
1.71	Coated combined hair and jute 56 oz.	—
2.09	Bonded urethane 4 lbs. density	1/2
2.15	Prime urethane 2.2 lbs. density	1/2

Total R-Value____________________

Values obtained from the Carpet and Rug Institute

Zone Size

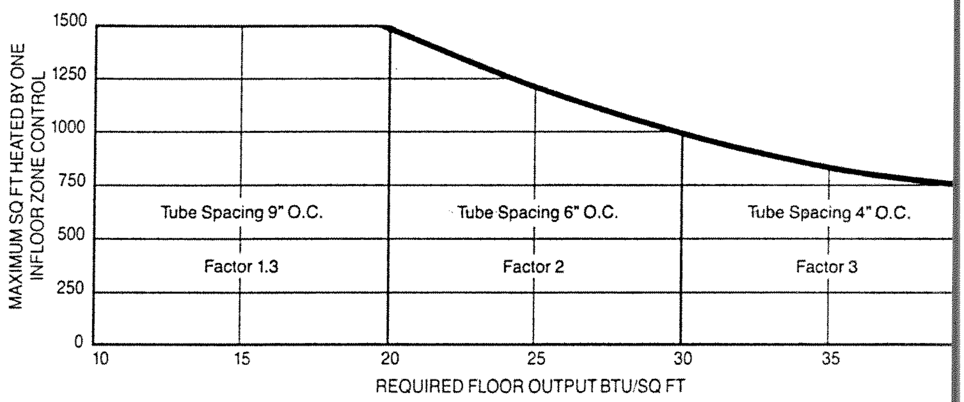
STEP 4 The Zone Size Chart will indicate the maximum square footage that can be serviced by one Zone Control. A zone may be any size up to this limit.

The chart also gives the recommended tube spacing based on heat loss. Tube spacing of 6 inches is most commonly used to achieve even floor surface temperatures. (When designing for floor warming only, plan tube spacing 6 inches on center and a floor output of 20 BTUs/square foot.)

The amount of tube required for a zone is calculated by multiplying the actual tube coverage area by the "factor" listed on the chart with the tube spacing.

Actual tube coverage area is the adjusted gross floor area minus the area six inches from all walls and cabinets.

Zone Size Chart



1. Based on your heat-loss calculation and floor covering considerations, locate the required floor output on the bottom of the chart.
2. Extend a line vertically until it intercepts the top line of the chart.
3. Read the tube spacing and the conversion factor from the area through which your line passed.
4. Draw a line horizontally from where your first line intersected the top of the chart to the left side of the chart and rea the maximum floor area one Infloor Zone Control can heat.

Actual Floor Output

STEP 5 The secondary loop water temperature is limited to 140°F. Generally, 85°F is accepted as the maximum surface temperature for comfort where there is prolonged foot contact with the floor. Perimeter areas outside normal traffic areas can operate at higher surface temperatures.

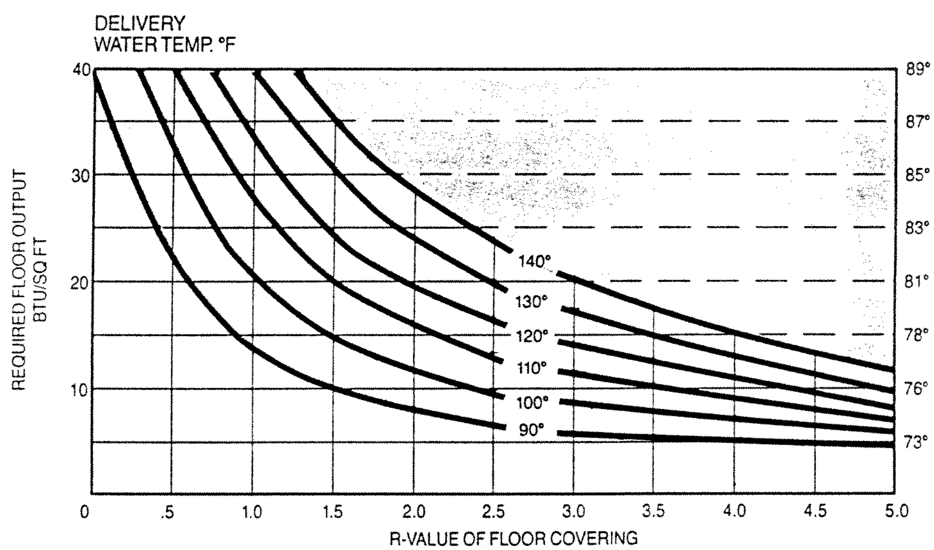
With these limits in mind, find the actual floor output from the chart.

Supplemental heat, if needed, may be used to make up the difference between the required floor output and the actual floor output. (See system diagrams on page 15).

Varying floor outputs can be averaged in a room to achieve a higher overall BTU/square foot output. High output tile floors under windows may be used to compensate for low output carpeted floor areas. (When designing for floor warming only, plan a floor output of 20 BTUs/square foot.)

Output Chart

Based on 70°F inside air temperature



1. Find the Required Floor Output on the left side of the chart and extend a line to the right to read the Floor Surface Temp
2. Locate your selected floor covering R-value on the bottom of the chart. (R-value from R-value Table) Extend a line u cross your first line.
3. Read the Delivery Water Temperature at the intersection of the two lines.
4. If the lines intersect beyond the 140°F line, do one or more of the following:
   a. Select a floor covering with a lower R-value.
   b. Reduce the heat loss of the area to lower the Required Floor Output.
   c. Figure supplemental fin tube baseboard heating.
      — Extend a line up from your Floor Covering R-value until it intersects the 140° line.
      — From that point, extend the line to the left to find the actual output of the floor.
      — Subtract the actual output from your required output to find the amount of supplemental heat needed.

*For further information, see **1987 ASHRAE Handbook, HVAC Systems and Applications,** chapter 7.*

HORS NOTE:

following page from a design manual gives a simple chart to ermine the tube spacing with respect to tube size and the required t output intensity. There is not any adjustment for panel R value. claims that with high R value coverings,the floor surface perature will be more uniform. This is true and obvious from the analytical algorithm (see Equation 15,Section 1). Figure 6-1 ws a plot which exhibits this trend. It shows the significance of n T_p. This effect decreases with t_w (heating load). It further ws that the significance diminishes with M and the family of ves become flat beyond 12" O.C.

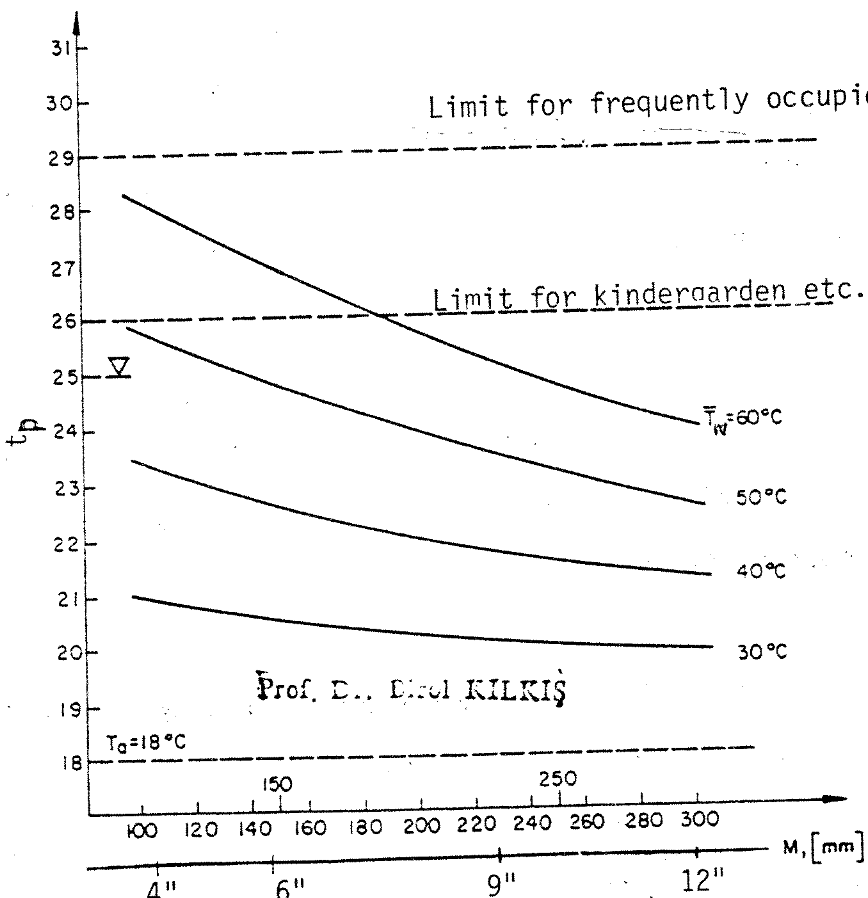


Fig.6-1. Significance of Hose spacing on t_p.

This manual further concludes that wider hose spacings may be used under these circumstances. This is not so,because any high R value increases the temperature drop across the floor and this offsets the former effect. Otherwise to add heavy carpeting would save hose material. Consequently,either the mean water temperature should be increased and/or tube spacing(M) be decreased for the same heat output intensity.

It further states that "heat <u>output</u> per foot of tube increases with spacing while floor output decreases". The latter part is obviously correct (see Figure 6-2),but let us analyze the first statement by challenging the new algorithm:

The heat that can be actually delivered per ft² is proportional to the following term (see Equation 13 in Section 1):

$$q_{delivered} \quad \alpha \quad \frac{(2 \cdot W \cdot \eta + D_o)}{M}$$

Then

$q_{delivered}/\text{ft of hose} = q_{delivered}/(\approx A_p/M)$

Therefore;

$$q_{delivered}/\text{ft of hose} \quad \alpha \quad \frac{(2 \cdot W \cdot \eta + D_o)}{M \cdot (A_p/M)}$$

Here A_p is the panel area (ft²). For the unit panel area it is 1 ft². M on right hand side of the equation cancels.

This yields the following equation:

$$q_{delivered}/\text{ft of hose} \quad \alpha' \quad (2 \cdot W \cdot \eta + D_o)$$

For a given tube diameter D_o,heat delivered by each foot of tube will be a function of the fin efficiency η, where η is an inverse function of W (net hose spacing). Referring to Figure A-2 in Appendix 2, there two distinct regions. For the region A the product W·∩ actually reases with decreasing W for a given fin coefficient. W is the net spacing between tubes and is a function of M. refore in region A, q delivered per ft of hose increases. But this s not hold in region B where the curve starts to become flat. this region fin efficiency is approximated by 1/(m·W) very sely. Replacing the above equation with this term gives:

livered/ft of hose α' $(2/m + D_o)$

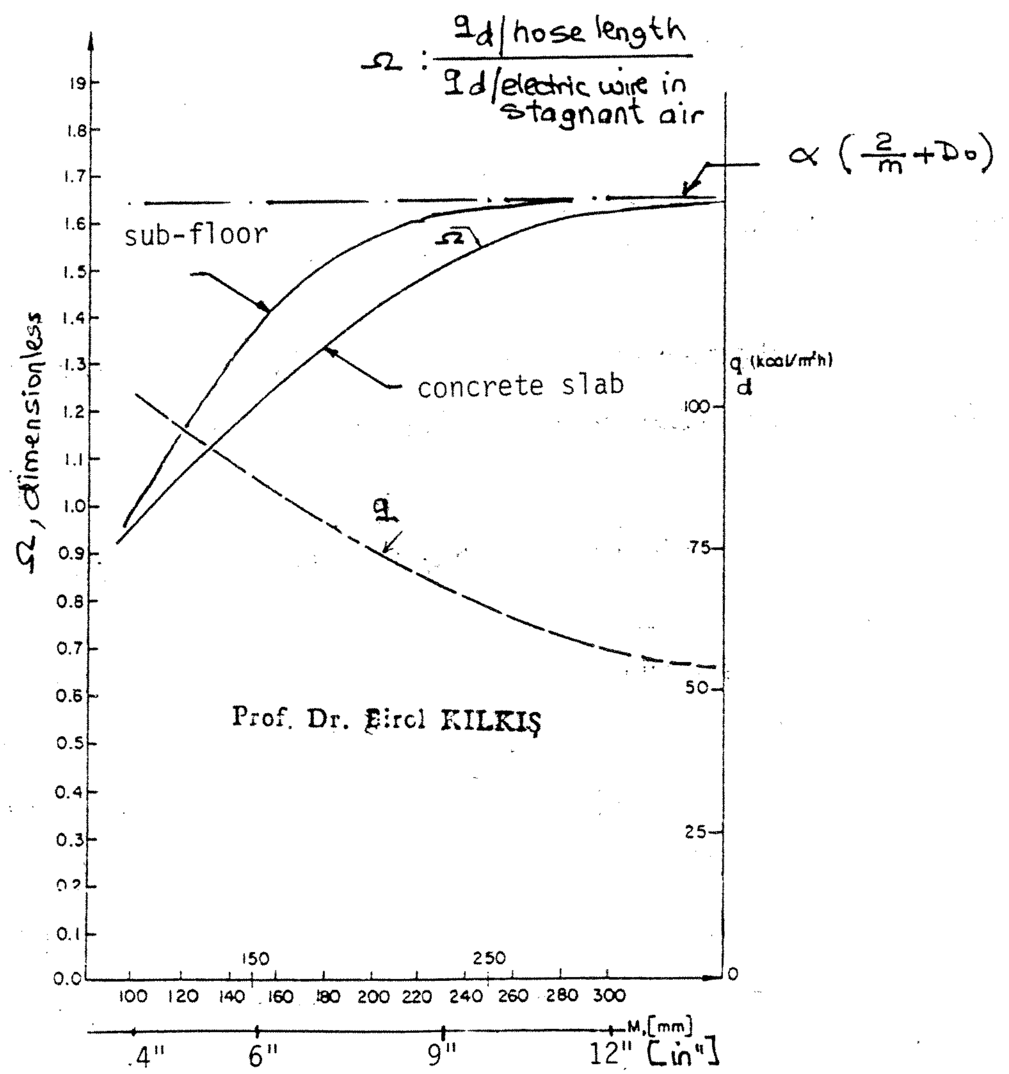


Fig.6- 2. Significance of Hose Spacing on Heat Output

Therefore if $m \cdot W$ is greater than 2, the heat delivered per ft of hose is practically constant. There may be many design cases in region B: A joisted sub-floor m:5 ft^{-1},D_o:3/8" (0.031 ft),M:12"

Figure 6-2 shows the significance of M on q_u and the heat delivered per foot of hose in a dimensionless form. Here Ω is the ratio of the heat delivered per foot of hose to the heat output of a metal wire with the same length suspended in the stagnant air at ambient conditions. This curve indicates that Ω approaches the asymptote given by the proportion mentioned above. A joisted sub-floor panel reaches this asymptote earlier.

Therefore,one can not generalize the statement that the heat delivered per foot of hose increases for every case of floor heating. Especially in plywood sub floors this condition will not hold for most of the practical design cases.

The same catalog also argues that the significance of hose spacing diminishes with the R value of floor coverings. This is in principle true but the rate of diminishing is not that high. Again this depends where the actual design is on Figure A-2 in Appendix 2 and what the ratio r_c/r_p is. Also a high R value on the floor generally means a higher back and edge heat loss (q_b). This will increase the temperature drop across the tube wall (see Equation A-6,Appendix 2) and this will add on to the mean water temperature required.

g. Determine the tube size to be used.

1. Use the size tubing provided by the manufacturer whose system you choose to install. If the manufacturer supplies more than one size of tubing, select the size to use based on the following criteria:

   a. If more than one size tubing will work for the type of projects you typically install, do a cost comparison between them. Estimate the total amount of tubing, manifolds, and labor for the job with each tubing size. Use the tubing with the lowest overall installed cost.

   b. Zone size: The larger the tubing diameter, the larger the heating zone it can serve. Follow the manufacturer's recommendations.

   c. Slab thickness: In slab-on-grade, any size tubing can be used because the mass is thick enough over the tubing. With topping slabs (over slab-on-grade or suspended wood floor), tube size may be important. The smaller the tube size, the thinner the topping slab can be.

2. Choose one tubing size that works for your jobs and stick with it as much as possible. Facilitates design and installation.

h. Determine tube spacing.

From Chart 3.

1. Once the required floor output has been determined and the tubing size selected, use the Chart to determine the maximum tube spacing that will provide the required output.

CHART 3 : TUBE SPACING (o.c.)

REQUIRED FLOOR OUTPUT (BtuH/sq.ft.)	TUBE DIAMETER (I.D.) 1/4"	3/8"	1/2"	5/8"	3/4"
10-20		9"	15"	15"	15"
20-30		6"	12"	12"	12"
30-40		4"	9"	9"	9"

2. Using the maximum tube spacing minimizes material and labor costs.

3. If thin topping slab, use 9" o.c. or greater tube spacing only under wood floor or carpet. High R-value of wood and carpet result in greater lateral heat movement, so can get even floor surface temperature with wider spacing than if bare floor, vinyl, or tile is used. In latter case, 9" o.c. spacing will result in noticeable floor surface temperature variation.

4. In general, tube spacing under carpet can be greater, as floor coverings reduce the significance of spacing.

5. Tube spacing most important in room with high heat loss and bare floor.

6. Heat output per foot of tube increases with spacing, while floor output per sq. ft. decreases.

AUTHOR'S NOTE:

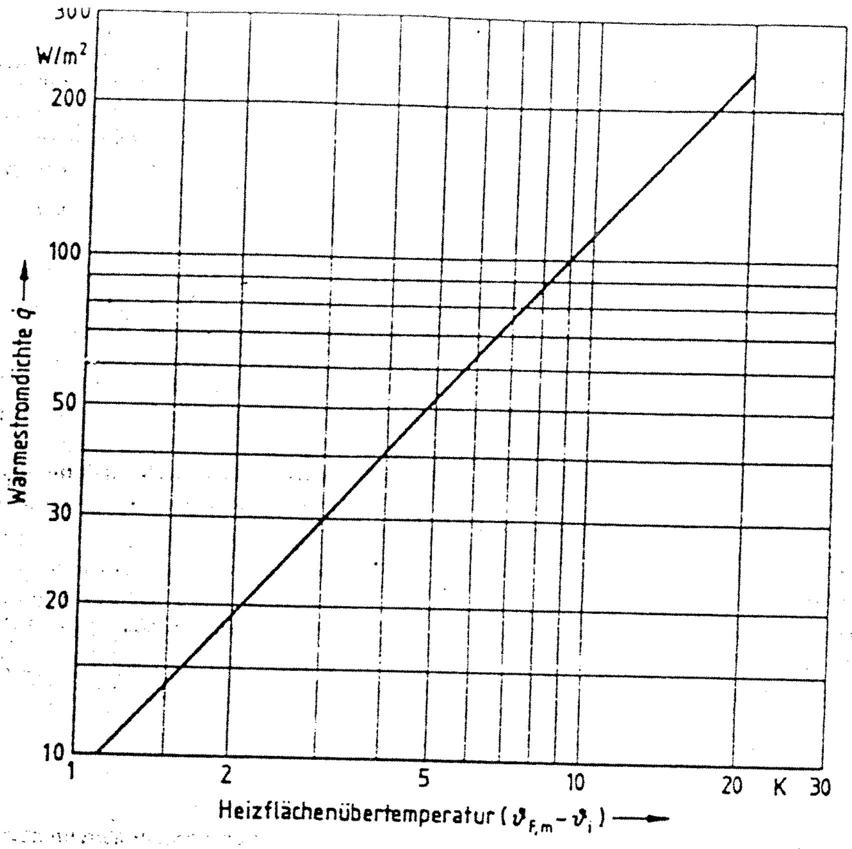
The following page gives the design mean water temperature in terms of required floor heat output and R-value of the floor covering. It also gives the floor surface temperature. It is quite interesting that t_w is not related to M at all. The floor temperature listed does not have any correction with respect to t_a or AUST either. r_c value is not related to other panel thermal resistances too.

CHART 2: DESIGN WATER TEMPERATURE

FLOOR OUTPUT (BtuH/sq.ft.)	FLOOR SURFACE TEMP. (°F)	DESIGN WATER TEMPERATURE									
	90										
40	89	110	130	149							
	88										
35	87	103	122	140	152						
	86										
30	85	97	114	130	143	156					
	84										
25	83	93	106	121	132	142	154				
	82										
20	81	90	99	110	121	130	140	150	160		
	80										
	79										
15	78	90	92	100	107	116	123	130	140	149	
	77										
10	76	90	90	90	95	101	105	110	115	121	131
	75										
	74										
5	73	90	90	90	90	90	90	90	90	90	90
	72										
		0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0

R-VALUE OF FLOOR COVERING

1. Find the Required Floor Surface Temperature on the left side of the chart. Extend a line to the right to the Design Water Temperature.
2. Find the Floor Covering R-Value on the bottom of the chart. Extend a line up to cross the first line.
3. Find the Design Water Temperature at the intersection of the two lines.



ϑ_i Norm-Innentemperatur in °C

$\vartheta_{F,m}$ mittlere Fußbodentemperatur in °C

$\dot{q}$ Wärmestromdichte in W/m²

$$\dot{q} = 8{,}92\ (\vartheta_{F,m} - \vartheta_i)^{1,1}$$

Bild 1. Basiskennlinie

This is the analytical algorithm from DIN Standard.

5 Kennlinienfeld und Norm-Wärmestromdichte $\dot{q}_N$

Zur Darstellung des Prüfergebnisses und als Arbeitsdiagramm für die Auslegung dient das Kennlinienfeld (siehe z. B. Bild 2). Jedes System und jede Rohrteilung hat ein eigenes Kennlinienfeld, das für den jeweiligen flächenbezogenen Norm-Heizmittelstrom $\frac{(\dot{m}_N)}{A}$ (Norm-Spreizung sp_N = 10 K) aufgenommen wird, jedoch über diesen hinaus für den im Kennlinienfeld angegebenen Bereich gültig ist [1]).

Die systemeigene Norm-Wärmestromdichte $\dot{q}_N$ wird bestimmt durch die erreichte mittlere Heizflächentemperatur $\vartheta_{F,m}$ bei maximaler Fußbodenoberflächentemperatur $\vartheta_{F,max}$ = 29 °C.

Die mittlere Heizflächentemperatur $\vartheta_{F,m}$ ist wegen der Verlegung der Rohre mit endlichem Abstand, der Temperaturspreizung zwischen Vor- und Rücklauf sowie des örtlich unterschiedlichen Wärmeübergangs kleiner als $\vartheta_{F,max}$ und somit ist auch $\dot{q}_N < \dot{q}_B$.

Die Norm-Wärmestromdichte und die Norm-Heizmittelübertemperatur ergeben sich für $R_{\lambda,B}$ = 0 und $(\vartheta_{F,max} - \vartheta_i)$ = 9 K aus dem Kennlinienfeld wie im Teil 1 dieser Norm, Abschnitt 8.4 beschrieben.

6 Auslegungs-Wärmestromdichte und Auslegungs-Heizmittelübertemperatur

Die Auslegungs-Wärmestromdichte $\dot{q}$ und die Auslegungs-Heizmittelübertemperatur $\Delta\vartheta$ lassen sich unter Beachtung der maximal zulässigen Heizflächentemperatur $\vartheta_{F,max}$ und des jeweiligen Wärmeleitwiderstands des Fußbodenbelages $R_{\lambda,B}$ für den im Kennlinienfeld angegebenen Bereich der flächenbezogenen Heizmittelströme $\left(\frac{\dot{m}}{A}\right)$ aus dem Kennlinienfeld entnehmen. Ansonsten ist wie in Abschnitt 7.2 beschrieben vorzugehen. Für die Heizmittelübertemperatur gilt

$$\Delta\vartheta = (\vartheta_V - \vartheta_R)/\left(\ln \frac{\vartheta_V - \vartheta_i}{\vartheta_R - \vartheta_i}\right) \qquad (3)$$

1) Gleiches gilt für ein Kennlinienfeld, das mit einem flächenbezogenen Heizmittelstrom $\frac{\dot{m}_{5K}}{A}$ entsprechend 5 K Spreizung aufgenommen wird.

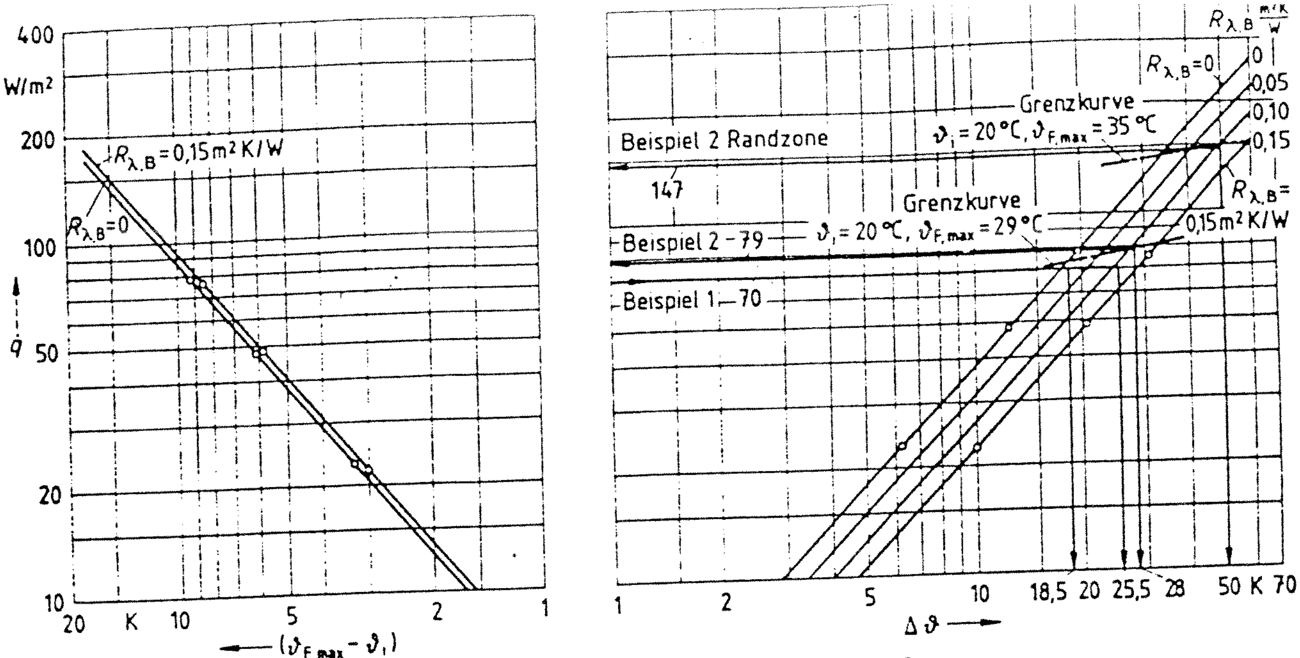


Bild 2. Kennlinienfeld gültig für (0,5 bis 2,0) · $(\dot{m}_N/A)$ mit $(\dot{m}_N/A)$ = 6,55 kg/hm².

Beispiel 1: A_F = 30 m², $\dot{Q}_N^*$ = 2100 W, ϑ_i = 20 °C,
erforderliche Wärmestromdichte $\dot{q}$ = 2100 W/30 m² = 70 W/m²

a) Fußbodenbelag Teppich mit $R_{\lambda,B}$ = 0,1 m² K/W. Da die Auslegungs-Wärmestromdichte $\dot{q}$ = 70 W/m² unter der Grenzkurve ϑ_i = 20 °C und $\vartheta_{F,max}$ = 29 °C liegt, wird die maximal zulässige Fußbodentemperatur nicht überschritten (siehe Bild 2).

   Für die Heizmittelübertemperatur ergibt sich $\Delta\vartheta$ = 25,5 K.

b) Fußbodenbelag aus PVC mit $R_{\lambda,B}$ = 0,01 m² K/W. Auch hier liegt die Auslegungs-Wärmestromdichte von $\dot{q}$ = 70 W/m² unterhalb der Grenzkurve.

   Für die Heizmittelübertemperatur ergibt sich $\Delta\vartheta$ = 18,5 K.

Beispiel 2: A_F = 30 m², $\dot{Q}_N^*$ = 2850 W, ϑ_i = 20 °C,

$$\text{erforderliche Wärmestromdichte } \dot{q} = \frac{\dot{Q}_N}{A_F} = 95 \text{ W/m}^2$$

$R_{\lambda,B}$ = 0,1 m² K/W.

Die Wärmestromdichte $\dot{q}$ = 95 W/m² ist nicht zulässig, da der Wert oberhalb der Grenzkurve liegt.

Die hier maximal zulässige Wärmestromdichte $\dot{q}$ = 79 W/m² wird bei $\Delta\vartheta$ = 28 K erbracht (siehe Bild 2).

Als Abhilfe muß eine Zusatzheizung mit

$$\dot{Q}_Z = \dot{Q}_N - \dot{q} \cdot A_F = 2850 \text{ W} - 79 \text{ W/m}^2 \cdot 30 \text{ m}^2 = 480 \text{ W}$$

oder eine stärker beheizte Randzone gleicher zusätzlicher Leistung vorgesehen werden.

Die maximal zulässige Wärmestromdichte für die Randzone ergibt sich für $\vartheta_{F,max}$ = 35 °C und eine Heizmittelübertemperatur $\Delta\vartheta_R$ = 50 K zu $\dot{q}_R$ = 147 W/m² (siehe Bild 2). Der von der gesamten Heizfläche A_F = 30 m² als stärker beheizte Randzone auszuführende Teil ergibt sich zu

$$A_R = \frac{\dot{Q}_N^* - A_F \cdot \dot{q}}{\dot{q}_R - \dot{q}} = \frac{2850 \text{ W} - 30 \text{ m}^2 \cdot 79 \text{ W/m}^2}{(147 - 79) \text{ W/m}^2} = 7{,}06 \text{ m}^2.$$

7 Wärmestrom des beheizten Fußbodens nach unten

Zur Reduzierung des Wärmestroms durch die Decke an darunter befindliche Räume müssen an den Wärmeleitwiderstand des Fußbodenbelags sowie an die Wärmedämmung unterhalb der Heizebene bestimmte Anforderungen gestellt werden.

Der Wärmeleitwiderstand des Fußbodenbelags darf nicht größer als $R_{\lambda,B}$ = 0,15 m² K/W sein.

Bei der Berechnung des Wärmestroms nach unten sind folgende Möglichkeiten zu unterscheiden.

7.1 Wärmestrom an darunter befindliche, gleichartig beheizte Räume

Für den Fall, daß nichts anderes vereinbart wird, ist eine Wärmedämmung von R_λ = 0,75 m² K/W unter der Heizebene einzubringen. Andernfalls ist nachzuweisen, daß der berechnete Wärmestrom nach unten nicht größer als 10% des Wärmestroms nach oben ist.

Die Berechnung ist ohne Fußbodenbelag wie nachfolgend beschrieben durchzuführen.

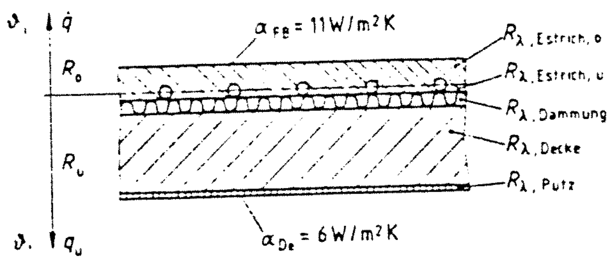


Bild 3.

---

2) $\dot{Q}_N^*$ ist der für Fußbodenheizung bereinigte Norm-Wärmebedarf, bei dem der Wärmeverlust an darunter befindliche Räume unterhalb der Heizfläche nicht in Rechnung gestellt ist.

Der flächenbezogene Auslegungs-Heizmittelstrom $\left(\frac{\dot{m}}{A}\right)$ ist aus dem im Kennlinienfeld angegebenen Bereich zu wählen.

Die Vorlauftemperatur ϑ_V ergibt sich aus Gleichung (15)

$$\vartheta_V = \frac{e^{\frac{sp}{\Delta\vartheta}} \cdot (sp + \vartheta_i) - \vartheta_i}{e^{\frac{sp}{\Delta\vartheta}} - 1} \qquad (15),$$

oder näherungsweise aus Gleichung (15 a)

$$\vartheta_V = \vartheta_i + \Delta\vartheta + \frac{sp}{2} + \frac{sp^2}{12\,\Delta\vartheta} \qquad (15a)$$

Die Rücklauftemperatur ist $\vartheta_R = \vartheta_V - sp$.

Gleichung (15a) nach sp aufgelöst, ergibt eine Beziehung für die Spreizung bei vorgegebener Vorlauftemperatur ϑ_V.

$$sp = -3\,\Delta\vartheta + \sqrt{9\,\Delta\vartheta^2 - 12\,\Delta\vartheta \cdot (\vartheta_i + \Delta\vartheta - \vartheta_V)} \qquad (16)$$

Für den Fall einer enger belegten Randzone des gleichen Heizkreises läßt sich die Wärmestromdichte $\dot{q}_R$ der Randzone aus dem zur entsprechenden Rohrteilung gehörenden Kennlinienfeld für die gegebene Heizmittelübertemperatur $\Delta\vartheta$ entnehmen.

Der Gesamt-Heizmittelstrom in der Fußbodenheizung eines Raumes ergibt sich mit den Wärmeströmen nach oben und unten aus Gleichung (17):

$$\dot{m}_G = \frac{(A_F - A_R)\ \dot{q} + A_R\ \dot{q}_R}{c_W\ (\vartheta_V - \vartheta_R)} \left(1 + \frac{\dot{q}_u}{\dot{q}}\right) \qquad (17)$$

8.2 Berücksichtigung des Heizmittelstrom-Einflusses bei der Auslegung

Bei Auslegung mit flächenbezogenen Heizmittelströmen $\left(\frac{\dot{m}}{A}\right)$, die außerhalb des im Kennlinienfeld angegebenen Bereiches liegen, kann der Einfluß des Heizmittelstromes nicht mehr vernachlässigt werden. Für diesen Fall muß der Heizmittelstromeinfluß bei der wärmetechnischen Prüfung ermittelt werden und als Diagramm für Korrekturfaktor f_m wie in Bild 5 vorhanden sein.

Das Diagramm (Bild 5) gibt an, wie sich bei konstanter Heizmittelübertemperatur die Wärmestromdichte in Abhängigkeit des Heizmittelstroms ändert.

Die unter Berücksichtigung des Heizmittelstromeinflusses notwendige Heizmittelübertemperatur erhält man aus dem Kennlinienfeld, wenn dort der Wert für eine Wärmestromdichte gemäß

$$\dot{q} = \frac{\dot{q}}{f_m} \qquad (18$$

entnommen wird.

Zur Begrenzung der maximalen Fußbodentemperatur darf die maximale Auslegungs-Vorlauftemperatur in der Verweilfläche A_V den Wert

$$\vartheta_{V,max} = \Delta\vartheta_{max} + \frac{sp_N}{2} + \vartheta_{i,N}$$

nicht überschreiten.

Beispiel: $A_F = 20\ m^2$, $\dot{Q}_N^* = 1000$ W, $R_{\lambda,B} = 0{,}1\ m^2$ K/W
$\dot{q} = 50\ W/m^2$, $(\dot{m}_N/A) = 6{,}55\ kg/hm^2$.

Auslegung mit $\left(\frac{\dot{m}}{A}\right) = 0{,}3 \cdot \left(\frac{\dot{m}_N}{A}\right) = 1{,}97\ kg/hm^2$.

Der Korrekturfaktor f_m ergibt sich aus Bild 5 zu $f_m = 0{,}91$

Mit der umgerechneten Wärmestromdichte $\dot{q}^* = 55\ W/m^2$ erhält man aus dem Kennlinienfeld die Heizmittelübertemperatur $\Delta\vartheta = 20$ K.

Für die Spreizung ergibt sich nach Gleichung (14):

$$sp = \frac{\dot{q}}{c_W\ (\dot{m}/A)} = \frac{50\ W/m^2 \cdot J/sW \cdot 3600\ s/h}{4180\ J/kgK \cdot 1{,}97\ kg/hm^2} = 21{,}9$$

Die Vorlauftemperatur nach Gleichung (15)

$$\vartheta_V = \frac{e^{\frac{21{,}9}{20}} \cdot (21{,}9 + 20) - 20}{e^{\frac{21{,}9}{20}} - 1} = 52{,}9\ ^oC$$

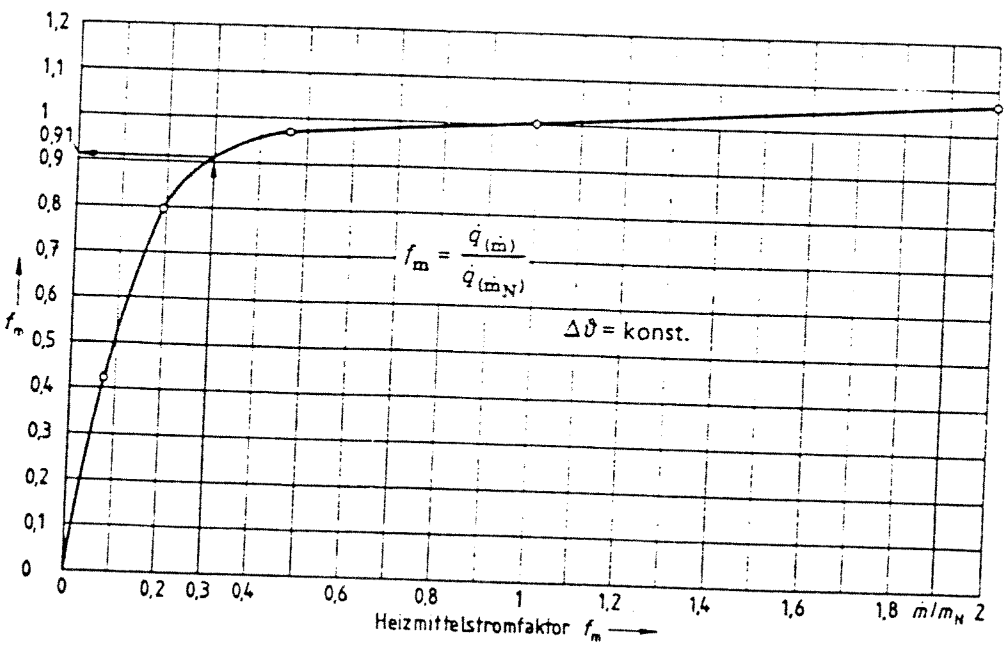
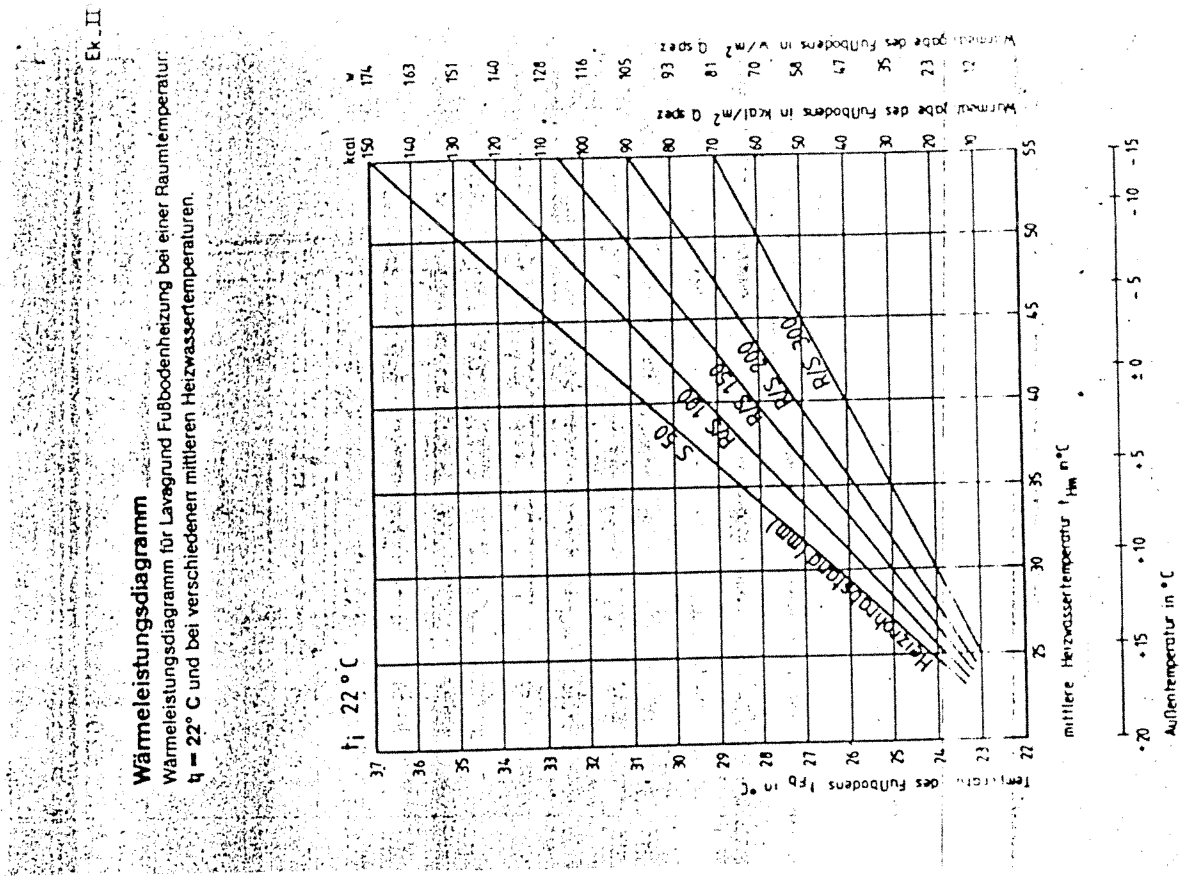
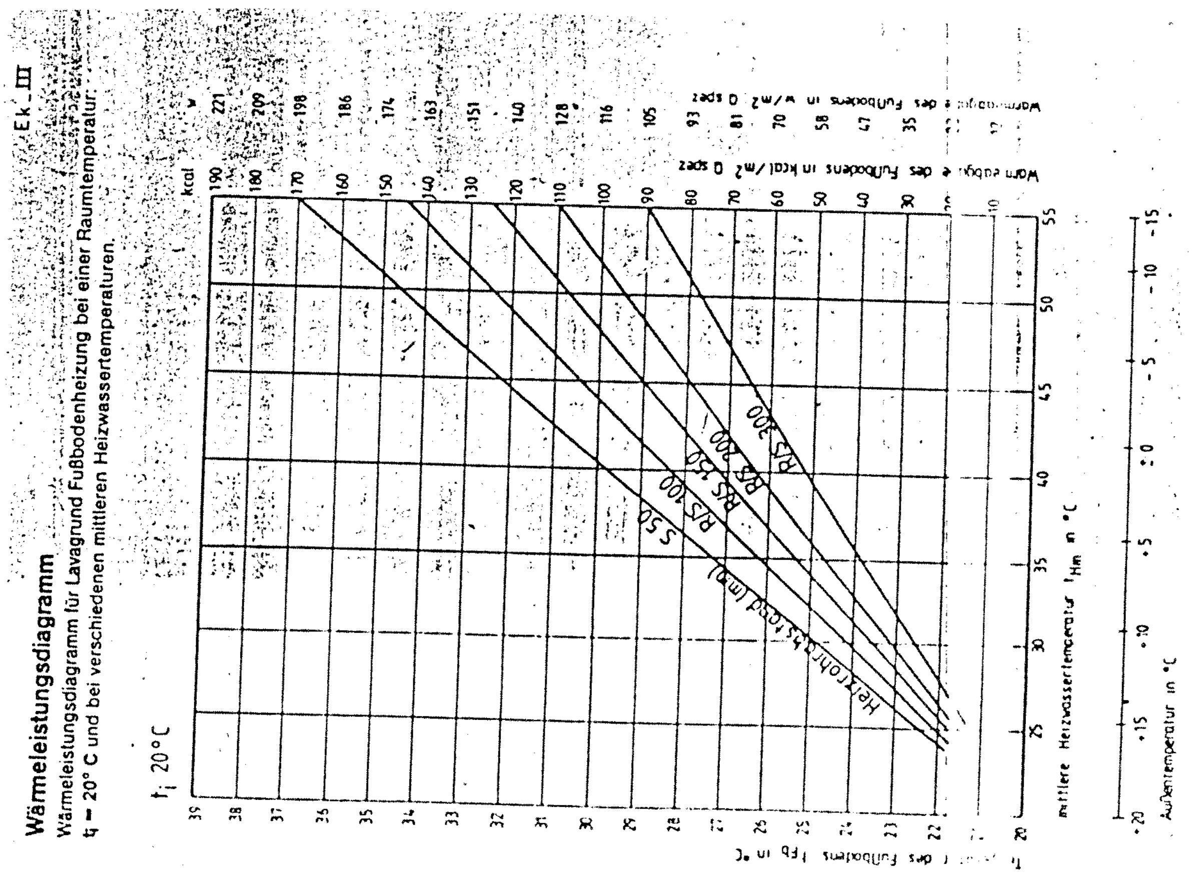


Bild 5. Heizmittelstromfaktor f_m



This is a typical nomograph from a German Manufacturer. There is a set of nomographs for different indoor air temperatures. It gives t_w for a given spacing and heating load intensity. It also gives t_p.

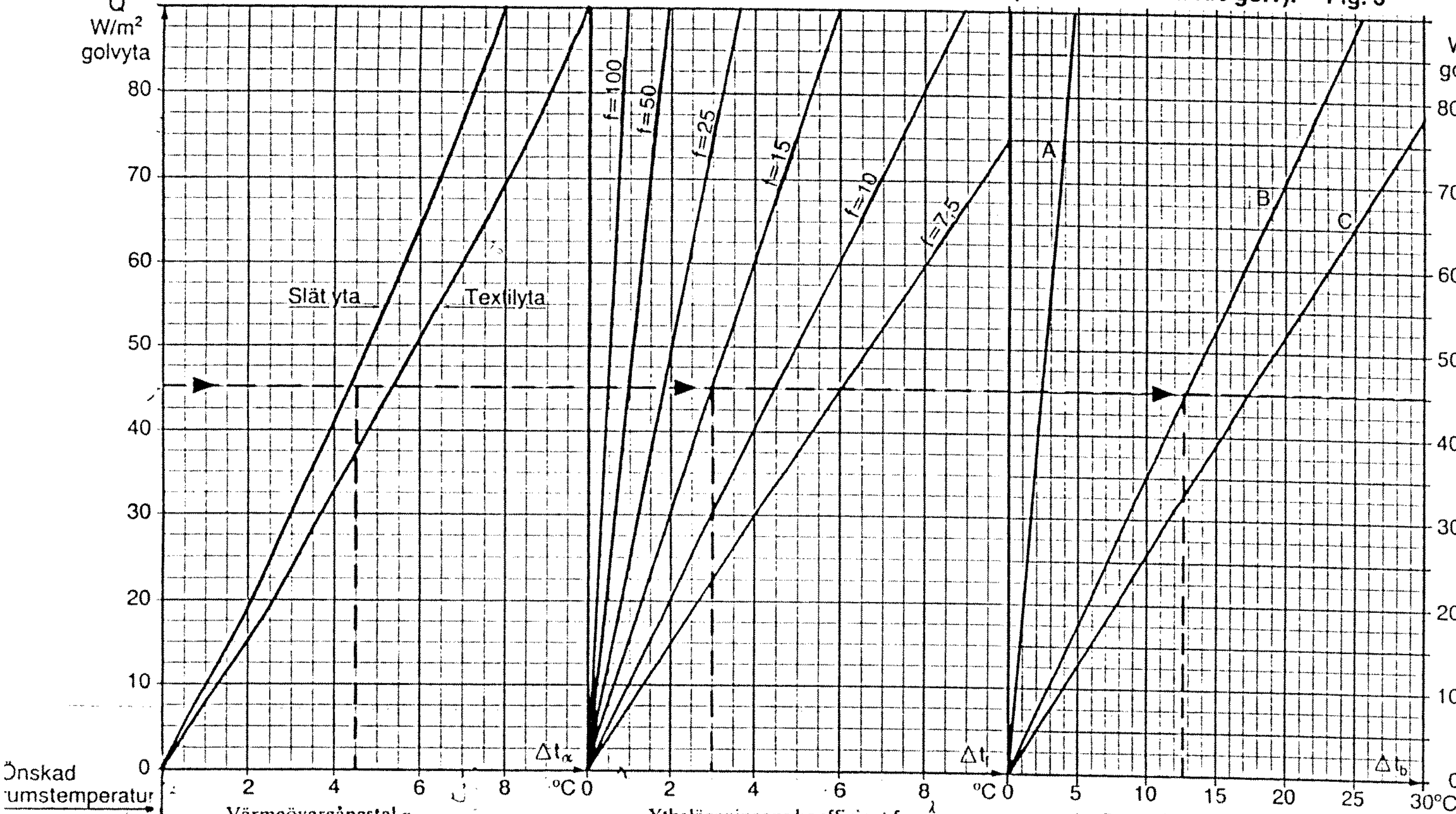
A guide about the relationship between outdoor air temperature and mean water temperature accompanies the t_w axis.



Värmeövergång från golvyta. Fig. 1

Temperaturfall gm golvbeläggning. Fig. 2

Temperaturfall i bjälklag. (Inkluderar bärande golv). Fig. 3



Värmeövergångstal α
$\Delta t\alpha = t_{golv} - t_{rum}$ °C
$Q = \alpha \cdot \Delta t\alpha$ W/m²

Ytbeläggningens koefficient $f = \frac{\lambda}{\delta}$.
λ = värmeledn.tal W/m°C
δ = tjocklek i m

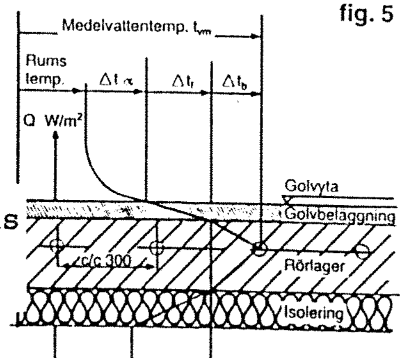
A = Betongbjälklag
B = 16 mm spånskiva med 80% al-plåt
C = 22 mm spånskiva med 80% al-plåt

Exempel (1) på framräkning av medelvattentemp.
Energibehov Q=45 W/m²
Önskad rumstemp.=20°C
Medelvattentemp. t_{vm}=t rum+$\Delta t\alpha + \Delta t_f + \Delta t_b$;
Ur diagram: $\Delta t\alpha$ =4,5°; Δt_f=3,0°; Δt_b=12,5°
· t_{vm}=20,0+4,5+3,0+12,5=40,0°C

Ex. på f-värden:

Plastmatta, 1,7 mm	f=100
Klinker, 20 mm	f= 50
Cork-o-Plast, 3,2 mm	f= 32
Nålfilt, 4,6 mm	f= 17,5
Parkett, 13 mm	f= 10
Helt.matta, 9 mm	f= 7,5

fig. 5

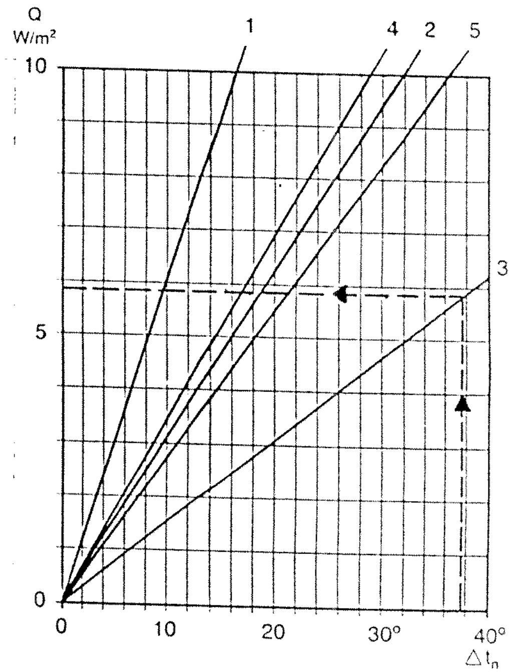


This nomograph aims to include more inputs but needs a set of graphical manipulations.

Värmeförluster nedåt genom bjälklaget vid olika isolering. Fig. 4

Bjälklagsisolering

1	Bjälklagsskiva	50 mm
2	Bjälklagsskiva	100 mm
3	Bjälklagsskiva	200 mm
4	Marksskiva	50 mm
5	Marksskiva	100 mm



Förutsättning för preliminär förlustberäkning, nedåt:
Betongbjälklag, A, antages ha samma temperatur i hela bjälklaget och bjälklag B och C samma temp. under glespanelen som på bärande golvet.
t_b=Rumstemp.+$\Delta t\alpha + \Delta t_f$
och $\Delta t_n = t_b - t_n$
t_n=temp. under bjälklaget.

Exempel på framtagning av värmeförlust nedåt.
Med temperaturer enl. exempel (1) och en temperatur under bjälklaget av −10°C blir förlusten nedåt vid 200 mm bjälklagsisolering:
Δt_n=Rumstemp + $\Delta t\alpha + \Delta t_f$ — (−10);
Δt_n=20+4,5+3,0 − (−10)=20+4,5+3,0+10=37,5°C
Ur diagram för förluster nedåt och kurva 3 får vi för

STUNGSTABELLEN

de Heizungsanlagen (Altanla- für eine maximale Vorlauf- ır von 70° C ausgelegt sind, irekt mit dem ›System 70‹ oder ergänzt werden. achrüstung von Fußboden- n kann ein einzelner Heizkreis 3ad) auch unmittelbar am ıen Heizkörper mit ange- werden.

Für die wärmetechnische Auslegung des Systems gilt ansonsten die gleiche Vorgehensweise wie bei Neuanlagen. Bei der Erweiterung bestehender Anlagen ist darauf zu achten, daß die Auslegung und der Vordruck des Ausdehnungsgefäßes nach DIN 4807 (E) unter den veränderten Bedingungen (größeres Gesamtwasservolumem) nochmals überprüft und nötigenfalls korrigiert wird.

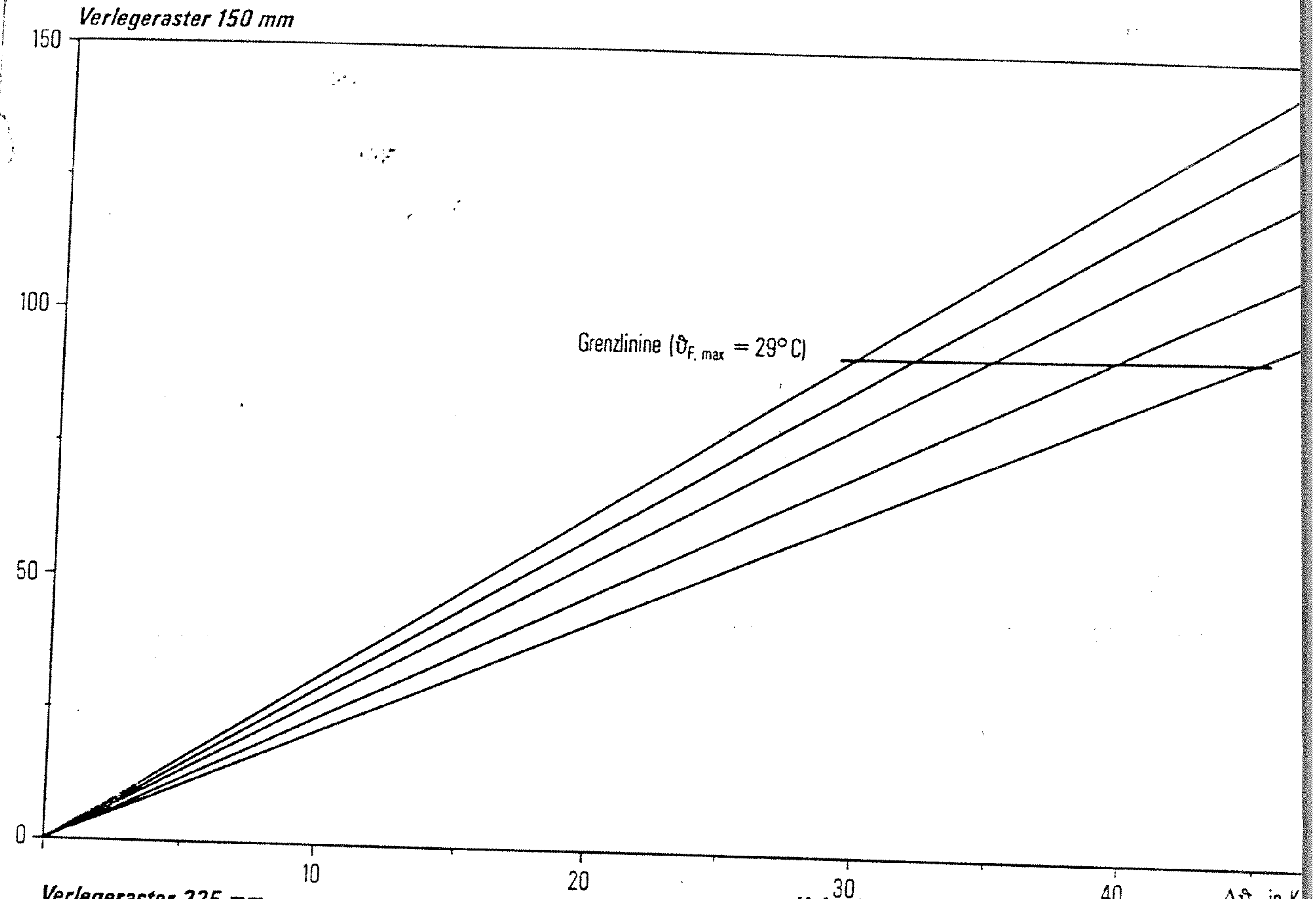
ıstabelle Randzone

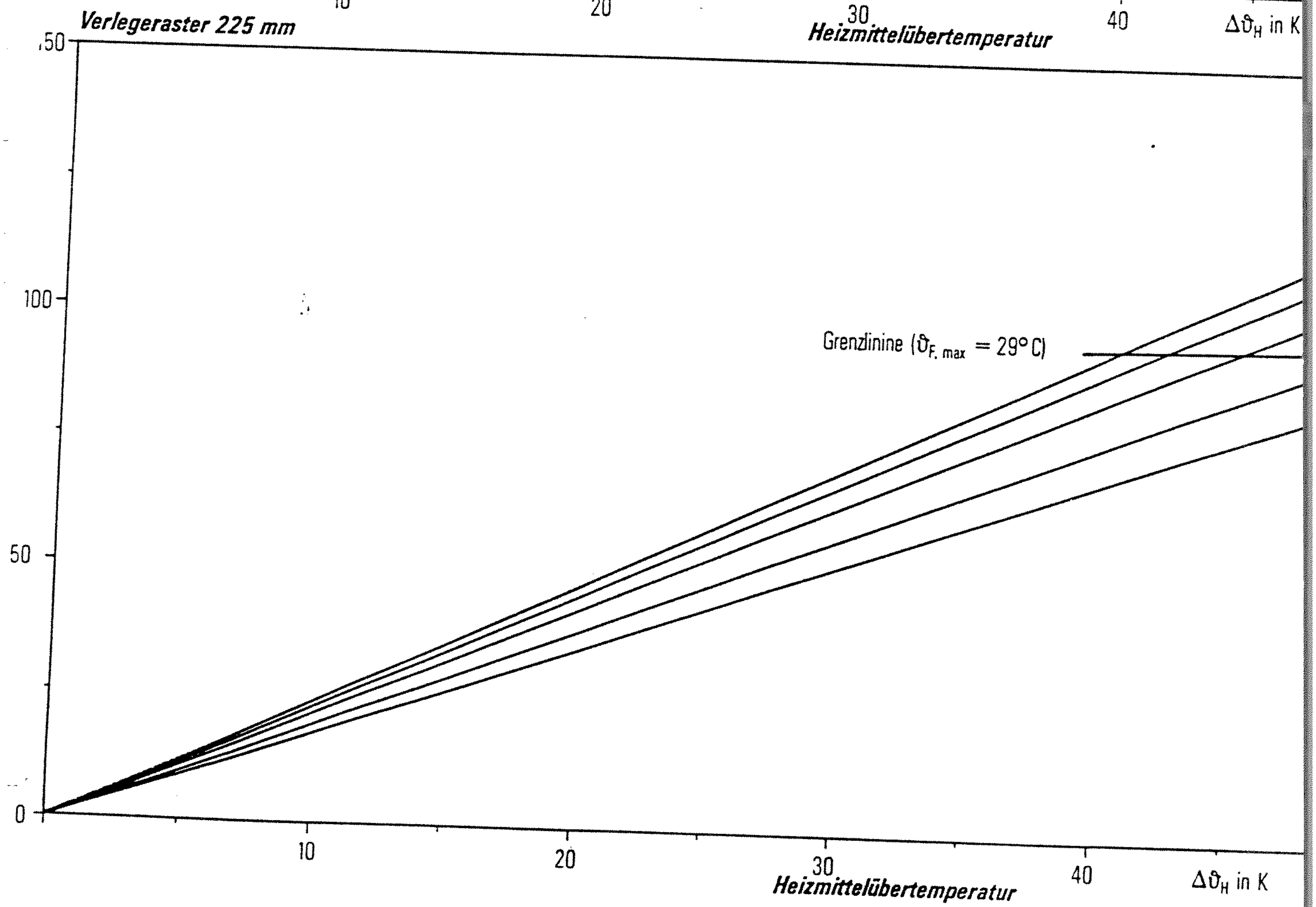
Raumtemperatur in °C				20				22				24			
Fußbodenbeläge R_λ in $\frac{m^2 \cdot K}{W}$				0,02	0,05	0,10	0,15	0,02	0,05	0,10	0,15	0,02	0,05	0,10	0,15
ır	Verlege-abstand RA in mm	Verlegte Rohr-menge m/m^2	Max. Heiz-kreisfläche in m^2 100 m	Wärmestromdichte $\dot{q}$ in W/m^2 bei einer Spreizung von $(\vartheta_V - \vartheta_R) = 15$ K											
	75	13	7	99	88	74	63	91	30	67	58	82	72	60	52
	150	6,5	14	66	61	54	48	60	55	49	44	54	50	44	39
	75	13	7	122	107	90	77	113	99	83	72	104	92	77	66
	150	6,5	14	81	74	65	59	75	69	61	54	69	63	56	50
	75	13	7	144	127	106	91	135	119	100	86	126	111	93	80
	150	6,5	14	96	88	77	69	90	82	73	65	84	77	68	61
	75	13	7	166	146	123	105	157	138	116	100	148	131	110	94
	150	6,5	14	110	101	89	80	104	96	84	76	98	90	80	71
	75	13	7	188	166	139	119	179	158	132	114	170	150	126	108
	150	6,5	14	125	115	101	91	119	109	96	86	113	104	92	82

R_λ in $\frac{m^2 \cdot K}{W}$

liesen
losaik-Parkett
VC, Linoleum
mm Teppich

Die Leistungsdaten sind den folgenden Diagrammen entnommen und berücksichtigen nicht die max. Oberflächentemperaturen 29, 33 bzw. 35° C.
Anhaltswert für die Randzone:
Bei ca. 175 W/m^2 wird die mittlere Oberflächentemperatur von 35° C


Verlegeraster 150 mm
150
100
50
0
Grenzlinine ($\vartheta_{F,max}$ = 29° C)
10
20
30
40
$\Delta\vartheta_H$ in K
Heizmittelübertemperatur

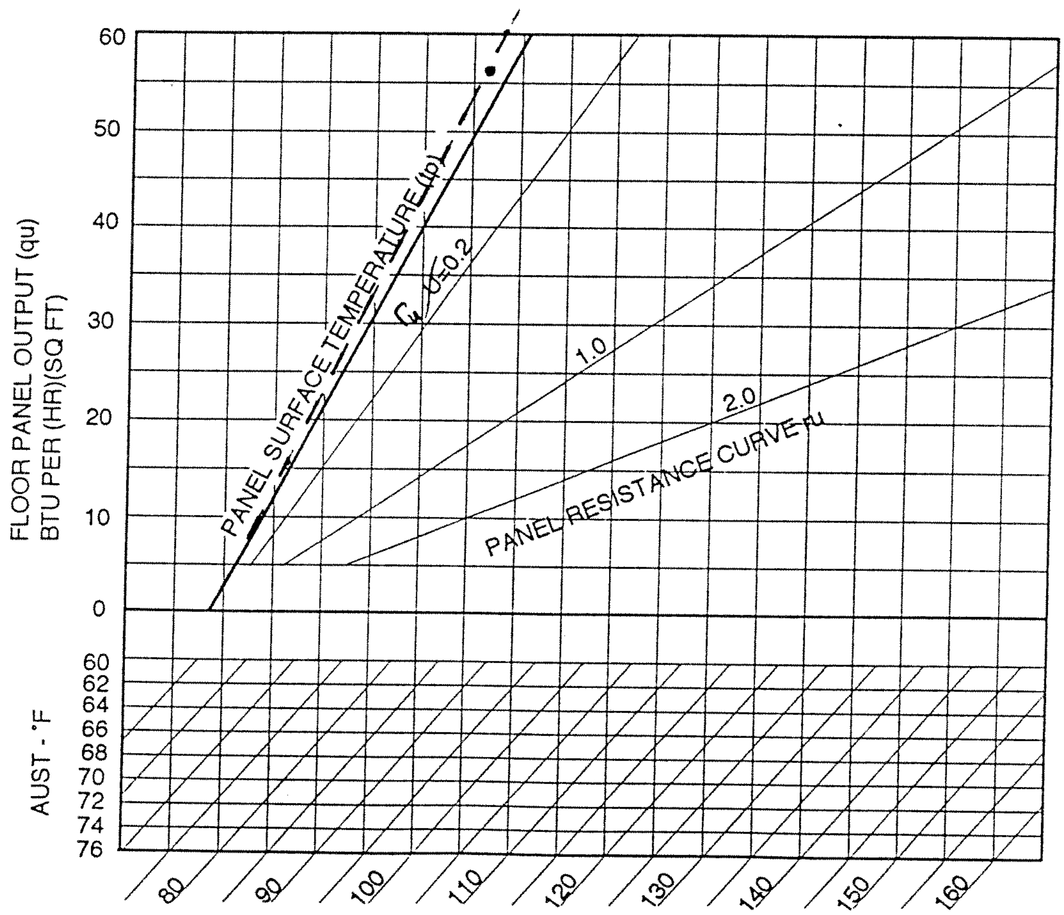

Verlegeraster 225 mm
150
100
50
0
Grenzlinine ($\vartheta_{F,max}$ = 29° C)
10
20
30
40
$\Delta\vartheta_H$ in K
Heizmittelübertemperatur

AUTHOR'S NOTE

This nomograph aims to improve Figure 8 of Chapter 6 in ASHRAE Handbook by replacing the air temperature by AUST. The panel surface temperature line seems to be a few degrees below the ASHRAE equation,because the power 1.3 was used in convection equation while ASHRAE equation uses the power of 1.31.

This nomograph does not modify Figure 8 with respect to its other shortcomings and it can not be used for ceiling heating or cooling. It still needs to calculate $r_u(M)$ from tables or obtain from manufacturer's catalogs. The restrictions of using $r_u(M)$ without distinguishing its split were explained earlier.

Proposed Floor Panel Design Graph

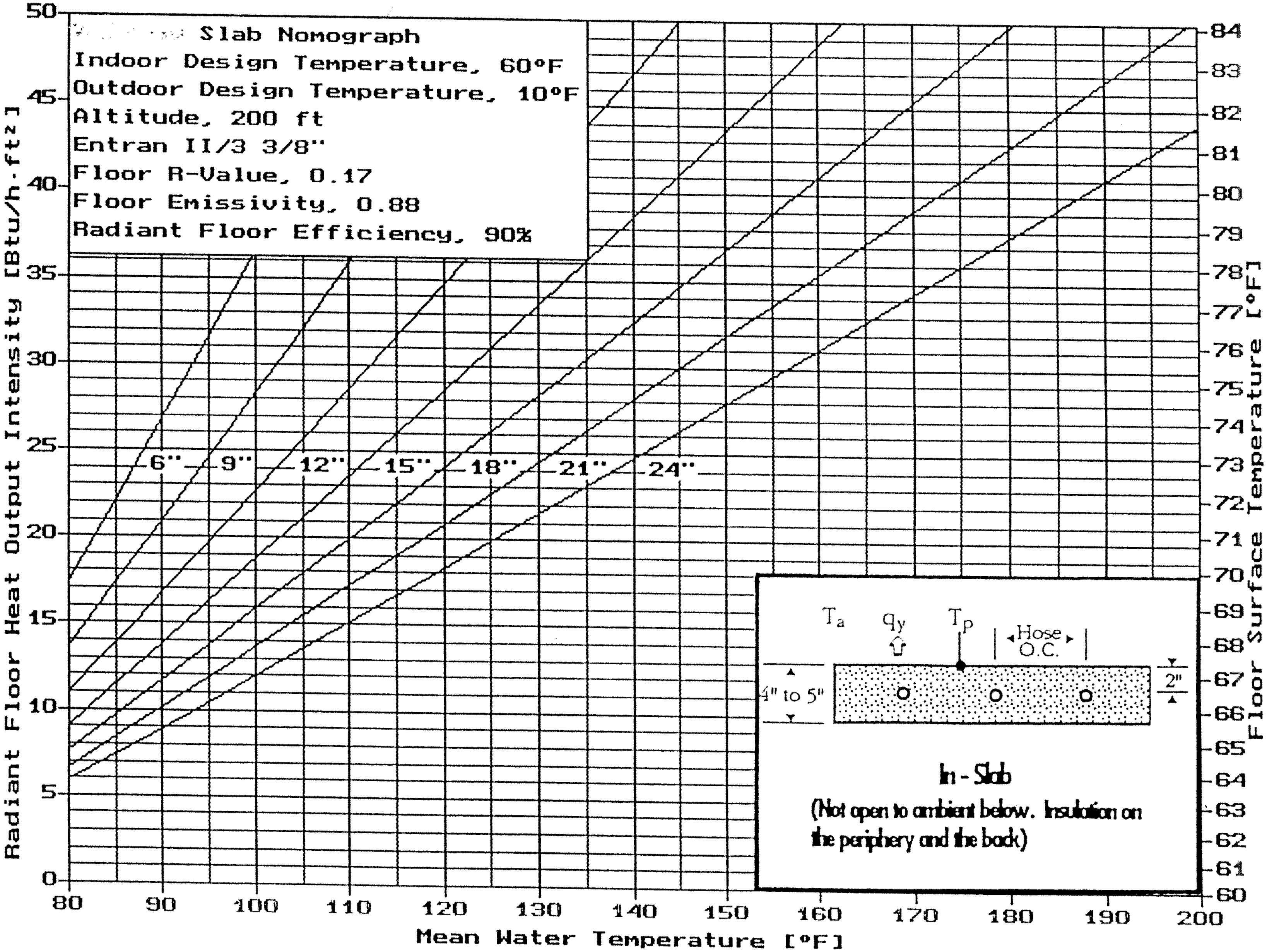


Lawrence V. Drake

August 4, 1993

AUTHOR'S NOTE

The following nomograph is for a bare slab, and for a given indoor temperature. It gives t_p and t_w for a set of hose spacings M. Altitude adjustment and room size adjustments are also possible for the computer generated nomograph. Adjustment for the floor covering R-value is made through an equation or a chart.


Slab Nomograph
Indoor Design Temperature, 60°F
Outdoor Design Temperature, 10°F
Altitude, 200 ft
Entran II/3 3/8"
Floor R-Value, 0.17
Floor Emissivity, 0.88
Radiant Floor Efficiency, 90%
6"
9"
12"
15"
18"
21"
24"
Radiant Floor Heat Output Intensity [Btu/h·ft²]
Mean Water Temperature [°F]
Floor Surface Temperature [°F]
Ta
qy
Tp
Hose O.C.
4" to 5"
2"
In - Slab
(Not open to ambient below. Insulation on the periphery and the back)

2.1.3 In-slab Heating Nomographs

The arrangement and the use of in-slab heating nomographs are the same with the following exception:

. Not any floor covering above the concrete slab is accounted for in these nomographs. Therefore, any floor covering (including additional slab thickness above the basic 2" slab above the piping center plane) present in a design must be encountered by adjusting the required mean water temperature by the following approximation :

$$T_W(\text{adjusted}) \approx T_W(\text{nomograph}) + q_y \cdot R_{total} \qquad [15]$$

Figure 6 provides the graphical representation of Eqn.15. On this figure, K represents the temperature penalty term $q_y \cdot R_{total}$, in ^{o}F.

Example 5:

A radiant panel which consists of a concrete slab having piping with 3/8" I.D, on a 12" spacing O.C. The required radiant heat output intensity, q_y is 35 Btu/h ft². There is **1/4"** thick ceramic tile flooring on the slab. In addition to this, **3/8"** thick carpet with k=0.05 is laid on the floor. The slab is **3"** thick above the piping center plane. Equivalent radiant comfort temperature is required to be 68 ^{o}F.

For ceramic tile;
$k = 0.40, \quad R = 1/4"/ (12 \cdot 0.40) = 0.052$
For carpet;
$R = 3/8"/ (12 \cdot 0.05) = 0.625$

The nomographs were prepared for 2" slab thickness from pipe center to the bare slab surface.(see the figure on right hand bottom on in-slab nomographs). Therefore 1" extra slab has to be treated separately:

For 1" extra concrete layer in the slab ;

$k = 0.81, \quad R = 1"/ (12 \cdot 0.81) = 0.10$

Therefore;

$$R_{total} = 0.052 + 0.625 + 0.10 \approx 0.78$$

Using the appropriate nomograph, T_w reading is 127 ^{o}F (see Figure 7). After adjusting for R_{total}, through Eqn.15;

$$T_w(\text{adjusted}) = 127 + 35 \cdot 0.78 \approx 155\ ^{o}F$$

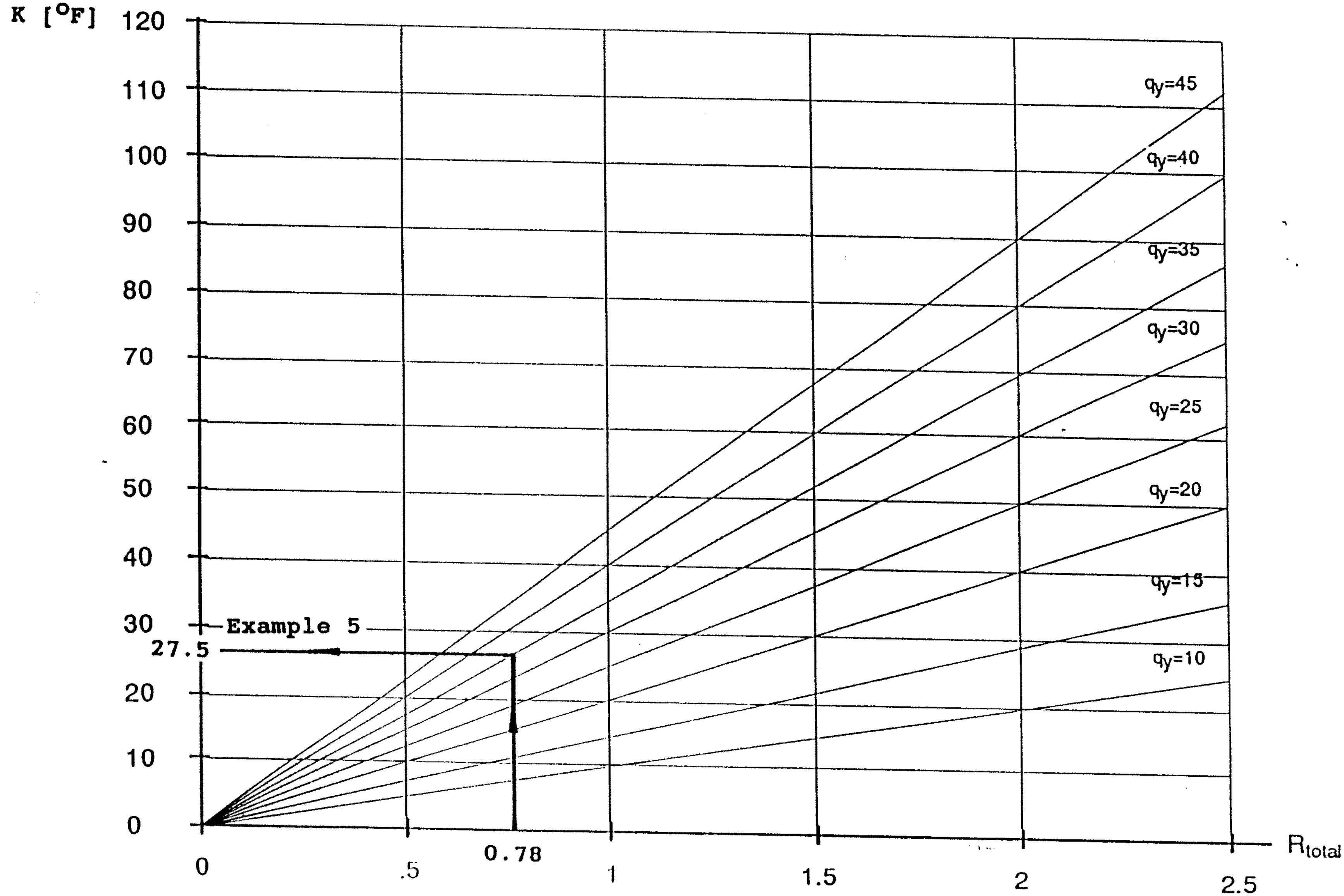


Figure 6. Diagram for adjusting nomograph readings for T_w in slab heating

(Adjusted $T_w = T_w + K$)

Appendix 7. Revisions on Figures of Chap.6,ASHRAE Handbook,1992

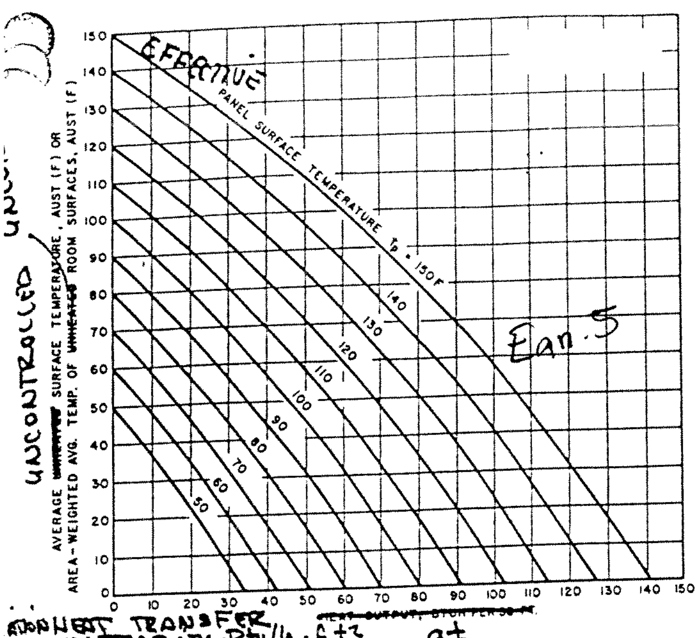


Fig. 1 Radiation Heat Transfer ~~from~~ Heated Ceiling, Floor, or Wall Panel

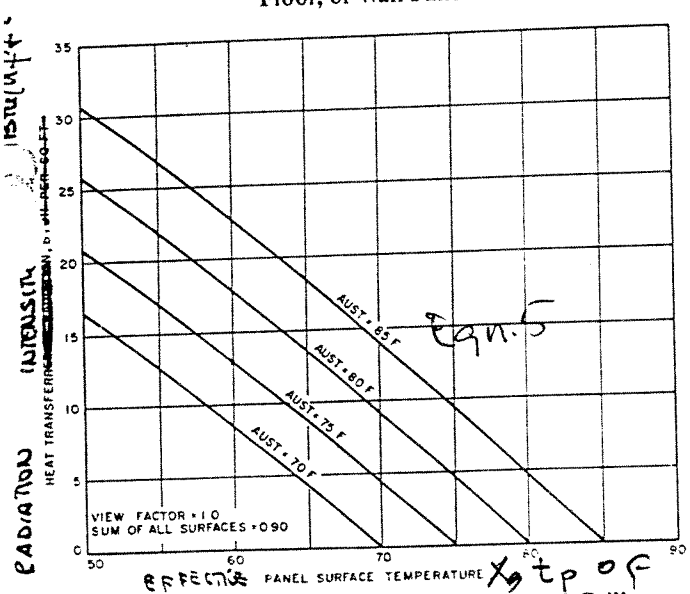


Fig. 2 Heat Removed by Radiation ~~from~~ Cooled Ceiling or Wall Panel
at

temperature at the 5-ft level will closely approach the average uncooled surface temperatures (AUST). In structures where the main heat gain is through the walls or where incandescent lighting is used, the wall surface temperatures tend to rise considerably above the room air temperature.

Convection Transfer

The convection coefficient q_c is defined as the heat transferred by convection in Btu/h·ft²·°F difference between air and panel temperatures. Heat transfer convection values are not easily established. Convection in panel systems is usually natural; that is, air motion is generated by the warming or cooling of the boundary layer of air. In practice, however, many factors such as a room's configuration interfere with or affect natural convection. Infiltration, the movement of persons, and mechanical ventilating systems can introduce some forced convection that will disturb the natural process.

Parmelee and Huebscher (1947) included the effect of forced convection on heat transfer from panels as an increment to be added to the natural convection coefficient. However, increased heat transfer from forced convection should not be used, because the increments are unpredictable in pattern and performance, and forced convection does not significantly increase the total capacity of the panel system.

Convection in a panel system is a function of the panel surface temperature and the temperature of the airstream layer directly below the panel. The most consistent measurements are obtained when the air layer temperature is measured close to the region where the fully developed stream begins, usually 2 to 2.5 in. below the panels.

Min *et al.* (1956) determined natural convection coefficients 5 ft above the floor in the center of a 12 ft by 24 ft room. Equations (6) to (11), derived from this research, can be used to calculate heat transfer from panels by natural convection.

Natural convection from a heated ceiling

$$q_c = 0.041\,(t_p - t_a)^{1.25}/D_e^{0.25} \tag{6}$$

Natural convection from a heated floor or cooled ceiling

$$q_c = 0.39\,(t_p - t_a)^{1.31}/D_e^{0.08} \tag{7}$$

Natural convection from a heated or cooled wall panel

$$q_c = 0.29\,(t_p - t_a)^{1.32}/H^{0.05} \tag{8}$$

where

q_c = heat transfer by natural convection, Btu/h·ft²
t_p = temperature of panel surface, °F
t_a = temperature of air, °F
D_e = equivalent diameter of panel (4 area/perimeter), ft
H = height of wall panel, ft

Schutrum and Humphreys (1954) measured panel performance in furnished test rooms that did not have uniform temperature surfaces and found no variations large enough to be significant in heating practice. Schutrum and Vouris (1954) established that the effect of room size was also usually insignificant. The convection equations can therefore be simplified to:

Natural convection from a heated ceiling

$$q_c = 0.021\,(t_p - t_a)^{1.25} \tag{9}$$

Natural convection from a heated floor or cooled ceiling

$$q_c = 0.32\,(t_p - t_c)^{1.31} \tag{10}$$

Natural convection from a heated or cooled wall panel

$$q_c = 0.26\,(t_p - t_c)^{1.32} \tag{11}$$

Figure 3 shows heat output by natural convection from floor, wall, and ceiling heating panels as calculated from Equations (9) and (10).

Figure 4 shows heat removed by natural convection by cooled ceiling panels as calculated by Equation (10) and data from Wilkes and Peterson (1938) for specific panel sizes. An additional curve illustrates the effect of forced convection on the latter data. Similar adjustment of the ASHRAE data is inappropriate, but the effects would be much the same. For preliminary design, use 1 Btu/h·ft²·°F between room design and panel temperature.

Combined Heat Transfer (Radiation and Convection)

The combined heat transfer from a panel to a room can be determined by adding the radiant heat transfer from Figure 1 or 2 to the convective heat transfer from Figure 3 or 4, respectively. Figures 1 and 2 require calculating the AUST in the room. In calculating the AUST, the surface temperature of the interior walls is assumed to be the same as the room air temperature. The inside surface temperatures of outside walls and exposed floors or ceil-

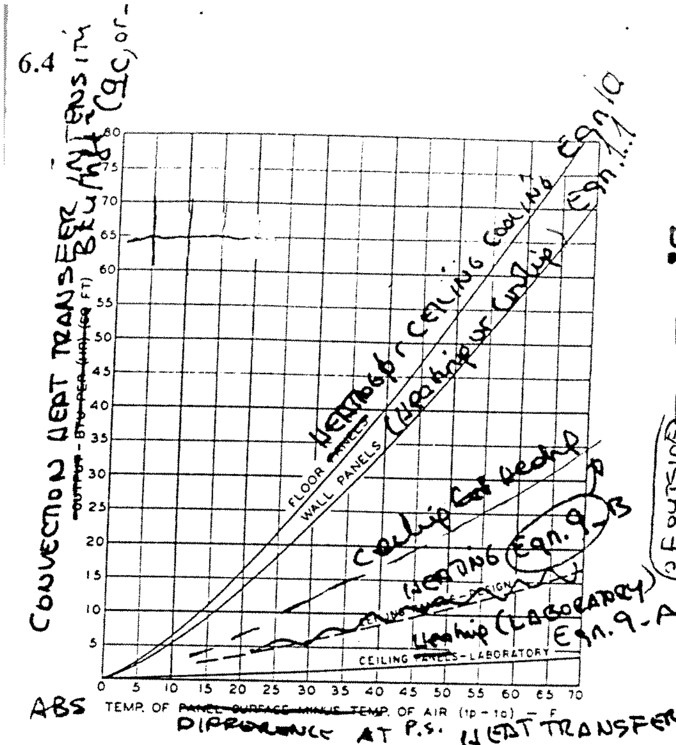


Fig. 3 ~~Heat Output by~~ Natural Convection ~~from~~ Floor ~~and~~ Ceiling Panels [Equations (10) and (11)]

and Wall (9)

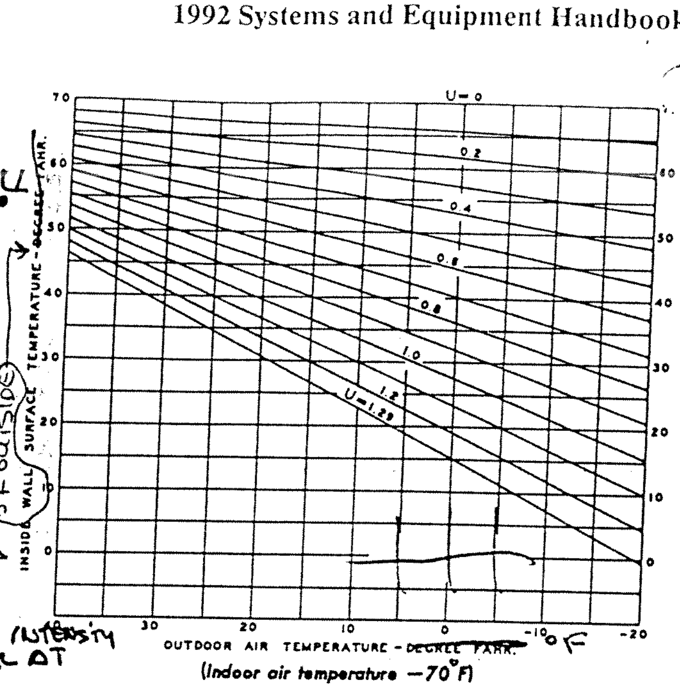


Fig. 5 Relation of Inside Surface Temperature to Overall Coefficient of Heat Transfer

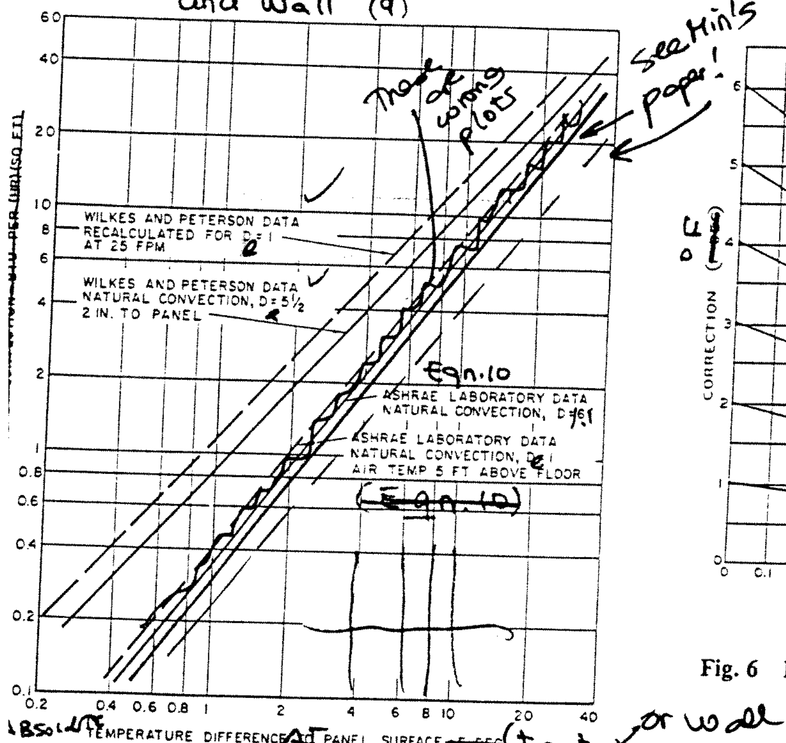


Fig. 4 Heat Removed by Natural Convection to Ceiling Cooling Panels [Equation (10)]

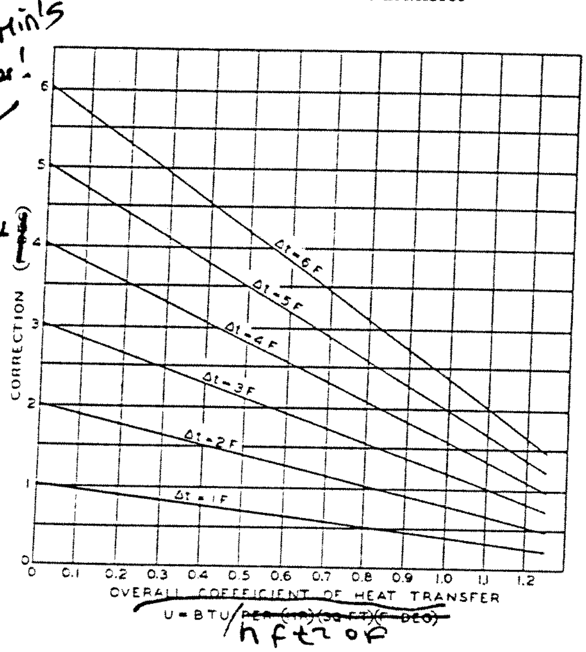


Fig. 6 Inside Wall Surface Temperature Correction for Air Temperatures Other than 70°F

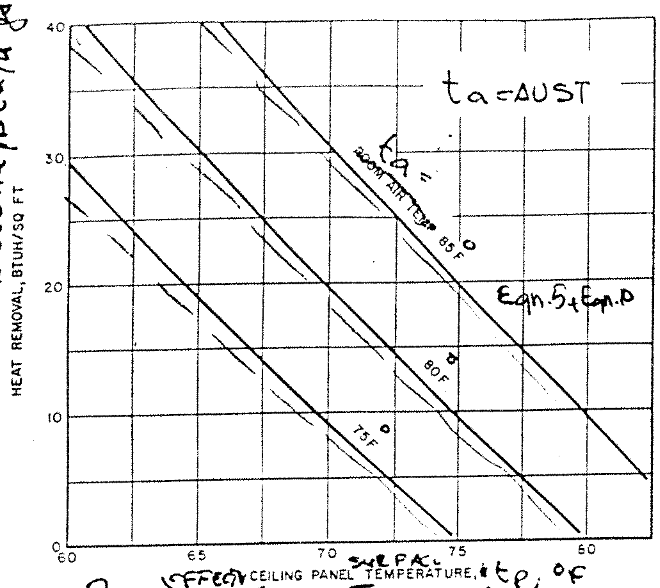
can be obtained from Figure 5 for a 70°F room temperature. ıre 6 gives corrections for other temperatures. Steinman *et al.* 9) noted that these temperatures may not be appropriate for ɔsures with large glass areas or a high percentage of outside and/or ceiling surface area. These cold surfaces have a lower 'T, which increases the radiant heat transfer.

ıe combined heat transfer for ceiling and floor panels used :at rooms in which the air temperature is 70 to 76°F can be directly from Figures 7 and 8, respectively. These diagrams ɣ to rooms in which the AUST does not differ greatly from room air temperatures. Tests by Schutrum *et al.* (1953a, 1953b) and simulation by Kalisperis (1985) based on a CAD program developed by Kalisperis and Summers (1985) show that the temperatures are almost equal.

The performance of radiant transfer equipment, in its various forms and applications to a particular space, may differ substantially from the outputs given in Figures 7 and 8. Refer to manufacturer's data for the specific system selected.

Figure 9 shows the combined radiation and convection transfer for cooling, as given in Figures 2 and 4. The data in Figure 9 do not include heat gains from sun, lights, people, or equipment: manufacturer's data includes these heat gains.

In suspended ceiling panel systems, heat transfers from the ceiling panel to the floor slab above (heating) and vice versa (cooling).



~~(Eqn. 5 + Eqn. 10)~~

Fig. ~~5~~ 7 Cooled Ceiling Panel Performance in Uniform Environment with No Infiltration and No Internal Heat Sources

The ceiling panel surface temperature is affected because of heat transfer to or from the panel and the slab by radiation and, to a much smaller extent, by convection. The radiation component can be approximated using Figure 1. The convection component can be estimated from Figure 3 or 4. In this case, the temperature difference is that between the top of the ceiling panel and the mid-space of the ceiling. The temperature of the ceiling space should be determined by testing, since it varies with different panel systems. However, much of this heat transfer is nullified when insulation is placed over the ceiling panel, which, for perforated metal panels, also provides acoustical control.

If lighting fixtures are recessed into the suspended ceiling space, radiation from the top of the fixtures raises the overhead slab temperature and transfers heat to the space by convection. This energy is absorbed at the top of the cooled ceiling panels by radiation, as in Figure 2, and by convection, generally in accordance with Equation (6). The amount the top of the panel absorbs depends on the system. Most manufacturers have information available. Similarly, panels installed under a roof absorb additional heat, depending on configuration and insulation.

GENERAL DESIGN CONSIDERATIONS

Radiant panel systems are similar to other air-water systems in the arrangement of the system components. Room thermal conditions are maintained primarily by direct transfer of radiant energy, rather than by convection heating and cooling. The room heating and cooling loads are calculated in the conventional manner. Manufacturers' ratings generally are for total performance and can be applied directly to the calculated room load.

Because the mean radiant temperature (MRT) within a panel heated space increases as the heating load increases, the controlled air temperature during this increase may be lowered without affecting comfort. In ordinary structures with normal infiltration loads, the required reduction in air temperature is small, enabling a conventional room thermostat to be used.

In panel heating systems, lowered night temperature can produce less satisfactory results with heavy panels such as concrete floors. These panels cannot respond to a quick increase or decrease in heating demand within the relatively short time required, resulting in a very slow reduction of the space temperature at night and a correspondingly slow pickup in the morning. Light panels, such as plaster or metal ceilings and walls, may respond to changes in demand quickly enough for satisfactory results from lowered night temperatures. Tests on a metal ceiling panel demonstrated the speed of response to be comparable to that of convection systems. However, very little fuel savings can be expected even with light panels unless the lowered temperature is maintained for long periods. If reduced nonoccupancy temperatures are employed, some means of providing a higher-than-normal rate of heat input for rapid warm-up is necessary; for example, fast-acting radiant ceiling panels (Berglund 1982).

Metal radiant heating panels, hydronic and electric, are applied to building perimeter spaces for heating in much the same way as finned-tube convectors. Metal panels are usually installed in the ceiling and are integrated into the ceiling design. The layout and arrangement of panels usually considers architectural design.

Partitions may be erected to the face of hydronic panels but not to the active heating portion of electric panels because of possible element overheating and burnout. Electric panels are often sized to fit the building module with a small removable filler or dummy panel at the window mullion to accommodate future partitions. Hydronic panels can run continuously.

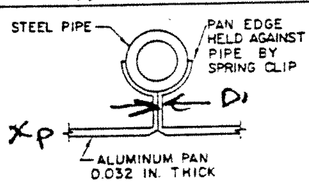
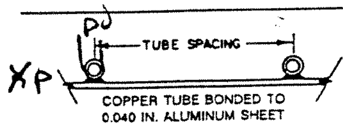
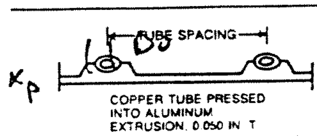
By cutting and fitting, field modification of hydronic panels is possible; however, modification should be kept to a minimum to keep installation costs down. Electric panels cannot be modified in the field.

Panel Thermal Resistance

The thermal resistance to heat flow may vary considerably among panel systems, depending on the type of bond between the water tube and the panel material. Corrosion between lightly touching surfaces, the method of maintaining contact, and other factors may change the bond with time. The actual thermal resistance of any proposed system should be verified by testing whenever practical. Tables 1 through 5 show some typical values for thermal resistance factors for various types of floor and ceiling panels.

The thermal resistance factors shown are for ferrous and nonferrous pipe and tube. Recently, the use of nonmetallic, plastic tubing has increased. Long lengths are available in coils up to 1000 ft, which simplify installation. Resistance and performance data should be obtained from the manufacturer.

Table 1 Thermal Resistance of ~~Metal~~ Ceiling Panels

Type of Panel	~~Spacing, in.~~	Thermal Resistance,[a] ft²·°F·h/Btu
	3	—
	6	0.31
	12	0.61
	4	0.071
	8	0.15
	5	0.071

~~[a]Manufacturer's data.~~

← Eqn Table 4

A2 more boxes for plaster ceilings from Table 4 fer to manufacturer's data for panel surface temperatures higher than those given in Figure 7.

Design the panel arrangement.

Thermal comfort requirements should be checked in the following steps (see Chapter 8 of the 1989 ASHRAE *Handbook—Fundamentals* and NRB 1981).

a. Determine occupant's clothing value (clo value) and metabolic rate (MET) (see Tables 1D and 4A, Chapter 8 of the 1989 ASHRAE *Handbook—Fundamentals*).
b. Determine the optimum operative temperature at the coldest point in the room (see Figure 15, Chapter 8 of the 1989 ASHRAE *Handbook—Fundamentals* or NRB Figure 2.3.22 for other values).
c. Determine the mean radiant temperature (MRT) at the coldest point in the room (see Fanger 1972).
d. From the definition of operative temperature, establish the optimum room design temperature at the coldest point in the room. If the optimum room design temperature varies greatly from the designated room design temperature, designate a new temperature.
e. Determine the MRT at the hottest point in the room.
f. Calculate the operative temperature at the hottest point in the room.
g. Compare the operative temperatures at the hottest and coldest points in the room. For light activity and normal clothing, the acceptable operative temperature range is 68 to 75 °F (see NRB 1981 for other ranges). If the range is not acceptable, the heating system must be modified.
h. Calculate radiant temperature asymmetry (NRB 1981). Acceptable ranges are < 18 °F for windows and < 9 °F for warm ceilings.

Determine water flow rate and pressure drop.

e application, design, and installation of panel systems have in requirements and techniques such as the following:

As with any hydronic system, look closely at the piping system design. Piping should be designed to ensure that water of the proper temperature and in sufficient quantity is available to every grid or coil at all times. Reverse-return systems should be considered to minimize balancing problems.

Individual panels can be connected for parallel flow using headers, or for sinuous or serpentine flow. To avoid flow irregularities within a header-type grid, the water channel or lateral length should be greater than the header length. If the laterals in a header grid are forced to run in a short direction, this problem can be solved by using a combination series-parallel arrangement. Serpentine flow will ensure a more even panel surface temperature throughout the heating or cooling zone.

Noises from entrained air, high-velocity, or high-pressure drop devices or from pump and pipe vibrations must be avoided. Water velocities should be high enough to prevent separated air from accumulating and causing air binding. Where possible, avoid automatic air venting devices over ceilings of occupied spaces.

Design piping systems to accept thermal expansion adequately. Do not allow forces from piping expansion to be transmitted to panels. Thermal expansion of the ceiling panels must be considered.

In circulating water systems, plastic, steel, and copper pipe or tube are used widely in ceiling, wall, or floor panel construction. Where coils are embedded in concrete or plaster, no threaded joints should be used for either pipe coils or mains. Steel pipe should be the all-welded type. Copper tubing should be soft-drawn coils. Fittings and connections should be minimized. Changes in direction should be made by bending. Solder-joint fittings for copper tube should be used with a medium temperature solder of 95% tin, 5% antimony, or capillary brazing alloys. All piping should be subjected to a hydrostatic test of at least three times the working pressure. Maintain adequate pressure in embedded piping while pouring concrete.

6. Placing the thermostat on a side wall where it can see the outside wall and the warm panel should be considered. The normal thermostat cover reacts to the warm panel, and the radiant effect of the panel on the cover tends to alter the control point so that the thermostat controls 2 to 3 °F lower when the outdoor temperature is a minimum and the panel temperature is a maximum. Experience indicates that radiantly heated rooms are more comfortable under these conditions than when the thermostat is located on a back wall.
7. If throttling valve control is used, either the end of the main should have a fixed bypass, or the last one or two rooms on the mains should have a bypass valve to maintain water flow in the main. Thus, when a throttling valve modulates, there will be a rapid response.
8. When selecting heating design temperatures for a ceiling panel surface, the design parameters are:
   a. Excessively high temperatures over the occupied zone will cause the occupant to experience a "hot head effect."
   b. Temperatures that are too low can result in an oversized, uneconomical panel and a feeling of coolness at the outside wall.
   c. Locate ceiling panels adjacent to perimeter walls and/or areas of maximum load.
   d. With normal ceiling heights of 8 to 9 ft, panels less than 3 ft wide at the outside wall can be designed for 235 °F surface temperature. If panels extend beyond 3 ft into the room, the panel surface temperature should be limited to the values as given in Figure 16. The surface temperature of concrete or plaster panels is limited by construction.
9. Floor panels are limited to surface temperatures of less than 85 °F for comfort reasons.
10. When the panel chilled water system is started, the circulating water temperature should be maintained at room temperature until the air system is completely balanced, the dehumidification equipment is operating properly, and building humidity is at design value.

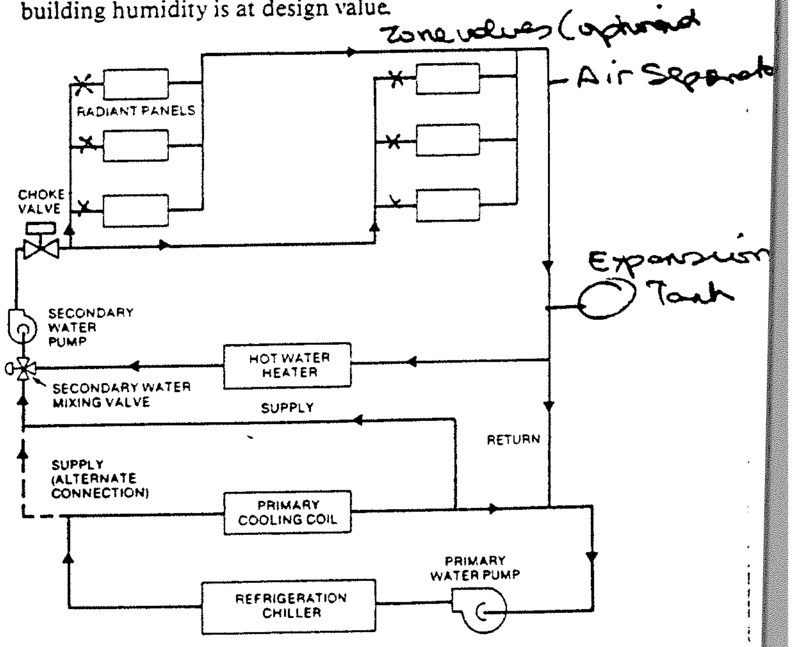


Fig. ~~11~~ 10 Primary/Secondary Water Distribution System with Mixing Control

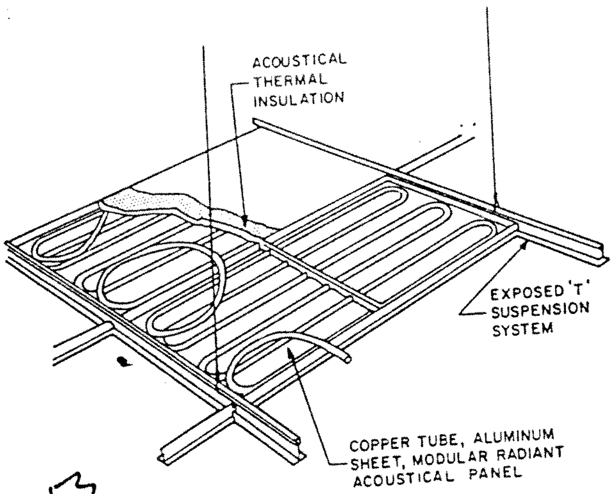


Fig. ~~14~~ 13 Metal Ceiling Panels Bonded to Copper Tubing

mal resistance and respond quickly to changes in space conditions. Table 1 shows thermal resistance values for various ceiling constructions.

Metal radiant ceiling panels can be used with any of the all-air cooling systems described in Chapter 2. Chapters 23 through 25 of the 1989 ASHRAE *Handbook—Fundamentals* describe how to calculate heating loads. Double glazing and heavy insulation in outside walls has reduced transmission heat losses. As a result, infiltration and reheat have become of greater concern. Additional design considerations are as follows:

1. Perimeter radiant heating panels, not extending more than 3 ft into the room, may operate at higher temperatures, as described under "Design Considerations" in the Heating and Cooling Systems section.
2. Hydronic panels operate efficiently at low temperature and are suitable for condenser water heat reclaim systems.
3. Locate ceiling panels adjacent to the outside wall and as close as possible to the areas of maximum load. The panel area within 3 ft of the outside wall should have a heating capacity equal to or greater than 50% of the wall transmission load.
4. Ceiling system designs based on passing return air through perforated modular panels into the plenum space above the ceiling are not recommended, because much of the panel heat transfer is lost to the return air system.
5. When selecting heating design temperatures for a ceiling panel surface or mean water temperature, the design parameters are:
   a. Excessively high temperatures over the occupied zone will cause the occupant to experience a "hot head effect."
   b. Temperatures that are too low can result in an oversized, uneconomical panel and a feeling of coolness at the outside wall.
   c. Give the technique in item 3 priority.
   d. With normal ceiling heights of 8 to 9 ft, panels less than 3 ft wide at the outside wall can be designed for 235 °F surface temperature. If panels extend beyond 3 ft into the room, the panel surface temperature should be limited to the values as given in Figure 16.
6. Allow sufficient space above the ceiling for installation and connection of the piping that forms the radiant panel ceiling.

Metal radiant acoustic panels provide heating, cooling, sound absorption, insulation, and unrestricted access to the plenum space. They are easily maintained, can be repainted to look new, and have a life expectancy in excess of 30 years. The system is quiet, comfortable, draft-free, easy to control, and responds quickly. The system is a basic air-and-water system. First costs are competitive with other systems, and a life cycle cost analysis often shows that the long life of the equipment makes it the lowest cost in the long run. The system has been used in hospitals, schools, office buildings, colleges, airports, and exposition facilities.

Metal radiant panels can also be integrated into the ceiling design to provide a narrow band of radiant heating around the perimeter of the building. The radiant system offers advantages over baseboard or overhead air in appearance, comfort, operating efficiency and cost, maintenance, and product life.

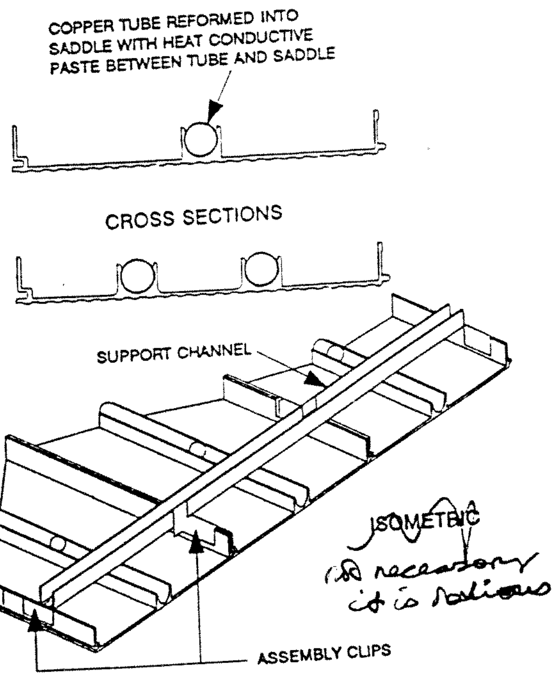


Fig. ~~15~~ 14 Extruded Aluminum Panels with Integral Copper Tube

DISTRIBUTION AND LAYOUT

Chapter 3 and chapters covering hydronic systems apply to radiant panels. Layout and design of metal radiant ceiling panels for heating and cooling begins early in the job. The type of ceiling chosen influences the radiant design and conversely, thermal considerations may dictate what ceiling type to be used. Heating panels should be located adjacent to the outside wall. Cooling panels may be positioned to suit other elements in the ceiling. In applications with normal ceiling heights, heating panels that

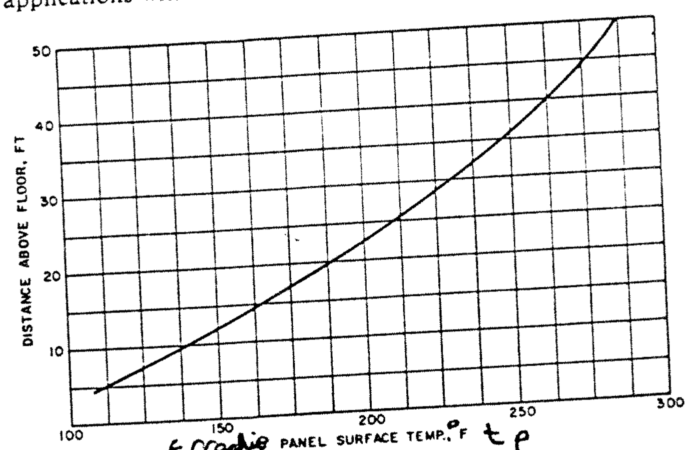


Fig. ~~16~~ 15 Suggested Design Ceiling Surface Temperatures at Various Ceiling Heights

Piping Embedded in Ceilings

exceed 160 °F should not be located over the occupied area. In hospital applications, valves should be located in the corridor outside patient rooms.

One of the following types of construction is generally used.

1. Pipe or tube is embedded in the lower portion of a concrete slab, generally within 1 in. of its lower surface. If plaster is to be applied to the concrete, the piping may be placed directly on the wood forms. If the slab is to be used without plaster finish, the piping should be installed not less than 0.75 in. above the undersurface of the slab. Figure 17 shows this method of construction. The minimum coverage must comply with local building code requirements.
2. Pipe or tube is embedded in a metal lath and plaster ceiling. If the lath is suspended to form a hung ceiling, the lath and heating coils are securely wired to the supporting members so that the lath is below, but in good contact with, the coils. Plaster is then applied to the metal lath, carefully embedding the coil as shown in Figure 18.
3. Smaller diameter copper or plastic tube is attached to the underside of wire lath or gypsum lath. Plaster is then applied to the lath to embed the tube, as shown in Figure 19.
4. Other forms of ceiling construction are composition board, wood paneling, etc., with warm water piping, tube, or channels built into the panel sections.

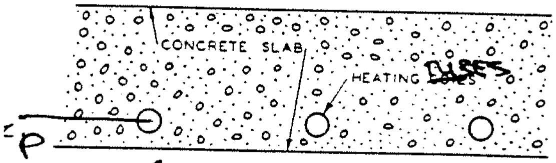


Fig. 16 ~~Coils~~ in Structural Concrete Slab
Tubes

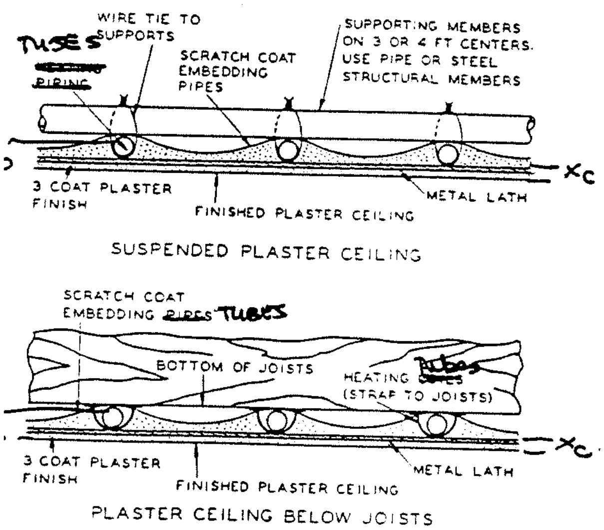


Fig. 17 ~~Coils~~ in Plaster above Lath
Tubes

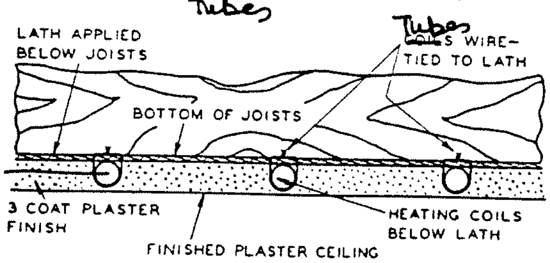


Fig. 18 ~~Coils~~ in Plaster below Lath
Tubes

Coils are usually the sinuous type, although some header grid-type coils have been used in ceilings. Coils may be plastic, ferrous, or nonferrous pipe or tube, with coil pipes spaced from 4.5 to 9 in. on centers, depending on the required output, pipe or tube size, and other factors.

Where plastering is applied to pipe coils, a standard three-coat gypsum plastering specification is followed, with a minimum of 0.38 in. of cover below the tubes when they are installed below the lath. Generally, the surface temperature of plaster panels should not exceed 120 °F. This can be accomplished by limiting the water temperature in the pipes or tubes in contact with the plaster to a maximum temperature of 140 °F. Insulation should be placed above the coils to reduce *reverse loss*, the difference between heat supplied to the coil and net useful output to the heated room.

To protect the plaster installation and to ensure proper air drying, heat must not be applied to the panels for two weeks after all plastering work has been completed. When the system is started for the first time, the water supplied to the panels should not be higher than 20 °F above the prevailing room temperature at that time and not in excess of 90 °F. Water should be circulated at this temperature for about two days, then increased at a rate of about 5 °F per day to 140 °F.

During the air drying and preliminary warm-up periods, there should be adequate ventilation to carry moisture from the panels. No paint or paper should be applied to the panels before these periods have been completed or while the panels are being operated. After paint and paper have been applied, an additional shorter warm-up period, similar to first-time starting, is also recommended.

Hydronic Wall Panels

Although embedded piping in walls is not as universally used as floor and ceiling panels, they can be constructed by any of the methods outlined for ceilings.

Hydronic Floor Panels

Interest has developed in radiant floor heating with the introduction of polybutylene tubing and new design techniques. The systems are energy-efficient and use low water temperatures available from solar collector systems.

Embedded Piping in Concrete Slab

Plastic, ferrous, and nonferrous pipe and tube are used in floor slabs that rest on grade. The coils are constructed as sinuous-continuous pipe coils or arranged as header coils with the pipes spaced from 6 to 18 in. on centers. The coils are generally installed with 1.5 to 4 in. of cover above them. Insulation is recommended to reduce the perimeter and reverse losses. Figure 20 shows the application of pipe coils in slabs resting on grade. Coils should be embedded completely and should not rest on an interface. Any supports used for positioning the heating coils should be nonabsorbent and inorganic. Reinforcing steel, angle iron, pieces of pipe

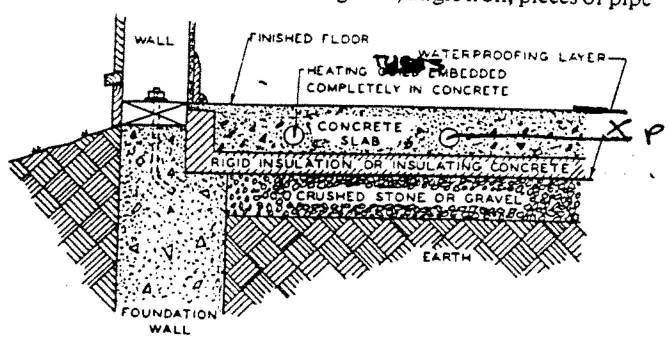


Fig. 20 ~~Coils~~ in Floor Slab on Grade
19 Tubes

or stone, or concrete mounds can be used. No wood, brick, concrete block, or similar materials should support coils. A waterproofing layer is desirable to protect insulation and piping.

Where coils are embedded in structural load-supporting slabs above grade, construction codes may affect their position. Otherwise, the coil piping is installed as described for slabs resting on grade.

The warm-up and start-up period for concrete panels are similar to those outlined for plaster panels.

Embedded systems may fail sometime during their life. Adequate valves and properly labeled drawings will help to isolate the point of failure.

Suspended Floor Piping

Piping may be applied on or under suspended wood floors using several methods of construction. Piping may be attached to the surface of the floor and be embedded in a layer of concrete or gypsum, mounted in or below the subfloor, or attached directly to the underside of the subfloor using metal panels to improve heat transfer from the piping. Whichever method is used for optimum floor output and comfort, it is important that the heat be evenly distributed throughout the floor. Pipes are generally spaced on 4- to 12-in. spacing. Wide spacing under tile or bare floors can cause uneven surface temperatures.

Figure 21 illustrates construction with piping embedded in concrete or gypsum. The thickness of the embedding material is generally 1 to 2 in. when applied to a wood subfloor. Gypsum products specifically designed for floor heating can generally be installed 1 to 1.5 in. thick because they are more flexible and crack resistant than concrete. When concrete is used, it should be of structural quality to reduce cracking due to movement of the wood frame or shrinkage. Embedding material must provide a hard, flat, smooth surface that can accommodate a variety of floor coverings.

As illustrated in Figure 22, tubing may also be installed in the subfloor. The tubing is installed on top of the rafters between the subflooring members. Heat transfer can be improved by the addition of metal heat transfer plates which spread the heat beneath the finished flooring. This construction is illustrated in the figure.

A third construction option is to attach the tube to the underside of the subfloor with or without metal heat transfer plates. The construction is illustrated in Figure 23.

Transfer from the hot-water tube to the surface of the floor is the important consideration in all cases. The floor surface temperature affects the actual heat transfer to the space. Any hin-

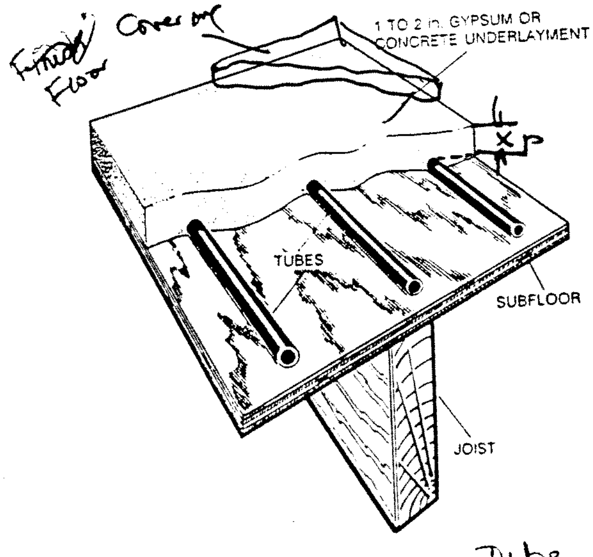
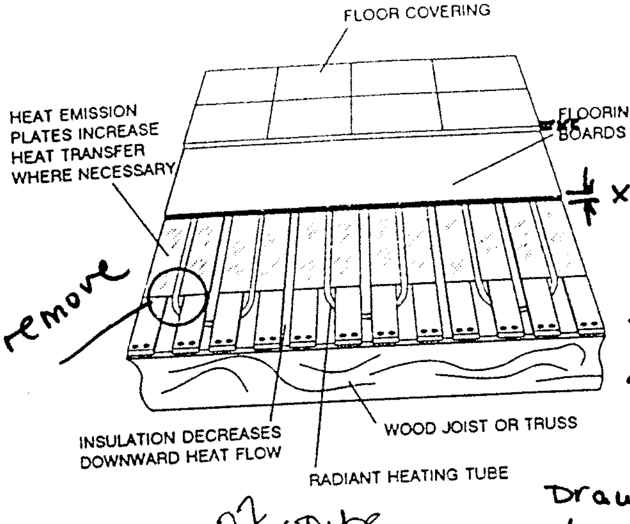


Fig. 21 Embedded ~~Pipe~~ Tube



? This arrow is not possible Drawing is technically wrong

Fig. ~~21~~ 22 ~~Pipe~~ Tube in Subfloor

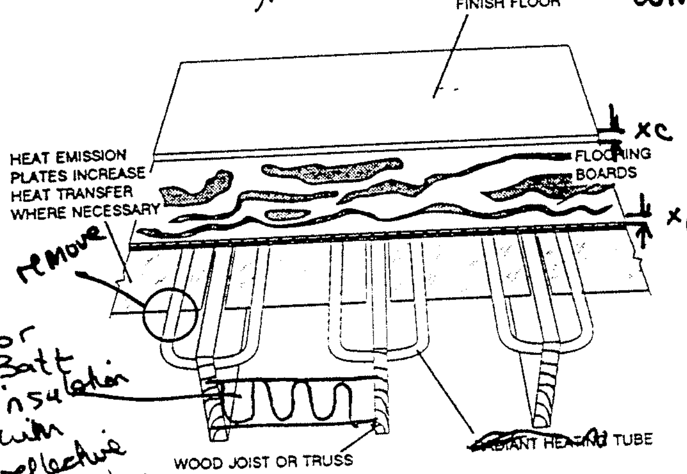


Fig. ~~22~~ 20 ~~Pipe~~ Tube under Subfloor

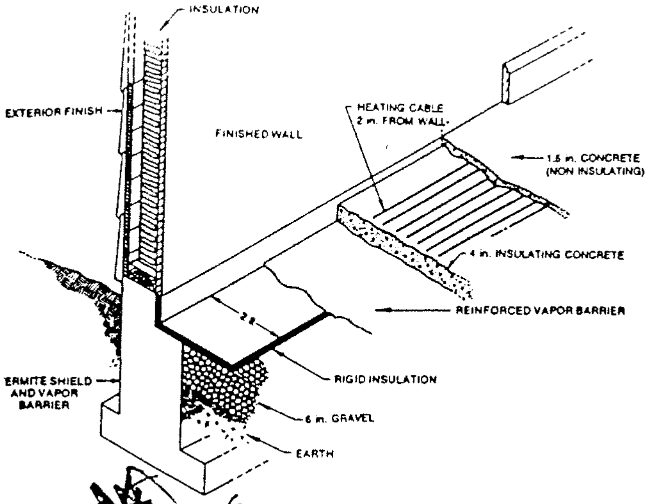
drance between the heated water tube and the floor surface will affect the efficiency of the system. As pointed out in the Panel Thermal Resistance section, the method that most effectively transfers and spreads the heat evenly through the subfloor with the least resistance produces the best results.

ELECTRICALLY HEATED SYSTEMS

Several different forms of electric resistance units are available for heating interior room surfaces. These include: (1) electric heating cables that may be embedded in concrete or plaster or laminated in drywall ceiling construction; (2) prefabricated electric heating panels to be attached to room surfaces; and (3) electrically heated fabrics or other materials for application to, or incorporation into, finished room surfaces.

Electrically Heated Ceilings

Prefabricated electric ceiling panels. A variety of prefabricated electric heating panels are available for either supplemental or full-room heating. These panels are available in sizes from 2 ft by 4 ft to 6 ft by 12 ft. They are constructed from a variety of materials such as gypsum board, glass, steel fiberglass, or vinyl. Different panels have rated inputs varying from 10 to 95 W/ft^2 for 120, 208, 240, and 277 V service. Maximum operating temperatures vary from about 100 to about 300°F, depending on watt density. National



26

Fig. ~~26~~ Electric Heating Cable in Concrete Slab

ıas not set, there is no adhesion problem between the first and econd pour, and a monolithic slab results. A variety of contours an be developed by using heater wire attached to glass fiber mats. ıllow for circumvention of obstructions in the slab.

MI electric heating cable can be installed in concrete slab using ither one or two pours. For single-pour applications, the cable ; fastened to the top of the reinforcing steel before the pour is :arted. For two-layer applications, the cable is laid on top of the ottom structural slab and embedded in the finish layer. Proper ɔacing between adjacent cable runs is maintained by using repunched copper spacer strips nailed to the lower slab.

ir-Heated Floors

Several methods have been devised to warm interior room surces by circulating heated air through passages in the floor. In ·me cases, the heated air is recirculated in a closed system. In hers, all or a part of the air is passed through the room on its ıy back to the furnace to provide supplementary heating and ntilation. Figure 27 indicates one common type of construction. ɔmpliance with applicable building codes is important.

CONTROLS

Automatic controls for panel heating may differ from those for ɔvective heating because of the thermal inertial characteristics the panel and the increase in the mean radiant temperature hin the space under increasing loads. However, lightweight

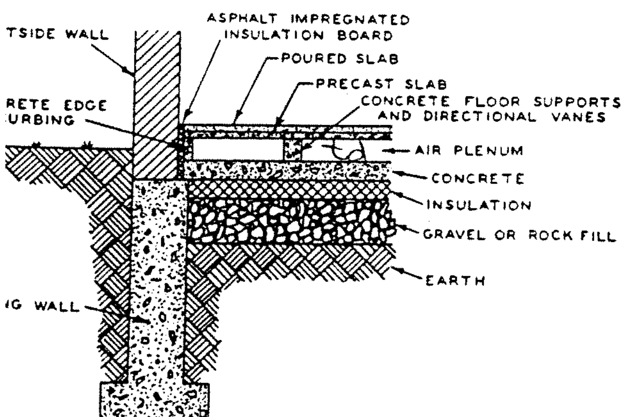


Fig. ~~27~~ Warm Air Floor Panel Construction

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systems using thin metal panels or thin underlay with low therı heat capacity may be successfully controlled with conventior control technology using indoor sensors. Many of the control principles for hot water heating systems described in Chapter 12 and Chapter 14 also apply to panel heating. Because radiant panels do not depend on air-side equipment to distribute energy, many control methods have been used successfully; however, a control interface between heating and cooling should be installed to prevent simultaneous heating and cooling.

High mass panels such as concrete radiant slabs require a control approach different from low mass panels. Because of thermal inertia, significant time is required to bring such massive panels from one operating point to another, say from vacation setback to standard operating conditions. This will result in long periods of discomfort from low temperature, then possibly periods of uncomfortable and wasteful overshoot. Careful economic analysis may reveal that a nighttime setback strategy is not warranted.

Once a slab is at operating conditions, the control strategy should endeavor to supply the slab with heat at the rate that heat is being lost from the space (MacCluer *et al.* 1989). For hydronic slabs with constant circulator flow rate, this equates to modulating the difference between the outgoing water and the returning water temperatures, accomplished via mixing valves, fuel modulation, or, for constant thermal power sources, via pulse-width modulation ("bang-bang" or "on-off" control). Slabs with embedded electric resistance cable can be controlled by pulse-width modulators such as the common round thermostat with anticipator or its solid state equivalent.

A related approach, outdoor reset control, has enjoyed wide acceptance. An outdoor reset control measures the outdoor air temperature, calculates the supply water temperature required for steady operation, and operates a mixing valve or boiler to achieve that supply water temperature. If the heating load of the controlled space is primarily a function of the outdoor air temperature, or indoor temperature measurement of the controlled space is impractical, then outdoor reset control alone is an acceptable control strategy. When other factors such as solar or internal gains are also significant, then indoor temperature feedback should be added to the outdoor reset.

In all radiant panel applications, precautions must be taken to prevent excessive temperatures. A manual boiler bypass or other means of reducing the water temperature may be necessary to prevent new panels from drying out too rapidly.

Cooling Controls

Controlling the panel water circuit temperature by mixing, heat exchange, or using the water leaving the dehumidifier is typical. Other considerations are listed in the General Design Conditions section. It is imperative to dry out the building space before starting the panel water system, particularly after extended down periods, such as weekends. Such delayed starting action can be controlled manually or by a device.

Panel cooling systems require the following basic areas of temperature control: (1) exterior zones, (2) areas under exposed roofs to compensate for transmission and solar loads, and (3) control of each typical interior zone to compensate for internal loads. For optimum results, each exterior corner zone and similarly loaded face zone should be treated as a separate subzone. Panel cooling systems may also be zoned to control temperature in individual exterior offices, particularly in applications where there is a high lighting load or for corner rooms with large glass areas on both walls.

The temperature control of the interior air and panel water supply should not be functions of the outdoor weather. The normal thermostat drift is usually adequate compensation for the slightly lower temperatures desirable during winter weather. This drift should be limited to result in a room temperature change of not

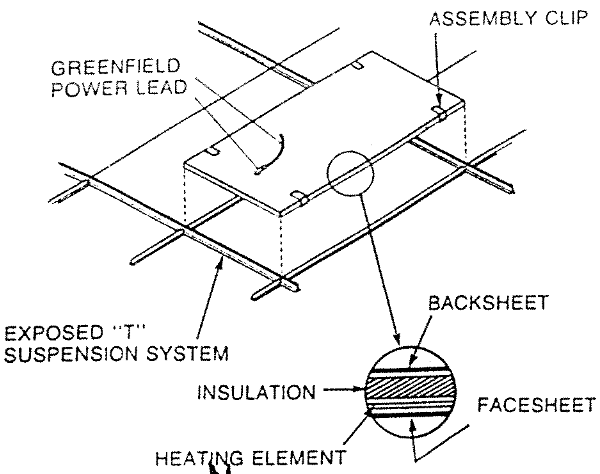

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Fig. 24 Electric Heating Panel

and local codes should be followed when placing partitions, lights, and air grilles adjacent to or near electric panels.

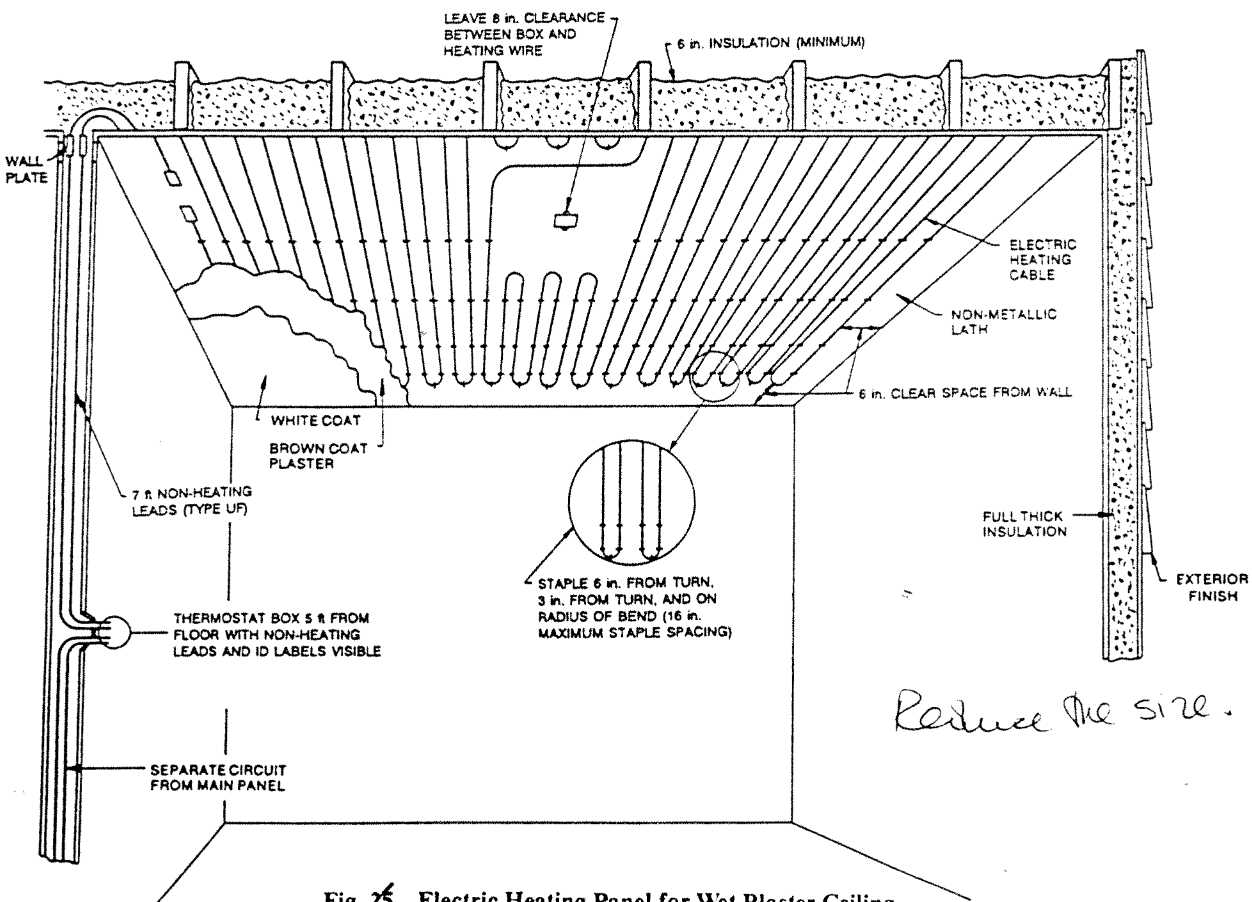
Panel heating elements may be embedded conductors, laminated conductive coatings, or printed circuits. Nonheating leads are connected and furnished as part of the panel. Some panels can be cut to fit available space; others must be installed as received. Panels may be either flush or surface mounted and, in some cases, are finished as part of the ceiling. Rigid panels that are about 1 in. thick and weigh from 6 lb to about 25 lb for a 2 ft by 4 ft panel are usually made for lay-in ceilings (see Figure 24). Panels may also be (1) surface-mounted on gypsum board and wood ceilings, or (2) recessed between ceiling joists. Panels range in size from wide to 8 ft long. The maximum output is 95 W/ft^2.

Electrical cables embedded in ceilings. Electric heating cables for embedded or laminated ceiling panels are factory-assembled units furnished in standard lengths of 75 to 1800 ft. These cable lengths cannot be altered in the field. The cable assemblies are normally rated at 2.75 W per linear foot and are supplied in capacities from 200 to 5000 W in roughly 200-W increments. Standard cable assemblies are available for 120, 208, and 240 V. Each cable unit is supplied with 7-ft nonheating leads for connection at the thermostat or junction box.

Electric cables for panel heating have electrically insulated coverings resistant to medium temperature, water absorption, aging effects, and chemical action with plaster, cement, or ceiling lath material. This insulation is normally a polyvinylchloride (PVC) covering which may have a nylon jacket. The outside diameter of the insulation covering is usually about 0.12 in.

For plastered ceiling panels, the heating cable may be stapled to gypsum board, plaster lath, or similar fire-resistant materials with rust-resistant staples (Figure 25). With metal lath or other conducting surfaces, a coat of plaster (brown or scratch coat) is applied to completely cover the metal lath or conducting surface before the cable is attached. After fastening on the lath and applying the first plaster coat, each cable is tested for continuity of circuit and for insulation resistance of at least 100,000 ohms measured to ground.

The entire ceiling surface is finished with a covering of thermally noninsulating sand plaster about 0.50 to 0.75 in. thick or other approved noninsulating material applied according to manufacturer's specifications. The plaster is applied parallel to the heating cable rather than across the runs. While new plaster is drying, the system should not be energized, and the range and



Reduce the size.

Fig. 25 Electric Heating Panel for Wet Plaster Ceiling

~~#~~ 25

PANEL HEATING AND COOLING

ANEL heating and cooling systems use controlled emperature surfaces in the floor, walls or ceiling; the erature is maintained by circulating water, air, or electricity igh a circuit embedded in or attached to the panel. They be combined with a central station air system of one-zone, tant temperature, constant volume design, or with dual- reheat, multi-zone or variable volume systems.

heating and cooling, room thermal conditions are tained primarily by radiation heat transfer, rather than by ection. A controlled temperature surface is referred to as a nt panel if 50% or more of the heat transfer at the surface is diation. This chapter is concerned with surfaces which are rimary sources of sensible heating and cooling within the itioned space, and whose temperatures are controlled.

gh-temperature surface radiant panels over 160°F gized by gas, electricity, or high-temperature water are issed in Chapter 15.

PANEL HEATING AND COOLING SYSTEMS

nost common forms of panels are as follows:

- Metal ceiling panels
- Embedded tubing or attached piping in ceilings, walls, or floors
- Electric ceiling panels
- Air-heated floors or ceilings
- Electrically heated ceilings or floors

ating applications usually consist of tube, pipe coils, or ric resistance elements embedded in masonry floors or er ceilings. This construction is suitable where loads are e and solar effects are minimized by building design. ever, in buildings where glass areas are large and load ges occur faster, the slow response, lag, and override effect asonry panels are unsatisfactory. Light metal panel systems ond quickly to load changes (Berglund *et al.*, 1982). These ften located in the ceiling because it is exposed to all other ces and objects in the room. It is not likely to be covered, e the floors, and higher surface temperatures can be used.

arm air and electric heating elements are two design epts used in systems influenced by local factors. The warm stem has a special cavity construction where air is lied to a cavity behind the panel surface. The air leaves the y through a normal diffuser arrangement and is supplied to oom. Generally, these systems are used as floor radiant ls in schools and in floors subject to extreme cold, such as overhang. Cold outdoor temperatures and heating medium temperatures must be analyzed with regard to potential damage to the building construction. Electric heating elements embedded in the floor or ceiling construction and unitized electric ceiling panels are used in various applications to provide both full heating and spot heating of the space.

Thermal comfort, as defined in ASHRAE *Standard* 55-1992, is "that condition of mind which expresses satisfaction with the thermal environment." No system is completely satisfactory unless the three main factors controlling heat transfer from the human body (radiation, convection, and evaporation) result in thermal neutrality.

Designers sometimes think that panel heating and cooling systems are desirable only for certain types of buildings and only in some climates. However, maintaining correct conditions for human comfort primarily by radiation is possible for even the most severe climatic conditions (Buckley, 1989).

Panel heating and cooling systems provide an acceptable thermal environment by controlling surface temperature within an occupied space, thus primarily affecting the radiant heat transfer. With a properly designed system, a person should not be aware that the environment is being heated or cooled. The mean radiant temperature (MRT) has a strong influence on body comfort. When the temperature of the surfaces comprising the building deviates excessively from the ambient temperature (particularly outside walls with large amounts of glass), convective systems sometimes have difficulty in counter-acting the discomfort caused by cold or hot surfaces. Heating and cooling panels neutralize these deficiencies and stabilize radiation losses or gains by the body.

Most building materials have surfaces with relatively high emittance factors and, therefore, absorb, re-radiate, and reflect radiant heat from active panels. Warm radiant panels are effective because heat transferred by radiation is absorbed and reflected by the irradiated surfaces, and not transmitted through the construction. Glass is opaque to the wavelengths emitted by active panels and, therefore, transmits little of the long-wave radiation to the outside. This is significant because all surfaces within the room tend to assume temperatures that result in an acceptable thermal comfort condition.

GENERAL EVALUATION

The principal advantages of radiant panel systems are:

- Comfort levels can be better than those of other conditioning systems, because radiant loads are treated directly, and air motion in the space is at normal ventilation levels.
- Space conditioning equipment is not needed at the outside

walls—simplifying the wall, floor, and structural systems.

- Almost all mechanical equipment may be centrally located —simplifying maintenance and operation.
- No space is required within the room for space conditioning equipment. This feature is especially valuable in hotels, hospital patient rooms, and other applications where space is at a premium, where maximum cleanliness is essential, or where dictated by legal requirements.
- Draperies and curtains can be installed at the outside wall without interfering with the space conditioning system.
- Cooling and heating can be simultaneous, without central zoning or seasonal changeover, when four-pipe systems are used.
- Supply air quantities usually do not exceed those required for ventilation and dehumidification.
- A 100% outdoor air system may be installed with less severe penalties in terms of refrigeration load because of reduced air quantities.
- A common central air system can serve both the interior and perimeter zones.
- The modular panel option provides flexibility to meet changes in partitioning.
- Wet surface cooling coils are eliminated from the occupied space, reducing the potential for septic contamination.
- The panel system can use the automatic sprinkler system piping (see NFPA 13, Chapter 5, Sections 5-6). The maximum water temperature must not fuse the heads.
- Space conditioning by panel heating and cooling systems and minimum supply air quantities provide a draft-free environment.
- Noise associated with fan coil or induction units is eliminated.
- Peak loads are leveled as the result of thermal energy storage in the panel, walls and partitions exposed to it.
- The systems are energy-efficient and give the opportunity to reduce conditioning costs (Brunk, 1993). It is possible to use water at low temperatures—available from solar collector systems, waste heat sources, condenser water heat reclaim systems, or heat pumps—by selecting the tube spacing accordingly. Solar absorption systems may be effectively employed for cooling (Kilkis, 1993-a).
- Can be safely used where high-temperature heating units are prohibited by the presence of hazardous or toxic materials.

HEAT TRANSFER AT PANEL SURFACES

A heated or cooled panel transfers heat to or from a room by radiation and natural convection. Some forced convection may also be present if air velocity is high.

Radiation Heat Transfer Intensity

Heat: (1) is transferred by radiation at the speed of light, (2) travels in straight lines and can be reflected; and (3) elevates the temperature of solid objects by absorption, but does not heat the gaseous medium (air) through which it travels.

Heat is exchanged continuously between all bodies in a building environment by radiation. The rate of radiation heat transfer depends on the following factors:

- Temperature (of the emitting surface and receiver)
- Emittance (of the radiating surface)
- Reflectance, absorptance, and transmittance (of the receiver)
- The view factor between the emitting surface and receiver (viewing angle of the occupant to the radiant source)

A critical factor is the structure of the body surface. In general, rough surfaces have low reflectance and high emittance/absorptance characteristics. Conversely, smooth or polished surfaces have high reflectivity and low absorptivity/emissivity characteristics.

One example of radiant heating is the feeling of warmth when standing in the sun's rays on a cool, sunny day. Some of the rays come directly from the sun and include the entire electromagnetic spectrum. Other rays from the sun impinge on surrounding objects and are absorbed or reflected. This generates a secondary source of radiant heat—at a wavelength corresponding to a combination of the temperatures of the objects and the reflected rays. If a cloud passes in front of the sun, there is an instant sensation of cold. This sensation is caused by the decrease in radiant heat received from the sun, although there is little, if any, change in the surrounding ambient air temperature.

The basic equation for a multi-surface enclosure with gray, diffuse isothermal surfaces is derived by radiosity formulation methods (Chapter 3 of the 1993 ASHRAE *Handbook—Fundamentals*). This equation may be written as:

$$q_r = J_p - \sum_{j=1}^{n} F_{pj} J_j \qquad (1)$$

where

q_r = net radiation heat transfer intensity on the panel surface, Btu/h · ft^2
J_p = radiosity of the panel surface, Btu/h · ft^2
J_j = radiosity of all other surfaces in room, Btu/h · ft^2
F_{pj} = radiation angle factor between panel surface and another surface in the room (dimensionless)
n = number of surfaces other than the panel

Equation (1) is applicable to simple and complex enclosures with varying surface temperatures and emittances. The net radiation transferred by the panels can be found by determining unknown J_j if the number of surfaces is small. More complex enclosures require computer calculations.

Radiation angle factors can be evaluated using the charts in Chapter 3 of the 1993 ASHRAE *Handbook—Fundamentals*. Fanger (1972) shows room-related angle factors, or they may b developed from algorithms in ASHRAE *Energy Calculations I* (1976).

Several methods have been developed to simplify Equation (1) by reducing a multi-surface enclosure to a two-surface approximation. In the MRT method, the radiant interchange in a room is modeled by assuming that each surface radiates to a fictitious surface that has a given area, emittance, and temperature—simulating the same heat transfer from the surfaces in the real multi-surface case (Walton, 1980). In ddition, angle factors do not need to be determined in the valuation of a two-surface enclosure. The MRT equation may e written as:

$$q_r = \sigma F_e \left[T_p^4 - T_r^4\right] \qquad (2)$$

where

q_r = net radiation heat transfer at the panel surface from/to room surfaces, Btu/h · ft²

T_p = effective surface temperature of heated (cooled) panel, °R

T_r = temperature of the fictitious surface (unheated or uncooled), °R

σ = Stefan-Boltzman constant, 0.1714×10^{-8} Btu/h · ft² · °R

The temperature of the fictitious surface is given by an mittance weighted average of all surfaces other than the panel.

$$T_r = \sum_{j\neq p}^{n} A_j \epsilon_j T_j \Big/ \sum_{j\neq p}^{n} A_j \epsilon_j \qquad (3)$$

where

A_j = area of surfaces other than panel

ϵ_j = surface emittances other than panel (dimensionless)

T_j = effective surface temperature of unheated (uncooled) surfaces, °R

When the emittances of an enclosure are closely equal and urfaces exposed to the panel are marginally unheated uncooled), then Equation (3) becomes the area-weighted verage temperature (AUST) of unheated (or uncooled) surfaces exposed to the panels.

The radiation interchange factor or Hottel equation for two-urface radiation heat exchange is given by:

$$F_e = \frac{1}{1/F_{p\text{-}r} + \left[(1/\epsilon_p) - 1\right] + A_p/A_r\left[(1/\epsilon_r) - 1\right]} \qquad (4)$$

where

F_e = radiation interchange factor (dimensionless)

$F_{p\text{-}r}$ = radiation angle factor from panel to fictitious surface (1.0 for flat panel)

A_p, A_r = area of panel surface and fictitious surface, respectively

ϵ_p, ϵ_r = emittance of panel surface and fictitious surface, respectively (dimensionless)

n practice, the emittance of nonmetallic or painted metal non-eflecting surfaces is about 0.9. When this emittance is used in Equation (4), the combined factor is about 0.87 for most rooms.

Substituting this value in Equation (2), the constant becomes bout 0.15. Min *et al.* (1956) showed that this constant was 0.152 in their test room. The radiation equation for heating or cooling becomes:

$$q_r = 0.15 \cdot 10^{-8}\left[\left(t_p + 460\right)^4 - \left(AUST + 460\right)^4\right] \qquad (5)$$

where

t_p = effective panel surface temperature, °F

AUST = area-weighted average temperature of uncontrolled surfaces in the room, °F

According to Equation (5), q_r is positive in heating and negative in cooling. This also establishes the general sign convention for this chapter, which states that any heat flux from the panel is regarded positive. Otherwise, it is negative.

The actual transfer by radiation in a room may be somewhat different from that given by Equation (5) because of non-uniform temperatures, irregular room surfaces, variations in surface emittance of materials, etc. However, the equation is accurate to within 10% when used in conventional heating and cooling calculations. Stemman (1989) found that the accuracy of Equations (2) and (5) was improved by adding back geometric angle factors to these equations. These factors allow the accuracy of the MRT method to approach that of Equation (1) and still maintain a first order temperature solution.

Radiation heat transfer calculated from Equation (5) is given in Figure 1. The values apply to ceiling, floor, or wall panel heat output. Heat removed by a cooling panel for a range of practical temperatures is given in Figure 2.

In many specific instances where normal multi-story commercial construction and fluorescent lighting are used, the room temperature at the 5-ft. level will closely approach the average uncooled surface temperatures (AUST). In structures where the main heat gain is through the walls or where incandescent lighting is used, the wall surface temperatures tend to rise considerably above the room air temperature.

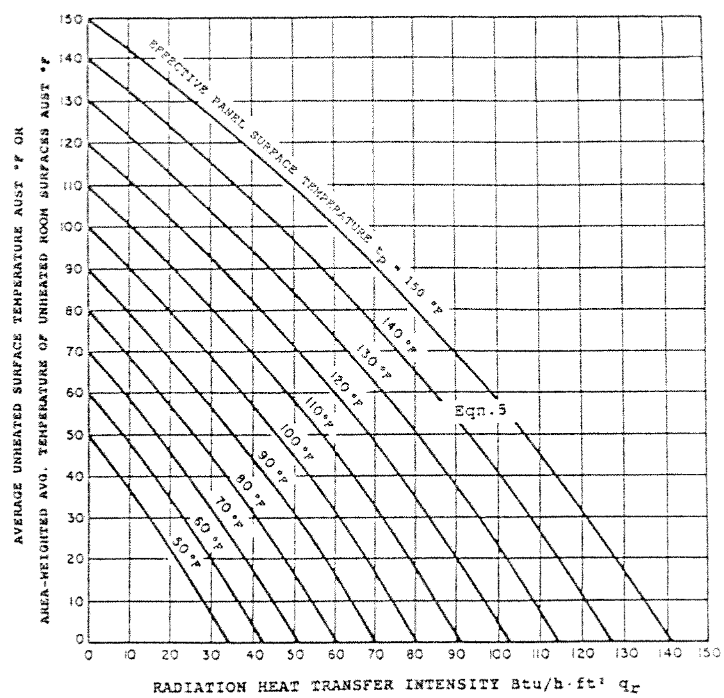


Fig. 1 Radiation Heat Transfer Intensity at Heated Ceiling, Floor, or Wall Panel Surface

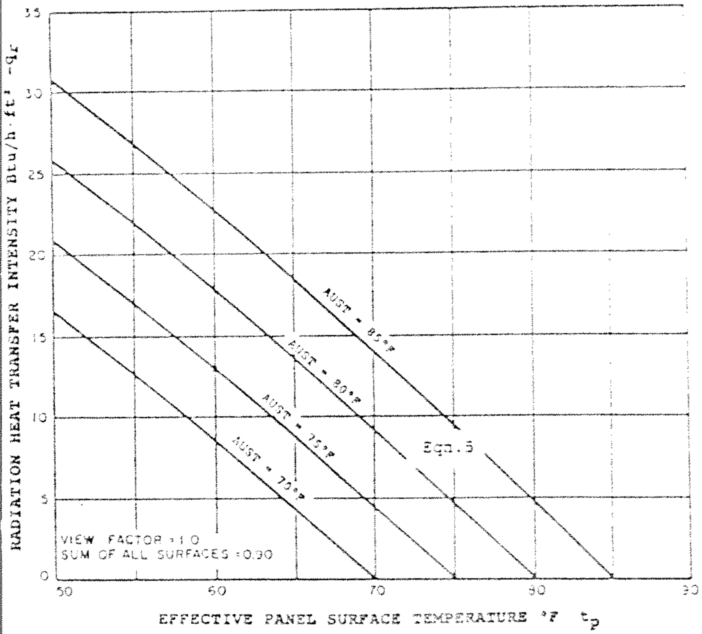


Fig. 2 Heat Removed by Radiation at Cooled Ceiling or Wall Panel Surface

Convection Heat Transfer Intensity

The convection heat transfer intensity q_c is defined as the heat transferred by convection in Btu/h ·ft² · °F difference between air and effective panel surface temperatures. Convection heat transfer values are not easily established. Convection in panel systems is usually natural; that is, air motion is generated by the warming or cooling of the boundary layer of air. In practice, however, many factors such as a room's configuration interfere with or affect natural convection. Infiltration, the movement of occupants, and mechanical ventilating systems can introduce some forced convection that will disturb the natural process.

Parmelee and Huebscher (1947) included the effect of forced convection on heat transfer from panels as an increment to be added to the natural convection coefficient. Harris and Sartain (1959) proposed a similar approach after observing rather strong downwash of air near cold window surfaces. However, increased heat transfer from forced convection should not be systematically used because the increments are unpredictable in pattern and performance, and forced convection does not significantly increase the total capacity of the panel system—except in rooms with large glass areas.Convection in a panel system is proportioned to the difference between the effective panel surface temperature and the temperature of the airstream layer directly wetting the panel. The most consistent measurements are obtained when the air layer temperature is measured close to the region where the fully developed stream begins, usually 2 to 2.5 in. away from the panel surface.

Min *et al.* (1956) determined natural convection coefficients 5 ft. above the floor in the center of a 12 ft. by 24 1/2 ft. room.

Natural convection from an "all-heated" ceiling,

$$q_c = 0.041 \left(t_p - t_a\right)^{1.25} / D_e^{0.25} \qquad (6)$$

Natural convection from a heated floor or cooled ceiling,

$$q_c = 0.39 \left(t_p - t_a\right)^{1.31} / D_e^{0.08} \qquad (7)$$

Natural convection from a heated or cooled wall panel,

$$q_c = 0.29 \left(t_p - t_a\right)^{1.32} / H^{0.05} \qquad (8)$$

where

q_c = heat transfer intensity by natural convection, Btu/h · ft²
t_a = temperature of airstream layer, °F
D_e = equivalent diameter of panel (4 area/perimeter), ft
H = height of wall panel, ft

Schutrum and Humphreys (1954) measured panel performance in furnished test rooms that did not have uniform temperature surfaces and found no variations large enough to be significant in heating practice. Schutrum and Vouris (1954) established that the effect of room size was also usually insignificant. Very large spaces like hangars and warehouses are exceptions. In this case, for the natural convection heat transfer intensity q_c, Equations (6) and (7), should be used. Otherwise, convection equations can be simplified to:

Natural convection from an all-heated ceiling,

$$q_c = 0.020 \left(t_p - t_a\right)^{1.25} \qquad (9A)$$

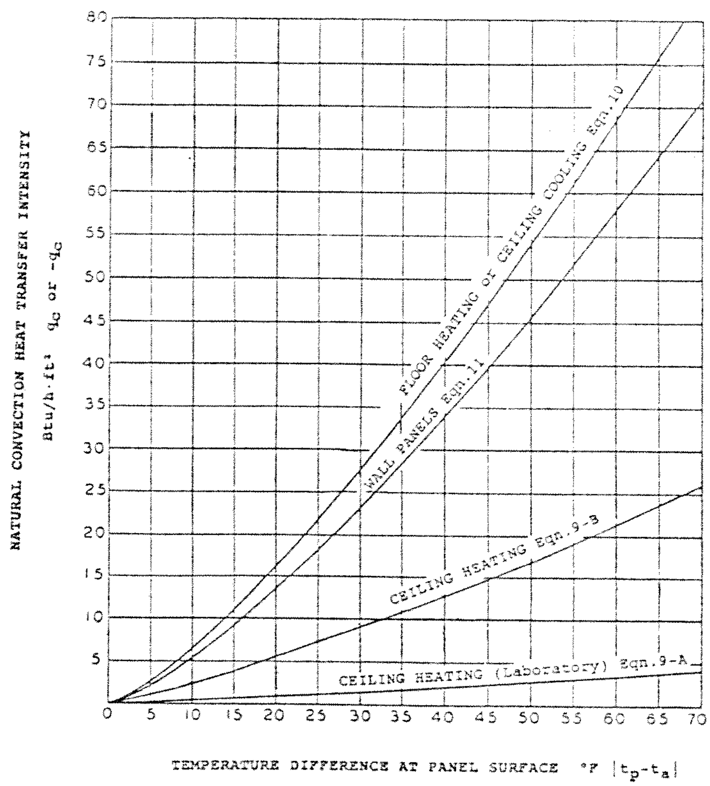


Fig. 3 Natural Convection Heat Transfer Intensity at Floor, Ceiling, and Wall Panel Surfaces [Equations (9) (10) and (11)]

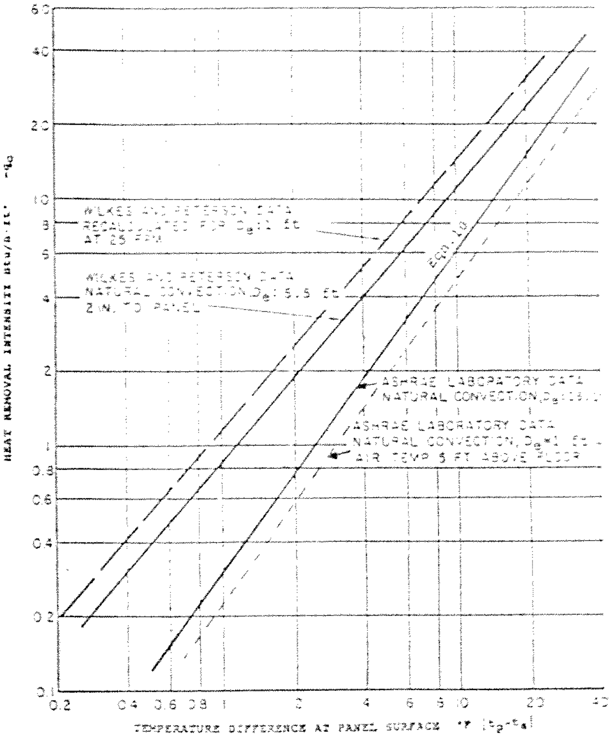


Fig. 4 Heat Removal by Ceiling Cooling Panels with Natural Convection [Equation (10)]

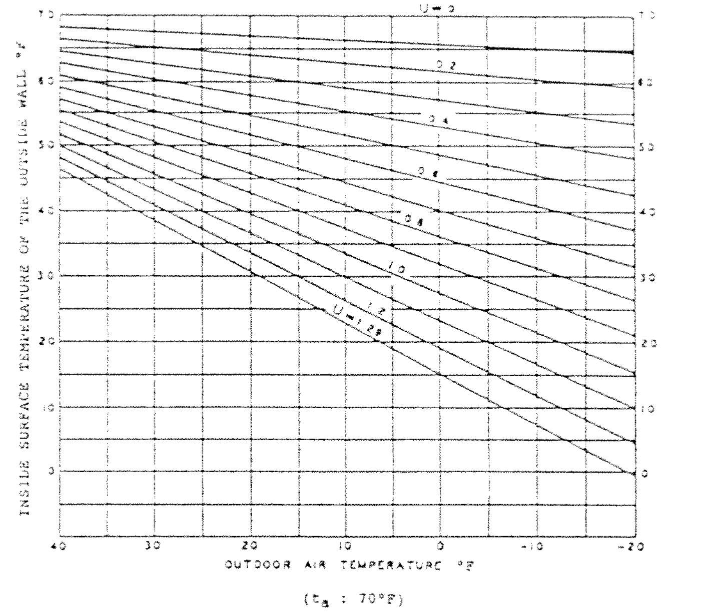


Fig. 5 Relation of Inside Surface Temperature to Overall Coefficient of Heat Transfer

Natural convection from a heated ceiling may be enhanced by leaving "cold strips" (unheated ceiling sections) between the ceiling panels. These strips help in initiating the natural convection (Raiss and Roedler, 1958). In this case, Equation (9A) may be replaced by the following equation (Kollmar and Liese, 1957):

$$q_c = 0.13\left(t_p - t_a\right)^{1.25} \qquad (9B)$$

For large spaces, this equation should be adjusted with the coefficient: $(16.1/D_e)^{0.25}$

Natural convection from a heated floor or cooled ceiling,

$$q_c = 0.31\left(t_p - t_a\right)^{1.31} \qquad (10)$$

Natural convection from a heated or cooled wall panel,

$$q_c = 0.26\left(t_p - t_a\right)^{1.32} \qquad (11)$$

Under normal conditions t_a can be taken as the indoor design temperature. In spaces with large glass areas, t_a may be taken equal to AUST (Drake, 1993).

In cooling t_p is less than t_a, therefore, q_c is also negative. Figure 3 shows heat transfer by natural convection at floor, wall, and ceiling heating panels as calculated from Equations (9A) (9B) (10) and (11).

Figure 4 compares heat removal by natural convection at cooled ceiling panel surfaces, as calculated by Equation (10) and data from Wilkes and Peterson (1938) for specific panel sizes. An additional curve illustrates the effect of forced convection on the latter data. Similar adjustment of the ASHRAE data is inappropriate, but the effects would be much the same.

Combined Heat Transfer Intensity

The combined heat transfer intensity on the panel surface can be determined by adding the radiant heat transfer from Figure 1

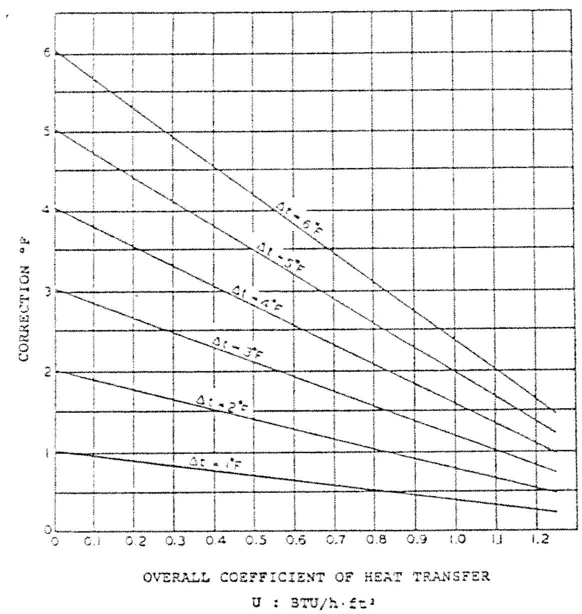


Fig. 6 Inside Wall Surface Temperature Correction for Air Temperatures Other than 70°F

or 2 to the convective heat transfer from Figure 3 or 4, respectively. Alternatively, equations for q_r and q_c can be employed, so that:

$$q = q_r + q_c \qquad (12)$$

Following is the terminology for q:

- = q_u in floor heating (heat output)
- = q_d in ceiling heating (heat output)
- = $-q_d$ in ceiling cooling (heat removal)
- = q_s in wall heating (heat output)
- = $-q_s$ in wall cooling (heat removal)

calculating the AUST in the room. In calculating the AUST, the surface temperature of the interior walls is assumed to be the same as the room air temperature. The inside surface temperatures of outside walls and exposed floors or ceilings can be obtained from Figure 5 for a 70°F room temperature. Figure 6 gives corrections for other temperatures.

Tests by Schutrum *et al.* (1953a, 1953b), and simulation by Kalisperis (1985) based on a CAD program developed by Kalisperis and Summers (1985), show that the temperatures are almost equal. Steinman *et al.* (1989) noted that these temperatures may not be appropriate for enclosures with large glass areas or a high percentage of outside wall and/or ceiling surface area. These surfaces decrease (increase in cooling) AUST, which increases the heat transfer by radiation.

Figure 7 shows the combined radiation and convection heat removal intensity for cooling, as given in Equations (5) and (10). The data in Figure 7 do not include heat gains from sun, lights, people, or equipment: manufacturer's data includes these heat gains.

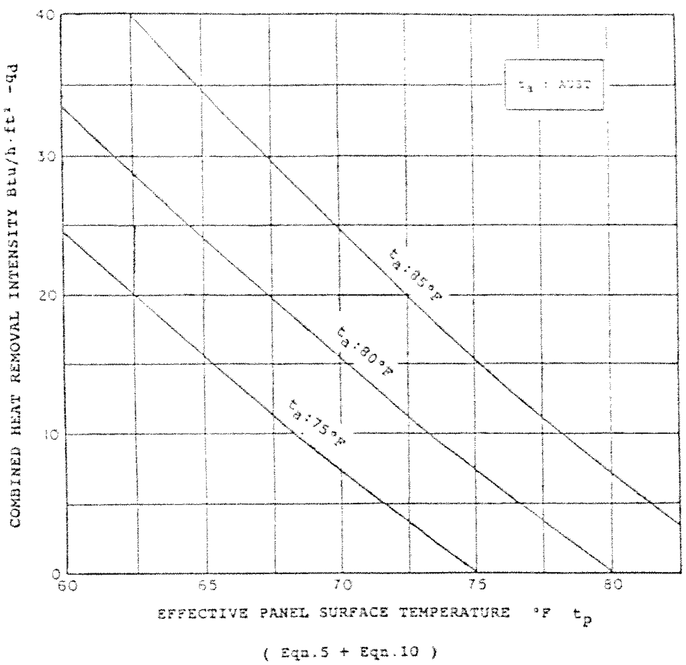


Fig. 7 Cooled Ceiling Panel Performance in Uniform Environment with No Infiltration and No Internal Heat Sources

Panel Thermal Resistance

Heat transfer in the panel to the surface is an important consideration. Any hindrance in between will affect the efficiency of the system. The thermal resistance to heat flow may vary considerably among panel systems, depending on the type of bond between the tubing (or wiring) and the panel material. Corrosion or adhesion defects between lightly touching surfaces, the method of maintaining contact, and other factors may change the bond with time. The actual thermal resistance of any proposed system should be verified by testing whenever practical. Specific resistance and performance data sh be obtained from the manufacturer, whenever available. Basically, there are four types of thermal resistance:

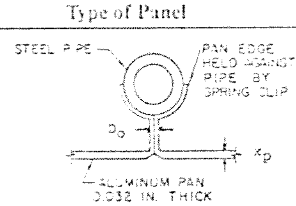
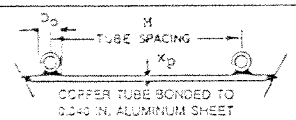
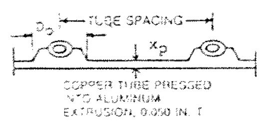
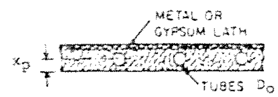
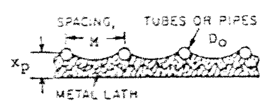
- Thermal resistance of the tube wall in a hydronic syste
- Thermal resistance between the tube (electric wire) an the panel material, r_s
- Thermal resistance of the panel material, r_p
- Thermal resistance of the panel covers, r_c

The sum of these thermal resistances is the panel thermal resistance, r_u.

$$r_u = r_t + r_s + r_p + r_c \tag{13}$$

Generally, if the tubes or electric cables are embedded in th slab, r_s may be neglected. However, if the tubes are attach the panel, r_s may be significant depending upon the quality bonding. Table 1 gives typical r_s values for various ceiling panels.

Table 1. Thermal Resistance of Ceiling Panels

Type of Panel	Thermal Resistance ft²·°F·h/Btu r_p	r_s
	x_p/k_p	0.20
	x_p/k_p	0.05
	x_p/k_p	0.05
	$(x_p-D_o/2)/k_p$	≈0
	$(x_p-D_o/2)/k_p$	≤0.05

r_p may be calculated if the characteristic panel thickness (ft) and the thermal conductivity of the panel material, k_p (Btu/h · ft· °F) are known.

If the tubes (electric wires) are embedded in the panel:

$$r_p = \frac{(x_p - D_o/2)}{k_p} \tag{14A}$$

Here, D_o is the outer diameter of the tube (electric cable) Hydronic floor heating by a heated slab or gypsum-plaste ceiling heating are typical examples.

the tubes are attached to the panel:

$$r_p = \frac{x_p}{t_p} \tag{14B}$$

letal ceiling panels (see Table 1) and tubes under subfloor (see igure 20) are typical examples.

The following expression is a simplified version of the hermal resistance equation for a tube with circular cross ection and a thermal conductivity of k_t (Btu/h · ft · °F).

$$r_t = \frac{0.116 \cdot D_o/D_i - 0.107}{k_t} \quad (1.1 \le D_o/D_i \le 2) \tag{15}$$

where D_i is the inside diameter of the tube, in ft. In metallic ubes, r_t may be ignored.

ecently, the use of nonmetallic tubes has increased. k_t values of different tubing material is given in Table 2. Long engths are available up to 600 ft, which simplify installation. ome manufacturers have also developed splicing or fusion echniques for shorter tube lengths.

able 2. Thermal Conductivity of Typical Tube Material

laterial	k_t (Btu/h · ft · °F)
teel	48.50
opper	17.00
tainless Steel	9.20
ow density polyethylene (LDPE)	0.18
ligh density polyethylene (HDPE)	0.24
ross-linked polyethylene (VPE)	0.22
extile reinforced rubber heat transfer hose (HTRH)	0.17
olypropylene block co-polymer (PP-C)	0.13
olypropylene random co-polymer (PP-RC)	0.14
olybutylene (PB)	0.13

Panel coverings like carpets and pads on the floor can have a ronounced effect on the performance of the panel systems. he added thermal resistance r_c reduces the panel surface heat ansfer. In order to re-establish the required performance, the emperature of the water must be increased (decreased in ooling). Thermal resistance of a panel covering is:

$$r_c = \frac{x_c}{k_c} \tag{16}$$

Here, x_c and k_c is the thickness (ft) and thermal conductivity Btu/h · ft · °F) of each panel covering. If there is more than one overing, individual r_c values should be added. Table 3 gives ypical r_c values for floor coverings.

'anel Heat Losses or Gains

Heat transfer from or to (in cooling) the back and edges of anels is considered a panel loss (gain). Panel heat losses gains) are part of the building load if the heat transfer exceeds he building. Then these losses should be considered in sizngthe boiler (chiller). If the heat transfer is limited to another onditioned space, the panel loss (gain) is a source of heat cool) for that space instead. In either case, the magnitude

Table 3. Thermal Resistance of Floor Coverings

Description	Thermal resistance, r_c ft² · °F · h/Btu
Bare concrete, no covering	0.00
Asphalt tile	0.05
Rubber tile	0.05
Light carpet	0.60
Light carpet with rubber pad	1.00
Light carpet with light pad	1.40
Light carpet with heavy pad	1.70
Heavy carpet	0.80
Heavy carpet with rubber pad	1.20
Heavy carpet with light pad	1.60
Heavy carpet with heavy pad	1.90
3/8" hardwood	0.54
5/8" wood floor (oak)	0.57
1/2" oak parquet and pad	0.68
Linoleum	0.12
Marble floor and mudset	0.18
Rubber pad	0.62
Prime urethane underlayment, 3/8"	1.61
48-oz. waffled sponge rubber	0.78
Bonded urethane, 1/2"	2.09

of panel loss should be determined. Panel loss (gain) per square ft. of panel area is the panel heat loss (gain) intensity, q_b. Following the sign convention adopted, it is negative in cooling.

Panel heat loss (gains) should be kept to a reasonable amount by insulation. For example, a floor panel may overheat the basement below, and a ceiling panel may cause the temperature of a floor surface above it to be too high for comfort unless it is properly insulated.

In suspended ceiling panel systems, heat is transferred from the ceiling panel to the floor slab above (heating) and vice versa (cooling). This decreases the desired performance. For example, the ceiling panel surface temperature is affected because of heat transfer to or from the panel and the slab by radiation and, to a much smaller extent, by convection. The radiation component can be approximated using Figure 1. The convection component can be estimated from Figure 3 or 4. In this case, the temperature difference is that between the top of the ceiling panel and the mid-space of the ceiling. The temperature of the ceiling space should be determined by testing, since it varies with different panel systems. However, much of this heat transfer is nullified when insulation is placed over the ceiling panel, which, for perforated metal panels, also provides acoustical control.

If lighting fixtures are recessed into the suspended ceiling space, radiation from the top of the fixtures raises the overhead slab temperature and transfers heat to the space by convection. This energy is absorbed at the top of the cooled ceiling panels by radiation, as in Figure 2, and by convection, generally in accordance with Equation (9A). The amount of heat the top of the panel absorbs depends on the system. Most manufacturers have information available. Similarly, panels installed under a roof absorb additional heat, depending on configuration and insulation.

The heat loss from most panels can be calculated by using the coefficients given in Chapter 22 of the 1993 ASHRAE *Handbook—Fundamentals*. These coefficients should not be used to determine the downward heat loss from panels built on grade because the heat flow from them is not uniform (Sartain and Harris, 1956; ASHAE 1956, 1957). The heat loss from panels built on grade can be estimated from Figure 8 or from data in Chapter 25 of the 1993 ASHRAE *Handbook—Fundamentals*.

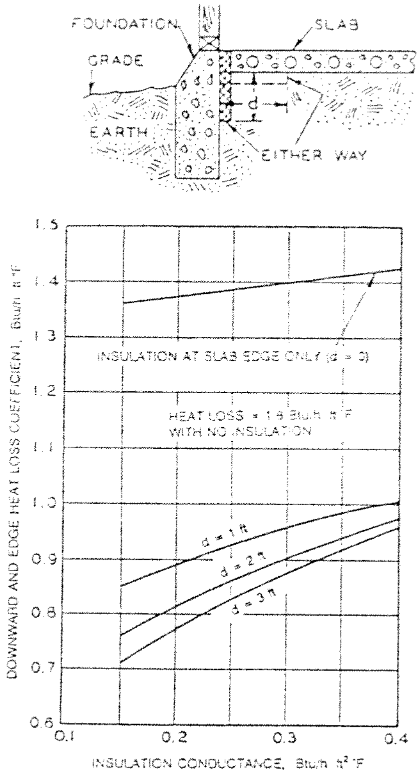


***Fig. 8** Downward and Edgewise Heat Loss Coefficient for Concrete Floor Slabs on Grade*

DESIGN OF RADIANT PANELS

The control of the panel surface temperature is accomplished by either hydronic or electric circuits. The required effective surface temperature, t_p, can be calculated by using applicable heat transfer equations for q_r and q_c. At a given AUST and t_a, it can be directly obtained from Figure 9, on its right hand vertical axis. Either way, AUST must be determined or assumed first. In a hydronic radiant panel, the mean water (brine) temperature necessary to maintain a required heat transfer intensity primarily depends upon the required effective surface temperature, the tube spacing and the panel resistance. Mean water (brine) temperature (t_w) readings in Figure 9 will correspond to skin temperature of the cable in electric resistance heating. After t_p is determined, the following simplified algorithm (Kilkis and Sager, 1993, TS 10464, 1993), which was used to generate Figure 9, may be directly used:

$$t_d \cong t_a + \frac{(t_p - t_a) \cdot M}{(2 \cdot W \cdot \eta + D_o)} + q \cdot (r_p + r_c + r_s) \qquad (17)$$

where

t_d = the mean skin temperature of the tubing (electric cable), °F

q = combined heat transfer intensity on the panel surface, Btu/h · ft^2

t_a = indoor design temperature, °F

D_o = outer diameter of the tube, in ft, or characteristi contact width of the tube with the panel (see Table 1).

M = spacing of the circuit on centers, ft

$2 \cdot W$ = net spacing between tubing (electric cables), $(M - D_o)$, ft

η = fin efficiency, dimensionless

$$\eta = \frac{\tanh(f \cdot W)}{(f \cdot W)} \qquad (18)$$

$$\eta \cong 1/(f \cdot W) \qquad \textit{if } (f \cdot W) > 2$$

f : fin coefficient, ft^{-1}

$$f \cong \left[\frac{q}{(t_p - t_a) \cdot 2 \cdot \sum_{i:1}^{n} \cdot k_i \cdot x_i}\right]^{1/2} \qquad t_p \neq t_a \qquad (19)$$

Here, n is the number of layers with different materials, including the panel itself. k_i and x_i are the thermal conduct (Btu/h · ft · °F) and characteristic thickness (ft) of each laye

For a hydronic system, the required mean water (brine) temperature is:

$$t_w = (q + q_b) \cdot M \cdot r_t + t_d \qquad (20)$$

Here, q_b is the intensity of the back and edge heat losses ir heated panel (positive) or gains (negative) in a cooled pan These equations may be easily adapted for a small comput program.

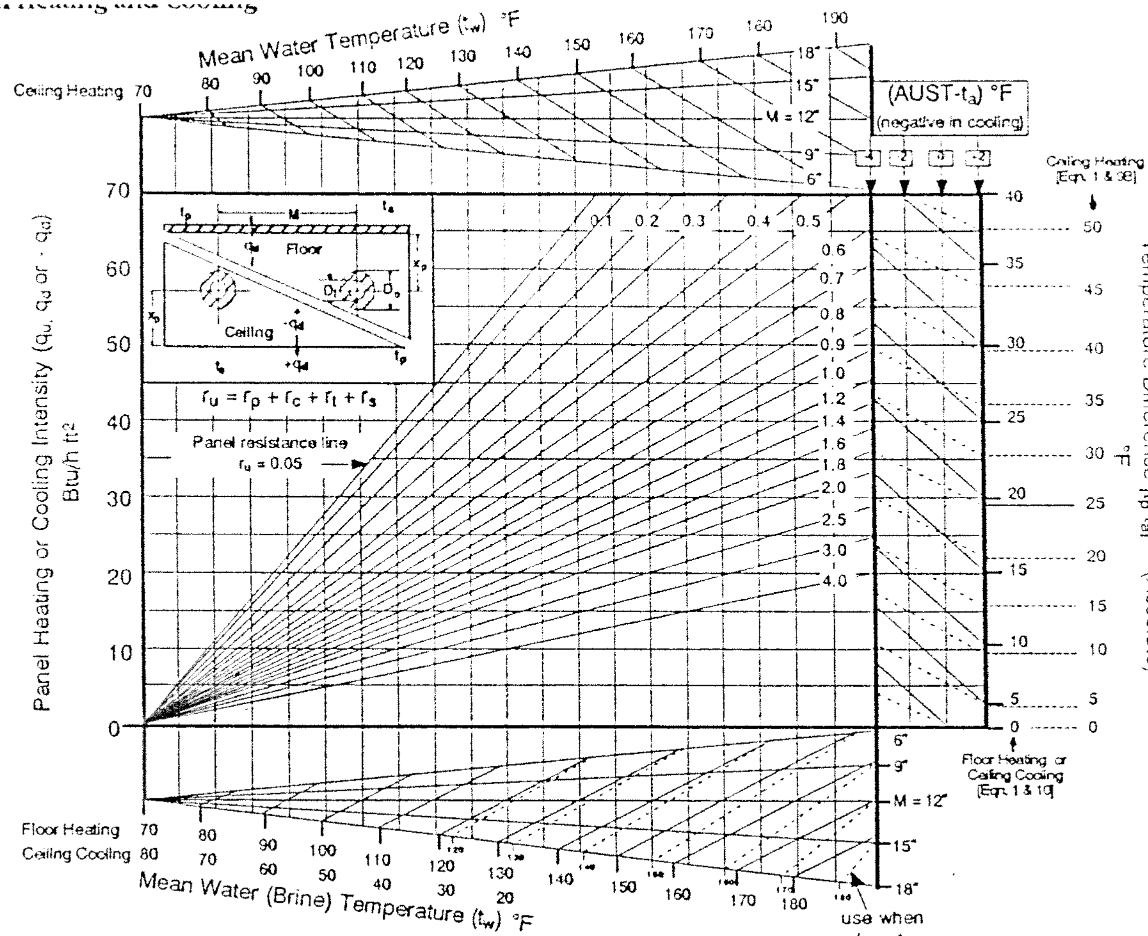
GENERAL DESIGN CONSIDERATIONS

The sensible heating and cooling loads are calculated in similar way for conventional systems. However, it should b noted that:

- Indoor design air temperature may be chosen lower (i heating) or higher (in cooling) for an equivalent huma thermal comfort. See Chapter 51 in ASHRAE *Handboo* 1995 *Applications*, and Buckley, *et. al.*, (1987).
- Natural infiltration rate may be lower due to a more uniform air temperature, pressure, and velocity field maintained in the conditioned space.
- Peak loads are leveled (shaved) to the extent of therm energy stored in the panel, walls, and partitions expos to it.

Because the mean radiant temperature (MRT) within a p heated space increases as the heating load increases, the controlled air temperature during this increase may be low without affecting comfort. In ordinary structures with norm infiltration loads, the required reduction in air temperature small—enabling a conventional room thermostat to be use

In panel heating systems, lowered night temperature can duce less satisfactory results with heavy panels such as concrete floors. These panels cannot respond to a quick increase or decrease in heating demand within the relativel



ig. 9 Combined Nomograph For Cooling and Heating Panel Design

ıort time required, resulting in a very slow reduction of the ɔace temperature at night and a correspondingly slow pickup ı the morning. Light panels, such as plaster or metal ceilings ıd walls, may respond to changes in demand quickly enough ɔr satisfactory results from lowered night temperatures. Tests n a metal ceiling panel demonstrated the speed of response to e comparable to that of convection systems. However, very tle fuel savings can be expected—even with light panels— nless the lowered temperature is maintained for long periods. reduced non-occupancy temperatures are employed, some ıeans of providing a higher-than-normal rate of heat input for ıpid warm-up is necessary; for example, fast-acting radiant eiling panels (Berglund, 1982).

Metal radiant heating panels, hydronic and electric, are ɔplied to building perimeter spaces for heating in much the ıme way as finned-tube convectors. Metal panels are usually ıstalled in the ceiling and are integrated into the ceiling design. he layout and arrangement of panels usually considers ːchitectural design.

Partitions may be erected to the face of hydronic panels, but ot to the active heating portion of electric panels because of ossible element overheating and burnout. Electric panels are ften sized to fit the building module with a small removable ller or dummy panel at the window mullion to accommodate ıture partitions. Hydronic panels can run continuously.

By cutting and fitting, field modification of hydronic panels is ossible; however, modification should be kept to a minimum ɔ keep installation costs down. Electric panels cannot be ıodified in the field.

Thermal expansion or contraction of panels, and other material in or adjacent to it, should be anticipated. Mean water temperature in hydronic systems or the cable temperature in electrically-heated systems must be selected according to the temperature limits of the materials in contact with the panel.

Oxygen-induced or bacteriological corrosion in hydronic systems may deteriorate the performance. Suitable additives should be considered.

Placing the thermostat on a side wall where it can see the outside wall and the panel should be considered. The normal thermostat cover reacts to the panel, and the radiant effect of the panel on the cover tends to alter the control point so that the thermostat controls 2 to 3°F lower when the outdoor temperature is at a minimum and the panel temperature is at a maximum in heating. Experience indicates that rooms heated with radiant panels are more comfortable under these conditions than when the thermostat is located on a back wall.

Floor panels are limited to surface temperatures of up to 84°F for comfort reasons in frequently occupied surfaces. (ASHRAE *Standard* 52-1992). For surfaces with little or no occupation, like perimeter bands not extending more than 3 ft. into the room, this limit is 95°F (DIN 4725, 1990).

In applications with normal ceiling heights, heating panels that exceed 160°F should not be located over the occupied area. Refer to Figure 15 for selecting effective ceiling surface temperatures.

Other factors to consider when using panel cooling systems:

- Evaluate the panel system to take full advantage of optimizing the physical building design.

- Select recessed lighting fixtures, air diffusers, hung ceilings, and other ceiling devices to provide the maximum ceiling area possible for use as ceiling panels.
- The air-side design must be able to maintain humidity levels at or below design conditions at all times to eliminate any possibilities of condensation on the panels. This becomes more critical if space dry- and wet-bulb temperatures are allowed to drift as an energy conservation measure, or if duty cycling of the fans is used.
- Do not place cooling panels in or adjacent to high humidity areas.

HYDRONIC PANEL SYSTEMS

Design Considerations

Hydronic radiant panel systems are similar to other air-water systems in the arrangement of the system components.

Chapter 3 and other chapters in this handbook covering hydronic systems apply to radiant panels. Layout and design of radiant panels for heating and cooling begins early in the job. The type of panel location type and construction chosen influences the design and, conversely, thermal considerations may dictate what type to be used.

Hydronic radiant panels can be used with two- and four-pipe distribution systems. Figure 10 shows the arrangement of a typical system. It is common to design for a 20°F temperature drop for heating across a given grid and a 5°F rise for cooling, but other temperature differentials may be used, if applicable.

Panel design requires determination of the panel area, panel type, supply water temperature, water flow rate, pressure drop, boiler (chiller) load, and the panel arrangement. Panel performance is directly related to room conditions. Air-side design must also be established. Heating and cooling loads may be calculated by procedures covered in Chapters 22 through 27 in the 1993 ASHRAE *Handbook—Fundamentals*. Provisions as listed in General Design Considerations, if applicable, may be applied. The general design procedure is as follows.

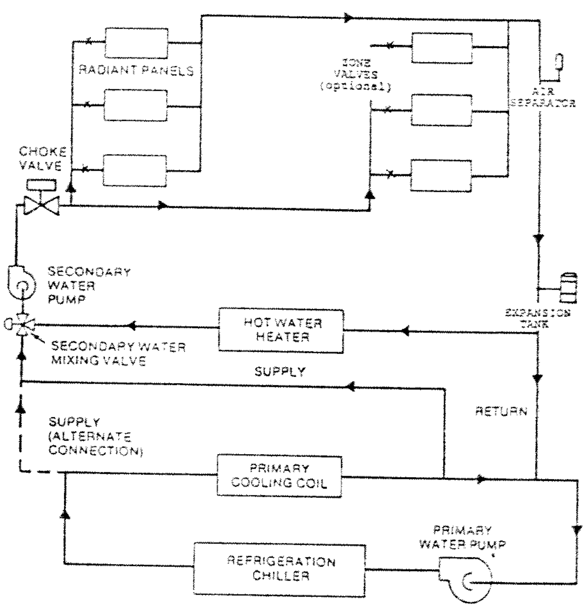


Fig. 10 Primary/Secondary Water Distribution System with Mixing Control

For cooling:

1. Determine room design dry-bulb temperature, rela humidity, and dew point. Determine surface tempe tures of unheated surfaces.
2. Calculate room sensible and latent heat gains.
3. Establish minimum supply air quantity.
4. Calculate latent cooling available from the air.
5. Calculate sensible cooling available from the air.
6. Determine panel cooling load.
7. Determine panel area for cooling.
8. Determine required effective surface temperature o panel. Refer to Figure 9, or use the equations.
9. Select the spacing of cooling circuits.
10. Determine mean water (brine) temperature for coo

For heating:

1. Designate room design dry-bulb temperature for he
2. Calculate room heat loss.
3. Determine panel area for heating.
4. Determine surface temperatures of unheated surfac Refer to Figures 5 and 6 to find surface temperature exterior walls and exposed floors and ceilings. Inte walls are assumed to have surface temperatures eq the room air temperature. Calculate AUST.
5. Design the panel arrangement.
6. Select the spacing of heating circuits.
7. Determine the mean water temperature.
8. Thermal comfort requirements should be checked i the following steps (see Chapter 8 of the 1993 ASHI *Handbook—Fundamentals*, and NRB, 1981).
   a. Determine occupant's clothing value (clo value) a metabolic rate (MET). (See Tables 1D and 4A, Ch of the 1993 ASHRAE *Handbook—Fundamentals*)
   b. Determine the optimum operative temperature at coldest point in the room (see Figure 15, Chapter the 1993 ASHRAE *Handbook—Fundamentals*, o Figure 2.3.22 for other values).
   c. Determine the mean radiant temperature (MRT) a coldest point in the room (see Fanger, 1972).
   d. From the definition of operative temperature, esta the optimum room design temperature at the colc point in the room. If the optimum room design te ature varies greatly from the designated room des temperature, designate a new temperature.
   e. Determine the MRT at the hottest point in the roo
   f. Calculate the operative temperature at the hottest in the room.
   g. Compare the operative temperatures at the hottes coldest points in the room. For light activity and n clothing, the acceptable operative temperature ra 68 to 75°F (see NRB, 1981, for other ranges). If the range is not acceptable, the heating system must b modified.
   h. Calculate radiant temperature asymmetry (NRB, 1 Acceptable ranges are <18 °F for windows and < 9 for warm ceilings.
9. Determine water flow rate and pressure drop. Refer manufacturer's specifications for the pressure drop characteristics of specific products. Pressure drop ir

parallel circuits of uneven lengths tend to balance (Kilkis, 1993-b; Hansen, 1985). Pump sizing should be made accordingly.

The application, design, and installation of hydronic panel stems have certain requirements and techniques such as the llowing:

1. As with any hydronic system, look closely at the tubing or piping system design. It should be designed to ensure that water of the proper temperature, and in sufficient quantity, is available to every grid or coil at all times. Reverse-return systems may be considered to minimize balancing problems.
2. Individual panels can be connected for parallel flow using headers, or for sinuous or serpentine flow. To avoid flow irregularities within a header-type grid, the water channel or lateral length should be greater than the header length. If the laterals in a header grid are forced to run in a short direction, this problem can be solved by using a combination series parallel arrangement. Serpentine flow will ensure a more even panel surface temperature throughout the heating or cooling zone.
3. Noises from entrained air, high-velocity, or high-pressure drop devices, or from pump and pipe vibrations must be avoided. Water velocities should be high enough to secure a turbulent flow and to prevent separated air from accumulating and causing air binding. Recommended water velocity range is between 3 ft/sec. and 7 ft/sec. Where possible, avoid automatic air-venting devices over ceilings of occupied spaces.
4. The expansion tank must be adequately sized. An undersized expansion tank may lead to leakages at manifolds and enhance oxygen-induced corrosion of metal surfaces (Metzner, 1986).
5. Design piping systems to accept thermal expansion adequately. Do not allow forces from piping expansion to be transmitted to panels. Thermal expansion of the ceiling panels must be considered.
6. In circulating water systems, rubber, plastic tube, steel, and copper pipe are widely used in ceiling, wall, or floor panel construction. Where coils are embedded in concrete or plaster, no threaded joints should be used for either pipe coils or mains. Steel pipe should be the all-welded type. Copper tubing should be soft-drawn coils. Fittings and connections should be minimized. Changes in direction should be made by bending. Solder-joint fittings for copper tube should be used with a medium temperature solder of 95% tin, 5% antimony, or capillary brazing alloys. All piping should be subjected to a hydrostatic test of at least three times the working pressure. Maintain adequate pressure in embedded piping while pouring concrete.
7. If throttling valve control is used, either the end of the main should have a fixed bypass, or the last one or two rooms on the mains should have a bypass valve to maintain water flow in the main. Thus, when a throttling valve modulates, there will be a rapid response. In hospital applications, valves should be located in the corridor, outside patient rooms.

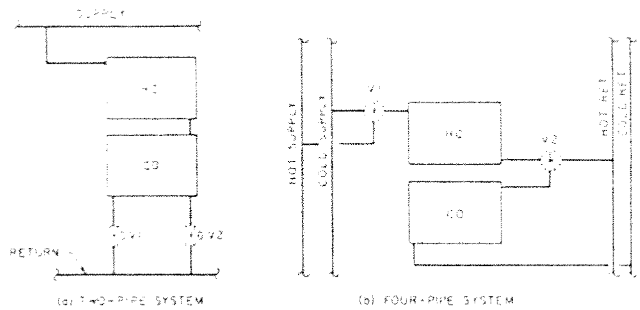


Fig. 11 Split Panel Piping Arrangement for Two-Pipe and Four-Pipe Systems

8. When the panel chilled water system is started, the circulating water temperature should be maintained at room temperature until the air system is completely balanced, the dehumidification equipment is operating properly, and building humidity is at design value.
9. When the panel area for cooling is greater than the area required for heating, a two-panel arrangement (Figure 11) can be used. Panel HC (heating and cooling) is supplied with hot or chilled water year-round. When chilled water is used, the controls function activates panel CO (cooling only), and both panels are used for cooling.
10. To prevent condensation on the room side of cooling panels, the panel water supply temperature should be maintained at least 1°F above the room design dew-point temperature. This minimum difference is recommended to allow for the normal drift of temperature controls for the water and air systems, and also to provide a factor of safety for temporary increase in space humidity.
11. Selection of summer design room dew point below 50°F is generally not economical.
12. The most frequently applied method of dehumidification utilizes cooling coils. If the main cooling coil is six rows or more, the dew point of the air leaving will approach the temperature of the water leaving. The cooling water leaving the dehumidifier can then be used for the panel water circuit.
13. Several chemical dehumidification methods are available to control latent and sensible loads separately. In one application, cooling tower water is used to remove heat from the chemical drying process, and additional sensible cooling is necessary to cool the dehumidified air to the required system supply air temperature.
14. When chemical dehumidification is used, hygroscopic chemical-type dew-point controllers are required at the central apparatus and at various zones to monitor dehumidification.

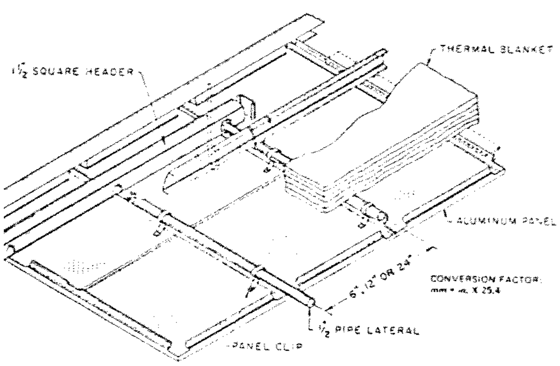
When cooled ceiling panels are used with a variable air volume (VAV) system, the air supply rate should be near maximum volume to assure adequate dehumidification before the cooling ceiling panels are activated.

Design operable windows to discourage unauthorized opening.

Pump and tube sizing must ensure a turbulent flow in the coils.

ronic Ceiling Panels

tal ceiling panels. Metal ceiling panels can be integrated
system that heats and cools. In such a system, a source of
midified ventilation air is required in summer, so the system
sed as an air-water (hybrid) system (Wilkins and Kosonen,
. Also, various amounts of forced air are supplied year-
l. When metal panels are applied for heating only, a
ation system may be required, depending on local codes.
ceiling panel systems are an outgrowth of the perforated
suspended acoustical ceilings. These radiant ceiling sys-
re usually designed into buildings where the suspended
ical ceiling can be combined with panel heating and
ng module, which provides extensive flexibility for zoning
ntrol; or the panels can be arranged as large continuous
or maximum economy. Some ceiling installations require
panels to cover only a portion of the room, and compati-
atching acoustical panels for the remaining ceiling area.
e types of metal ceiling systems are available. The first
s of light aluminum panels, usually 12 in. by 24 in.,
d in the field to 0.5-in. galvanized pipe coils. Figure 12
es a metal ceiling panel system that uses 0.5-in. pipe
, on either 6-, 12-, or 24-in. centers, hydraulically
ted in a sinuous- or parallel-flow welded system.
um ceiling panels are clipped to these pipe laterals and act
ating panel when warm water is flowing, or as a cooling
hen chilled water is flowing.
econd type of panel consists of a copper coil secured to
ninum face sheet to form a modular panel. Modular
re available in sizes up to about 36 in. by 60 in., and are



Metal Ceiling Panels Attached to Pipe Laterals

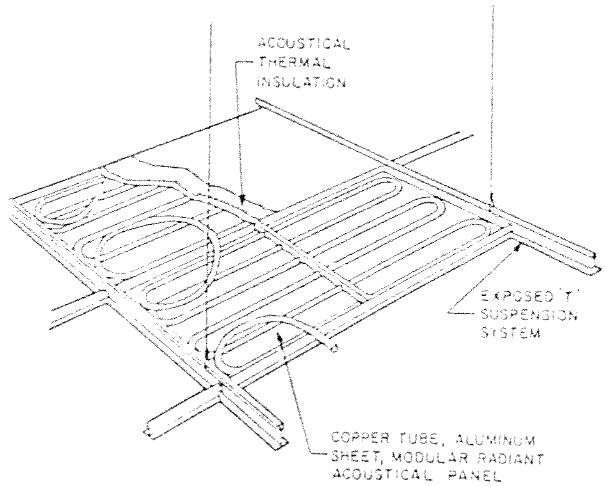


Fig. 13 Metal Ceiling Panels Bonded to Copper Tubing

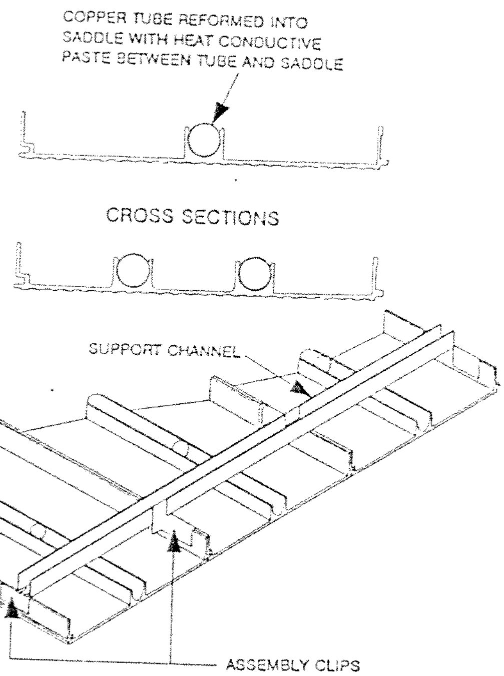


Fig. 14 Extruded Aluminum Panels with Integral Copper Tube

held in position by various types of ceiling suspension systems—most typically, a standard suspended T-bar 24 in. by 48 in. exposed grid system. Figure 13 illustrates a metal panel using copper tubing bonded to an aluminum panel. Metal ceiling panels can be perforated so that the ceiling becomes sound-absorbent when acoustical material is installed on the back of the panels. The acoustical blanket is also required for thermal reasons, so that the reverse loss or upward flow of heat from the metal ceiling panels is minimized.

The third type of panel is an aluminum extrusion face sheet with a copper tube mechanically fastened into a channel housing on the back of the face sheet. Extruded panels can be manufactured in almost any shape and size. Extruded aluminum panels are often used as long, narrow panels at the outside wall, and are independent of the ceiling system. Panels 15 or 20 in. wide usually satisfy the heating requirements of a typical office building. Lengths up to 20 ft. are available. Figure 14 illustrates metal panels using a copper tube pressed into an aluminum extrusion, although other methods of securing the copper tube have proven equally effective.

Performance data for extruded aluminum panels vary with extrusion surface configuration, copper tube/aluminum contact, and test procedures used. Hydronic ceiling panels have a low thermal resistance and respond quickly to changes in space conditions. Table 1 gives the thermal resistance values of various ceiling panels.

Metal radiant ceiling panels can be used with any of the all-air cooling systems described in Chapter 2. Double glazing and heavy insulation in outside walls have reduced transmission heat losses. As a result, infiltration and reheat have become a greater concern.

Additional design considerations are as follows:

1. Locate ceiling panels adjacent to the outside wall and as close as possible to the areas of maximum load. The panel area within 3 ft. of the outside wall should have a heating capacity equal to or greater than 50% of the wall transmission load.
2. Ceiling system designs based on passing return air through perforated modular panels into the plenum space above the ceiling are not recommended, because much of the panel heat will be lost to the return air.
3. When selecting heating design temperatures for a ceiling panel surface temperature, the design parameters are:
   a. Excessively high temperatures over the occupied zone will cause the occupant to experience a "hot head effect".
   b. With normal ceiling heights of 8 to 9 ft., panels less than 3 ft. wide at the outside wall can be designed for 235°F surface temperature. If panels extend beyond 3 ft.. into the room, the panel surface temperature should be limited to the values as given in Figure 15. The surface temperature of concrete or plaster panels is limited by construction.
4. Allow sufficient space above the ceiling for installation and connection of the piping that forms the radiant panel ceiling.
5. Any allowance of unheated ceiling sections, "cold strips", will enhance the heat output by convection. [See Equation (9B) versus (9A).]

Metal radiant acoustic panels provide heating, cooling, sound absorption, insulation, and unrestricted access to the plenum space. They are easily maintained, can be repainted to look new, and have a life expectancy in excess of 30 years. The system is a basic air-and-water system—easy to control, and responds quickly. First costs are competitive with other systems, and a life cycle cost analysis often shows that the long life of the equipment makes it the lowest cost in the long run. The system has been used in hospitals, schools, office buildings, airports, and exposition facilities.

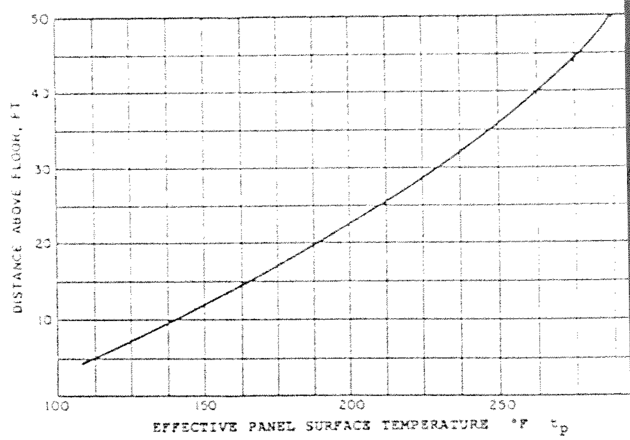


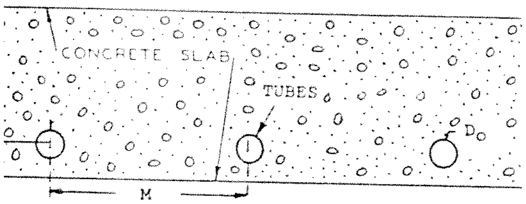
Fig. 15 Suggested Design Ceiling Surface Temperat
at Various Ceiling Heights

Metal radiant panels can also be integrated into the ceili design to provide a narrow band of radiant heating aroun perimeter of the building. The radiant system offers advan over baseboard or overhead air in appearance, comfort, operating efficiency, cost, maintenance, and product life.

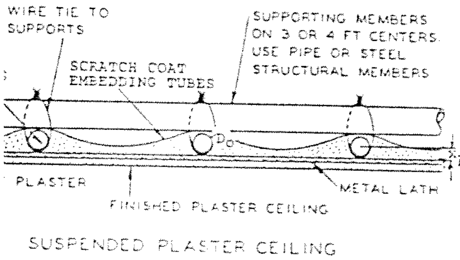
Piping or Tubing Embedded in Ceilings. One of the following types of construction is generally used:

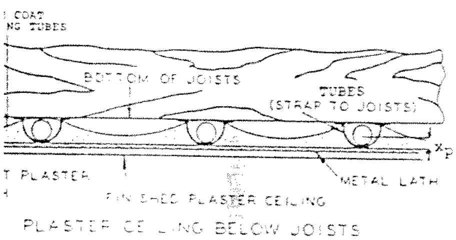
1. Pipe or tube is embedded in the lower portion of a concrete slab, generally within 1 in. of its lower surf. If plaster is to be applied to the concrete, the piping be placed directly on the wood forms. If the slab is t be used without plaster finish, the piping should be installed not less than 0.75 in. above the undersurfa the slab. Figure 16 shows this method of constructio The minimum coverage must comply with local bui code requirements.
2. Pipe or tube is embedded in a metal lath and plaster ceiling. If the lath is suspended to form a hung ceilin the lath and heating coils are securely wired to the supporting members so that the lath is below, but in good contact with, the coils. Plaster is then applied t metal lath, carefully embedding the coil as shown in Figure 17.
3. Smaller diameter copper or plastic tube is attached t underside of wire lath or gypsum lath. Plaster is then applied to the lath to embed the tube, as shown in Figure 18. See also Table 1 for plaster ceiling panels.
4. Other forms of ceiling construction are composition board, wood paneling, etc., with warm water piping tube, or channels built into the panel sections.

Coils are usually the sinuous type, although some heade grid-type coils have been used in ceilings. Coils may be pl rubber tube, ferrous, or nonferrous pipe spaced from 4.5 t in. on centers—depending on the required output, pipe o size, and other factors.

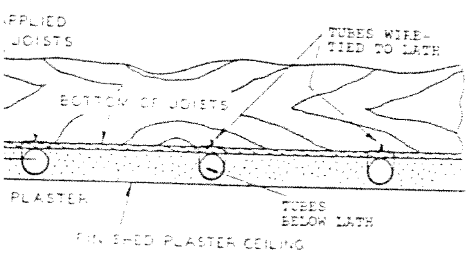


6 Tubes in Structural Concrete Slab





7 Tubes in Plaster Above Lath



Tubes in Plaster Below Lath

'e plastering is applied to coils, a standard three-coat
1 plastering specification is followed—with a minimum
in. of cover below the tubes or pipes when they are
d below the lath. Generally, the surface temperature of
panels should not exceed 120°F. This can be accom-
by limiting the water temperature in the pipes or tubes in contact with the plaster to a maximum temperature of 140°F. Insulation should be placed above the coils to reduce *reverse loss*, the difference between heat supplied to the coil and net useful output to the heated room.

To protect the plaster installation and to ensure proper air drying, heat must not be applied to the panels for two weeks after all plastering work has been completed. When the system is started for the first time, the water supplied to the panels should not be higher than 20°F above the prevailing room temperature at that time, and not in excess of 90°F. Water should be circulated at this temperature for about two days, then increased at a rate of about 5°F per day to 140°F.

During the air drying and preliminary warm-up periods, there should be adequate ventilation to carry moisture from the panels. Paint or paper should not be applied to the panels before these periods have been completed, or while the panels are being operated. After paint and paper have been applied, an additional, shorter warm-up period (similar to first-time starting) is also recommended.

Hydronic Floor Panels

Interest has increased in radiant floor heating with the introduction of non-metallic tubing and new design, application, and control techniques. Whichever method is used for optimum floor output and comfort, it is important that the heat be evenly distributed throughout the floor. Spacing is generally 4- to 12-in. on center for the coils. Wide spacing under tile or bare floors can cause uneven surface temperatures.

Embedded Piping or Tubing in Concrete Slab. Plastic, rubber tube, ferrous, and nonferrous pipe are used in floor slabs, too. The coils are constructed as sinuous-continuous pipe coils, or arranged as header coils with the pipes spaced from 6 to 18 in. on centers. The coils are generally installed with 1.5 to 4 in. of cover above them. Insulation is recommended to reduce the perimeter and back losses. Figure 19 shows the application of pipe coils in slabs resting on grade. Coils should be embedded completely, and should not rest on an interface. Any supports used for positioning the heating coils should be non-absorbent and inorganic. Reinforcing steel, angle iron, pieces of pipe or stone, or concrete mounds can be used. No wood, brick, concrete block, or similar materials should support coils. A waterproofing layer is desirable to protect insulation and piping.

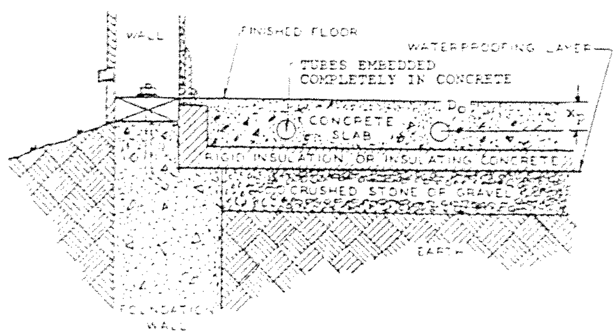


Fig. 19 Tubes in Floor Slab on Grade

Where coils are embedded in structural load-supporting slabs above grade, construction codes may affect their position. Otherwise, the coil piping is installed as described for slabs resting on grade.

The warm-up and start-up periods for concrete panels are similar to those outlined for plaster panels.

Embedded systems may fail at some point during their lifetime. Adequate valves and properly labeled drawings will help to isolate the point of failure.

Suspended Floor Piping or Tubing. Piping or tubing may be applied on or under suspended wood floors using several methods of construction. Coils may be in a layer of concrete or gypsum on the floor, or mounted in or below the subfloor, or attached directly to the underside of the subfloor. Metal plates or batt insulation with a reflective layer will improve heat diffusion in the subfloor. Both alternatives are illustrated in Figure 20. In design calculations, fin coefficient may be taken similar to the case with metal plates if a batt insulation with a reflective layer is used..

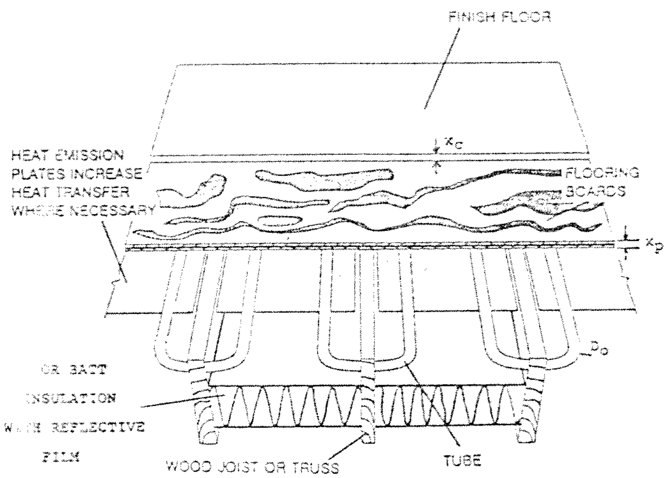


Fig. 20 Tube Under Subfloor

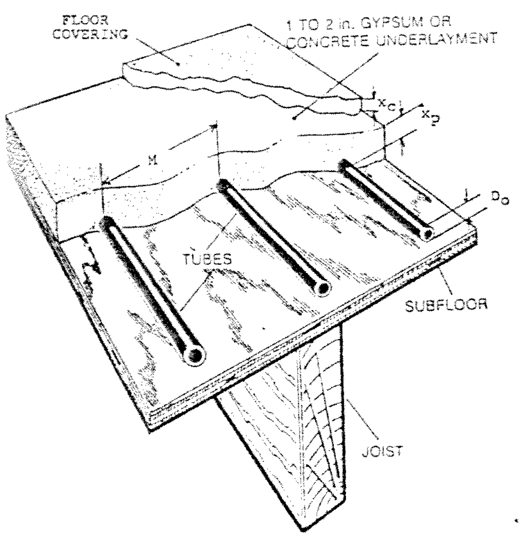


Fig. 21 Embedded Tube in Thin Slab

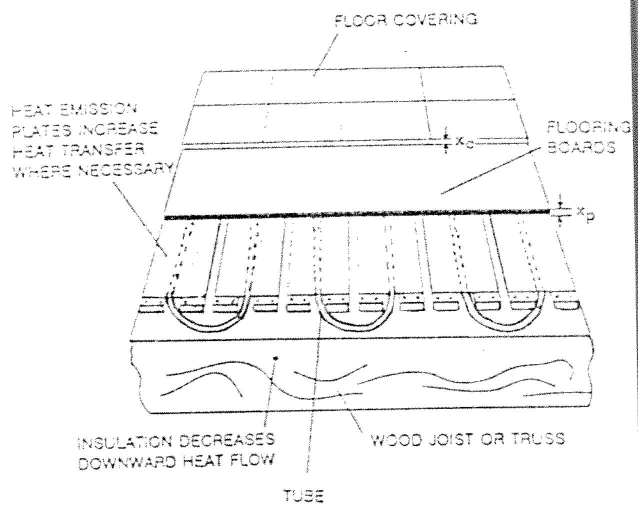


Fig. 22 Tube In Subfloor

Figure 21 illustrates construction with tubing embedded concrete or gypsum. The thickness of the embedding mat is generally 1 to 2 in. when applied to a wood subfloor.

Gypsum products specifically designed for floor heatin generally be installed 1 to 1.5 in. thick because they are m flexible and crack resistant than concrete. When concrete used, it should be of structural quality to reduce cracking to movement of the wood frame or shrinkage. Embeddin material must provide a hard, flat, smooth surface that ca accommodate a variety of floor coverings. The extra struc load due to embedding material must be considered in th design and layout of the subfloor.

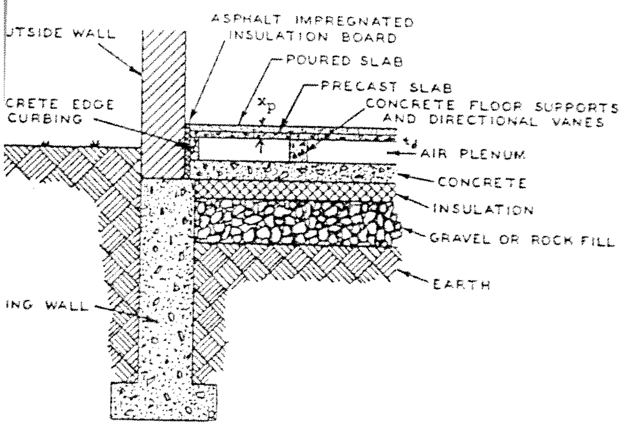
Figure 22 illustrates that tubing may be installed in the subfloor. The tubing is installed on top of the rafters betw the subflooring members. Heat diffusion and surface temperature uniformity can be improved by the addition metal heat transfer plates, which spread the heat beneath finished flooring.

Hydronic Wall Panels

Although piping embedded in walls is not as widely us floor and ceiling panels, it can be constructed by any of th methods outlined for ceilings or floors. Its design is simila other hydronic panels (see Equations 17 to 20). Heat trans the surface of wall panels is given by Equations (5) and (1

AIR-HEATED OR COOLED PANELS

Several methods have been devised to warm or cool int room surfaces by circulating air through passages in the fl walls, or even the ceiling. In some heating cases, the heat is recirculated in a closed system. In others, part or all of t is passed through the room, on its way back to the furnac provide supplementary heating and ventilation. Figure 23 illustrates one common type of construction. Compliance applicable building codes is important.



3 *Air Heated Floor Panel Construction*

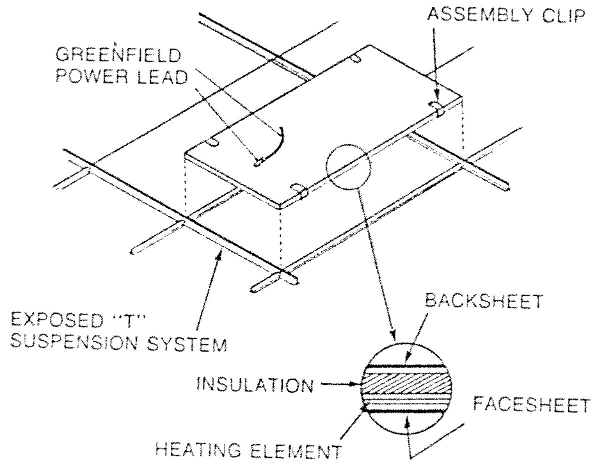


Fig. 24 Electric Heating Panel

rinciple, all the heat transfer equations for the panel e and the design algorithm, explained in the section gn of Radiant Panels", are applicable. In these systems, ver, the fluid (air) moving in the duct has a virtually uous contact with the panel. Therefore, η is about 1,), and M is unity. Equation (17) gives the required skin rature of the plenum, t_d. The design of air side can be l out by following the principles given in Chapters 23 and he 1993 ASHRAE *Handbook—Fundamentals*.

ELECTRICALLY HEATED SYSTEMS

lesign calculations are similar to the hydronic heating is, except the cable layout. Several different forms of resistance units are available for heating interior room es. These include: (1) electric heating cables that may be lded in concrete or plaster or laminated in drywall ceiling iction; (2) prefabricated electric heating panels to be d to room surfaces or floor joist space; and (3) electric- ated fabrics or other materials for application to, or incor- on into, finished room surfaces.

rically Heated Ceilings

abricated electric ceiling panels. A variety of icated electric heating panels are available for either mental or full-room heating. These panels are available from 2 ft. by 4 ft. to 6 ft. by 12 ft. They are constructed variety of materials such as gypsum board, glass, steel iss, or vinyl. Different panels have rated inputs varying) to 95 W/ft^2 for 120, 208, 240, 277, and 347V service. um operating temperatures vary from about 100 to about depending on watt density. National and local codes be followed when placing partitions, lights, and air idjacent to or near electric panels.

l heating elements may be embedded conductors, ed conductive coatings, or printed circuits. Non-heating eads are connected and furnished as part of the panel.

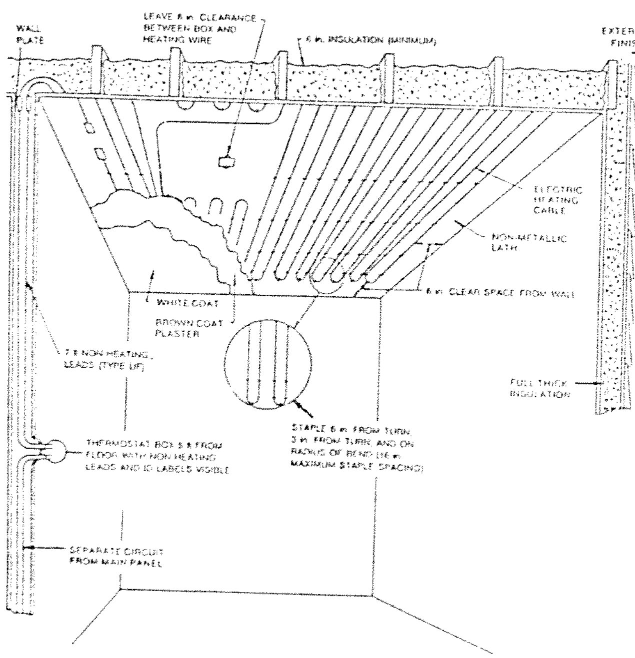


Fig. 25 Electric Heating Panel for Wet Plaster Ceiling

Some panels can be cut to fit available space; others must be installed as received. Panels may be either flush or surface mounted and, in some cases, are finished as part of the ceiling. Rigid panels that are about 1 in. thick, and weigh from 6 lb to about 25 lb for a 2ft. by 4 ft. panel, are usually made for lay-in ceilings (see Figure 24). Panels may also be (1) surface-mounted on gypsum board and wood ceilings, or (2) recessed between ceiling joists. Panels range in size from 4 ft. wide to 8 ft. long. The maximum output is 95 W/ft^2.

Electrical cables embedded in ceilings. Electric heating cables for embedded or laminated ceiling panels are factory-assembled units furnished in standard lengths of 75 to 1800 ft. These cable lengths cannot be altered in the field. The cable assemblies are normally rated at 2.75 W per linear foot and are supplied in capacities from 200 to 5000 W in roughly 200-W increments. Standard cable assemblies are available for 120, 208, 240, and 347V. Each cable unit is supplied with 7-ft. non-heating leads for connection at the thermostat or junction box.

Electric cables for panel heating have electrically-insulated coverings resistant to medium temperature, water absorption, aging effects, and chemical action with plaster, cement, or ceiling lath material. This insulation is normally a polyvinylchloride (PVC) covering which may have a nylon jacket. The outside diameter of the insulation covering is usually about 0.12 in.

For plastered ceiling panels, the heating cable may be stapled to gypsum board, plaster lath, or similar fire-resistant materials with rust-resistant staples (Figure 24). With metal lath or other conducting surfaces, a coat of plaster (brown or scratch coat) is applied to completely cover the metal lath or conducting surface before the cable is attached. After fastening on the lath and applying the first plaster coat, each cable is tested for continuity of circuit and for insulation resistance of at least 100,000 ohms measured to ground.

The entire ceiling surface is finished with a covering of thermally non-insulating sand plaster about 0.50 to 0.75 in. thick, or other approved non-insulating material applied according to manufacturer's specifications. The plaster is applied parallel to the heating cable rather than across the runs. While new plaster is drying, the system should not be energized, and the range and rate of temperature change should be kept low by other heat sources or by ventilation until the plaster is thoroughly cured. Vermiculite or other insulating plaster causes cables to overheat and is contrary to code provisions.

For laminated drywall ceiling panels, the heating cable is placed between two layers of gypsum board, plasterboard, or other thermally non-insulating fire-resistant ceiling lath. The cable is stapled directly to the first (or upper) lath, and the two layers are held apart by the thickness of the heating cable. It is essential that the space between the two layers of lath be *completely* filled with a non-insulating plaster or similar material. This fill holds the cable firmly in place and improves heat transfer between the cable and the finished ceiling. Failure to fill the space completely between the two layers of plasterboard may allow the cable to overheat in the resulting voids and cause cable failure. The plaster fill should be applied according to manufacturer's specifications.

Electric heating cables are ordinarily installed with a 6-in. non-heating border around the periphery of the ceiling. An 8-in. clearance must be provided between heating cables and the edges of the outlet or junction boxes used for surface-mounted lighting fixtures. A 2-in. clearance must be provided from recessed lighting fixtures, trim, and ventilating or other openings in the ceiling.

Heating cables or panels must be installed only in ceiling areas that are not covered by partitions, cabinets, or other obstructions. However, it is permissible for a single run of isolated embedded cable to pass over a partition.

The *National Electric Code* requires that all general power and light wiring be run above the thermal insulation, or at least 2 in. above the heated ceiling surface, or that the wiring be derated.

In drywall ceiling construction, the heating cable is alw installed with the cable runs parallel to the joist. A 2.5-in. ance between adjacent cable runs must be left-centered u each joist for nailing. Cable runs that cross over the joist n be kept to a minimum. Where possible, these crossings sl be in a straight line at one end of the room.

For cable having a watt density of 2.75 W/ft, the minim permissible spacing is 1.5 in. between adjacent runs. Som manufacturers recommend a minimum spacing of 2 in. fo wall construction.

The spacing between adjacent heating cables can be determined by using Equation (21):

$$M = A_n/C \tag{21}$$

where

M = cable spacing, ft.
A_n = net panel area, ft^2
C = length of cable, ft

A_n in Equation (21) is the net panel area available after deducting the area covered by the non-heating border, lig fixtures in the ceiling, cabinets, and other obstructions. Fo simplicity, Equation (21) contains a slight safety factor, an small lighting fixtures are usually ignored in determining n ceiling area.

Resistance of the electric cable must be adjusted accord its temperature at design conditions:

$$R' = R \cdot \frac{(1 + \alpha_e \cdot t_d)}{(1 + \alpha_o \cdot t_d)} \tag{22}$$

Here,

R' = electrical resistance per ft of electric cable at standard temperature, ohm/ft..
α_e = thermal coefficient for the material resistivity, °
α_o = thermal expansion coefficient, °F^{-1}
t_d = skin temperature of the electric cable at operati conditions, °F. [See Equation (17).]

The 2.5-in. clearance required under each joist for nailin drywall applications occupies one-fourth of the ceiling are the joists are 16 in. on center. Therefore, for drywall constr tion, the net area A_n must be multiplied by 0.75. Many inst tions have a spacing of 1.5 in. for the first 2 ft. from the col wall. Remaining cable is then spread over the balance of t ceiling.

Electrically Heated Wall Panels

Cable embedded in walls similar to ceiling construction occasionally found in Europe. Because of possible damag from nails driven for hanging pictures or from building alt ations, most codes in the United States prohibit such panel Some of the prefabricated panels described in the precedi section are also used for wall panel heating.

Electrically Heated Floors

Electric heating cable assemblies, such as those used for ceiling panels, are sometimes used for concrete floor heati systems. Since the possibility of cable damage during insta

.s greater for concrete floor slabs than for ceiling panels, assemblies must be carefully installed. After the cable has placed, all unnecessary traffic should be eliminated until oncrete covering has been placed and hardened.

eformed mats are sometimes used for electric floor slab ng systems. These mats usually consist of PVC-insulated ng cable woven in, or attached to, metallic or glass fiber 1. Such mats are available as prefabricated assemblies in y sizes from 2 to 100 ft^2, and with various watt densities ing from 15 to 25 W/ft^2. When used with a thermally ed cavity beneath the floor, a heat storage system is ided, which may be controlled for off-peak heating.

neral-insulated (MI) heating cable is another effective iod of slab heating. MI cable is a small-diameter, highly ble, flexible heating cable composed of solid electric- ance heating wire or wires surrounded by tightly com- ed magnesium oxide electrical insulation and enclosed by tal sheath. MI cable is available in stock assemblies in a ty of standard voltages, watt densities, and lengths. A cable nbly consists of the specified length of heating cable, -proof hot-cold junctions, 7-ft. cold sections, UL-approved ittings, and connection leads. Several standard MI cable ructions are available, such as single conductor, twin con- r, and double cable. Custom-designed MI heating cable iblies can be ordered for specific installations.

ier outer-covering materials that are sometimes specified ectric floor heating cable include (1) silicone rubber, ad, and (3) tetrafluoroethylene (Teflon).

a given floor heating cable assembly, the required cable ig is determined from Equation (21). In general, cable lensity and spacing should be such that floor panel watt y is not greater than 15 W/ft^2. Higher watt densities (up to ft^2) are often specified for the 2-ft border next to cold Check with the latest issue of the *National Electric Code,* ther applicable codes, to obtain information on maximum watt density and other required criteria and parameters.

or heating cable installation. When PVC-jacketed c heating cable is used for floor heating, the concrete slab in two pourings. The first pour should be at least 3 in. ind, where practical, should be insulating concrete to downward heat loss. For a proper bond between the the finish slab should be placed within 24 hours of the our, with a bonding grout applied. The finish layer should east 1.5 in. and not more than 2 in. thick. This top layer iot be insulating concrete. At least 1 in. of perimeter insu- should be installed as shown in Figure 26.

ormed mats can be embedded in the concrete in a uous pour. The mats are positioned in the area between sion and/or construction joints and electrically connected iction box. The slab is poured to within 1.5 to 2 in. of the d level. The surface is rough screeded and the mats in position. The final cap is applied immediately. Since t pour has not set, there is no adhesion problem between t and second pour, and a monolithic slab results. A of contours can be developed by using heater wire d to glass fiber mats. Allow for circumvention of tions in the slab.

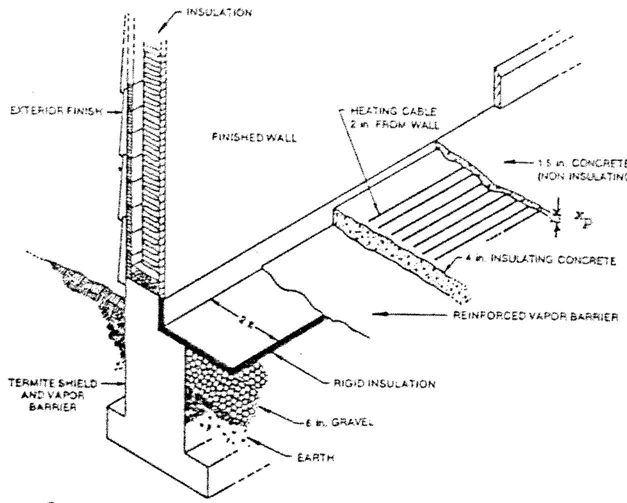


Fig. 26 Electric Heating Cable in Concrete Slab

The cable is installed on top of the first pour of concrete, not closer than 2 in. from adjoining walls and partitions. Methods of fastening the cable to the concrete include:

1. Staple the cable to wood nailing strips fixed in the surface of the rough slab. The predetermined cable spacing is maintained by daubs of cement, plaster of paris, or tape.
2. In light or uncured concrete, staple the cable directly to the slab using hand-operated or powered stapling machines.
3. Nail special anchor devices to the first slab to hold the cable in position while the top layer is being poured.

MI electric heating cable can be installed in concrete slab using either one or two pours. For single-pour applications, the cable is fastened to the top of the reinforcing steel before the pour is started. For two-layer applications, the cable is laid on top of the bottom structural slab and embedded in the finish layer. Proper spacing between adjacent cable runs is maintained by using prepunched copper spacer strips nailed to the lower slab.

CONTROLS

Automatic controls for panel systems may differ from those for other space conditioning systems because of the thermal inertial characteristics of the panel and the increase in the mean radiant temperature within the space under increasing loads. However, lightweight systems using thin metal panels or thin underlay with low thermal heat capacity may be successfully controlled with conventional control technology using indoor sensors. Many of the control principles for hydronic systems described in Chapter 12 and Chapter 14 also apply to panel systems. Because radiant panels do not depend on air-side equipment to distribute energy, many control methods have been used successfully; however, a control interface between heating and cooling should be installed to prevent simultaneous heating and cooling.

High mass panels, such as concrete radiant slabs, require a control approach different from low mass panels. Because of thermal inertia, significant time is required to bring such massive panels from one operating point to another, say from vacation setback to standard operating conditions. This will result in long periods of discomfort from low temperature, then possibly periods of uncomfortable and wasteful overshoot. Careful economic analysis may reveal that a nighttime setback strategy is not warranted.

Once a slab is at operating conditions, the control strategy should endeavor to supply the slab with heat at the rate that heat is being lost from the space (MacCluer *et al.*, 1989). For hydronic slabs with constant circulator flow rate, this equates to modulating the difference between the outgoing water and the returning water temperatures, accomplished via mixing valves, fuel modulation, or, for constant thermal power sources, via pulse-width modulation ("bang-bang" or "on-off" control). Slabs with embedded electric resistance cable can be controlled by pulse-width modulators, such as the common round thermostat with anticipator or its solid state equivalent.

A related approach, outdoor reset control, has enjoyed wide acceptance. An outdoor reset control measures the outdoor air temperature, calculates the supply water temperature required for steady operation, and operates a mixing valve or boiler to achieve that supply water temperature. If the heating load of the controlled space is primarily a function of the outdoor air temperature, or indoor temperature measurement of the controlled space is impractical, then outdoor reset control alone is an acceptable control strategy. When other factors such as solar or internal gains are also significant, then indoor temperature feedback should be added to the outdoor reset.

In all radiant panel applications, precautions must be taken to prevent excessive temperatures. A manual boiler bypass or other means of reducing the water temperature may be necessary to prevent new panels, and covering materials like wood, from drying out too rapidly.

Cooling Controls

Controlling the panel water circuit temperature by mixing, heat exchange, or using the water leaving the dehumidifier is typical. Other considerations are listed in the General Design Conditions section. It is imperative to dry out the building space before starting the panel water system—particularly after extended down periods, such as weekends. Such delayed starting action can be controlled manually or by a device.

Panel cooling systems require the following basic areas of temperature control: (1) exterior zones, (2) areas under exposed roofs to compensate for transmission and solar loads, and (3) control of each typical interior zone to compensate for internal loads. For optimum results, each exterior corner zone and similarly loaded face zone should be treated as a separate subzone. Panel cooling systems may also be zoned to control temperature in individual exterior offices, particularly in applications where there is a high lighting load, or for corner rooms with large glass areas on both walls.

The temperature control of the interior air and panel water supply should not be functions of the outdoor weather. The normal thermostat drift is usually adequate compensation for the slightly lower temperatures desirable during winter weather. This drift should be limited to result in a room temperature change of not more than 1.5°F. Control of the interior zones is best accomplished by devices that reflect t[...] actual presence of the internal load elements. Frequently, ti[...] clocks and current-sensing devices are used on lighting fee[...]

Because air quantities are generally small, constant volum[...] supply air systems should be used. With the apparatus arra[...] to supply air at an appropriate dew point at all times, comf[...] able indoor conditions can be maintained throughout the y[...] with a panel cooling system. As with all systems, to prevent[...] condensation on window surfaces, the supply air dew poin[...] should be reduced during extremely cold weather accordin[...] the type of glazing installed.

Electric Heating Slab Controls

For comfort heating applications, the effective surface temperature of the floor slab is held to a maximum of 84°F [...] occupied surfaces. Therefore, when the slab is the primary heating system, thermostatic controls sensing air temperatu[...] should not be used to control temperature; instead, the hea[...] system should be wired in series with a slab-sensing thermo[...] The remote sensing thermostat in the slab acts as a limit sw[...] to control maximum surface temperatures allowed on the s[...] The ambient sensing thermostat controls the comfort level. [...] supplementary slab heating, as in kindergarten floors, a ren[...] sensing thermostat in the slab is commonly used to tune in [...] desired comfort level. Indoor-outdoor thermostats are used [...] vary the floor temperature inversely with the outdoor temp[...]ture. If the heat loss of the building is calculated for 70 to 0[...] and the floor temperature range is held from 70 to 84°F wit[...] remote sensing thermostat, the ratio of outdoor temperatur[...] slab temperature is 70:15, or approximately 5:1. This means [...] a 5°F drop in outdoor temperature requires a 1°F increase i[...] slab temperature. An ambient sensing thermostat is used to [...] the ratio between outdoor and slab temperatures.

A time clock is used to control each heating zone if off-p[...] slab heating is desirable.

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SIMULATION OF RADIANT FLOOR HEATED ROOM

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Abstract

Radiant floor heating systems are highly energy efficient provided that a proper design and analysis is made. According to the thermophysical attributes of the system, the heating loads may be substantially reduced. One of the primary elements is the uniformity of temperature, velocity and pressure fields in a floor heated space. This uniformity mainly reduces the infiltration heat losses. The Turkish standards were prepared which quantitatively reflects this and other attributes. The qualitative algorithm given in Turkish standards for customizing the heating loads was largely realized by modeling typical spaces heated by different heating systems. This analysis involved the use of ANSYS 5.0 and FLOTRAN (Computational Fluid Dynamics) package. For different boundary conditions which reflects alternative heating systems and floor heating panels, indoor velocity, pressure and temperature fields were solved and processed for heat loss parameters. This paper gives the details of this application and summarizes the customized heat loss algorithm given by Turkish Standards.

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FLOTRAN solver. 75.6 W/m^2 heat input (due to convection by floor heating) from floor to room was calculated at the end of ANSYS solution. Screen outputs of the solution result are indicated at Figures 6, 7, 8, 9, 10 and 11.

2- Convection due to Radiator Heating : The same model was used except, a radiator located under the window was used instead of floor heating (Figure 12). 95.2 W/m^2 heat input (due to convection by radiator heating) from radiator to room was calculated at the end of ANSYS solution. Screen outputs of the solution result are indicated at Figures 13, 14, 15, 16, 17 and 18.

For the above two convection analyses;

a) Material Properties;

The same conduction heat transfer coefficients given at section "radiation analysis" are also valid for convection analyses. Additionally, ANSYS uses air properties from FLOTRAN database.

b) Boundary Conditions;

Additional to boundary conditions defined at "radiation analysis" section, no slip condition for air at inner surfaces of walls, window and door was defined.

ii) Convection Analysis : This time radiation effect was excluded and the same room fill with air was modeled (Figure 1). At this stage two different models of the room was solved, first the room heated from floor and second the room heated by a radiator located under the window. So that, we can see the difference between effects of floor slab and radiator on air circulation. For this purpose natural convection of the air was solved by ANSYS to obtain temperature, pressure and velocity distributions on the air, and air circulation. As it is well known, ANSYS is a general purpose program package to make analyses on wide range of area (static, dynamic, thermal, fluid, magnetic and their coupled analysis). But, the problem here is a fluid dynamics problem which includes both thermal and fluid flow effects. It is very difficult to solve such kind of a problem with ANSYS subroutines. There would be some difficulties, if this problem was solved by ANSYS subroutines. For example, modeling of the room will be very difficult, since ANSYS wants at least one node inside the boundary layer thickness. This will increase the amount of nodes and elements. Very large disk space should be available on the computer, since size of the files created by ANSYS becomes very large (about 150 MBytes), and also computation time becomes very large (about 1 or 2 days). That's why the FLOTRAN subroutine (which is very useful and suitable for such kind of problems) was used to solve this natural convection problem. FLOTRAN brings us easiness for modeling of this problem, since it does not wants nodes in boundary layer thickness. It solves the problem in a rather short time (about 1 hour) with rather coarse mesh, a few amount of nodes and elements. As it can be understood from above lines, FLOTRAN is not an interactive separate finite element program package but is a solver subroutine of ANSYS. For this kind of applications, again the problem is modeled in ANSYS program, but it is solved using FLOTRAN solver subroutine not by ANSYS subroutines. At the end, results are obtained again in ANSYS's post-processing section.

1- Convection due to Floor Heating : The walls, window, door and inner space of the room were divided into finite element mesh with PLANE55 element. Additional to boundary conditions defined in Figure 2, no slip condition for air at inner surfaces of walls, window and door was defined. During solution, additional to temperature degree of freedom FLOTRAN adds pressure, velocity and energy degrees of freedom to the nodes of PLANE55 element (at inner space of room). Thus ANSYS calculates air motion in the room with the help of

Modeling of the Room

The room which was analyzed by ANSYS 5.0 has inner dimensions as 4 x 2.9 m. It's window is 1.3 m height located on the outer wall and door is 1.9 m height on the inner wall (Figure 1). The room was analyzed as two dimensional at two separate steps as follows:

i) Radiation Analysis : At this stage the room was modeled without including the air. Since the fluid medium is not important in radiation, inside of the room was considered as empty and the problem was solved by ANSYS 5.0 package to see the radiation effect of slab on the walls, window and door. For this problem PLANE55 is the most suitable element type of ANSYS to model the solid parts (walls, window and door). PLANE55 is 2-Dim, 4-node conduction heat transfer element. After dividing the walls, window and door into finite elements with PLANE55 element, boundary conditions were defined (Figure 2). 28.2 W/m^2 heat input (due to radiation) from floor to room was calculated at the end of ANSYS solution. Screen outputs of the solution result are indicated at Figures 3, 4 and 5. The constants used in the model for radiation analysis are as follows;

a) Material Properties;

Conduction heat transfer coefficient for outer walls	=	0.7 W/m K
Conduction heat transfer coefficient for ceiling	=	1.4 W/m K
Conduction heat transfer coefficient for inner walls	=	1.5 W/m K
Conduction heat transfer coefficient for window	=	3 W/m K
Conduction heat transfer coefficient for door	=	1.2 W/m K

b) Boundary Conditions;

- constant temperature (=26 °C) at floor surface
- convection heat transfer coefficient (20 W/m^2 K , T_{bulk} = -12 °C) on outer walls and window
- convection heat transfer coefficient (5 W/m^2 K , T_{bulk} = 18 °C) on floor surface of upper flat
- convection heat transfer coefficient (5 W/m^2 K , T_{bulk} = 15 °C) on inner walls
- convection heat transfer coefficient (5 W/m^2 K , T_{bulk} = 15 °C) on door surface

ADVANTAGES OF COMBINING HEAT PUMPS WITH RADIANT PANEL HEATING AND COOLING SYSTEMS

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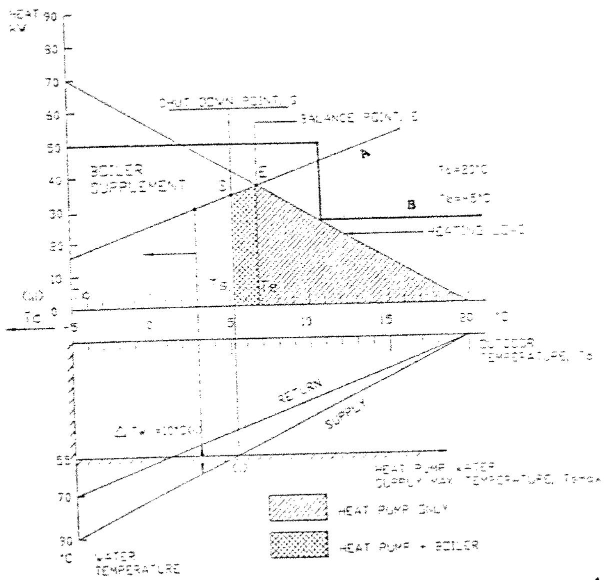
For space heating and cooling purposes,heat pumps are becoming a key element in energy conservation and protection of the environment. Their performance however,may be adversely effected by the temperature requirement of conventional space conditioning systems like forced-air or high-temperature hydronic heating systems. Usually a high temperature hydronic heating system is designed for 80°C mean water temperature. In cooling,the chilled water mean temperature is about 10°C. According to the basic nature of radiant panel systems, the temperature requirement in heating and cooling are very moderate and can be easily suited to the heat pump characteristics. The mean water temperature (T_W) requirement in heating may be 35°C,and in cooling may be up to 18°C. In order to demonstrate the attributes of radiant panel systems,an air to water (AWHP),a ground source (GSHP) and a solar absorption heat pump will be analyzed.

-Air to Water Heat Pump:

igure 1 depicts an air to water type heat pump serving a central eating system with radiators. A boiler supplements the heat pump ither in a parallel or tandem configuration below the equilibrium emperature (T_e) and takes over at T_s when the heat pump is forced to hut down due to one of the following additional constraints,imposed y this high temperature hydronic heating system:

-Constraint (i): Maximum water temperature that can be delivered by a heat pump (T_{smax}) is generally 55°C and in this example point (i) is at +5°C for a parallel mode. In a tandem mode, return water temperature governs the case.

-Constraint (ii): Condenser side water temperature drop (T_w) is generally limited to 10°C where a radiator heating system is normally designed for 20°C, unless the circulation pump is oversized. Otherwise, constraint (ii) will be effective in a parallel mode. In this illustration, constraint (i) occurs first and determines the Shut Down Point S. For parallel mode, T_s is +5°C. For tandem mode, it is +1°C. In order to avoid an early shut down, heat pump sizing must satisfy $T_e > T_s$ condition too.



T_a: Indoor air temp.

T_b: Outdoor design temp.

A- AWHP

B- Double speed GSHP

Figure 1. Heat outputs versus the heating load as a function of outdoor air temperature, T_o.

A radiant panel heating system will improve the performance and cost effectiveness of AWHP according to the following attributes:

1-Design and operation of a radiant panel already requires that

T_w should not exceed 55°C,while large radiant panel surfaces can deliver the required heat. This low temperature makes it more practical to use a 10°C temperature drop at design conditions. These will eliminate constraints (i) and (ii),and consequently T_s. Therefore, $T_e > T_s$ condition will be automatically satisfied which consequently enables to freely choose any optimum heat pump size. As a result,if other conditions are also satisfied,elimination of the supplementary boiler is possible:

2-T_a may be 2 to 3°C lower than usual without any sacrifice of the desired human comfort (Kilkis,1992). This moves the lower end of the load line in Figure 1 to the left. This also reduces transmission heat losses. In addition,a more uniform indoor air temperature and pressure distribution,and a slow air movement contribute to reduced natural infiltration heat losses. Due to heat (cool) storage in the panel and the exposed walls and partitions, the peak heating or cooling loads are further shaved. This enables to size the heat pump according to the leveled load instead of the peak load. This overall reduction of the heating load shifts the upper end of the load line downwards. The net result is while T_e moves towards the origin, deficit of the heat pump is minimized. Consequently the need for a supplementary boiler is either minimized or eliminated for a given heat pump capacity.

-COP of the heat pump is enhanced by lowering T_w.

he combined effect of these attributes are shown in Figure 2 and xplained in the case study given in the following sections. The heat ump can also be used for sensible cooling purposes through radiant ooling panels. The above attributes hold also true in cooling.

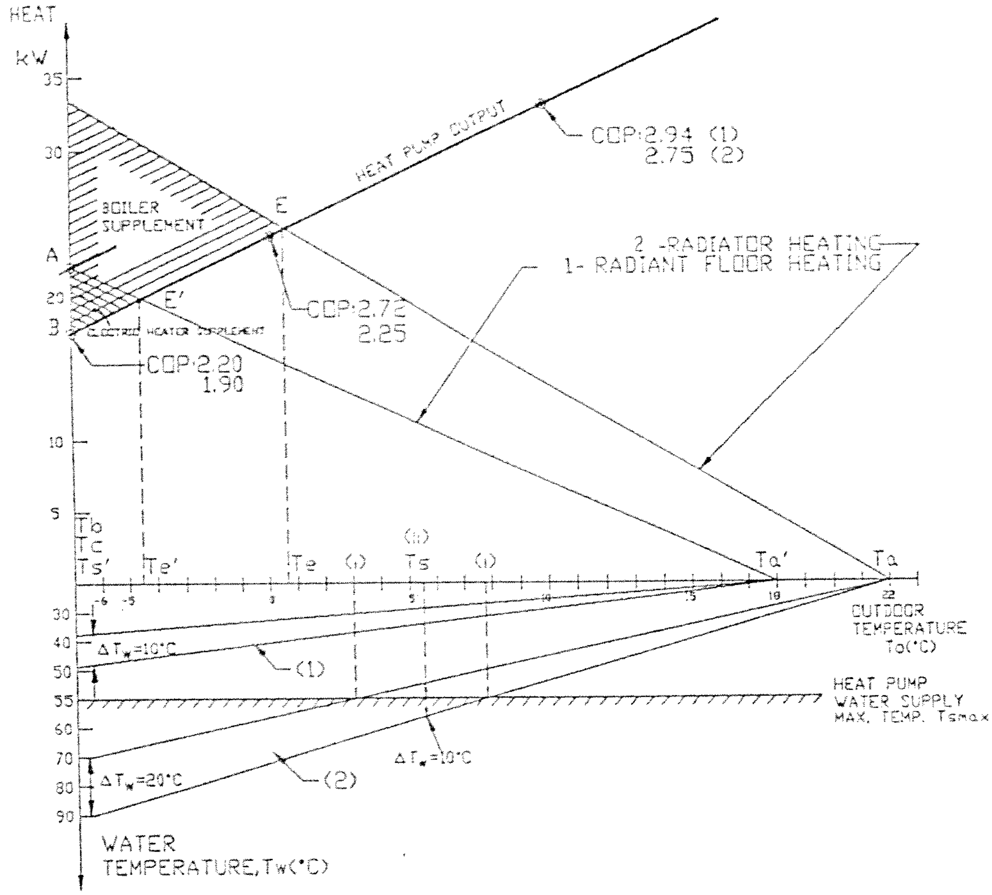


Figure 2. Improvement of the Performance with Radiant Panel Systems.

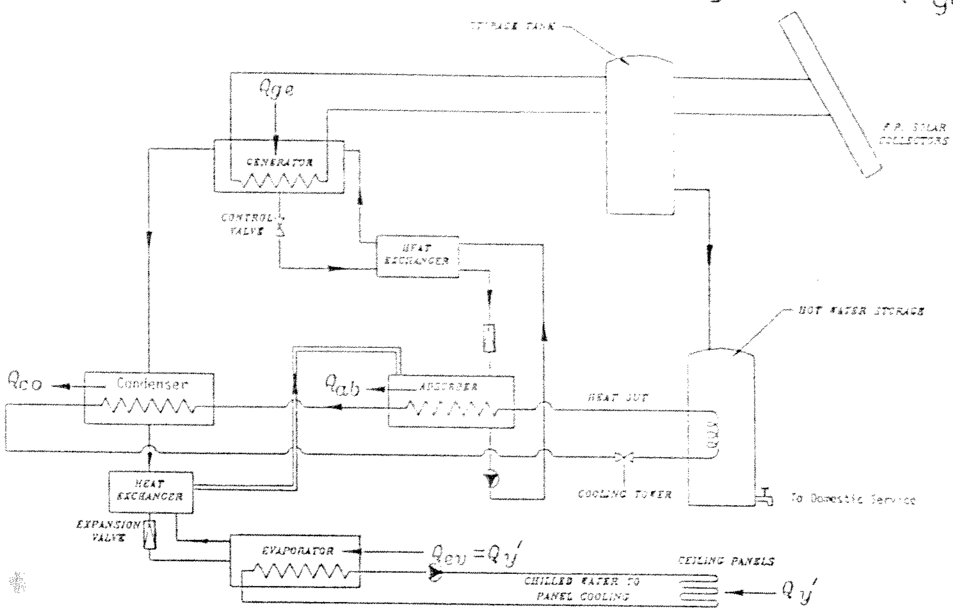
However,the need for handling the latent cooling load is important. This can be satisfied by splitting the cooling output to the panel circuit and a small air handler in appropriate and possibly variable proportions. Technically this means a hybrid system,but generally it proves to be economically feasible (Wilkins and Kosonen,1992).

B-Ground Source Heat Pump:

A ground source heat pump is virtually free from the effects of the outdoor temperature. However,in order to follow the heat load line closely,multiple speed heat pump may be desirable. Figure 2 shows a typical performance of a two speed GSHP. Similar to the case of AWHP,a ground source heat pump will enjoy the same attributes of a radiant panel system.

:- Solar Absorption Heat Pump:

ladiant ceiling cooling with solar energy is already an option at .east in technical terms (Kilkis,1993). A typical set up for a rater-ammonia absorption heat pump is shown in Figure 3. The most .mportant variable is the temperature of the generator (T_{ge}).



igure 3. General Arrangement of Solar Absorption Cycle Radiant Panel Cooling System.

his temperature directly affects both the heating and cooling COP. he optimum generator temperature however,depends on the temperature f the chilled water required by the cooling system on the evaporator ide (T_{ev}): while a radiant panel cooling system will permit to icrease T_{ev},COP increases,and optimum T_{ge} decreases which increases he solar collector efficiency. Figure 4 shows these in the cooling ode of a water-ammonia solar absorption system.

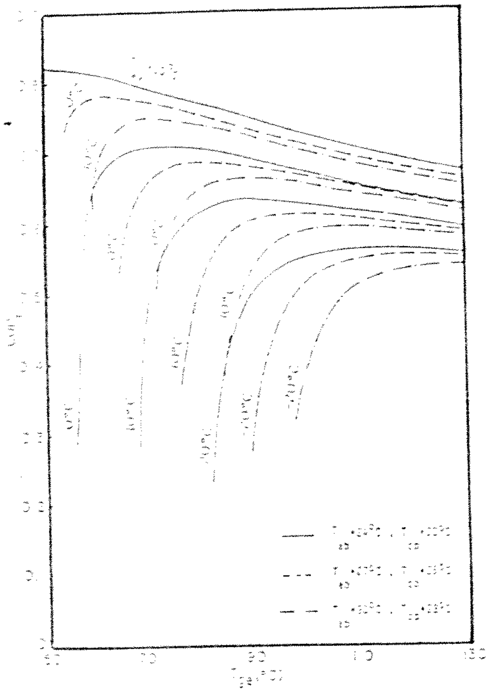


Figure 4. Change of COP_r with the Generator and Evaporator Temperatures (Ataer and Gogus,1990).

When the evaporator temperature (T_{ev}) increases to 10°C from 7½°C,COP_r increases to about 0.76 from 0.68 at an absorber temperature of (T_{ab}) 30°C,and a condenser temperature (T_{co}) of 28°C. The optimum T_{ge} drops from 70°C to 65°C respectively. The overall improvement will be even better if the evaporator temperature can be increased above 10°C,which is quite possible at an expense of reducing the tube spacing in the cooling panels. This is shown in the second case study.

CASE STUDY 1

A two story home has a conventional heating load of 33.6 kW at a design outdoor temperature of -7°C. The standard indoor design temperature is 22°C for a radiator heating system. For radiant floor heating,the peak load shaving factor is 0.88 for the given climate and building construction type (Kilkis,1992). For an equivalent thermal comfort,T_a is 18°C. The adjusted heating load with peak load shaving is 22.2 kW. Heated floor is concrete slab with a carpet.

ubber hose with 9.5 mm I.D and 16 mm O.D. at 200 mm spacing on enters is used. Design was performed with a computer program leveloped for design and analysis of radiant panel systems.

)esign T_W is 42°C. In order to minimize the cost, an air to water heat ›ump was selected with a heating capacity of 19 kW (Point B). Its ut-off point (T_C) is -7°C. This capacity will be only 3 kW less than he design load and can be easily supplemented by an electric heater ather than a boiler. It must be noted that by selecting the next ize of the heat pump, the electric heater can be eliminated (Point A). f a conventional heating system would be used, with the original heat oad line, the equilibrium point is +1°C. The supplementary boiler equirement is 11.4 kW (33.6 - 22.2), unless a much bigger heat pump s installed. The COP of the heat pump at design conditions will drop ɔ 1.9 from 2.20. The radiator system would also force the heat pump ɔ be shut down at +3°C due to constraint (i). This means that the ıpplementary boiler will be called more frequently. The two systems re compared in Figure 2.

ASE STUDY 2

ıe second floor is to be cooled in summer. Although the same air to ıter heat pump can be employed, for demonstration purposes, separate solar absorption heat pump will be used. With the extra iling insulation the sensible cooling load is 4.9 kW. With similar sign provisions, the design cooling load for the cooling panel is 0 kW. Indoor relative humidity will be maintained at 0.35 by a brid system. Indoor air temperature will be 24°C. Rubber hose with mm I.D and at 100 mm spacing on centers will be laid in concrete iling slabs. Under these conditions T_W is 18°C while the ceiling surface temperature is maintained at 20°C. This moderate temperature requirement translates to the performance by an increase of the COP_r and solar collector efficiency:

For the same sensible cooling load of 3 kW, COP increases to 0.83 from 0.68 and flat plate collector efficiency increases to 0.72 from 0.65, while the generator temperature decreases to 50°C from 70°C, when compared to a conventional fan coil cooling system.

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DEVELOPMENT of DESIGN SOFTWARES for RADIANT HEATING and COOLING of BUILDINGS

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ABSTRACT

Radiant panel heating and cooling systems may substantially contribute to energy conservation in space conditioning provided that a proper modeling, design and installation is made. The primary attributes include peak load shaving, decrease in heat losses (or gains in cooling) and natural infiltration when a radiant panel system is used [1]. Moderate temperatures that a typical radiant panel system operates, yield better tie-in capabilities with alternative energy sources with or without heat pumps [2,3]. In spite of these features, the attributes were overlooked and not properly acknowledged mainly due to the fact that not any dedicated algorithm was available before. This study had two primary aims in order to facilitate the proper and accurate design of radiant panel heating nd cooling systems. These were namely to develop a customized space onditioning load calculation algorithm and to couple it to a system esign algorithm with heating and cooling options. This was achieved y implementing a general computer code which involves the customized oad calculation and the design of the radiant panel system. The load alculation algorithm generally follows the ASHRAE guidelines,but ntroduces certain adjustments to various parameters in addition to he peak load shaving adjustment due to energy storage in the slab. t also enables to compare the space conditioning energy with nventional systems. The design algorithm relies on the fin theory ing developed prior to this study [1,2]. This paper gives the basic eory underlying the interactive computer program which is also

described in detail with the help of sample designs.

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AN ANALYTICAL DESIGN MODEL AND ITS COMPUTER IMPLEMENTATION FOR PANEL HEATING AND COOLING

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Abstract
Panel heating and cooling is becoming one of the key systems in efficient space heating and cooling. In spite of their very promising energy efficiency and operational attributes,there is not any simple but accurate model for the performance analysis and design. Although sophisticated numerical models are already available,their everyday use have not yet welcomed by the design engineers. This made it essential to develop a simplistic and a substitute analytical model without sacrificing the engineering accuracy. With this challenging objective,a modified composite fin model was developed. This model enables to perform a sufficiently accurate design,where all variables can be easily manipulated by the designer. Although the algorithm is simple enough to carry out the design manually,the optimization search may require several iterations in a typical design. For this purpose a computer program was implemented. Comparisons of typical case studies with finite element solutions substantiated the convenience and accuracy of this new model. This paper gives the basic theory of panel heating and cooling,describes the new model,the implementation and results of the computer program.

1. Nomenclature

- A_p Panel surface area, m^2
- A_r Total floor (ceiling) area in the heated (cooled) space, m^2
- A_u Total surface area of the heated (cooled) space, m^2
- AUST Area-weighted average temperature of the unheated (uncooled) surfaces at design conditions (excludes panel surfaces), °C
- c Temperature gradient factor, °C
- d Room position code, dimensionless
- D_e Equivalent diameter of the floor (ceiling), m
- D_i Inside diameter of the heat transfer hose, m
- D_o Outside diameter of the heat transfer hose (or electric resistance), m
- e Surface emittance,dimensionless
- F_r Radiation interchange factor,dimensionless
- h Altitude of the location above sea level, m
- k Thermal conductivity of the slab material, W/m K
- k_e Equivalent thermal conductivity of the composite slab for lateral heat diffusion, W/m K
- k_h Thermal conductivity of the hose material, W/m K
- k_i Thermal conductivity of each panel cover, W/m K
- L Thickness of the slab in the fin, m
- L_r Inside perimeter of the floor (ceiling), m
- M Hose spacing on centers, m
- m Fin coefficient, m^{-1}
- n_k Number of panel covers,dimensionless
- q_a Heat loss (gain) intensity from (to) the panel from its back, W/m^2

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COMPUTER AIDED DESIGN of RADIANT SUB-FLOOR HEATING SYSTEMS

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ABSTRACT

Radiant floor heating systems require moderate fluid temperatures. This enables effective and efficient utilization of low temperature - low intensity heat sources. Due to the specific nature of this type of heating, the heating load is decreased and its peak is shaved. With these prime features, it is also a favorable alternative for existing homes as a retrofit . A Computer Aided Design program was developed to design and analyze radiant heating systems for joisted sub-floors. It predicts the performance parameters like heating intensity at steady state conditions for various floor configurations and hose layouts, enabling the designer to choose optimum design variables for several objectives. This article provides a model study, gives the algorithm of the program and provides case studies with generalized design strategies.

1. NOMENCLATURE

A_p	Total floor heating panel area, m2
c	Room position code, dimensionless
cc	Stratification factor, dimensionless
C_D	Radiation coefficient, W/m2K4
COP	Coefficient of heat pump performance, dimensionless
D_{eq}	Equivalent diameter of the room perimeter, m
D_i, D_o	Inside and outside hose diameters, m
F_1	Convective heat transfer correction factor w.r.t. room size, dimensionless
FIF	Fin efficiency modification factor, dimensionless
h	Elevation, m
H	Air gap thickness between the sub-floor and batt insulation, m
k	Thermal conductivity, W/m K
k_{air}	Thermal conductivity of air, W/mK
k_{eq}	Equivalent thermal conductivity of the floor above hose center, (including any floor covering), W/m K
k_h	Thermal conductivity of the heat transfer hose, W/m K
k_i	Thermal conductivity of insulating material, W/m K
k_p	Thermal conductivity of sub-floor material, W/m
k_{pj}	Thermal conductivity of the joist material, W/m K
L	Total required hose length in the panel circuit, m
L_f	Length of the fin, m
L_r	Standard hose length as marketed, m
m	Fin coefficient, $(m)^{-1}$
M	Distance between adjacent hose centers o.c., m
n	Number of floor coverings, dimensionless
n_h	Number of hoses in the joist space, dimensionless
Q_{is}	Design heating load for conventional systems, kW
Q_y	Design heating load for floor heating, kW
p	Joist pitch o.c., m
P_H	Heat output of an air to water type heat pump, kW
q	Heating load intensity W/m^2
q_y	Panel heating intensity, W/m^2
r	Surface emissivity, dimensionless
r_a	Thermal resistance in downward direction, $(W/m^2K)^{-1}$
r_y	Thermal resistance in upward direction, $(W/m^2K)^{-1}$
T	Thickness of the sub-floor, m
T_a	Design indoor temperature for floor heating, °C
T_b	Maximum temperature on the heated floor surface, °C
T_c	Cut-off temperature, °C
T_d	Outside wall temperature of the hose, °C
T_e	Equilibrium temperature, °C
T_f	Local floor surface temperature, °C
t_f	Temperature fluctuation factor, $(T_b - T_{min}) / T_p$, dimensionless
T_i	Temperature of the space below, °C
TJ, Y	Joist dimensions, m
T_{min}	Minimum temperature on the heated floor surface, °C
T_o	Outdoor temperature, °C
T_p	"Uniform" floor surface temperature, °C

+ Member ASME, ASHRAE

T_s	Shut down temperature °C
T_{smax}	Maximum water supply temperature, °C
t_{total}	Total thickness of the floor (above hose center), m
T_u	Average wall inside temperature (excluding radiant floor panel), °C
T_w	Average water temperature, °C
u_c	Overall heat transfer coefficient, W/m²K
V	Water flow rate, m³/s
v_s	Water speed, m/s
W	Half clearance between adjacent hoses(m-D_O)/2, m
W1,W2	Clearance between each joist and the nearby hose, m
X	Heating efficiency, dimensionless
XI	Thickness of the batt insulating material, m
x_j	Thickness of the floor covering, m

Superscript

'	Radiant floor heating system variable

Greek Symbols

ΔT	Temperature drop, K
α_a	Convective heat transfer coefficient between back surface and air space below, W/m²K
α_y	Floor to air convective heat transfer coefficient on floor surface, W/m²K
α_w	Water-hose wall convection heat transfer coefficient, W/m²K
γ_w	Density of water, kg/m³
ΔP	Pressure drop in the heating panel circuit, kPa, mm H_2O
$\Delta P/L$	Unit pressure drop in the hose, kPa/m, mm H_2O/m
σ	Radiative heat transfer equation linearization coefficient, K³ (°C³)
η	Fin efficiency, dimensionless

2. INTRODUCTION

Various advantages of radiant floor (panel) heating were explained earlier (Kilkis, 1990a): It offers substantial energy savings and enables to utilize low temperature - low intensity heat sources like waste heat, geothermal energy or the ambient energy with or without the assistance of heat pumps (Kilkis, 1992). Although it seems that it is more suitable to implement such a system in a new building, it is usually quite simple to install the floor heating system in an existing building with minimum or no visible alterations. In existing residential homes in the United States, floor construction generally consists of a joisted sub-floor as shown in Figure 1.

In radiant floor panel heating systems, the temperature difference between the supply and return water, ΔT_w, is usually 10°C. This value exactly matches the condenser side (heating coil) water temperature difference, ΔT_{ws} in most of the ambient air to water type heat pumps. A conventional base board or radiator heating system contradicts this with a standard ΔT_w value of 20 K. Therefore it is difficult to employ a heat pump in a conventional heating system because of two main reasons, namely higher average water temperatures involved and higher temperature drops. Although it is not a rule to employ a heat pump in floor heating systems when one depends upon waste heat or ambient air, floor heating offers substantial advantages for heat pump utilization as shown in Figure 2a and summarized below for a case study involving a residential building with Q_{is} = 34 kW at design outdoor temperature (Kilkis, 1990b):

The heat pump operation shut down point S at T_o=T_s is determined by one of the following factors which comes first on T_o axis:

(i) - Maximum water supply temperature Tsmax that can be delivered by the heat pump. Usually it does not exceed 55°C.

(ii) - Condenser side water temperature drop ΔT_{ws} is usually limited to 10 K. The same figure indicates that the shut down temperature maybe as high as +8°C for a conventional space heater. These constraints virtually diminish in a radiant floor heating system.

(iii) - Cut off outdoor temperature T_c for defrost.

It must be noted that in this specific example, shut down point precedes Equilibrium Point E.

The equilibrium point E, between the heating demand, Q_{is}, and the net heat pump output P_H·X shifts from +1°C to -5°C, simply because the heating demand in floor heating, is reduced. This shift minimizes or eliminates the need for a boiler supplement at adverse climatic conditions.

Coefficient of performance (COP) as indicated at typical points on the same figure is systematically higher for floor heating case because the heat pump supply water temperature can be maintained at lower values. The dependence of COP on supply water temperature is shown in Figure 2b (Reay, 1979).

Some recent incentives for energy conservation, and off-peak hours electric power utilization (Vadasz and Weiner, 1987) further increased the interest in floor heating systems, because the heated floor and surrounding walls store heat during off-peak hours provided that it is kept idling (Kilkis, 1990a). In spite of this trend and obvious advantages, there is not any complete model to direct the development of a dedicated numerical design algorithm to achieve an optimum solution. An optimum design is essential both for comfort requirements and minimization of installation and running costs in order to reflect the above mentioned virtues of the floor heating system. This is very crucial in increasing the quality of design and application. In this study, an interactive computer program for a CAD environment was developed which is capable of modeling and simulating a sub-floor radiant heating system.

3. THEORY

3.1. Radiant Floor Heating Capacity

A special algorithm for calculating the floor heating load, Q_y, and determining the peak load shaving characteristics were developed earlier by Kilkis (1990a). For a predetermined radiant floor heating load and available panel area for the given enclosed space, heat load intensity will be:

$$q = Q_y / A_p \tag{1}$$

In order to meet this load , the panel heating intensity q_y has to be:

$$q_y \geq q \tag{2}$$

For a flat horizontal floor surface at a fictitious uniform temperature T_p , q_y can be expressed as the sum of radiative and convective terms:

$$q_y = \sigma C_D (T_p - T_u) + (1.0 - 2.257 \cdot 10^{-5}\, h)^{2.62} \cdot 2.67 \cdot (T_p - T_a)^{1.25} \cdot F_1 \tag{3}$$

The first term is the temperature linearized radiative heating intensity with;

$$\sigma = 0.0105\, (Tp+Tu) / 2 + 0.7955 \tag{4}$$

This equation is valid with 2% accuracy in the range of $15°C \leq (T_p + T_u)/2 \leq 30°C$

and $$C_D \cong r \cdot 5.67 \tag{5}$$

The material, texture, and color of the floor covering surface determines the emissivity, r.

The average wall inside temperature T_u is a function of the design outdoor temperature, T_o , and indoor design temperature, T_a, (ASHRAE, 1984) and the position of the room in the building. T_u can be approximated by the following Equation (Kilkis, 1991)

$$T_u \cong T_a - c\,\frac{15}{(25 + T_o)} \tag{6}$$

c is the room position as given in Table 1.

A typical wall temperature distribution including external walls and opaque surfaces is given in Figure 3 (Raiss and Roedler, 1969).

The elevation correction term in Equation 3, was developed by Oskay and et. al. (1977). In the convective term, h is the altitude, and F_1 is the correction for the room size:

$$F_1 = (4.96 / D_{eq})^{0.08} \tag{7}$$

where,

$$D_{eq} = 4 \cdot \text{total floor area / floor perimeter} \tag{8}$$

For a given q_y, the necessary floor temperature, T_p can be uniquely solved through Equation 3 provided that T_u is predicted and values of h and T_a are fixed. Several codes and standards impose a maximum upper limit of 29°C for T_p (ASHRAE, 1987).

3.2. Sub-Floor Heat Transfer Model

A typical plywood sub-floor, its several coverings, stapled hoses, and the batt insulation with a reflective foil is shown in Figures 1 and 4. Generally, hoses are equally spaced and 2 to 4 hoses are placed in the joist space. Major inputs for a floor heating design task are: Q_y, A_p, T_a, T_i , room dimensions, T_o, h, and the position of the room. Obviously for an existing home, several dimensions like T, Y, TJ, p are also fixed.

Once the "uniform" T_p value is determined, one needs to optimize the design within the limits of comfort, the floor construction, its dimensions, and the available water temperature at the source, so that other design parameters can be determined. The second important comfort criteria is the degree of surface temperature fluctuation due to presence of discrete hoses stapled to the sub-floor (see Figure 4). In fact, depending upon thermal conductivity of floor materials, indoor air temperature, required heat intensity thickness of the floor and hose spacing, the floor surface temperature exhibits a definite swing.

Designating this swing by ι_f:

$$\iota_f = \frac{T_b - T_{min}}{T_p} \tag{9}$$

One has to minimize this ratio in the feasible design domain. There is not any model to predict the temperature profile on the surface of a radiant sub-floor. Kilkis (1992) developed a fin model to predict the heat diffusion and surface temperature profile on a radiant slab. This model has been also adapted for sub-floor type floor heating systems

The independent design variables are:

(i) - Properties of the hose, and its dimensions D_o and D_i;
(ii) - Number of hoses in the joist space, n_h;
(iii) - Hose spacings from the joists W1, W2 (usually W1 = W2);
(iv) - Insulation material, insulation thickness, XI, and the air gap, H;
(v) - Any additional layer below the sub-floor; and
(vi) - Floor coverings: their material, thickness, j, and surface emissivity of the top covering.

The dependent design variables are:

(i) - Average water temperature, T_w;
(ii) - Clearance between adjacent hoses, 2W (M);
(iii) - Heating efficiency X, (see Equation 21);
(iv) - T_b, T_{min}, (ι_f);
(v) - Required water flow rate, V, and unit pressure drop, $\Delta P/L$; and
(vi) - Required hose length, L.

In this model, an analogy between the sub-floor having heat transfer hoses stapled under it and a solar collector plate with attached tubes is made. The definition of collector fin efficiency, η, as expressed by Kreider and Kreith (1981) is used:

$$u_c = \frac{q_y}{T_p - T_a} \tag{10}$$

$$m = \sqrt{\frac{u_c}{k_{eq} \cdot t_{total}}} \cdot FIF \tag{11}$$

Here FIF is the fin efficiency improvement factor. It takes into account the back radiation of heat from the reflective foil above the batt insulation. This radiation helps in distributing the heat more evenly in the sub-floor which is usually made of material with poor thermal conductivity like plywood. It is essential to carry out a separate analysis to predict FIF. Under normal circumstances of joist and batt insulation construction it has been calculated that it is around 1.6. In this analogy it is anticipated that a good contact between the hose and plywood surface exists. This can be accomplished by special adhesives or stapling techniques and largely depends upon the quality of installation also. It has been analytically determined that if 1 mm of air gap exists between the plywood and the hose, the heat output can be impaired by as much as 25 to 30%.

Here;

$$k_{eq} = \frac{t_{total}}{\sum_{j=1}^{n} x_j / k_j + T / k_p} \tag{12}$$

$$t_{total} = T + \sum_{j=1}^{n} x_j \tag{13}$$

Then,

$$\eta = \frac{\tanh(mW)}{mW} \tag{14}$$

Here W is the half spacing between adjacent hoses: $(M - D_o) / 2$.
If n_h and D_o is fixed, then:

$$M = (p - TJ - W1 - W2 - D_o) / (n_h - 1) \tag{15}$$

Setting W1 = W2, and imposing a symmetry of the hose spacing along neighbor joists:

$$W1 = W2 = \{p - n_h (D_o + TJ)\} / (2\, n_h) \tag{16}$$

Sub-floor is considered to have three fin sections, having lengths namely W1, W2 and W respectively;

$$T_b = T_a + \frac{p \cdot q_Y}{(n_h - 1) 2W \cdot \eta + n_h \cdot D_o + (W1 + TJ/2) \cdot \eta_1 + (W2 + TJ/2) \cdot \eta_2} \cdot \frac{1}{u_c} \tag{17}$$

η, η_1 and η_2 are calculated using Equation 14 .

The outside surface temperature of the hose will be;

$$T_d = \frac{q_y}{k_{eq} \cdot t_{total}} \tag{18}$$

The minimum floor surface temperature T_{min} will be:

$$T_{min} = T_a + \frac{2(T_b - T_a)}{e^{mL_f} + e^{-mL_f}} \tag{19}$$

Here L_f is the largest of W, W1 and W2 values.

Following the equation for a fin with insulated edge (Eckert and Drake,1959) the temperature profile can be calculated by the following equation. It must be noted that this profile is not sinusoidal.

$$T_f(x) = \frac{\cosh[m \cdot (M/2 - x)]}{\cosh[m \cdot M/2]} \cdot \{T_b - T_a\} + T_a \tag{20}$$

Heating efficiency X is defined by Kilkis (1990a) as the ratio of the intensity of heat delivered to the indoor space to the intensity of total heat released by the panel. Assuming that a good insulation is applied around the floor perimeter, edge losses are neglected:

$$X = \frac{q_y}{q_y + q_a} = \frac{1}{1 + q_a / q_y} \quad (T_i < T_a)$$
$$= \frac{1}{1 + r_y / r_a} \quad (\text{if } T_i = T_a) \tag{21}$$

$$\text{Here, } r_y = \frac{k_{eq}}{t_{total}} + \frac{1}{\alpha_y} \tag{22}$$

Making use of electrical analogy for conduction, and accounting for the stratification in the air gap, back radiation from the reflective foil and convection to the space below:

$$r_a = p \left\{ \frac{1}{\frac{cc \cdot k_{air}}{H}(p - TJ) + \frac{2k_{pj} \cdot TJ}{(H + XI + J)}} + \frac{XI}{k_i(p - TJ)} + \frac{1}{\alpha_a \cdot p} \right\} \tag{23}$$

Here cc is the air stratification correction term and is a function of H/(p-TJ). The water flow rate required to transfer the necessary heat and compensate the losses is:

$$V = \frac{Q}{3600 \cdot \gamma_w \cdot \Delta T_w \cdot X} \tag{24}$$

The water film heat transfer coefficient as corrected for water velocity v_s is (Kreider and Kreith, 1981):

$$\alpha_w = 1050\,(0.02\,(T_s + 273) - 4.06)\, v_s^{0.8} / D_i^{0.2} \tag{25}$$

where;

$$v_s = \frac{4V}{\pi D_i^2} \tag{26}$$

Finally,

$$T_W = T_d + \frac{q_y}{\pi} \frac{M}{X} \left\{ \frac{1}{\alpha_w D_i} + \frac{1}{2 k_h} \ln (D_o / D_i) \right\} \tag{27}$$

4. DEVELOPMENT OF THE COMPUTER PROGRAM

An interactive computer aided design program was developed in order to implement the model and provide the designer with a tool to analyze various alternatives within certain practical design constraints. Its algorithm is given in Figure 5. The program was written in Turbo Pascal language. Program enables the designer to input any dimensional proportion, material, floor insulation, hose properties, comfort conditions and heating demand. Through an iterative process bounded by some practical constraints given below, design can be optimized for a design objective. Practical constraints imposed on the program are:

(i) - $T_p \le 29°C$	(vii) - $2 \le n_h \le 4$
(ii) - $T_w \le 70°C$ (Rubber,60°C Thermoplastic)	(viii) - $q_y \le 110$ W/m^2
(iii) - $L < L_T$	(ix) - $X \ge 0.8$
(iv) - $Q_y \le 5000$ W (per heating panel)	(x) - $\Delta P \le 5000$ mm H_2O (49 kPa)
(v) - $0.5 \le v_s \le 1.5$ m/sec.	(xi) - $T_b \le 45°C$
(vi) - $M \ge 3.D_o$	(xii) - $T_d \le 70°C$

Here T_b is the maximum floor surface temperature (see Figure 4) and is limited by the generally accepted complaint limit for bare feet. The hose outside temperature at sub-floor contact line, T_d is limited by the safe temperature limit for sub-floor material. Besides the design variables, the program also provides necessary information for equipment selection and sizing, like the unit pressure drop, total hose length, and required number of panels in the heated room.

5. CASE STUDY

A case study was carried out with the following input data:

$Q_y = 2.3$ kW ; $A_p = 30$ m^2 ($q_y = 76.7$ W/m^2) ; h= 865m

Gross room floor dimensions = 5.6 m x 6.6 m ; $T_a = 18°C$; $T_o = -10°C$

For the given design requirements, corresponding T_p needs to be 23.6°C. (See Equation 3). This is already less than 29°C. The floor surface temperature fluctuation is the second comfort criteria. The surface temperature swing is a function of hose spacing.

Sub-floor heat transfer analysis:

The sub-floor construction dimensions are as follows:

H = 0.1016 m (4"), Y = 0.2032 (8") m, p = 0.406 m (16"),

TJ = 0.0381 m (1.5"), T = 0.01905 m (3/4"),

Floor Cover: 3/8" carpet with k = 0.069 W/m°C (nylon plush)

$T_i << T_a$ (unheated basement), sub-floor material = Plywood.

With the primary objective of minimizing the installation cost, the following cost effective factors were distinguished:

(i) - Total cost of the hose,

(ii) - Total cost of insulating material,

(iii) - Total pressure drop,

In order to gain an insight to the life cycle cost of various design parameters, several alternatives were tested and compared.

Figure 6 shows the change of the pressure drop in the panel w.r.t. number of hoses in the joist space (n_h) for various available standard hose diameters, D_i . Obviously, the total hose length in a radiant floor panel increases with n_h . Consequently the total pressure drop increases. A reduction in D_i increases the pressure drop in the panel . This pressure drop effects the pump rating, consequently its installation cost as well as its running costs. The lower scale indicates the required hose length, L, that is to be laid in the panel. As hose splicing has limited application, L must not exceed a maximum value. If this limit is exceeded, the panel must be subdivided into shorter circuits. However, the latter imposes additional costs like extra manifolds, valves, control units and extra hose between headers and the panel(s). For this case study, the economically feasible domain in general terms is indicated by the shaded area on the same figure. The total cost of the hose material will be the product of required hose length, L, and the current market unit price of the hose.

As revealed on the same figure, the only justification of increasing the number of hoses is the fact that average water temperature, T_w, decreases substantially. Depending upon the type of water heating plant, there is a varying degree of cost effectiveness related to the reduction of T_w . This effectiveness especially gains importance in heat pump operations, because the coefficient of performance (COP) has a rather strong dependence on T_w (See Figure 2b). In turn, increasing COP decreases the operating cost of the heat pump. Therefore depending upon the specific market conditions of equipment and energy, increasing n_h may be justified by the reduction of the plant running cost provided that the extra pumping installation cost will be recovered in a reasonable period. Another advantage of increasing n_h is the minimization of floor surface temperature fluctuations as shown on Figure 7.

The floor temperature fluctuation factor t_f, as defined by Equation 9 is 0.4 for $n_h = 3$. With a 3/4" I.D. hose, it improves to 0.15 with $n_h = 4$. Although there is not any comfort criteria related to this factor available in the literature, it is obvious that a minimum value is desirable, also from material and structural stability points of view. Regarding Equation 17, the peak temperature T_b is mainly a function of η and consequently M for a given n_h. Physically, heat flowing upwards through the sub-floor, can not distribute evenly in the lateral direction before it reaches the surface, because of rather poor thermal conductivity of the material like plywood. Fin efficiency can be improved if a conductive layer is attached to the bottom of sub-floor. For this purpose, Aluminum sheets of different thicknesses were tested, and typical results are plotted in Figure 8. It is evident that by employing a 5 mm Aluminum sheet, t_f improves to 0.3. However the heating efficiency is impaired by about 4%, because of increased downward radiant heat loss due to the introduction of a warm radiative surface to the joist space. This corresponds to about 5% increase in pumping requirement. Therefore such a comfort requirement may contradict with the operating costs. The impact of covering the conductive layer by an additional insulation of 0.02 m thickness, on heating efficiency did not prove to be cost effective, because X improved slightly, by 2%. During iterations of hose material selection, it has been observed that thermal conductivity plays an important role on the average water temperature, T_w, which is shown on Figure 9. In the practical range of various hose materials of the same wall thickness, it is clear that T_w can be reduced by about 6°C which is especially important for cases where alternative energy sources are going to be used. As can been seen, the curve starts to level off beyond a k_h value of about 0.30. Beyond this knee, there is not much advantage of employing a more conductive hose.

Figure 10 demonstrates the change of heating efficiency with an increase of the insulation thickness, where high density styrofoam was used for the case study. The heating efficiency, as defined, increases by the insulation thickness. However in practice it has a lesser degree of effect on water temperature and water flow rate which are the effective factors of the running cost. In this case study, the decrease in the required water flow rate was only 4 % in spite of increasing the insulating thickness by a factor of 5. The impact on T_w was even smaller.

After completing the iterations, the following design alternative was selected for an optimum cost:

$n_h = 3$

$D_o = 0.0161$ m, $D_i = 0.00952$ m (3/8") Number of circuit in the room: 3

Insulation thickness, XI = 0.0508 m (2")

The resulting operational variables as determined by the program were:

$T_w = 54°C$	$\Delta P = 22.2$ mm H_2O/m
$X = 87.9$ %	$L = 90$ m
$V = 2.13 \cdot 10^{-5}$ m³/s	$\Delta P \cong 2000$ mm H_2O

$v_s = 0.31$ m/sec.

$T_b = 29°C$, $T_{min} = 19.2°C$

$T_d = 50°C$

Standard hose length L_t varies between 75m to 125m in practice, so the hose length is acceptable. The design satisfies all of the 12 criteria, except the average water velocity. However, a careful air discharge system can eliminate the risks of possible air separation. Therefore the design should be considered to be satisfactory.

6. CONCLUSIONS

By using the computer aided design program, for various case studies, it was possible to make the following design generalizations:

(i) - Increasing the batt (underframe) insulation thickness is not always cost justifiable.

(ii) - To add a conductive layer beneath the sub-floor does not necessarily improve the overall design.

(iii) - Selection of the standard hose diameter depends upon specific market conditions and water temperature of the available source.

(iv) - Generally speaking, n_h may be kept at minimum if conventional boilers are used.

(v) - Thermal conductivity of the hose beyond a knee value does not improve the design much.

(vi) - Increasing the number of hoses, n_h, may be feasible if a low T_w value is desired and the floor surface temperature swing has to be minimized.

As seen, it is now possible to draw certain important rules which could only be guessed intuitively by the radiant floor heating industry before. Until now, designers usually had few mathematical tools, mostly in the form of general nomograms developed for slab type floors and had little chance to accurately analyze the system and iterate on it. The iterative technique was found to be sufficiently powerful to arrive at an optimum design at given specific market conditions with a better theoretical support and comprehensive tool.

This is especially true for radiant sub-floor heating. The unique design tool represented hereby will enable designers to enhance the virtues of radiant panel heating system by achieving optimum designs and gaining the ability to predict its performance and iterate on it if required.

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Figure Captions

Figure 1. A Typical Radiant Sub-floor Heating System.

Figure 2a. Comparison of the Heat Pump Performance for Floor Heating and Radiator Heating Cases (Kilkis, 1990, 1992)

Figure 2b. A typical relationship between COP and T_W.

Figure 3. A Typical Wall inside Temperature Distribution (Raiss, Roedler, 1969)

Figure 4. Fin Model for the sub-floor Radiant Heating System (Kilkis, 1991).

Figure 5. General Algorithm of the Computer Program.

Figure 6. Dependence of ΔP, L, and T_w on n_h, and Hose Dimensions.

Figure 7. Dependence of Floor Temperature Fluctuation, t_f, on the Number of Hoses between Adjacent Joists, n_h.

Figure 8. Improvement of t_f by a Thermally Conductive Layer Beneath the sub-floor.

Figure 9. Variation of Required Water Temperature with Hose Thermal Conductivity.

Figure 10. Variation of Heating Efficiency with Underfloor Insulation.

Table Captions

Table 1. Room Position Code, c (Kilkis, 1991).

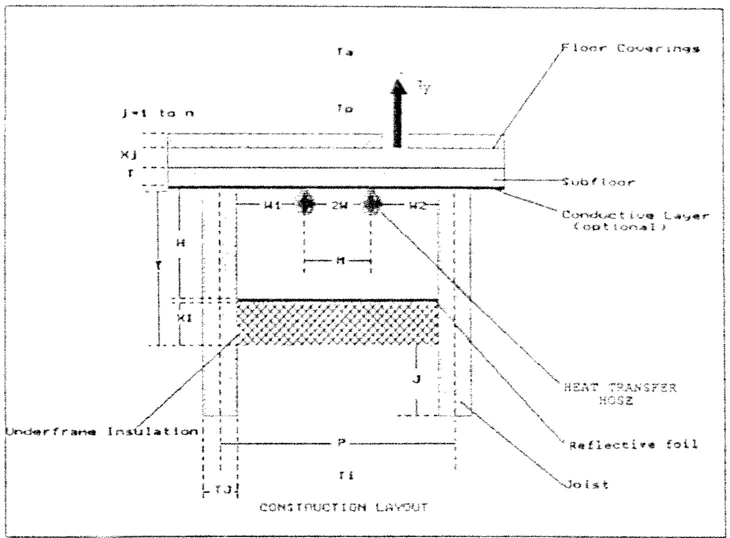


Figure 1.

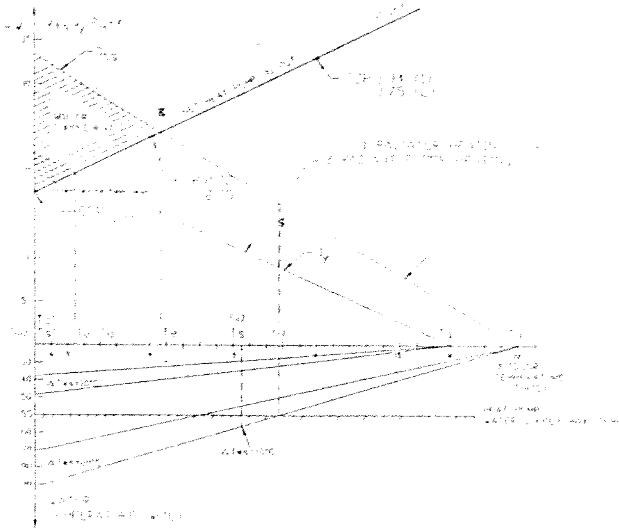


Figure 2-a.

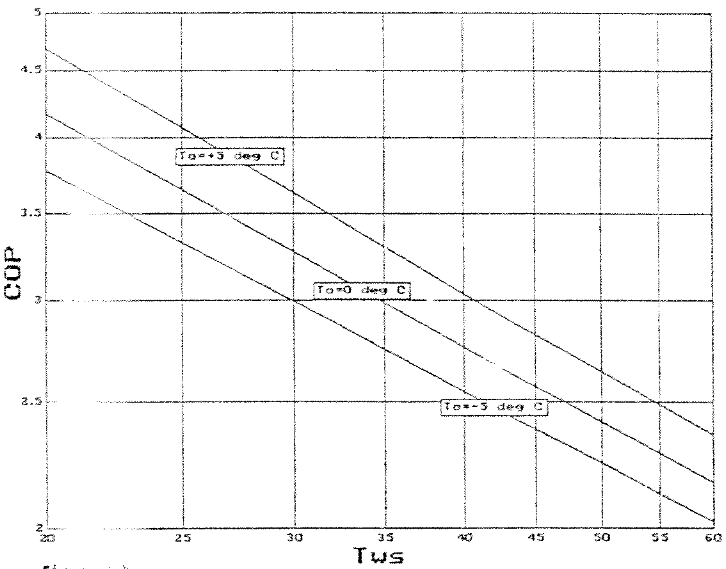


Figure 2-b.

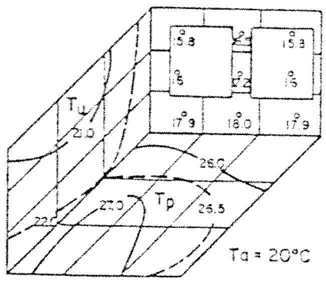


Figure 3.

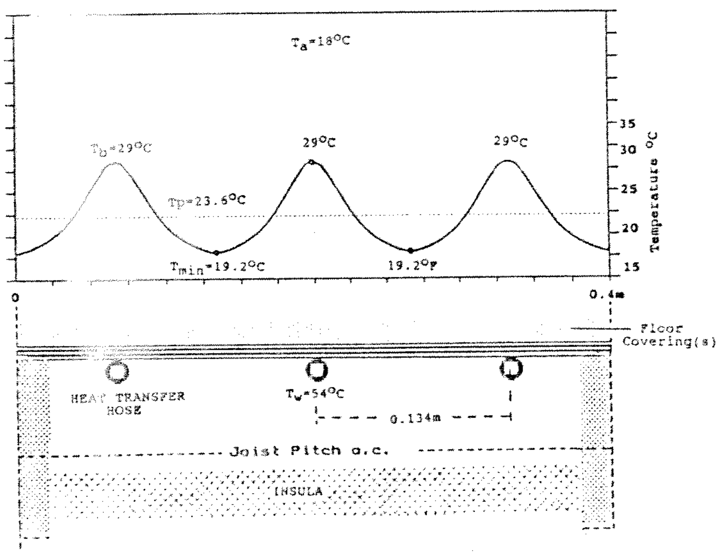
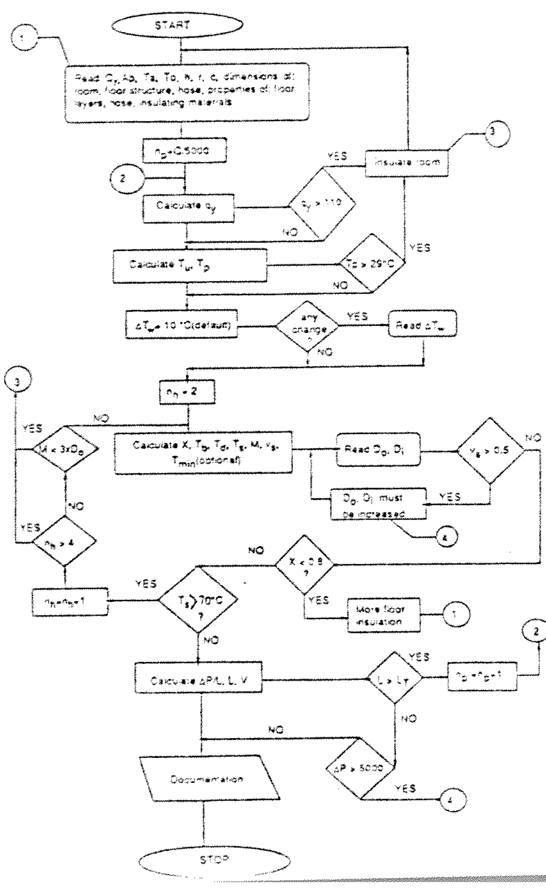


Figure 4.



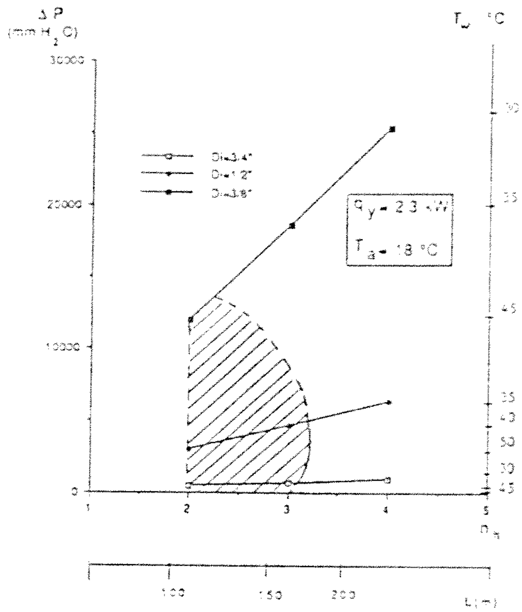


Figure 6.

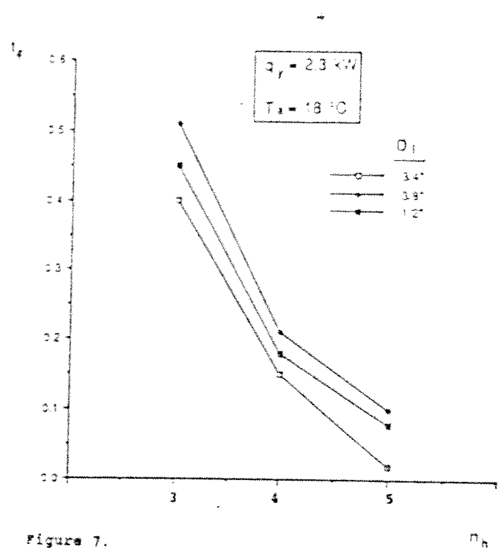


Figure 7.

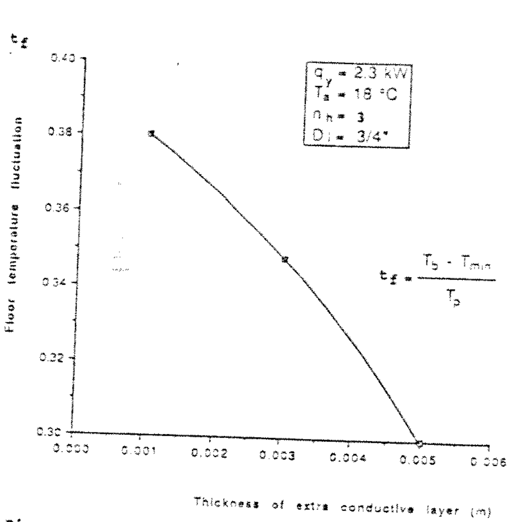


Figure 8.

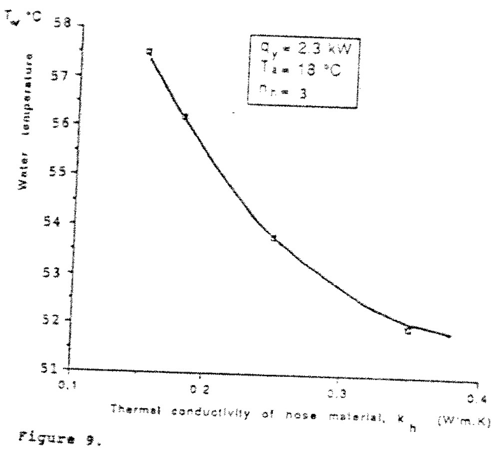


Figure 9.

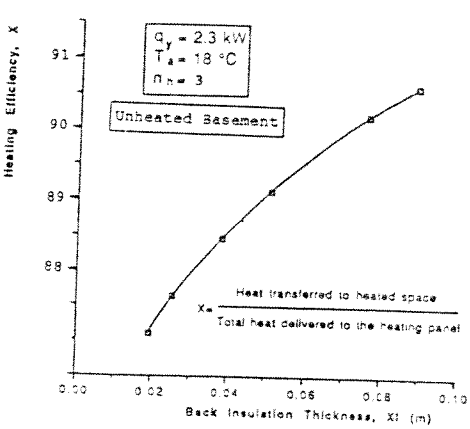


Figure 10.

Room Position Type, c					
Room is located at a:					
Intermediate floor			Top floor		
Inner Room	Facing Outdoor	Corner Room	Inner Room	Facing Outdoor	Corner Room
c:(1)					
0	1	2	1	2	3

(1) For rooms at basement, which are panel heated c=c-0.5

Table 1.

A NEW NOMOGRAPHIC APPROACH TO PANEL HEATING AND COOLING DESIGN

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EXTENDED ABSTRACT

Panel heating and cooling systems are becoming more popular and diversified due to their energy efficient attributes and better tie-in features with alternative energy sources and heat pumps. ASHRAE Handbook:HVAC Systems and Equipment is the only design reference in America and possibly in many other countries with the exceptions of design,product testing and material standards of some other nations for specific applications only,like floor heating (DIN Standards,1990,Turkish Standards,1993). In the same manner,current ASHRAE guidelines provide only limited information in this subject matter. Although panel heating and cooling is known to be rather design forgiving systems,in fact the thermo-hydraulic fundamentals of this type of space conditioning system are quite complicated.

In spite of the advancements in theory and analysis,and better understanding of the mechanism of panel heating and cooling,engineers tend to employ simple and easy design tools rather than more sophisticated and available numerical and analytical models. Today, there are various types of panels and applications. Therefore,any industrially successful algorithm needs to cover all the alternatives and options in a unified design approach.

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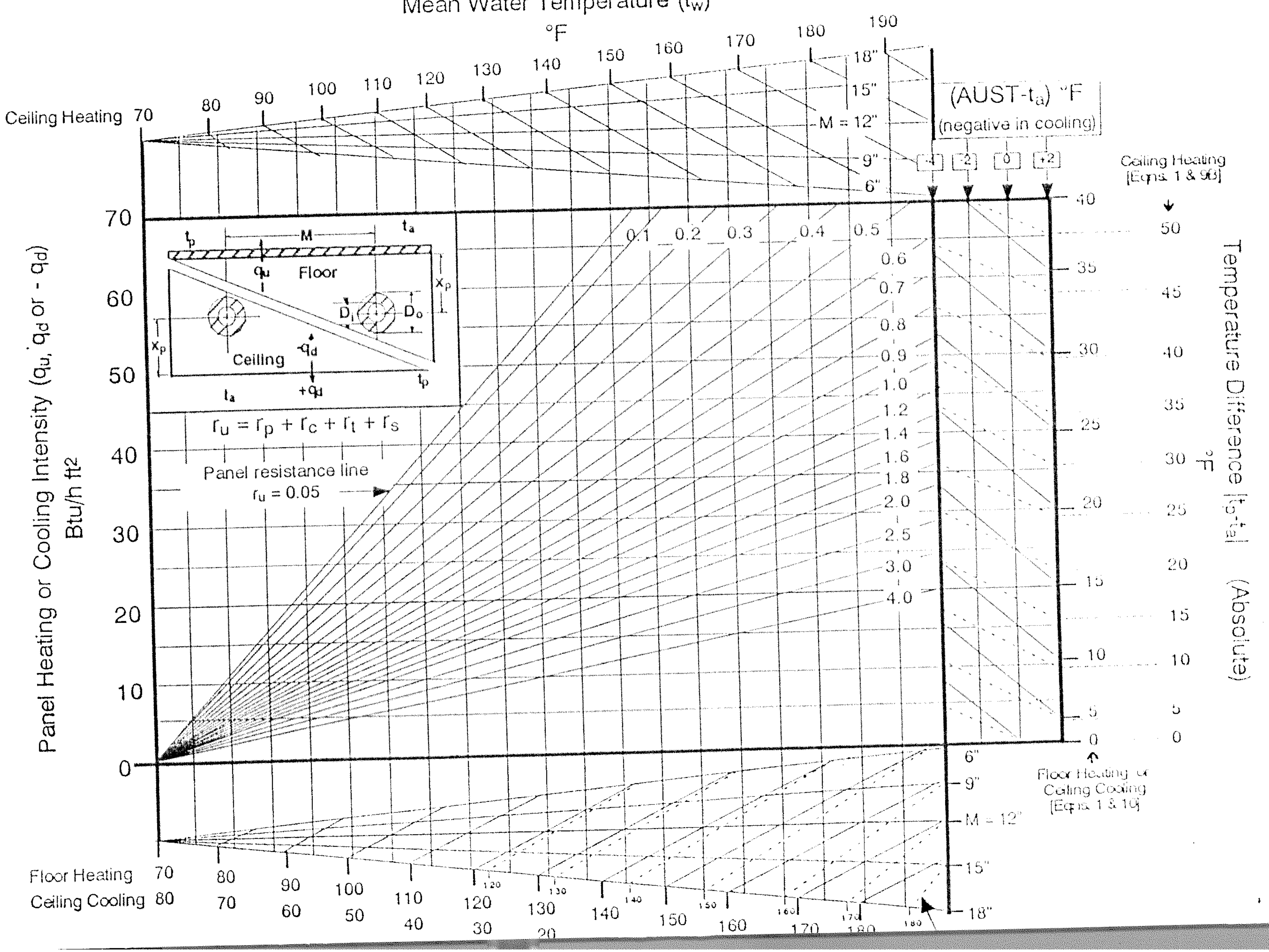
'his makes it even more difficult to develop a simplistic but diverse .lgorithm without sacrificing the accuracy and scope. In order to ieet this challenging requirement, first an analytical algorithm was leveloped which is applicable to any panel type and space :onditioning mode (Kilkis,1993,Kilkis and Sager,1994). This was ›rimarily accomplished by developing a new panel thermal resistance .nd defining a characteristic panel thickness. The model was tested 'ith finite element solutions and good agreement were obtained. A oftware program was also developed for computer aided design urposes (Kilkis and Coley,1994). This program and the algorithm were lso verified with some field results. Then,in order to further implify the design process a "Combo" design nomograph was developed oth in IP and SI units and proposed for ASHRAE publications in the uture (Kilkis,1993). Figure 1 shows the IP version of this new omograph. This single nomograph is capable of predicting the erformance of a design alternative for either panel heating or ooling for virtually any kind of panel type and position, provided hat its panel thermal resistance is defined and calculated. This omograph provides accurate information about the required mean fluid emperature,panel surface temperature,and heating or cooling capacity r a given panel type,construction,hose type and its spacing,indoor hermal conditions and panel thermal resistance. Therefore,in ntrast to other available nomographs in the literature,this unique mograph enables to directly manipulate all the relevant design rameters for an optimization search. This article gives the ndamentals of the analytical algorithm,the technique used in the nstruction of the nomograph and provides case studies with mparisons.

Figure1.Universal Design Nomograph for Heating and Cooling Panels



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ACKNOWLEDGEMENT

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Author is indebted to the generous and timely support of these institutions.

EK-6 BİLGİSAYAR YARDIMI İLE DÖŞEMEDEN ISITMA TASARIM PROGRAMI:TANITIM ve KULLANICI KILAVUZU

Giriş:

Döşemeden ısıtma tasarım programı önceki bölümlerde verilmiş olan algoritmayı takibederek yazılmış bir etkileşimli bilgisayar programıdır. Program,algoritmada ve ilgili Türk Standardında öngörülen kısıt ve tavsiye edilen sınırlar içersinde tasarımın bütünü ile tamamlanmasını sağlar. TURBO PASCAL dilinde yazılmış olup program ve gerekli alt yazılımları havi disket bu raporun 10.4 bölümünde sunulmuştur. Program ismi ISITMA2 olup,disket IBM uyumlu bir bilgisayara sürüldükten sonra bu isimle doğrudan çalıştırılabilir. Programın grafik ekran görüntüleri için EGA.VGA sürücü programı da aynı disket üzerindedir. Ancak bilgisayarın EGA.VGA kartı olması gerekir. Grafik ekran görüntülerinin yazıcı çıktısı için PZP veya benzeri bir yazılımın ise kullanıcı tarafından bilgisayara önceden yüklenmiş olması gerekir. Aksi taktirde sadece satır görüntüler PRINT komutu ile alınabilir. Programın UNIT1,UNIT2,UNIT3 ve UNIT4 adlı dört ana alt yazımına ilaveten şehir bilgilerinin saklandığı veri tabanı LOCS.DAT ve CLIMATE.DAT yazılımları program içersinden çağrılarak düzenlenebilir,ek yapılabilir.

Proje yöneticisi bu programın hazırlanması sırasında grafik ekranların düzenlenmesindeki çalışmaları için Yük.Müh.Meriç Sapçı'ya,veri tabanı düzenlenmesinde ise Michael Coley'e teşekkür eder.

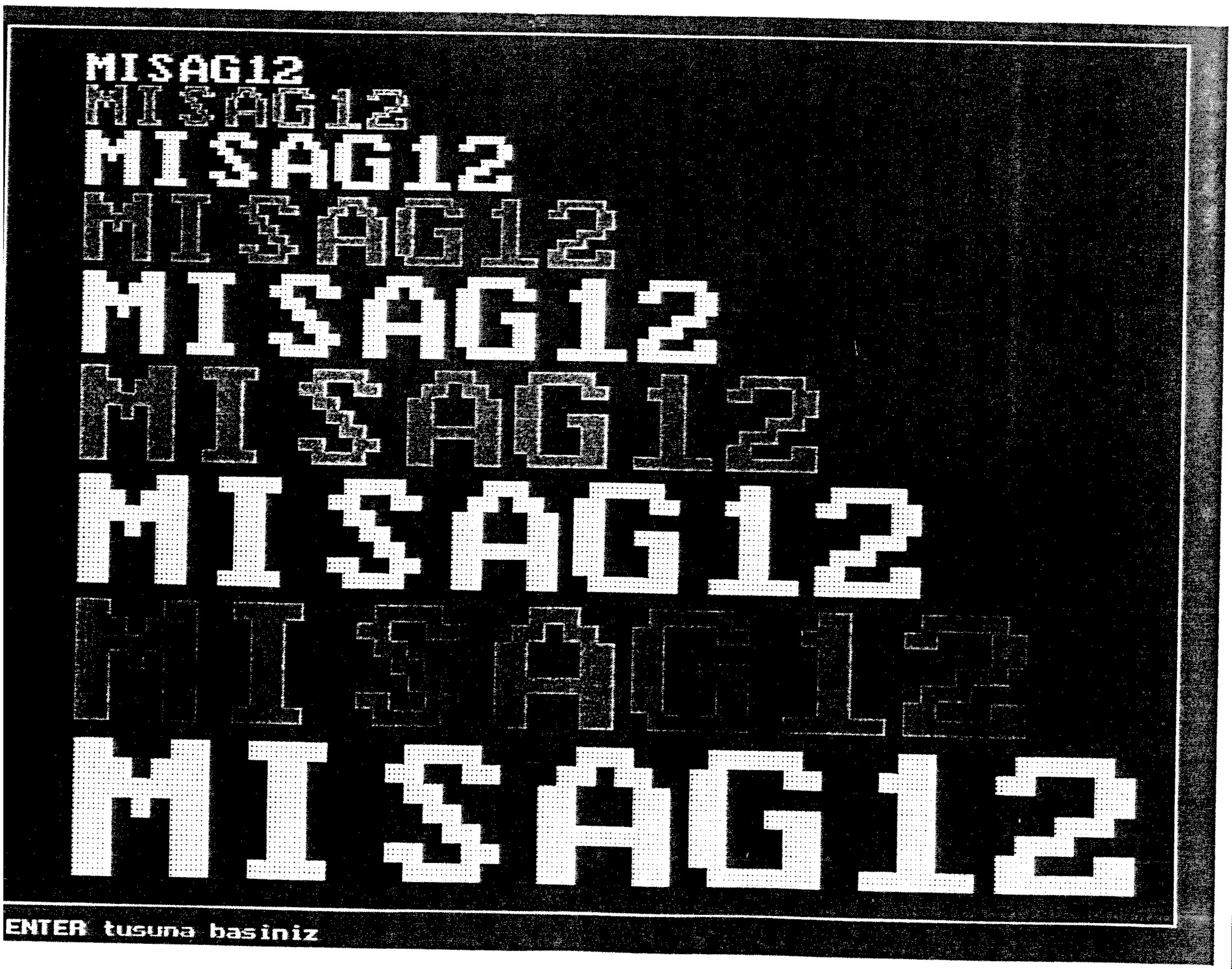
Program ANKARA ili için örnek bir tasarım için çalıştırılmıştı. Tasarımın bütün safhaları ekran görüntüleri ile birlikte verilen açıklamalar ile takibeden bölümde verilmiştir.

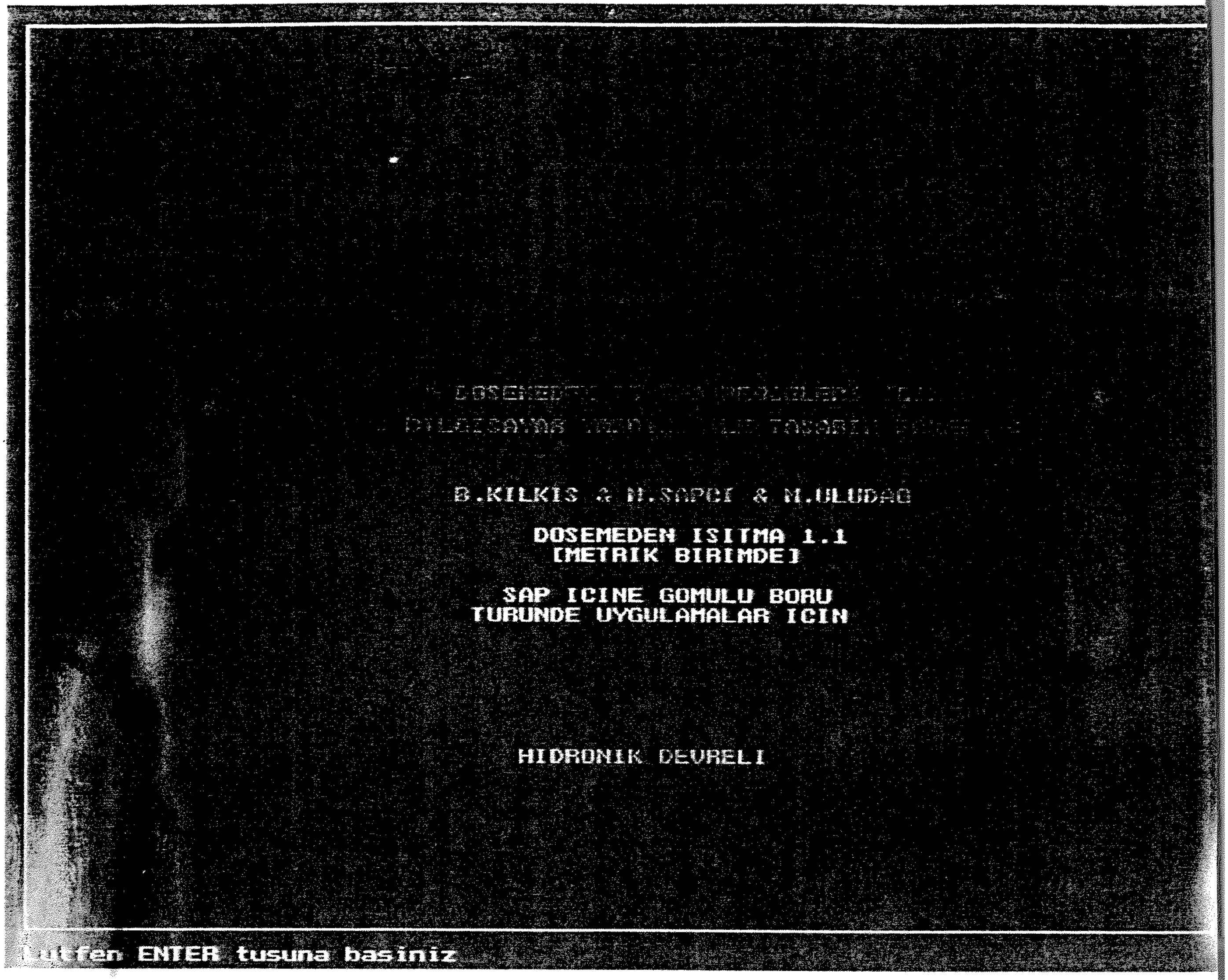
ÖRNEK TASARIM VE AÇIKLAMALAR

Bu tasarımda 15m x 10 m boyutlarında bir odanın döşemeden ısıtma sistemi tasarlanmıştır.

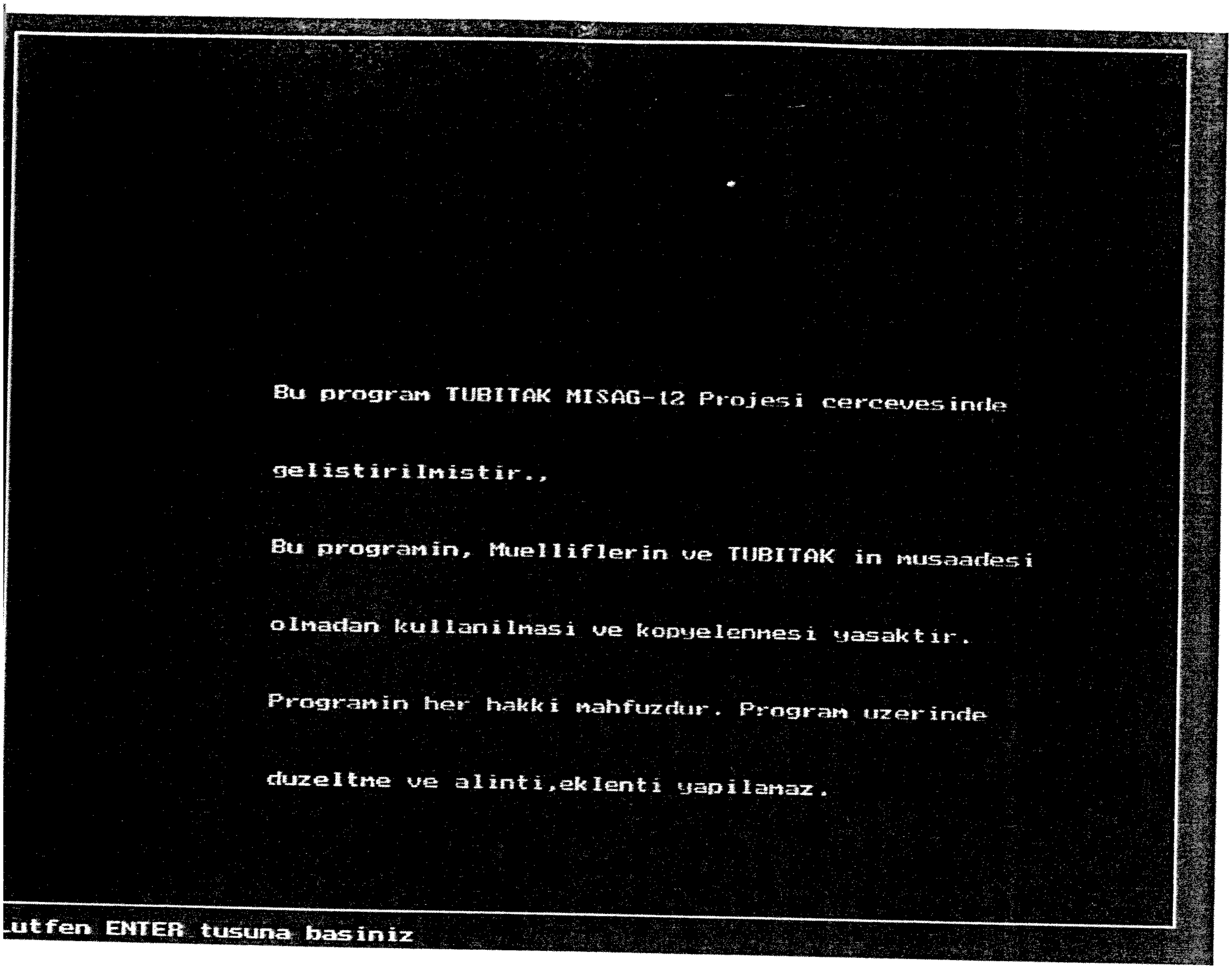
Programaçıldıktan sonra ilk ekran görüntüsü olarak MiSAG12 logosu ekrana gelir(Bakınız EKRAN NO:1).İkinci görüntüde ise tanıtım yapılır.Takibeden ekranda ise yazılımın telif hakları açıklanmştır.

EKRAN NO:1

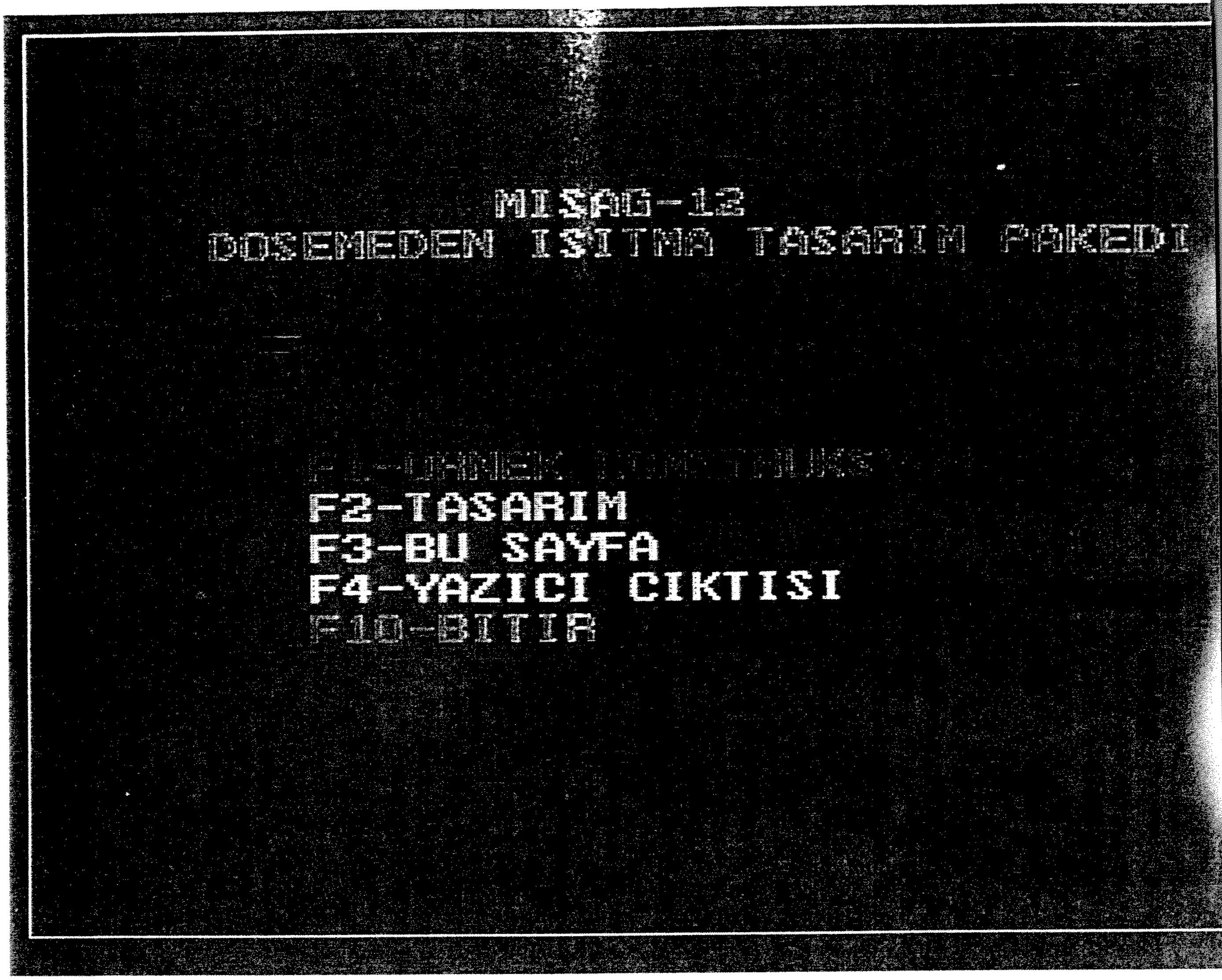

MISAG12
MISAG12
MISAG12
MISAG12
MISAG12
MISAG12
MISAG12
MISAG12
MISAG12
ENTER tusuna basiniz


DOSEMEDEN ISITMA 1.1
[METRIK BIRIMDE]
SAP ICINE GOMULU BORU
TURUNDE UYGULAMALAR ICIN
HIDRONIK DEVRELI
ENTER tusuna basiniz

EKRAN NO:2

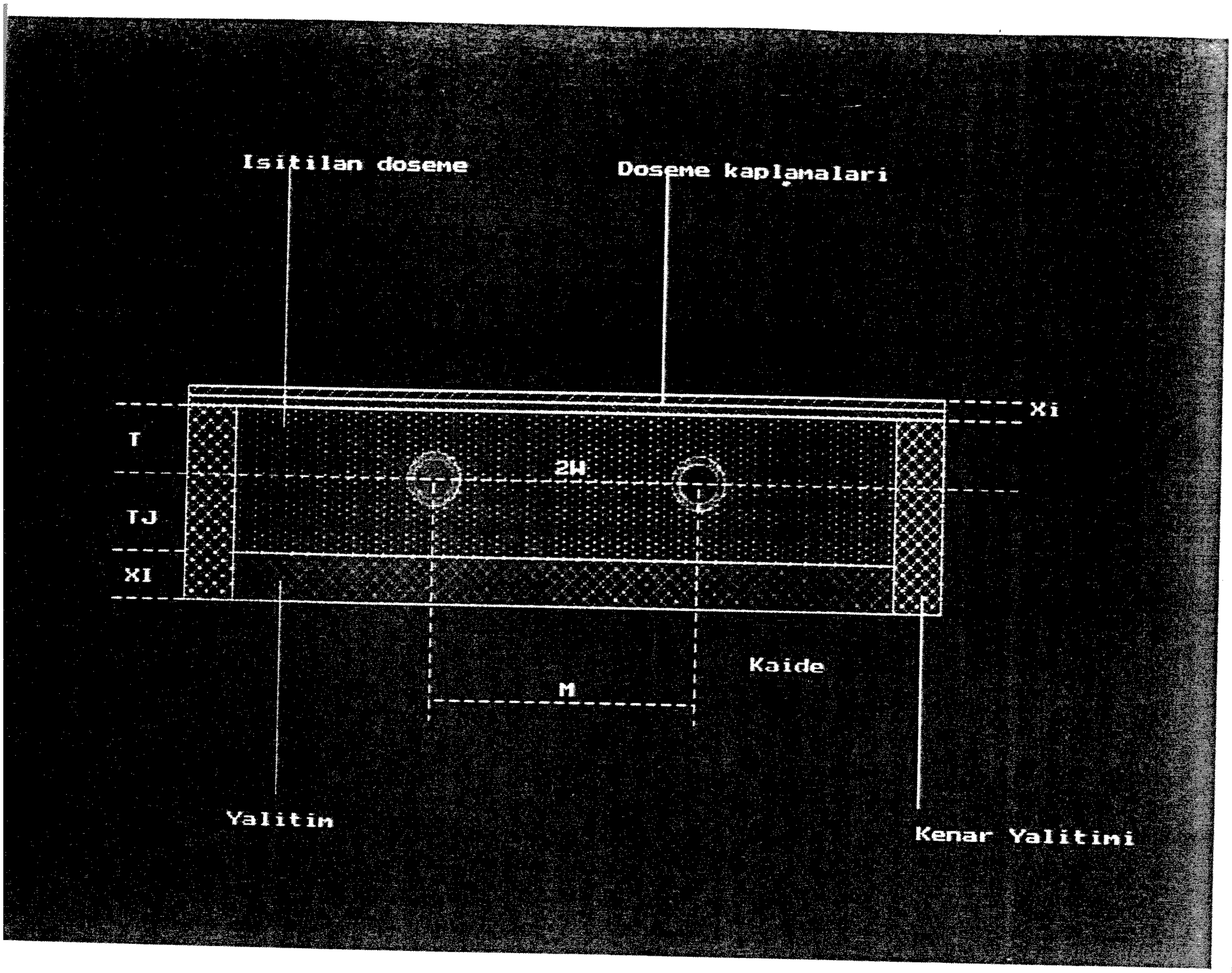

Bu program TUBITAK MISAG-12 Projesi cercevesinde
gelistirilmistir.,
Bu programin, Muelliflerin ve TUBITAK in musaadesi
olmadan kullanilmasi ve kopyelenmesi yasaktir.
Programin her hakki mahfuzdur. Program uzerinde
duzeltme ve alinti,eklenti yapilamaz.
Lutfen ENTER tusuna basiniz

3

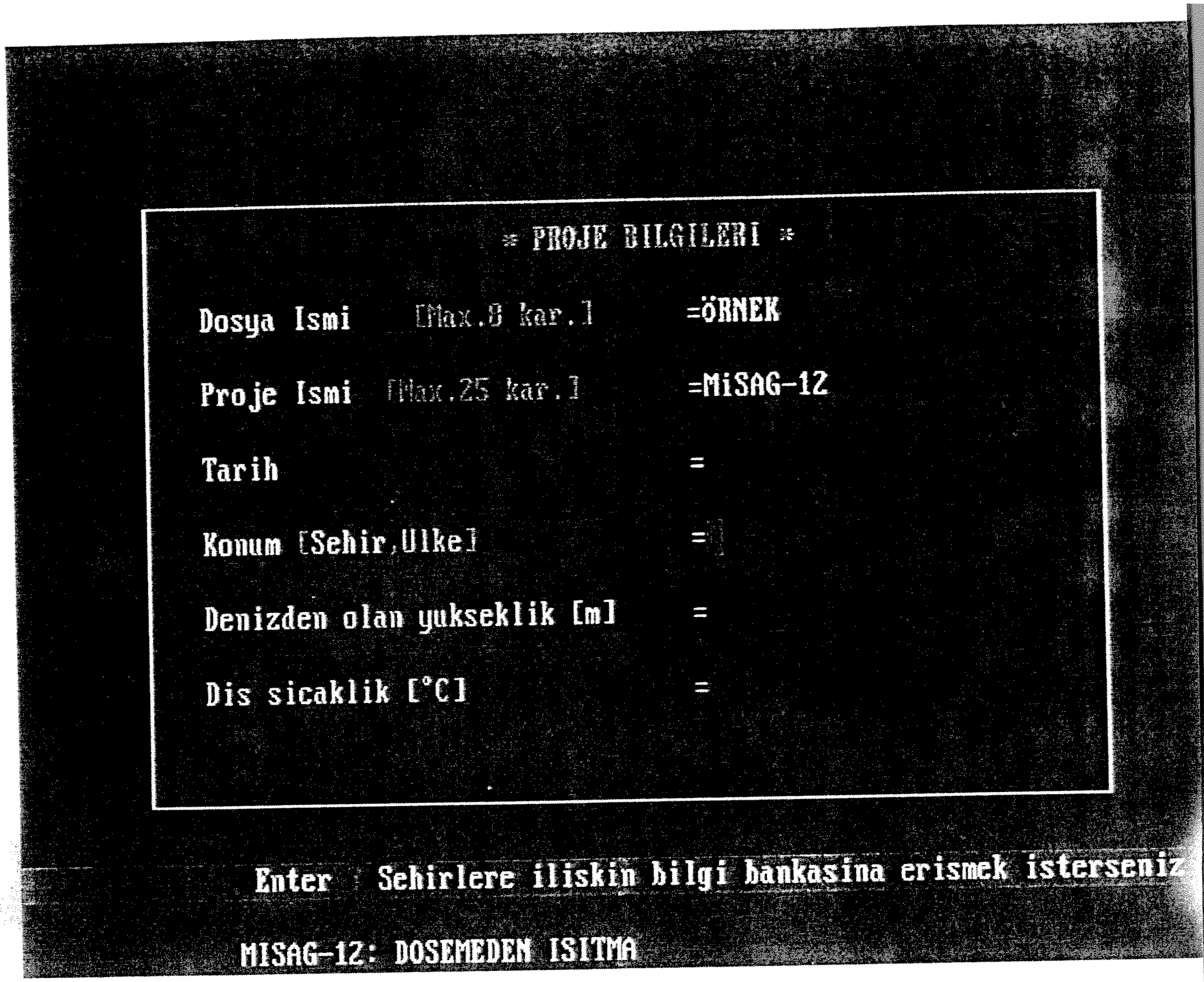


4

Bu ekranda F1 e basıldığında ısıtılan döşeme detayı ve simgeler verilir.(Ekran No:5).
F2 ile tasarım seansı başlar.(Ekran No:6). F10 ile programdan çıkılır.

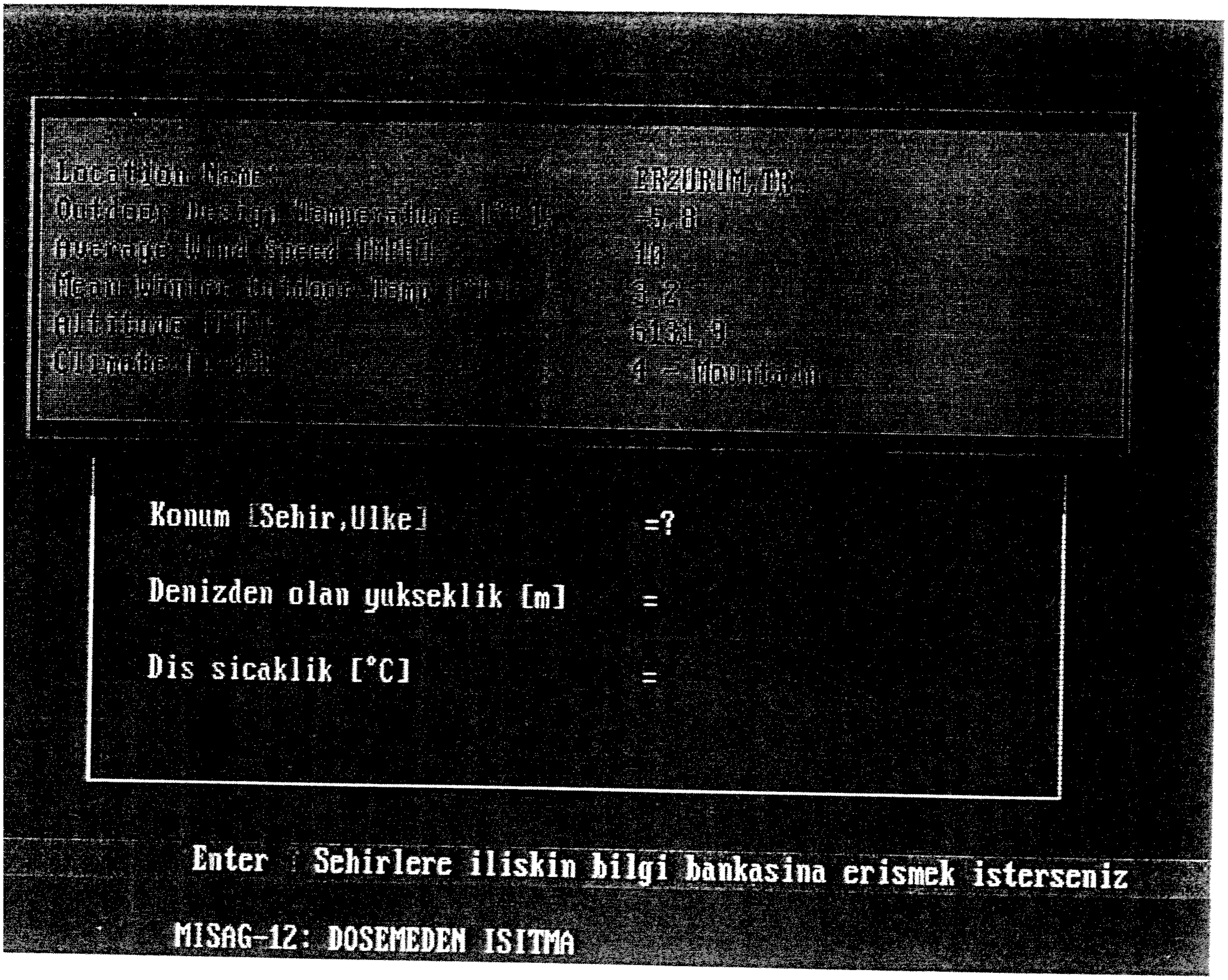


5 (F1 tuşu ile) opsyonel



6 F2 TUŞU ile

Bu ekranda proje bilgileri girilir.
Program ASHRAE ortak çalışmaları da gözetilerek ABD ve KANADA şehir bilgilerini de ihtiva eder. Bu bilgiler ASHRAE Handbook tablolarından alınmıştır. Bu veri tabanına girmek istendiğinde default olarak Sehir,Ülke satırındaki ? işareti ile girilir. (Ekran No:9).Bu ekranda ANKARA,TR satırında gerekli bilgiler görülmektedir. Bu bilgiler ortak kullanım nedeni ile IP biriminde ise de,programa bilgilerin aktarımında otomatik olarak metrik birimlere dönüşür. Başka bir konuma(satıra) yukarı veya aşağı tuşları ile gidilebileceği gibi istenen şehrin ilk bir kaç harfi yazılarak da gidilebilir. Ekran No 7 de ise yeni bir verinin giriliş tarzı anlatılmıştır. Bu ekrana ALT-E ve INSERT tuşları ile girilir.



ALT-E ve
7 INSERT ve tuşları ile

›kranda listede olmayan ERZURUM bilgileri girilmektedir.
JRUM,TR ilk satıra girilir. TR Türkiye kodudur. ABD
ırlerinde bu kod State kodudur. Kullanıcı diğer ülkeler için
›diği kodu seçebilir.
ı sonra sırası ile -21°C dış hesap sıcaklığı,günlük ortalama
sıcaklık (-16°C),rüzgar hızı ve denizden olan yükseklik
›9m)IP birimlerinde girilir. Ortalama dış sıcaklık
›oroloji bülten ve yayınlarından alınır.
m statüsü [1-4] Ülkemizin ısı iklim bölgelerine tekabül eder.
ğin 1 birinci iklim bölgesindeki bir kentimiz için
›rlidir:
IMAN
TEDİL
RA İKLİMİ
RT iklim ve/veya dağlık yöre.

RUM için 4 girilmiştir.
tuşu ile menüye dönülür.(Ekran No:8).

Select a Location to Edit	Temp	Wind	Mean	Alt	Climate
ELKO, NV	-2°F	4 MPH	8°F	5050 FT	Cold
ELLENSBURG, WA	6°F	10 MPH	16°F	1735 FT	Moderate
ELMIRA, NY	1°F	10 MPH	11°F	955 FT	Moderate
ELY, NV	-4°F	9 MPH	6°F	6253 FT	Cold
EMPORIA, KS	5°F	10 MPH	15°F	1210 FT	Moderate
ENID, OK	13°F	10 MPH	23°F	1307 FT	Moderate
ERIE, PA	9°F	0 MPH	19°F	731 FT	Sea Side
ERZURUM, TR	**-5.8°F**	**10 MPH**	**3.2°F**	**6131.9 F**	**Mountain**
ESCANABA, MI	-7°F	10 MPH	3°F	607 FT	Sea Side
ESCONDIDO, CA	41°F	10 MPH	51°F	660 FT	Sea Side
ESTEVAN, SK	-25°F	10 MPH	10°F	1884 FT	Moderate
EUGENE, OR	22°F	7 MPH	32°F	359 FT	Moderate
EUREKA/ARCATA, CA	33°F	5 MPH	43°F	218 FT	Sea Side
EVANSTON, WY	-3°F	10 MPH	7°F	6780 FT	Cold
EVERETT, WA	25°F	6 MPH	35°F	596 FT	Sea Side
FAIRBANKS, AK	-47°F	5 MPH	-37°F	436 FT	Moderate

Enter : Sehirlere iliskin bilgi bankasina erismek isterseniz

ESC Exit RETURN Edit INS Add DEL Delete | Use Arrow Keys to Move

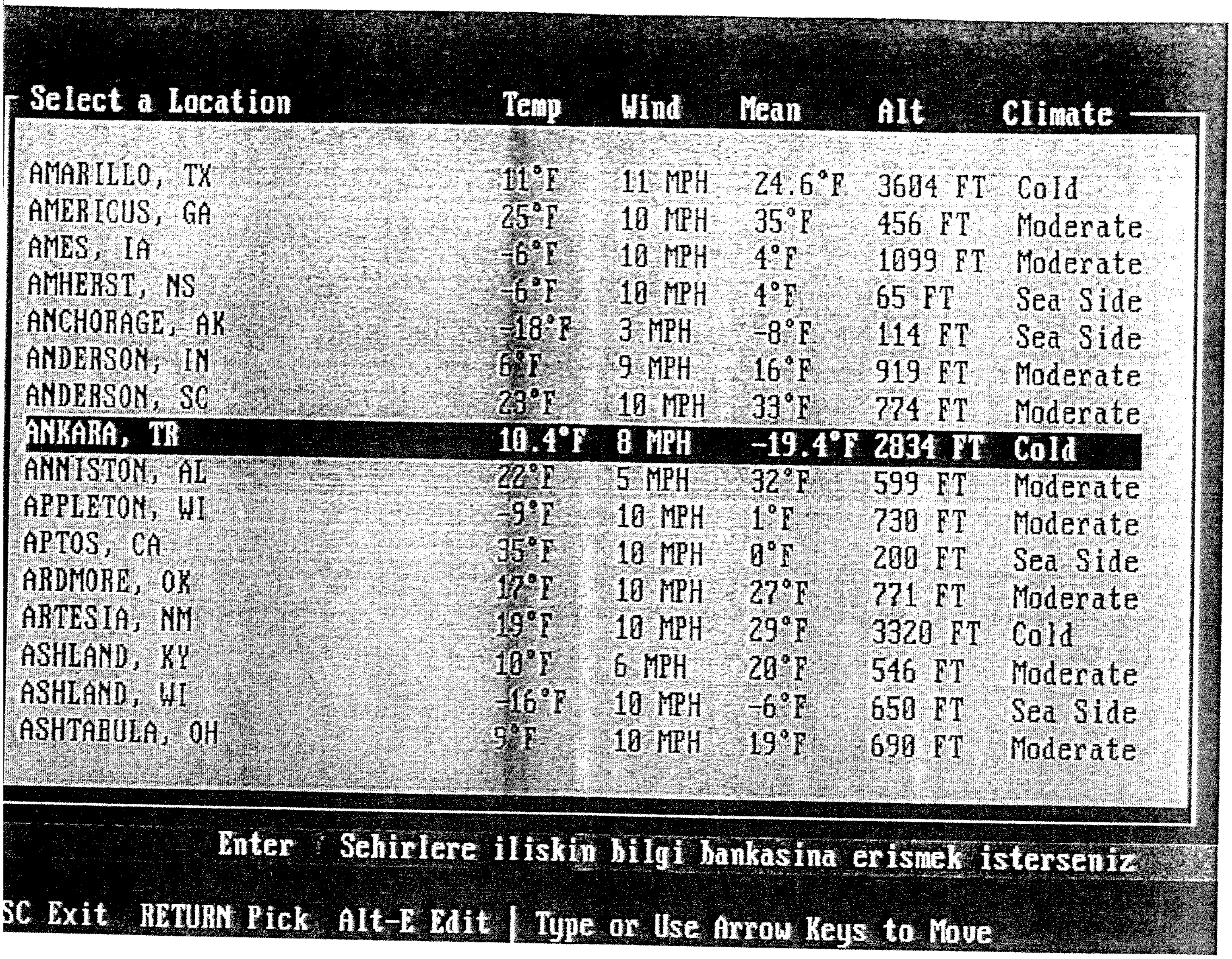
8 Yeni girilen şehir bilgileri (IPS biriminde)
NOT: DEL Tuşu ile tekrar bu bilgi silinebilir,
INSERT Tuşu ile başka bilgi girilebilir.
↑ tuşu veya ↓ ile şehirler taranabilir.
Şehir biliniyor ise,örneğin ANKARA,TR, sadece ANK yazılarak bu şehre ulaşıl
Bu ekran çalışması bitince ENT tuşu ile ana ekrana,seçilen şehir ile birlik
dönülür.

Bu ekranda ERZURUM bilgileri görülmektedir.

Eğer ENTER e basılır ise ERZURUM bilgileri Proje ekranına dönülerek,metrik birimlerde otomatik olarak gözükür:

* PROJE BILGILERI *

Dosya Ismi [Max.8 kar.]	=test
Proje Ismi [Max.25 kar.]	=
Tarih	=
Konum [Sehir,Ulke]	=ERZURUM, TR
Denizden olan yukseklik [m]	=1869
Tasinim duzeltme katsayisi	=0.893
Dis sicaklik [°C]	=-21
Devam edecekmisiniz ? [E/H]	=E



Select a Location

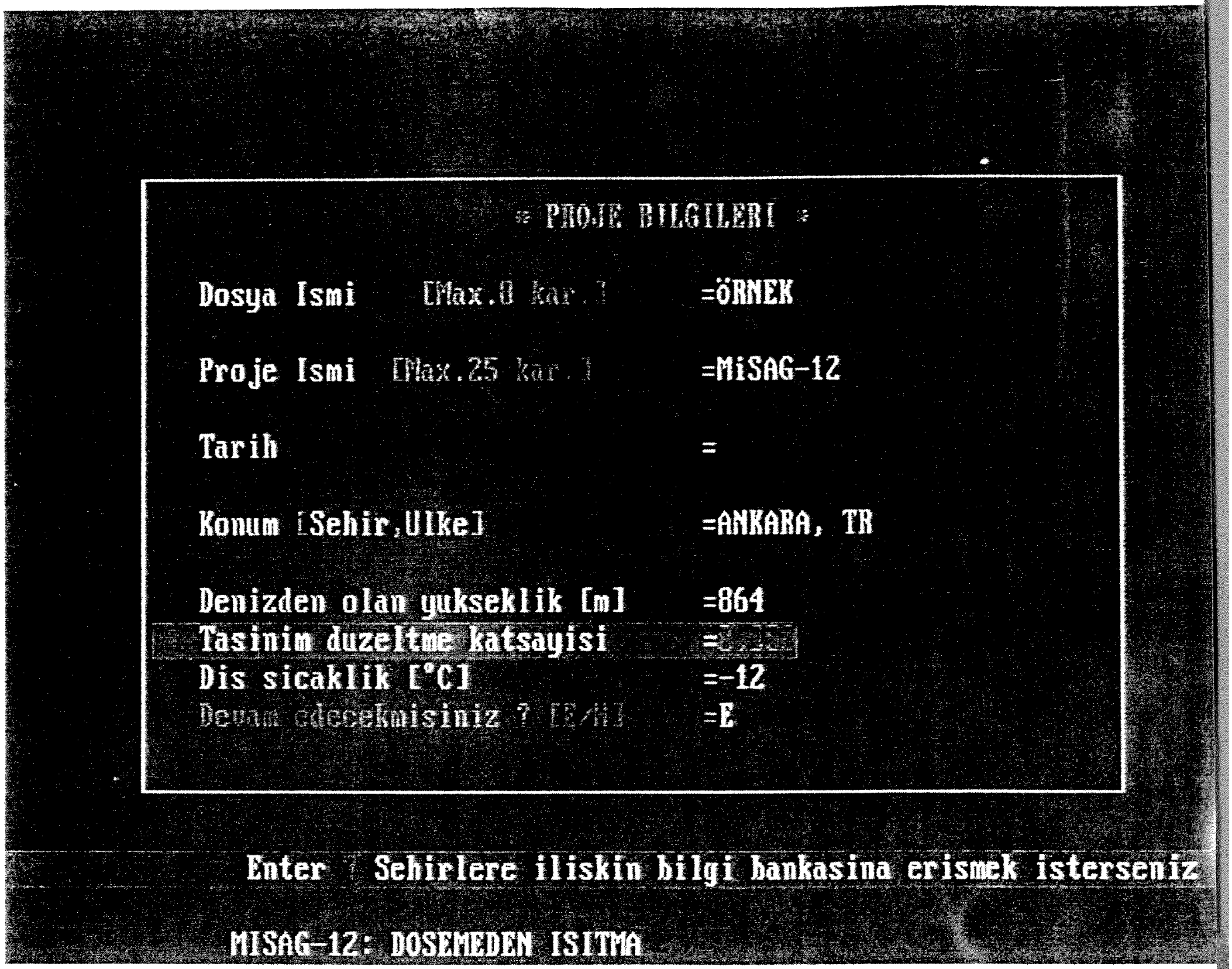
Select a Location	Temp	Wind	Mean	Alt	Climate
AMARILLO, TX	11°F	11 MPH	24.6°F	3604 FT	Cold
AMERICUS, GA	25°F	10 MPH	35°F	456 FT	Moderate
AMES, IA	-6°F	10 MPH	4°F	1099 FT	Moderate
AMHERST, NS	-6°F	10 MPH	4°F	65 FT	Sea Side
ANCHORAGE, AK	-18°F	3 MPH	-8°F	114 FT	Sea Side
ANDERSON, IN	6°F	9 MPH	16°F	919 FT	Moderate
ANDERSON, SC	23°F	10 MPH	33°F	774 FT	Moderate
ANKARA, TR	10.4°F	8 MPH	-19.4°F	2834 FT	Cold
ANNISTON, AL	22°F	5 MPH	32°F	599 FT	Moderate
APPLETON, WI	-9°F	10 MPH	1°F	730 FT	Moderate
APTOS, CA	35°F	10 MPH	0°F	200 FT	Sea Side
ARDMORE, OK	17°F	10 MPH	27°F	771 FT	Moderate
ARTESIA, NM	19°F	10 MPH	29°F	3320 FT	Cold
ASHLAND, KY	10°F	6 MPH	20°F	546 FT	Moderate
ASHLAND, WI	-16°F	10 MPH	-6°F	650 FT	Sea Side
ASHTABULA, OH	9°F	10 MPH	19°F	690 FT	Moderate

Enter : Sehirlere iliskin bilgi bankasina erismek isterseniz

SC Exit RETURN Pick Alt-E Edit | Type or Use Arrow Keys to Move

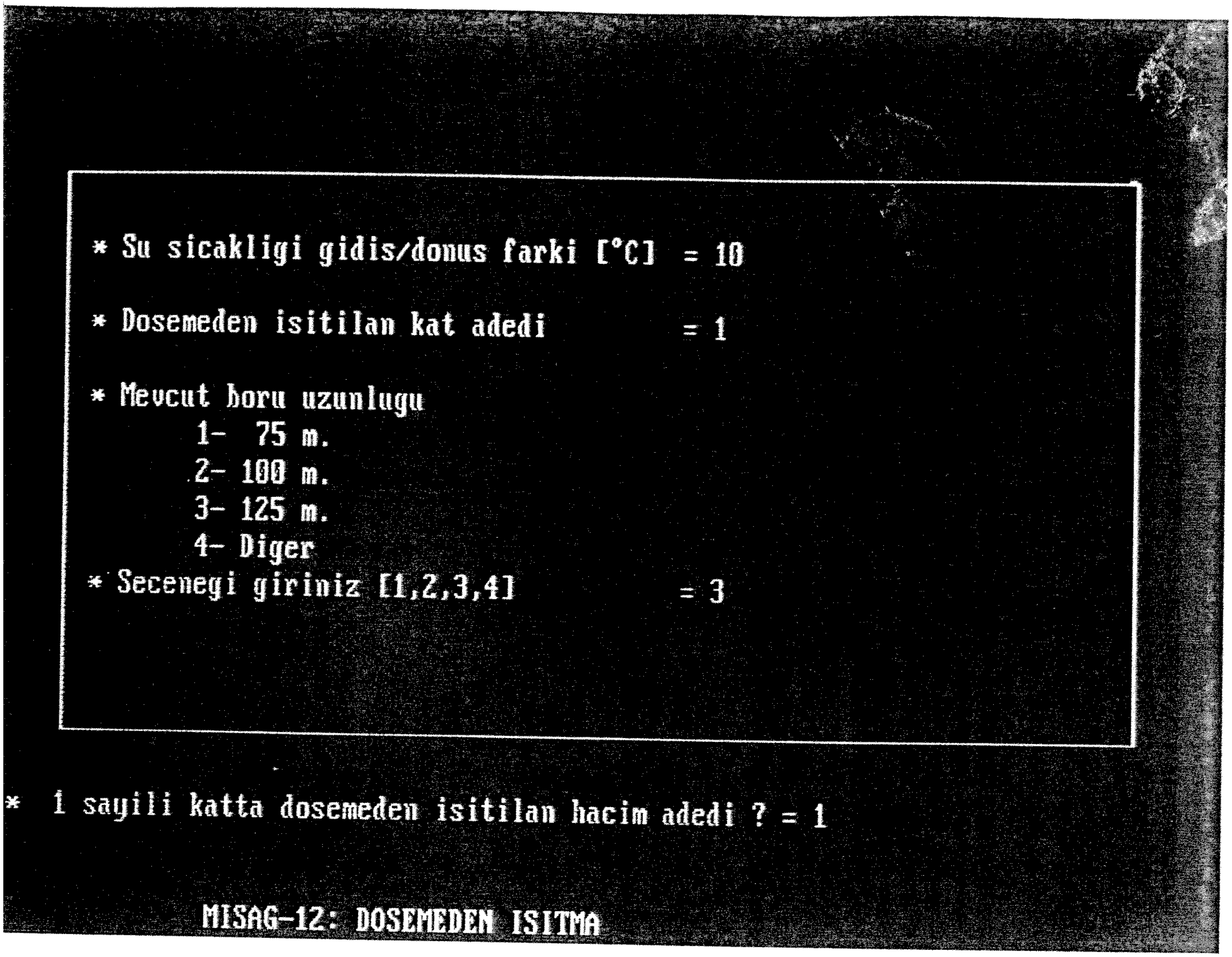
9 (?) TUŞU ile (opsyonel)

Ancak Projede Ankara kenti kullanılacağı için önce ANKARA satırına gidilerek (Ekran No:9) ENTER tuşuna basılmıştır.



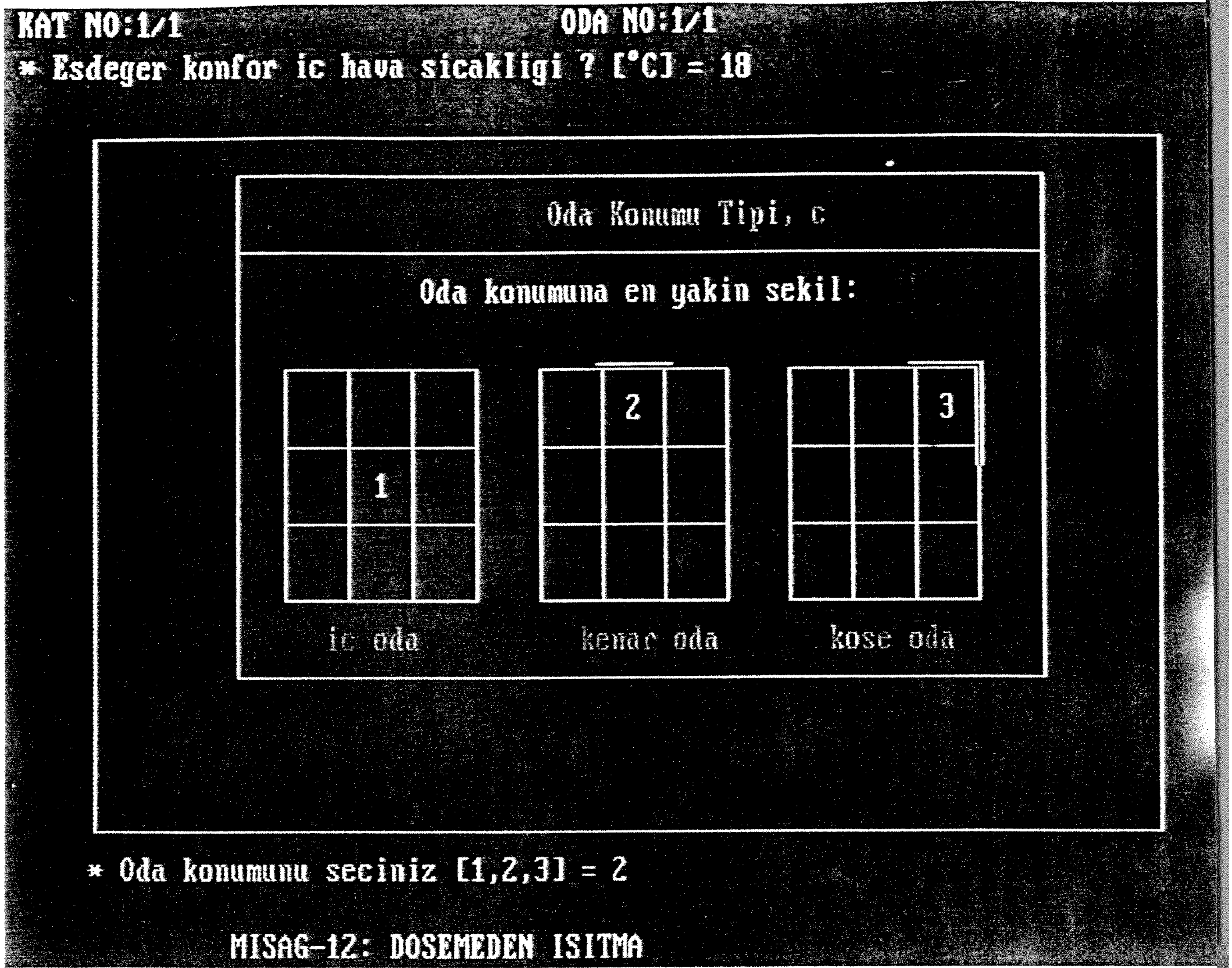
10

Bu durumda Proje ekranına otomatik olarak ANKARA,TR bilgileri gelir(Ekran No:10).
Bu ekranda programın hesapladığı yükseklik düzeltme katsayısı (0.950) de basılmıştır. Dış sıcaklık -12 defaultu aynen girilerek ve E tuşuna basılarak bu ekrandan çıkılır. H ile ekran başına dönülür.
Eğer veri tabanında mevcut bir şehir girilecek ise o zaman ? tuşu ile veri tabanına girilmeksizin proje ekranında şehir,ülke kodu girilmesi yeterlidir.



1

Bu ekranda sisteme gidiş ve dönüş akışkan sıcaklık farkı (DEFAULT:10°C) girilir. Bina tek katlı olup kat ededi 1 girilmiştir. Isı transfer borusunun stok uzunluğu menüden veya istenen herhangi bir değer m olarak girilir. Örnekte 125 m girilmiştir.
Örnek problemin kısa sürmesi için ısıtılan oda sayısı 1 olarak girilmiştir.



* Oda cati katinda mi ? [E/H] = H

* Oda bodrum katinda mi ? [E/H] = H

* Oda dosemesi dikdortgen (kare) mi ? [E/H] = E

12 Default olarak 18°C oda iç sıcaklığı girilmiştir. Odanın bina içersindeki konumu tablodan seçilerek (2:tek cephesi dışa bakıyor).Altta ufak pencerede sıra ile ekrana gelen üç soruya sırası ile H,H ve E girilmiştir. Son soruya H(ayır) cevabı girildiğinde program oda çevre uzunlugu ve/veya alanını sorar. Bu örnekte E girildiği için oda boyutları takibeden ekranda sorulmuştur. Oda boyutları 10m ve 15 m olarak girilmiştir.

KAT NO:1/1 ODA NO:1/1

* Odanin boyunu girin, x [m] : 15

* Odanin enini girin, y [m] : 10

* Degerler dogru mu ? [E/H] = E

Odanin olculerine gore tasinim duzeltmesi = 0.932
ENTER tusuna basiniz

MISAG-12: DOSEMEDEN ISITMA

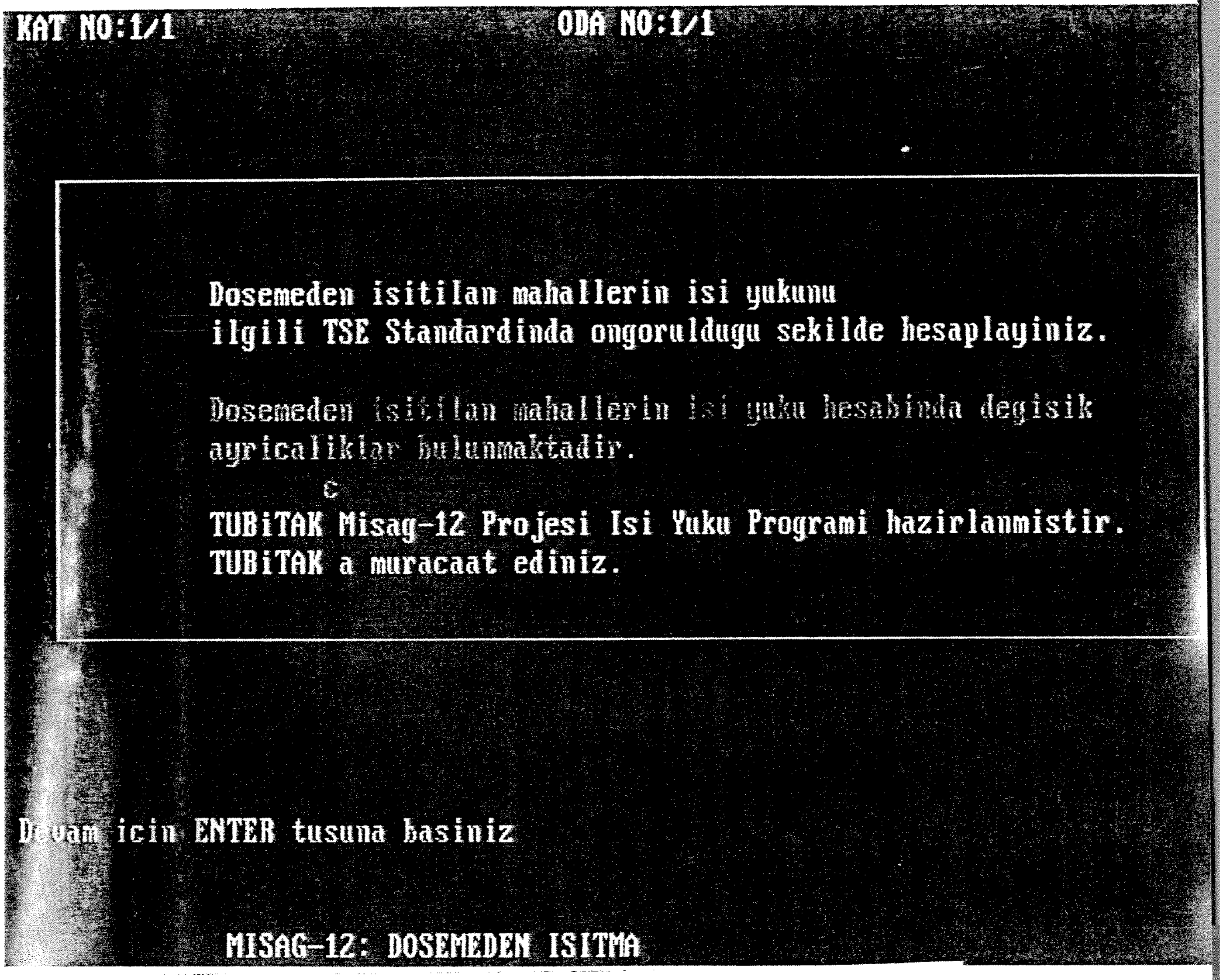
Isitilmayan yuzeylerin alan agirlikli ortalama sicakligi = 16.8 °C

Devam icin ENTER tusuna basiniz

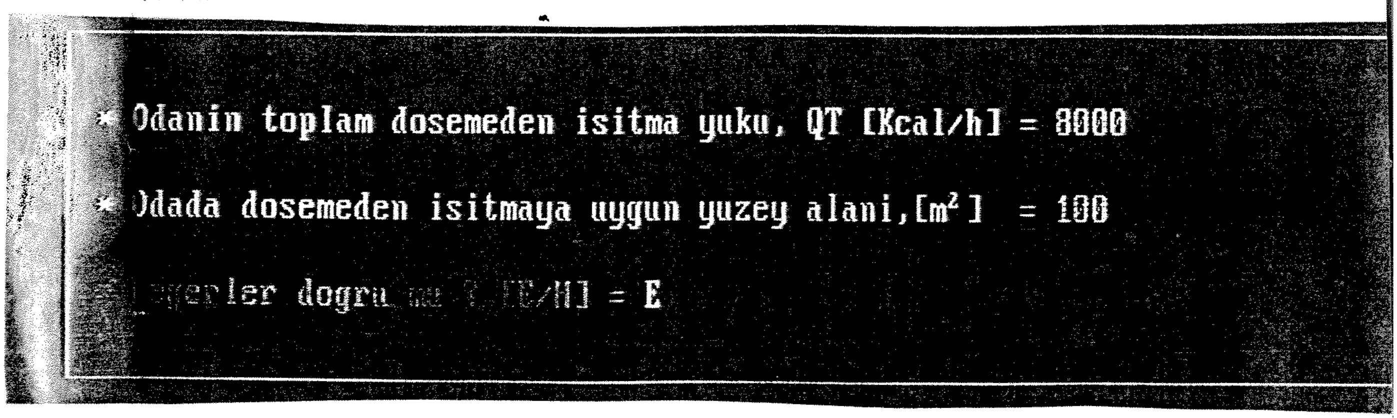
13

Oda boyutlarına göre hesap düzeltme katsayısı program tarafından 0.932 olarak tesbit edilmiştir.

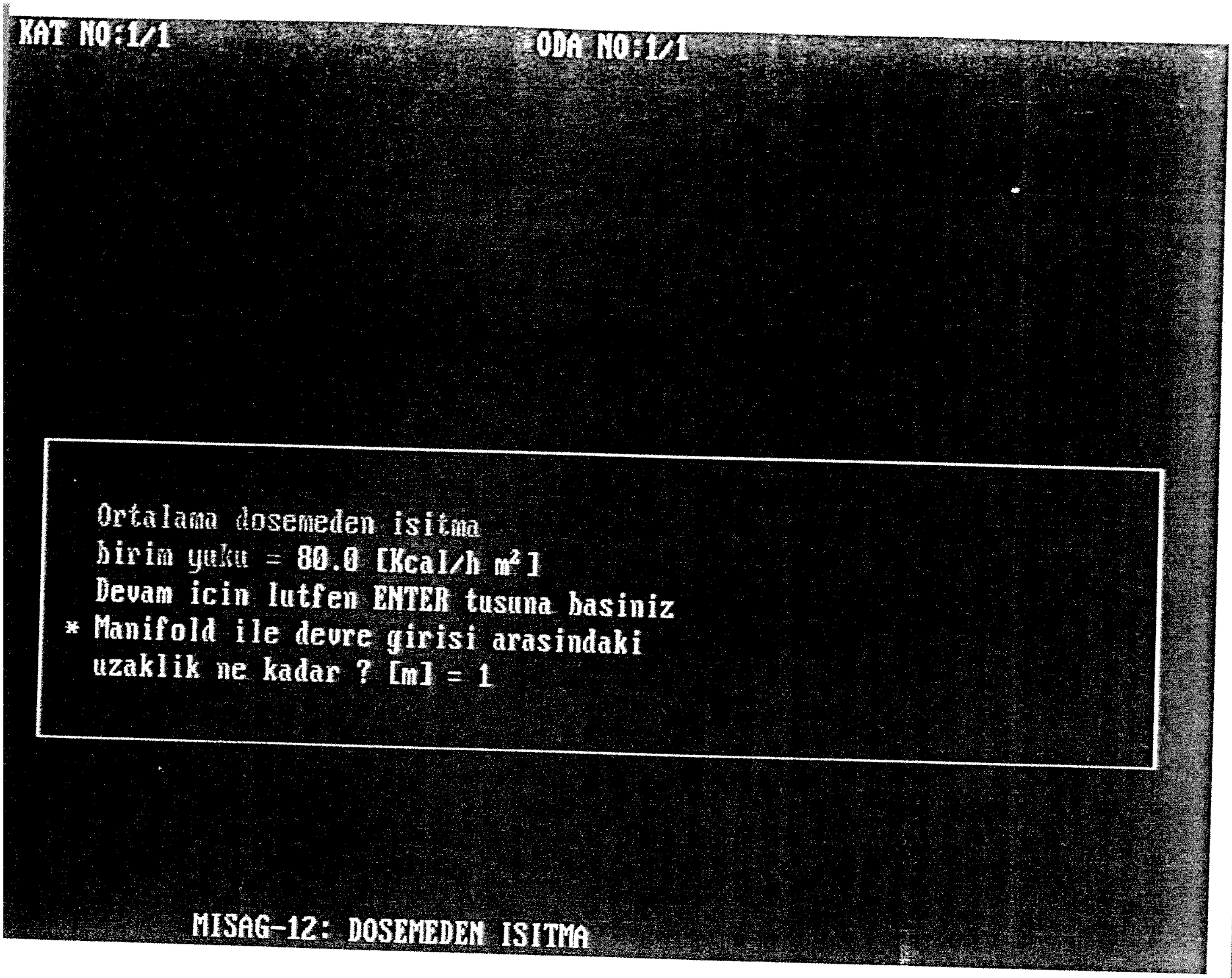
Isıtılmayan iç yüzeylerin ortalama sıcaklığı 16.8°C olarak hesaplanmıştır.

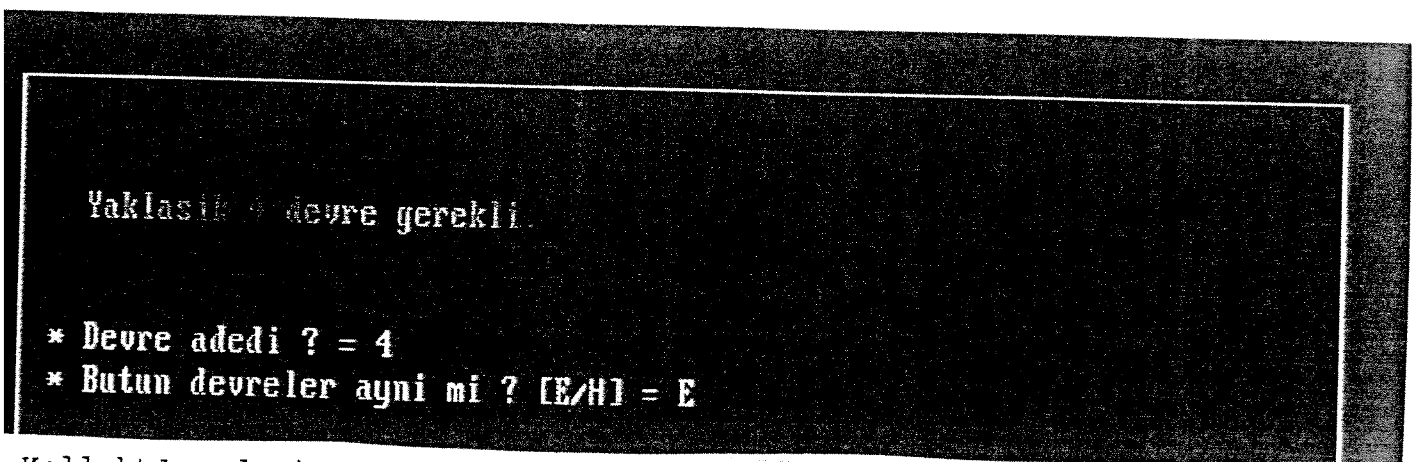


14 Bu ekranda ısı yükünün özel hesab edilmesi gereği hatırlatılmaktadır.



15 Bu ekranda odanın hazırlanan yeni TSE standardına göre hesaplanan (8000Kcal/h) değeri girilmiştir. Bu projede geliştirilmiş bulunan ısı yükü hesap programı raporun 10.4 bölümünde verilmiştir. Isıtmada kullanılabilir döşeme alanı 100 m dir.





Kollektör oda içersinde olduğu için yaklaşık 1 metre uzaklık girilmiştir.
Program gerekli hidrolik devre sayısını 4 olarak tavsiye etmiş,kullanıcıda bunu kabul etmiştir. Her devre birbirinin aynıdır. Aksi durumda program dört devreyi de peş peşe tasarımlayacaktır. E girildiği için tek bir devrenin tasarımı yeterli olacaktır.

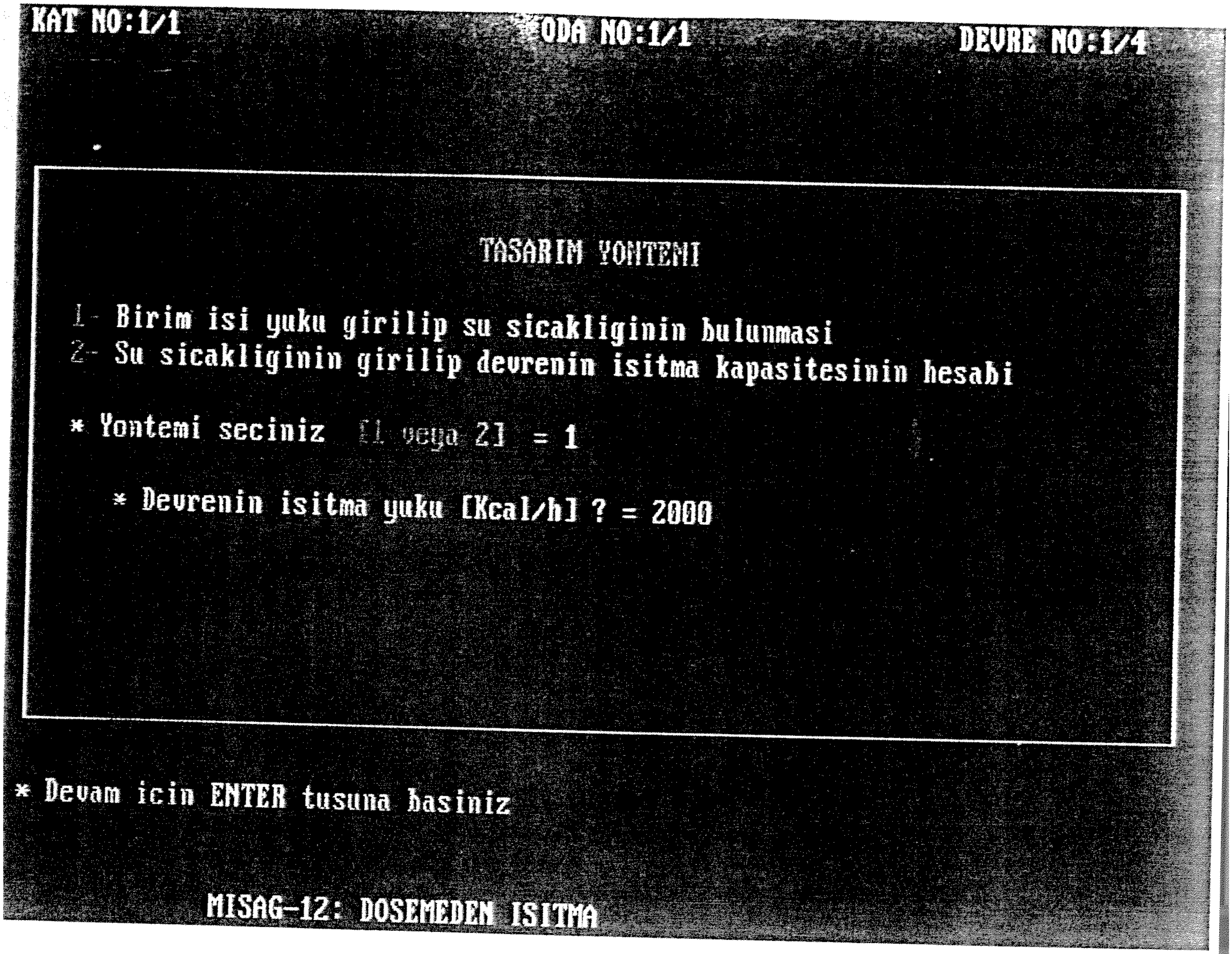
KAT NO:1/1 ODA NO:1/1 DEVRE NO:1/4

Devre numarasi[1] icin tasarim basliyor
* Manifold numarasi ? = 1
* Manifolda gidis/gelis toplam boru uzunlgu [m] ? = 3
* Devre icinde ayri modulasyon var mi ? [E/H] = H
* Devrenin doseme yuzeyinin boyu [m] ? = 5
* Devrenin doseme yuzeyinin boyu [m] ? = 5
Devrenin doseme yuzey alani = 25.00 m^2
* Bu alan dogrumu ? [E/H] = E

MISAG-12: DOSEMEDEN ISITMA

18

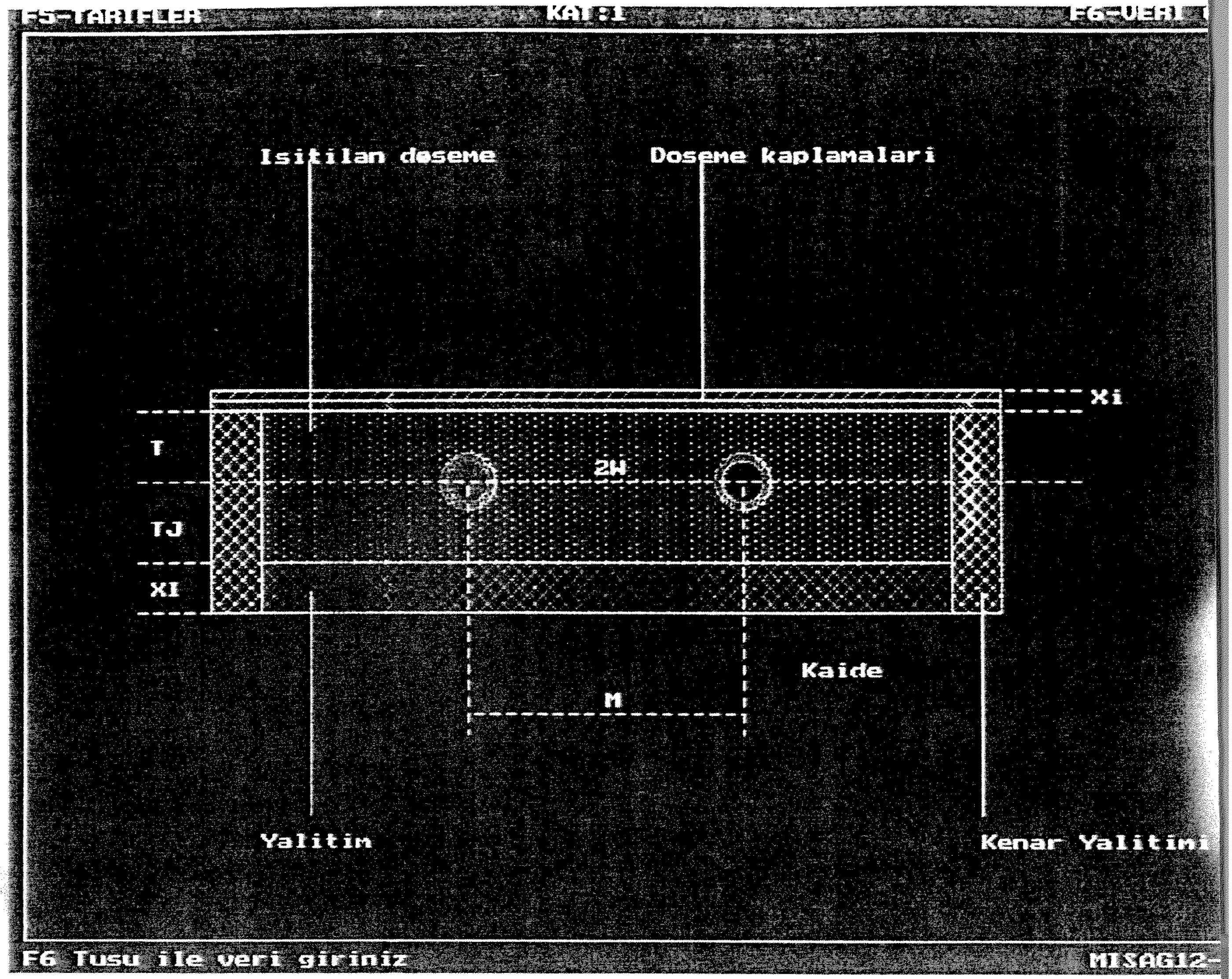
Bu ekranda tasarım statüsü artık ilk satırda (Kat no,Oda no,Devre no olarak)devamlı bir şekilde gösterilmektedir.
Devreden çıkan borunun manifolda(kollektör) dönüşü 2 m olduğundan toplam 3m girilmiştir.
Devre eni boyu(yüzeydeki) girilmiş,ve alan (25m^2)programca sorularak doğrulanmıştır.



.9

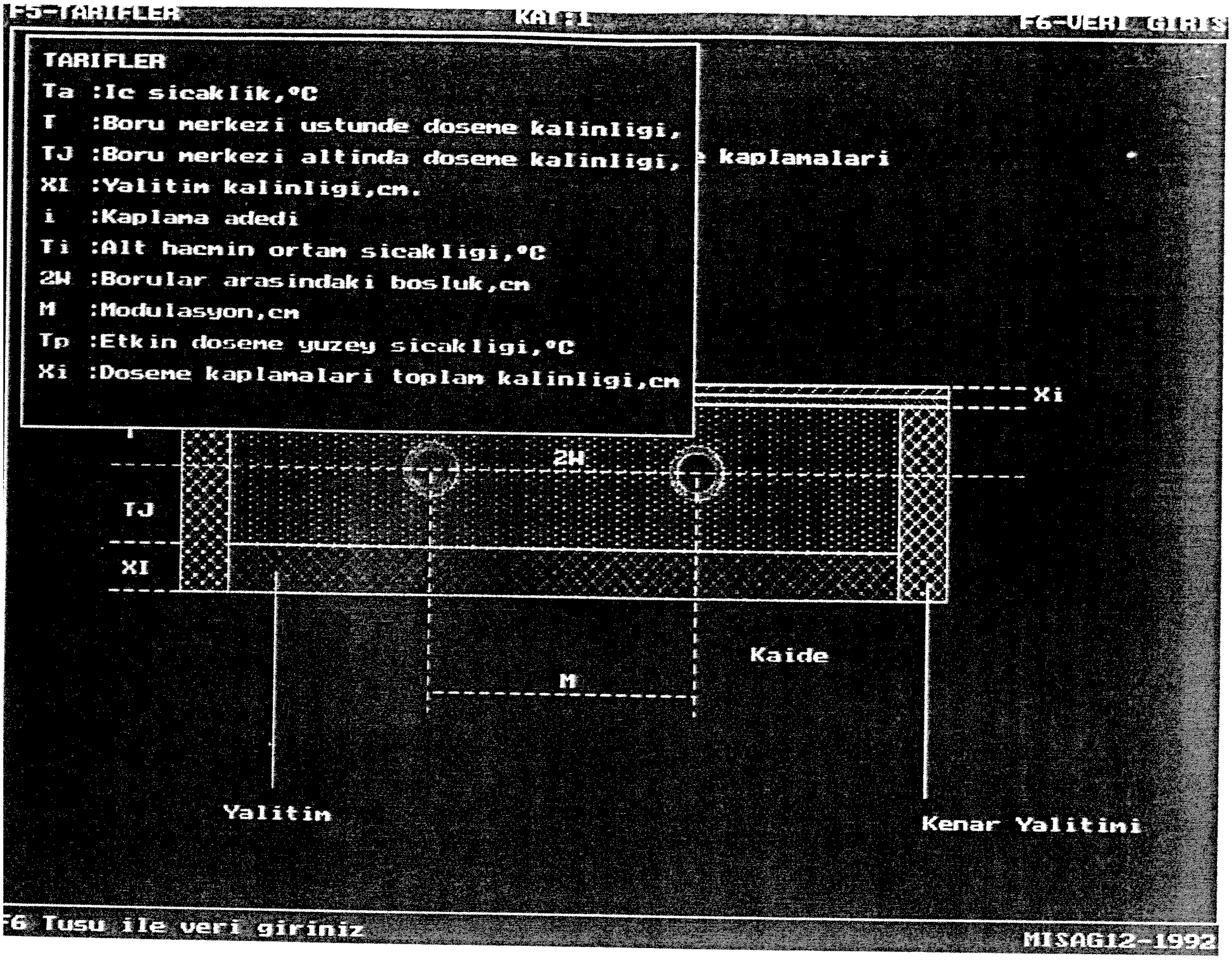
Tasarıma bu noktadan sonra iki şekilde devam edilebilir. ılk metod da birim ısı yükü(bu tasarımda 80 Kcal/hm^2) kullanılarak gerekli ortalama su sıcaklığı bulunur. İkinci metod da ise aksi geçerlidir.

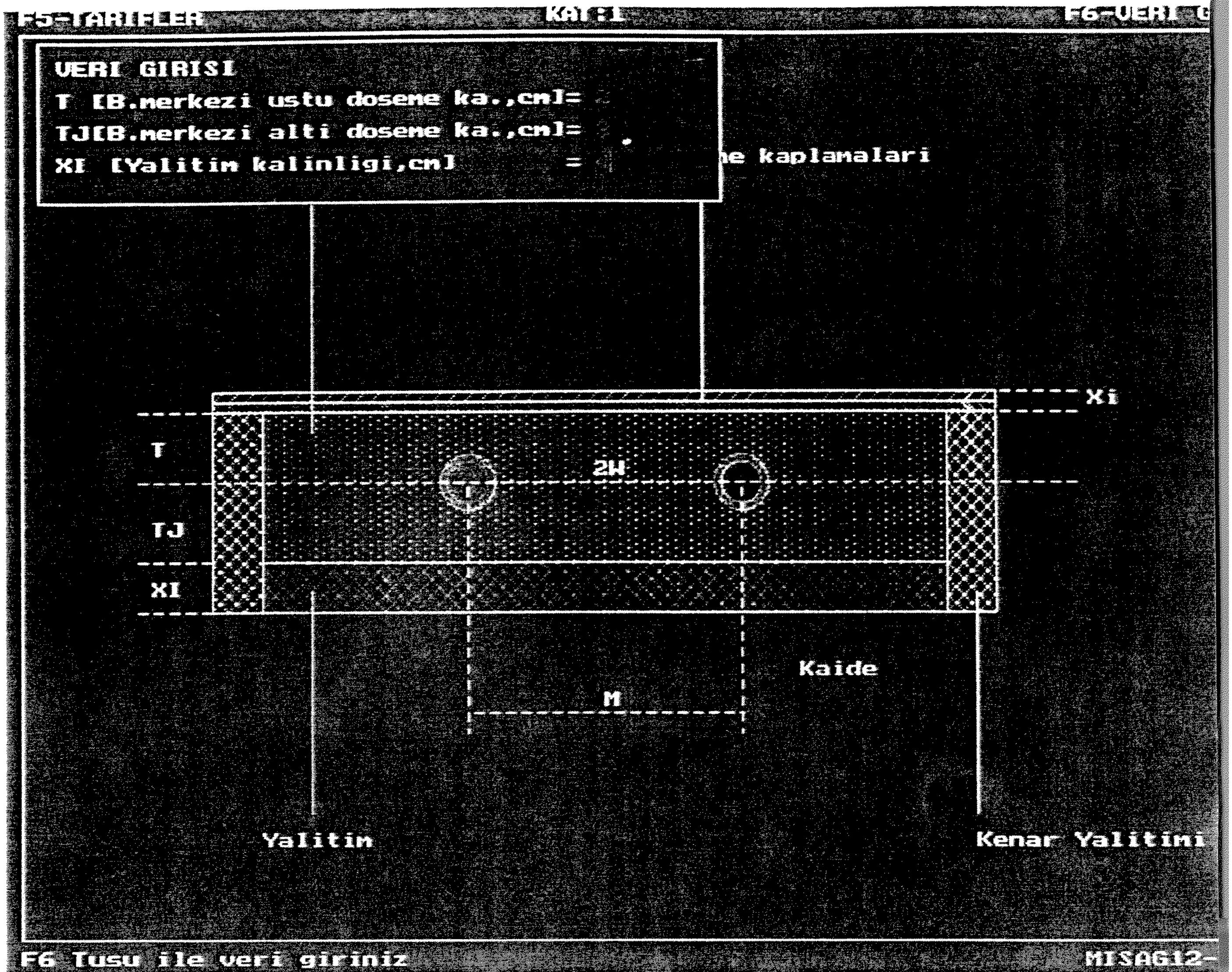
1/4 devresinin ısı yükü 2000Kcal/h girilmiştir(8000/4). "Şimdi döşeme bilgileri girilecek" ekran komutundan sonra bu görüntü ekrana gelir.



20

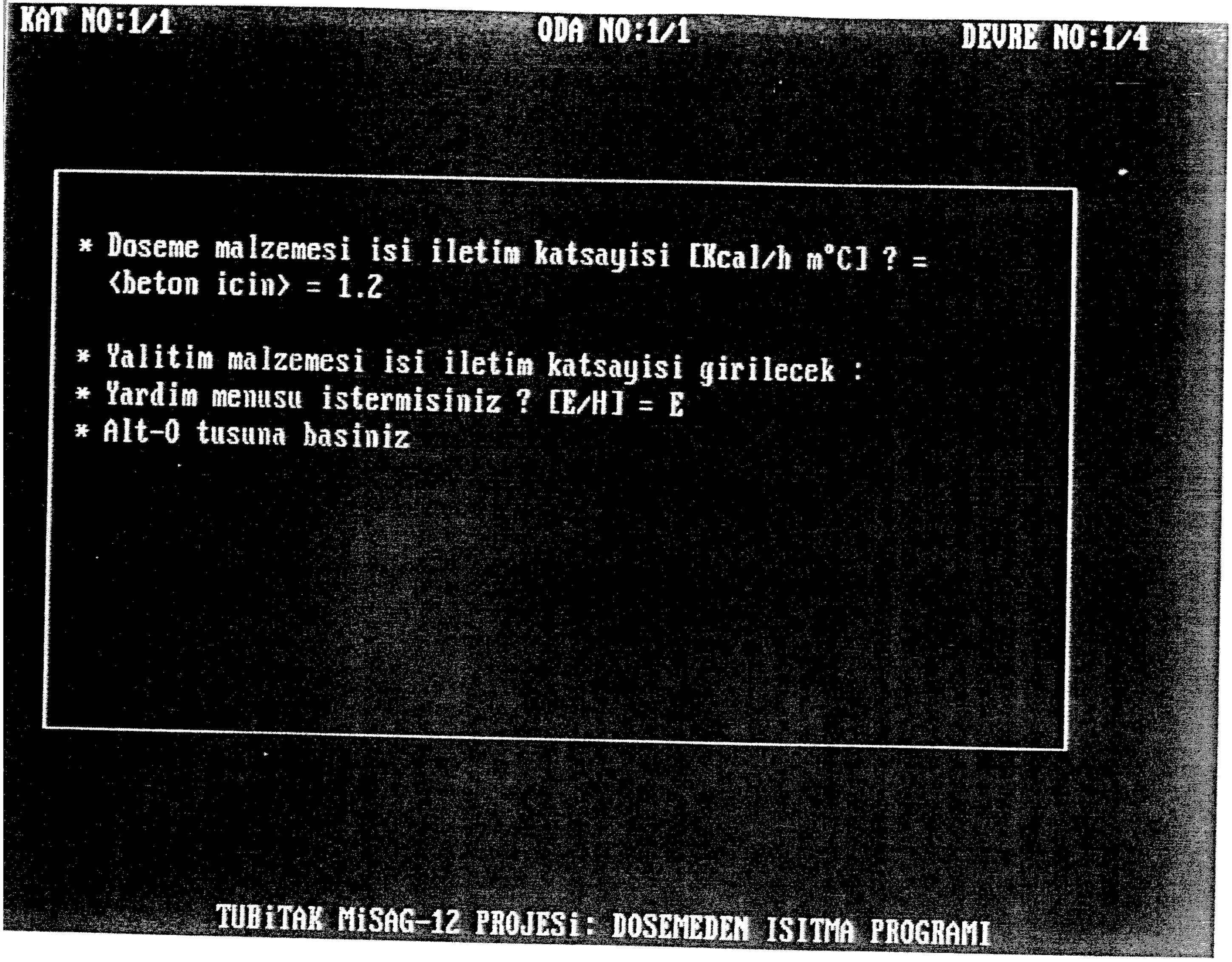
Bu ekranda F5 tuşu ile TARİFLER yardımcı penceresi açılır.(Ekran No:21).
F6 tuşu ile de veriler girilir.(Ekran No:22)


F5-TARIFLER
KAT:1
F6-VERI GIRIS
TARIFLER
Ta :Ic sicaklik,°C
T :Boru merkezi ustunde doseme kalinligi,
TJ :Boru merkezi altinda doseme kalinligi,
XI :Yalitim kalinligi,cm.
i :Kaplama adedi
Ti :Alt hacmin ortam sicakligi,°C
2W :Borular arasindaki bosluk,cm
M :Modulasyon,cm
Tp :Etkin doseme yuzey sicakligi,°C
Xi :Doseme kaplamalari toplam kalinligi,cm
e kaplamalari
Xi
T
2W
TJ
XI
Kaide
M
Yalitim
Kenar Yalitimi
F6 Tusu ile veri giriniz
MISAG12-1992



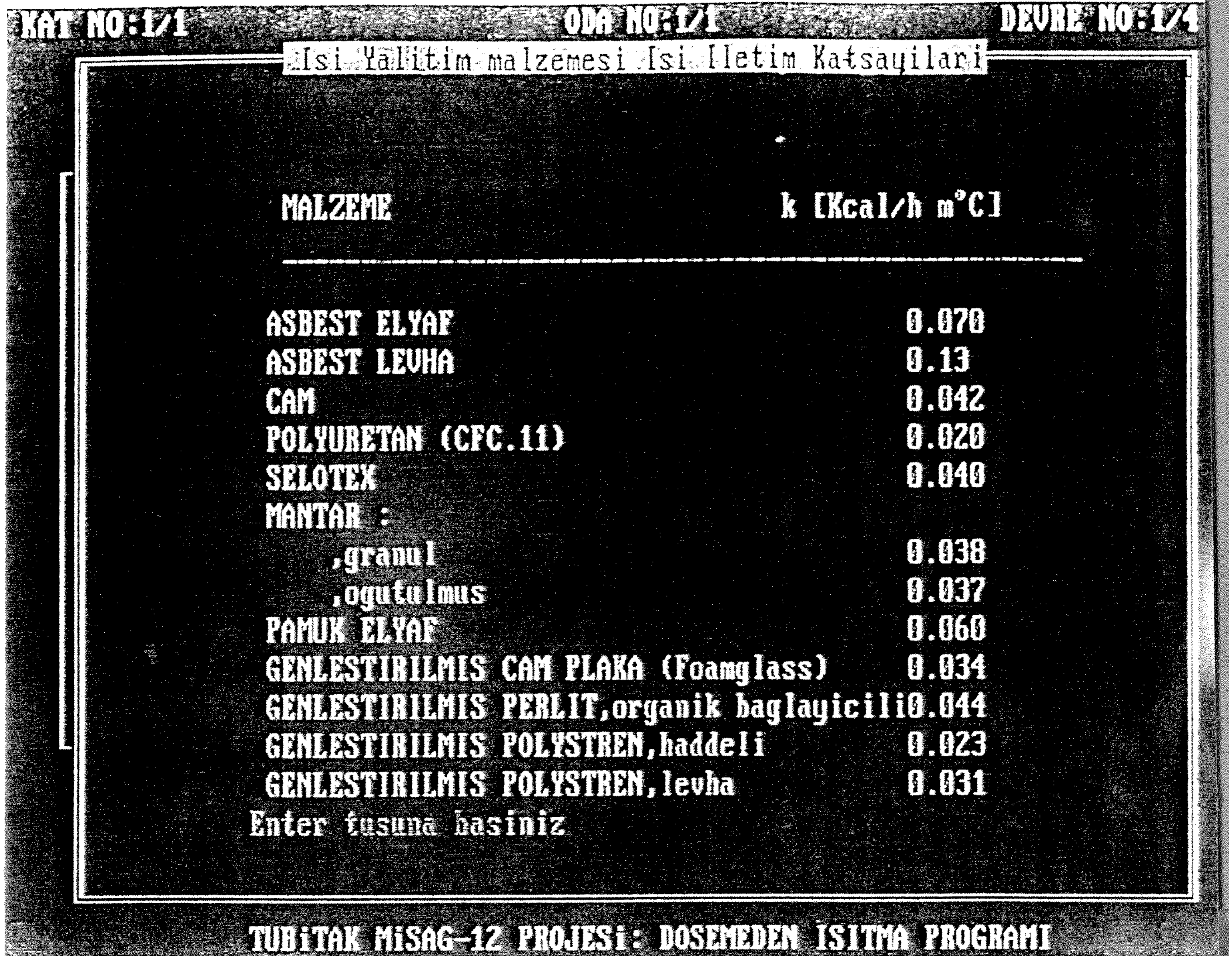
22

Bu ekranda T:2 cm,TJ:2 cm ve arka yalıtımının kalınlığı 4 cm olarak girilmiştir. Bundan sonra iki kez ENTER tuşu ile ana ekrana dönülür. Eğer ana ekrana geri dönmeyip bazı veriler değiştilmek istenirse ilk ENTER den sonra F6 ya basılır,yeni değerler girilir. Bilahare CTRL Q tuşu ile ana ekrandan çıkılır.



3

Beton için 1.2 değeri ısı iletim katsayısı olarak girilmiştir. Yalıtım malzemesi için yardım listesi ALT-O (sıfır değil) tuşuna basılır. Bu liste Ekran No:24 de kısmen gösterilmiştir.



KAT NO:1/1 ODA NO:1/1 DEVRE NO:1/4

Isi Yalitim malzemesi Isi Iletim Katsayilari

MALZEME	k [Kcal/h m°C]
ASBEST ELYAF	0.070
ASBEST LEVHA	0.13
CAM	0.042
POLYURETAN (CFC.11)	0.020
SELOTEX	0.040
MANTAR :	
,granul	0.038
,ogutulmus	0.037
PAMUK ELYAF	0.060
GENLESTIRILMIS CAM PLAKA (Foamglass)	0.034
GENLESTIRILMIS PERLIT,organik baglayicili	0.044
GENLESTIRILMIS POLYSTREN,haddeli	0.023
GENLESTIRILMIS POLYSTREN,levha	0.031

Enter tusuna basiniz

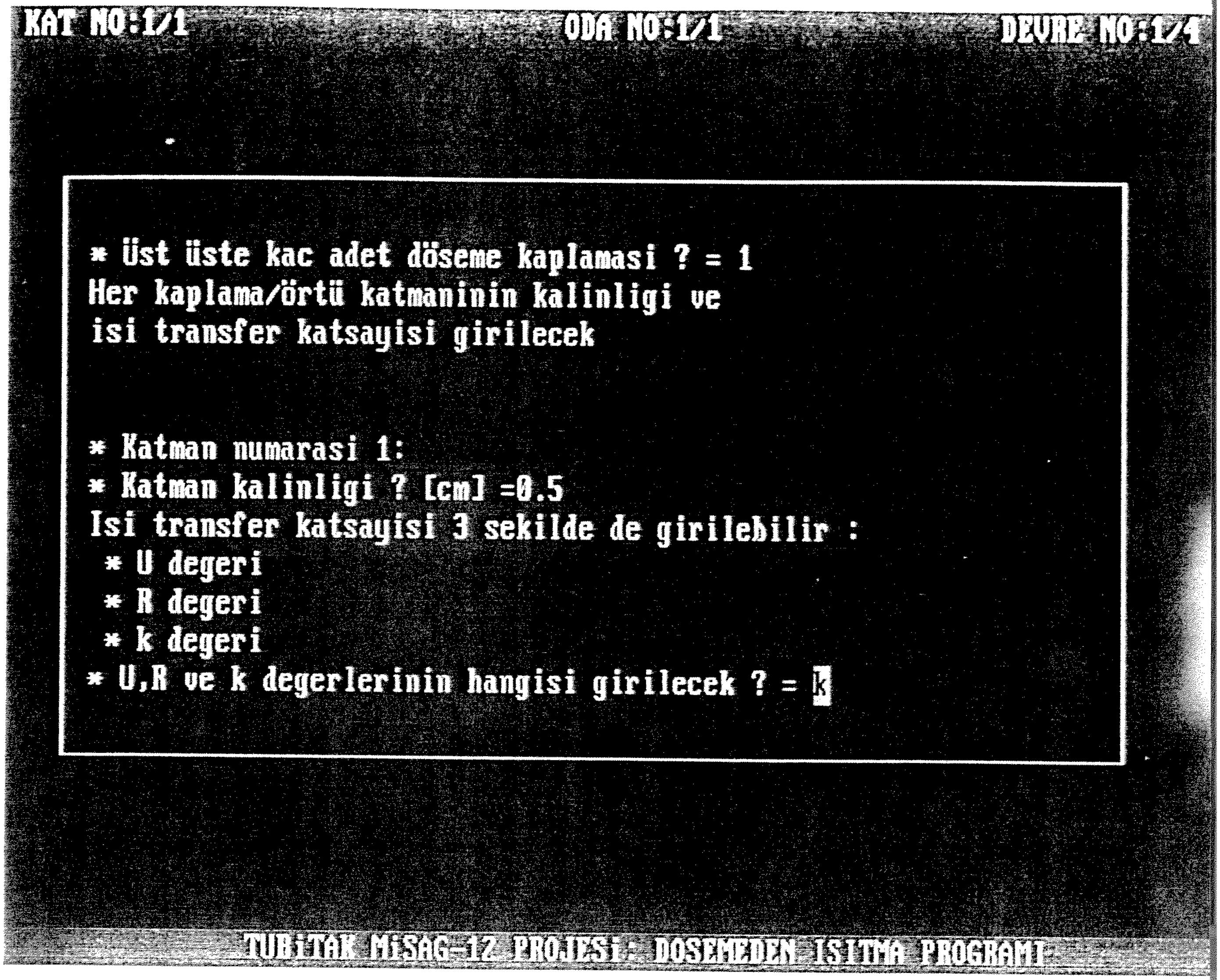
TUBiTAK MiSAG-12 PROJESi: DOSEMEDEN ISITMA PROGRAMI

24

Bu listenin devamı için ENTER, tablo başına dönmek için ALT-O menüden çıkmak için ALT-C tuşuna basılır. Bu liste otomatik bir menü olmadığı için seçilen malzemenin k değeri akılda tutulmalıdır. Bu değer ana ekranda kullanıcı tarafından girilecektir.

:5

Bu ekranda strapor için 0.033 Kcal/h m°C değeri girilmiştir.
Döşemede kaplama vardır(Evet)



26

Bir tane kaplama (Halı) vardır. Kalınlık 0.5 cm dir. Malzemenin k,R(ısıl direnç) veya U(ısı transfer katsayısı) değerlerinden her hangi biri girilebilir. k değeri gerileceği zaman yardım listesi istendiğinde ALT-O tuşuna basılır. H(ayır) cevabı ile kullanıcı bildiği k katsayısını girmiştir.

KAT NO:1/1 ODA NO:1/1 DEVRE NO:1/4

* Katman numarasi 1:
* Katman kalinligi ? [cm] =0.5
Isi transfer katsayisi 3 sekilde de girilebilir :
* U degeri
* R degeri
* k degeri
* U,R ve k degerlerinin hangisi girilecek ? = k
* Yardim menusu istermisiniz ? [E/H] = H
* k degeri [Kcal/h m°C] = 0.06

* Deger dogru mu ? [E/H] = E

TUBiTAK MiSAG-12 PROJESi: DOSEMEDEN ISITMA PROGRAMI

:7

KAT NO:1/1 ODA NO:1/1 DEVRE NO:1/4

YAYINIM KATSAYISI

Ahsap Parke (parlak)	0.88
Ahsap Parke (mat)	0.90
Linolyum (Acik renkli)	0.90
Linolyum (Koyu renkli)	0.92
Sunta	0.90
Mermer (Acik renkli)	0.89
Mermer (Koyu renkli)	0.91
Siyah lastik	0.95
Beton	0.88
Hali (Uzun bukleli)	0.90
Hali (Kisa bukleli)	0.95
Seramik karo	0.87
Asfalt	0.98
Kirmizi tugla,karo(tas/cimento)	0.95
Toprak satih.	0.87

* Doseme yuzeyi tasinim katsayisi (e)= 0.9

TUBiTAK MiSAG-12 PROJESi: DOSEMEDEN ISITMA PROGRAMI

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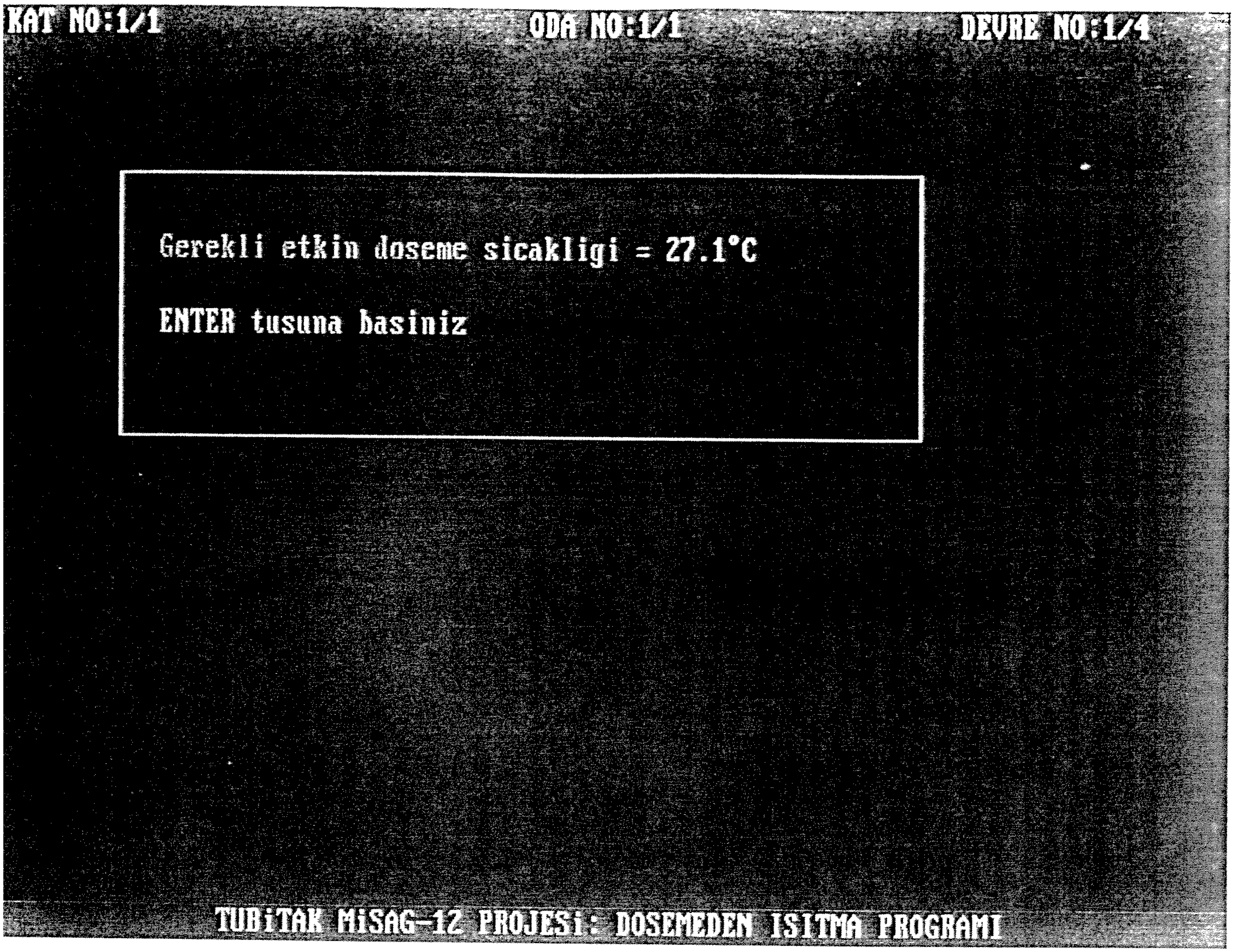
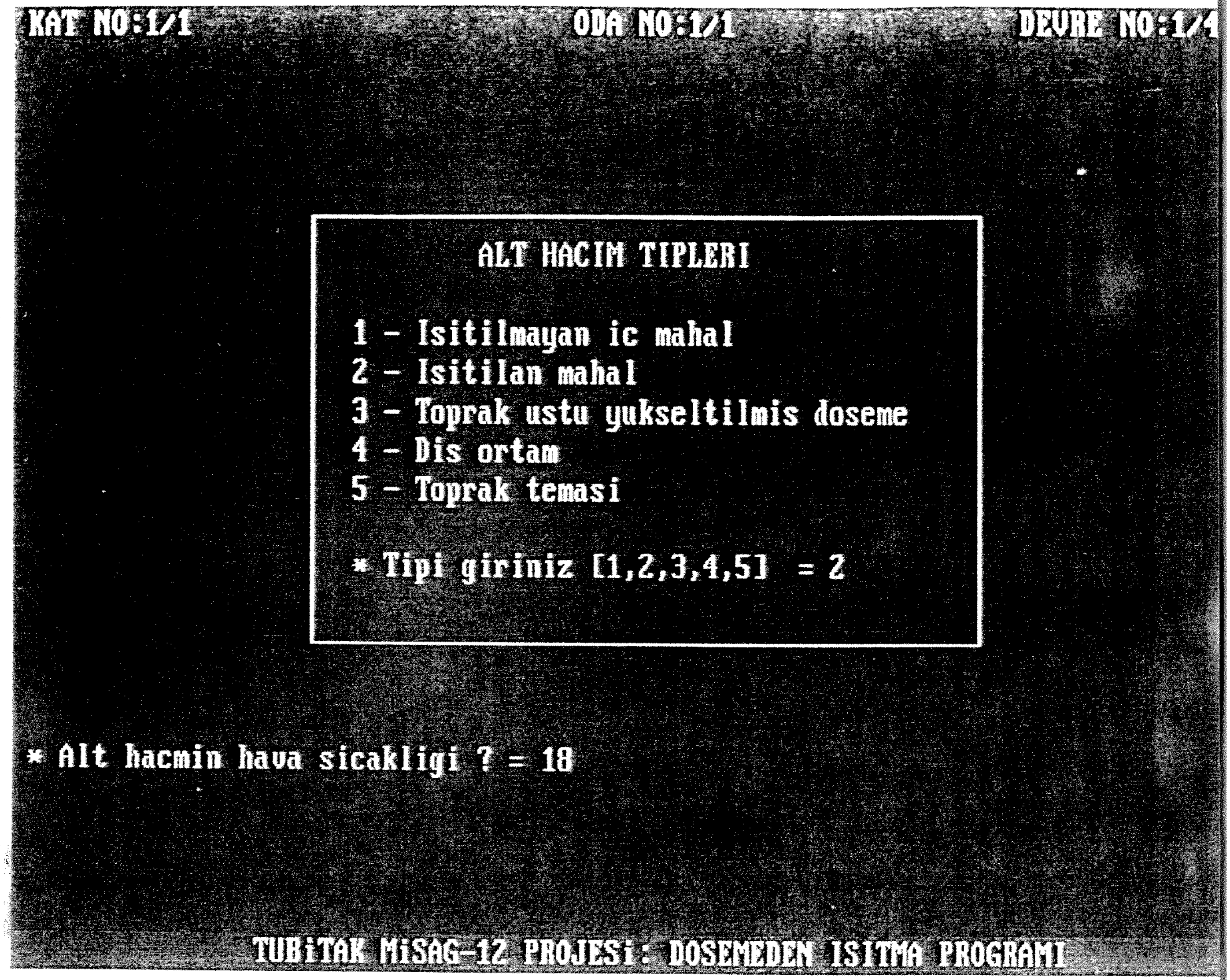
Halı için 0.9 değeri seçilmiştir.



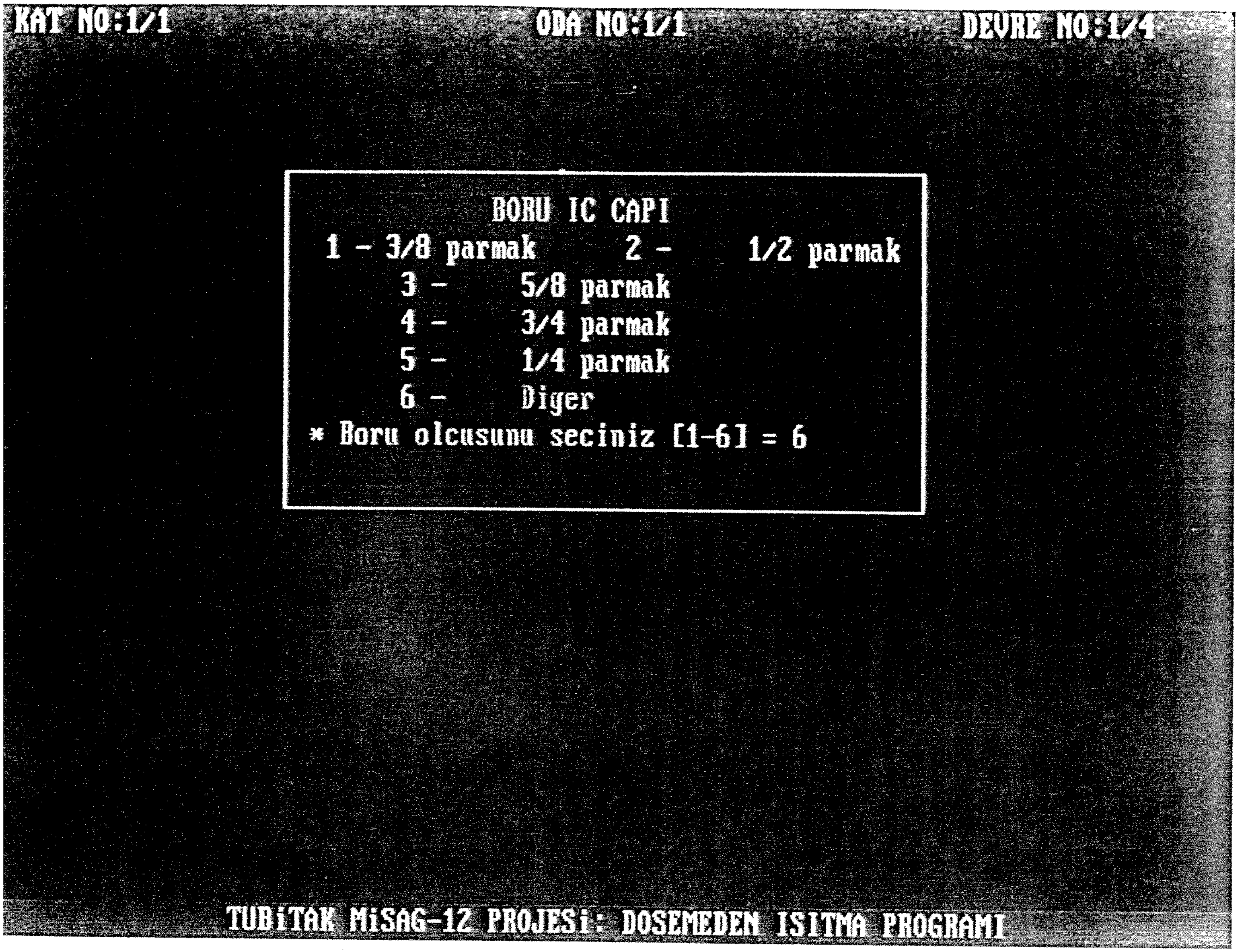
29

Program,ara ekran bilgisi olarak gerekli döşeme sıcaklığını 27.1 olarak vermiştir.

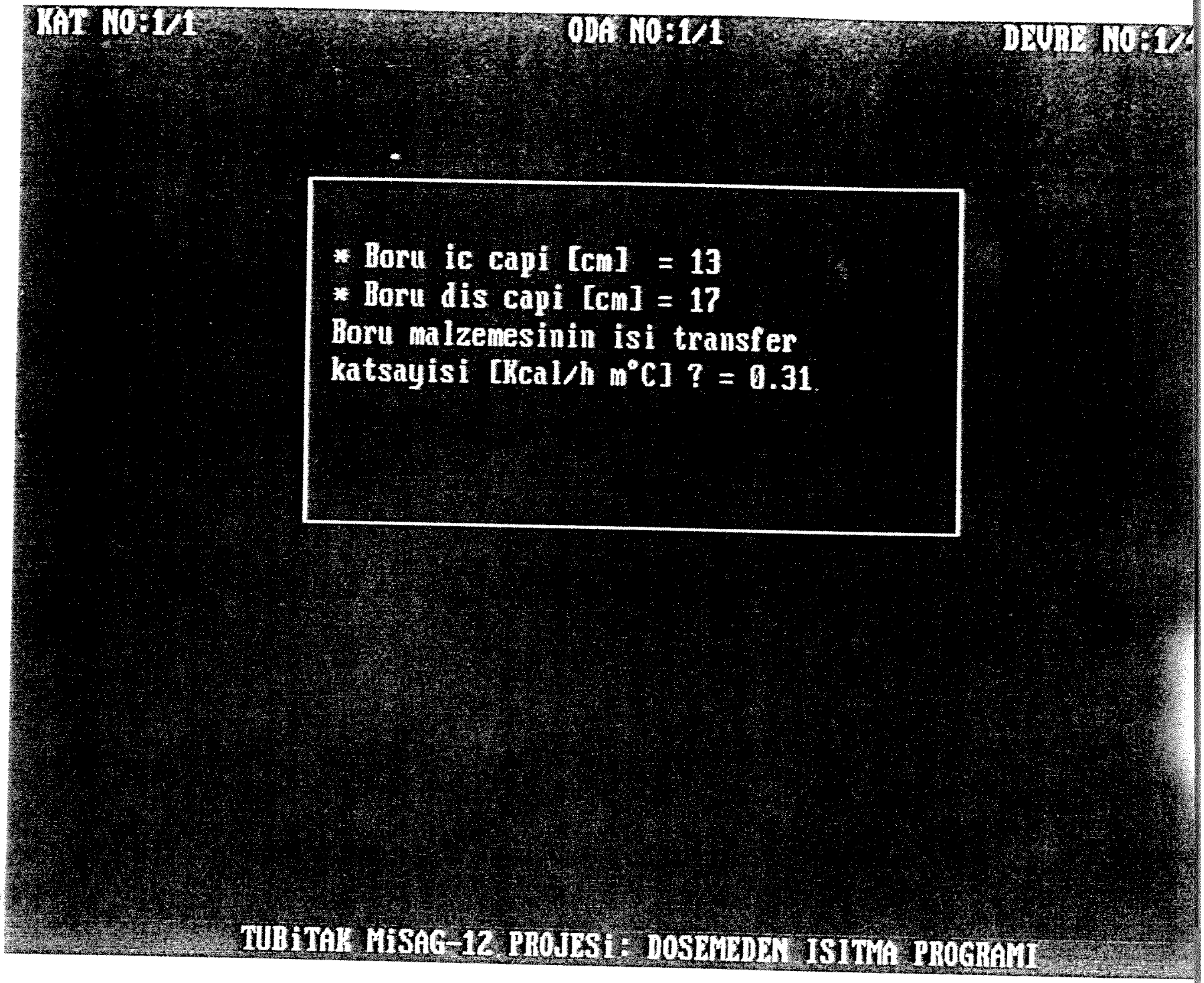


30

Alt hacim menüsüden "ISITILAN MAHAL" (2) girilmiş bu mahallin sıcaklığıda 18°C olarak girilmiştir.



Boru çapı kullanıcı tarafından özel girilecektir(seçenek 6).



32

Bu ekranda bir senaryo düzenlenmiştir:
YANLIŞ olarak cm yerine değerler mm olarak girilmiştir. Bu durumda arkadaki ekran görüntüye gelmiştir. Zira, akışkan hızı çok düşüktür.

KAT NO:1/1 ODA NO:1/1 DEVRE NO:1/4

SU HIZI KONTROLU

Su hizi cok dusuk.

vw = 0.01 [m/sn]

Su hizi 0.3 ≤ vs ≤ 1.5 m/sn. arasinda olmali

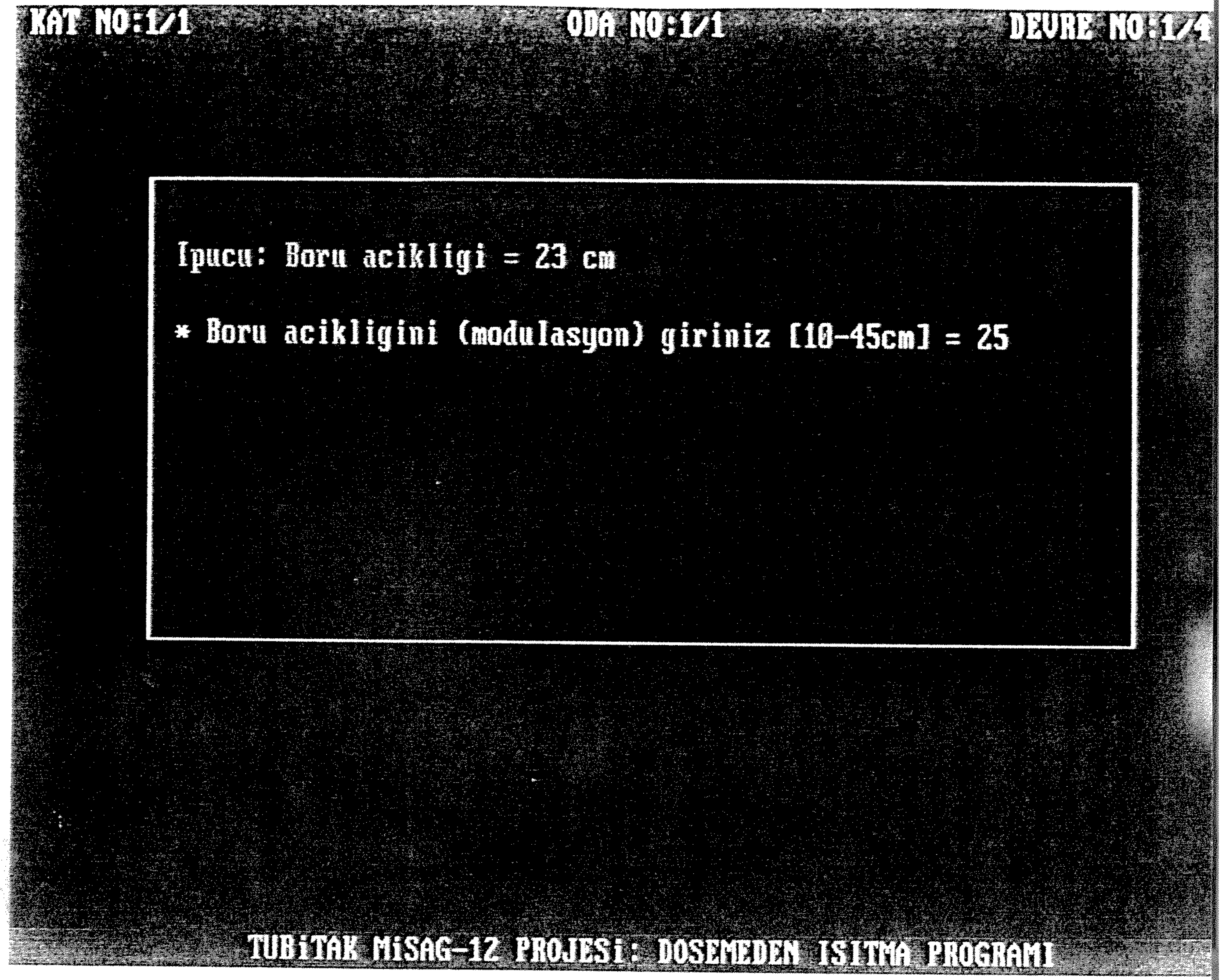
1- Boru capini azaltin,
2- Devre adedini azaltin
   [HATIRLATMA : Gerekli boru uzunlugu artacaktir]
3- Tasarima devam etmek istiyorum
   [Devrede hava problemi olabilir]

* [1,2,3] seceneklerinden birisisini seciniz = 1

TUBiTAK MiSAG-12 PROJESi: DOSEMEDEN ISITMA PROGRAMI

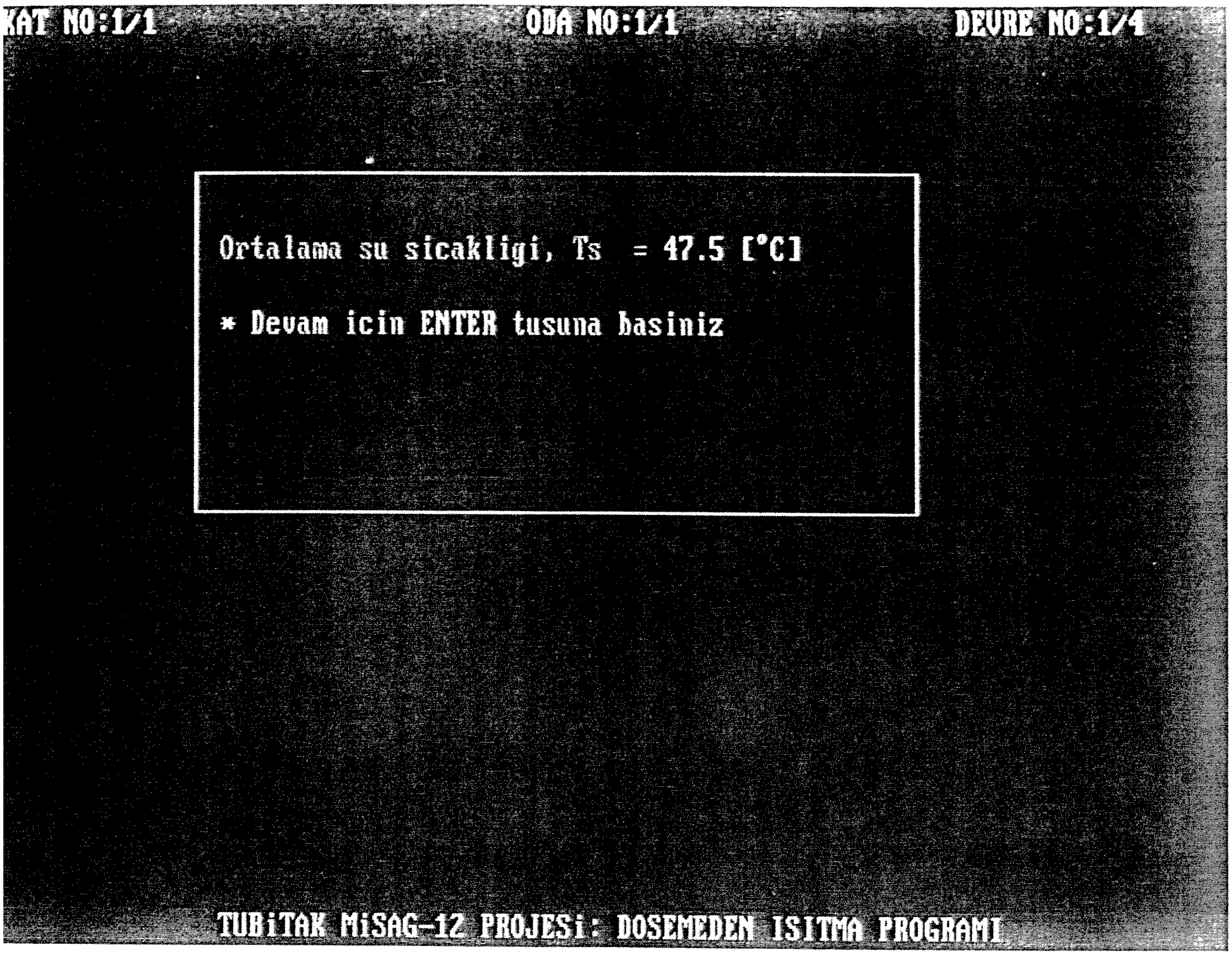
3

Bu ekrandan (1) seçilerek geri dönülen evvelki ekranda 1.3 ve 1.7 cm girilmiştir.



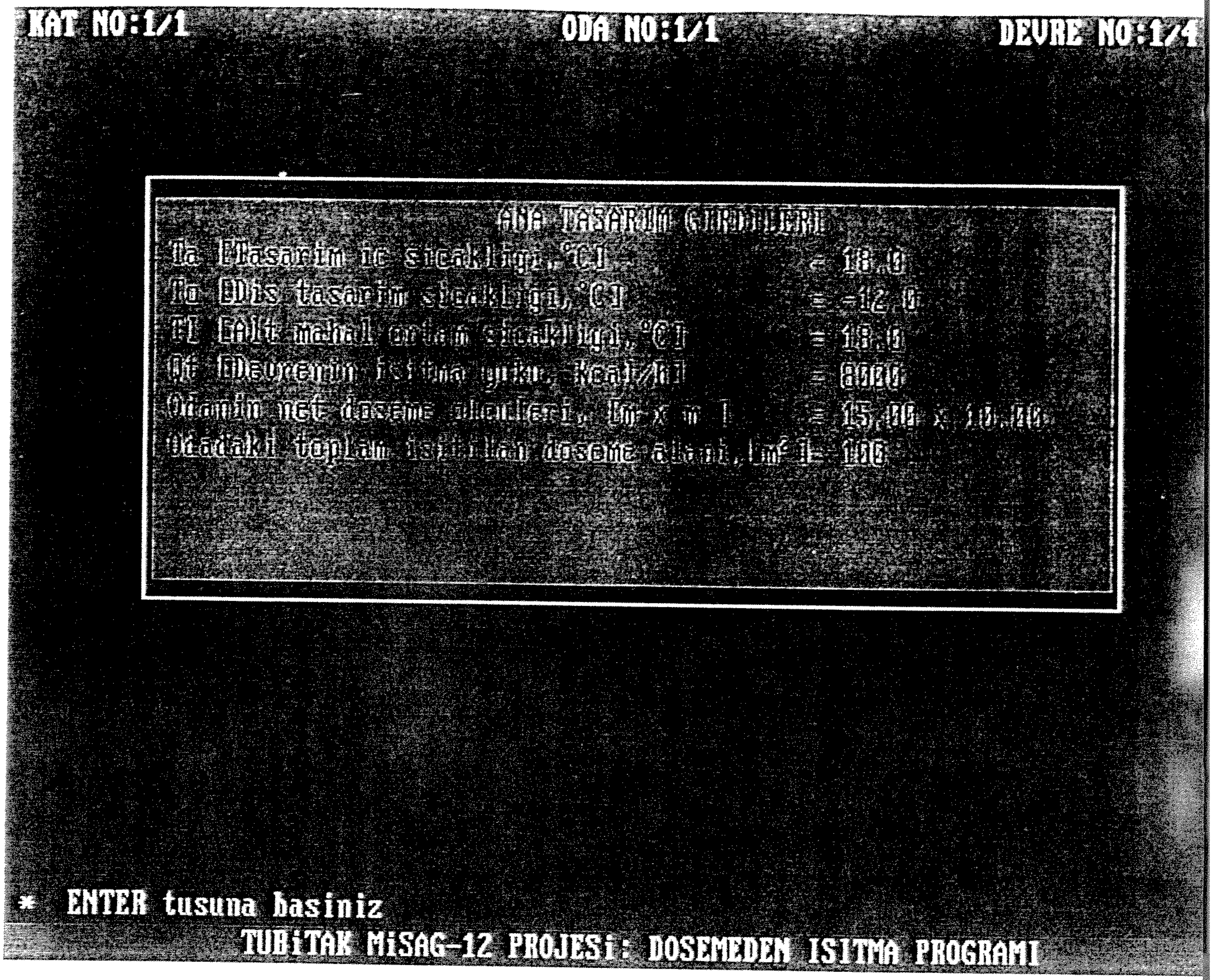
34

Ekranda "Boru Aralığı Girilecek" uyarısından sonra,kullanıcı ipucu istemiştir.ipucu 23 cm olup kullanıcı bunu 25 cm olarak girmiştir.



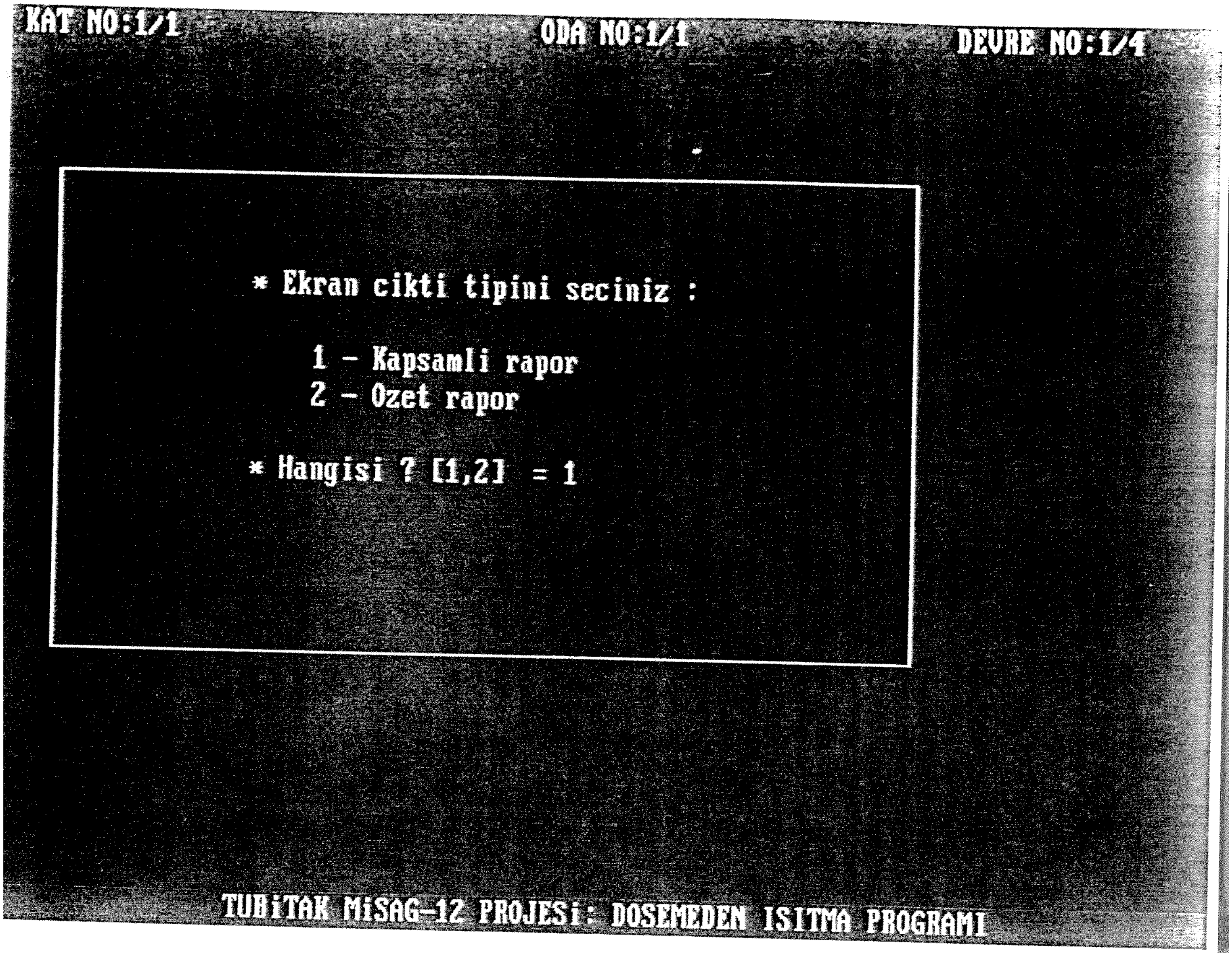
35

Program bilgi ekranında gerekli akışkan sıcaklığı verilmektedir. Eğer 70oC üzerinde ise kullanıcıya boru aralığı ve panel bilgilerinde düzeltme imkanı geri dönüşler ile verilir.



36

Tasarım bitmiştir. Bu ekranda bazı önemli veriler listelenmektedir.


KAT NO:1/1
ODA NO:1/1
DEVRE NO:1/4
* Ekran cikti tipini seciniz :
1 - Kapsamli rapor
2 - Ozet rapor
* Hangisi ? [1,2] = 1
TUBiTAK MiSAG-12 PROJESi: DOSEMEDEN ISITMA PROGRAMI

KAT NO:1/1 ODA NO:1/1 DEVRE NO:1/4

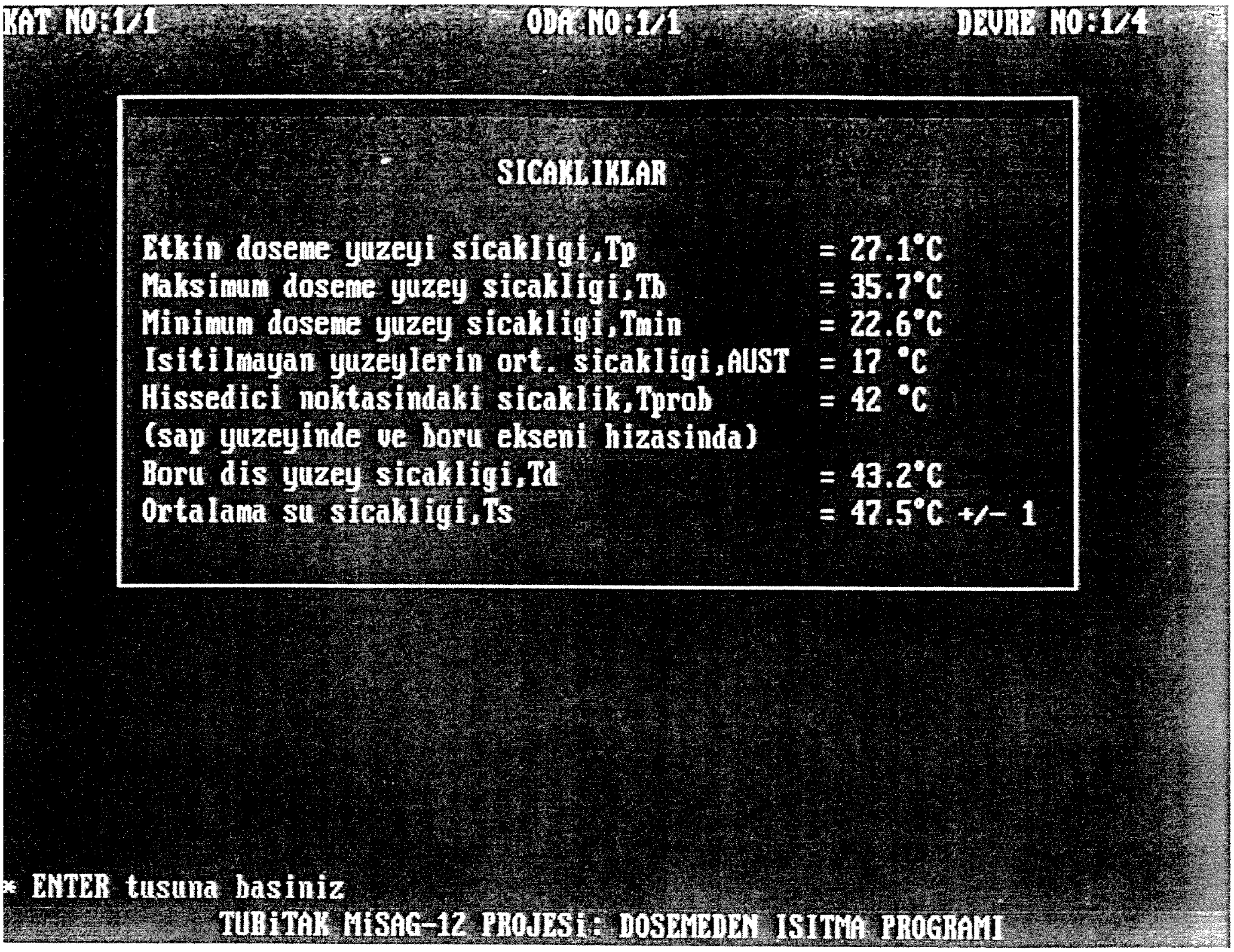
OLCULER

T [Boru merkezi uzerinde doseme kalinligi]	=	2.00 cm
TJ[Boru merkezi altinda doseme kalinligi]	=	2.00 cm
T+TJ [Toplam doseme kalinligi]	=	4.00 cm
XI [Yalitim kalinligi]	=	4.00 cm
DT [Gidis/Donus su sicaklik farki]	=	10.0 °C
di [Boru ic capi]	=	1.300 cm
Doseme kaplama/ortu adedi	=	1
Doseme kaplama/ortu toplam isil direnci	=	0.08hm²°C/Kcal
Boru acikligi [modulasyon]	=	25.00 cm
Isitilan dosemenin kanatcik verimi	=	49.83 %

* ENTER tusuna basiniz

TUBİTAK MİSAG-12 PROJESİ: DOSEMEDEN ISITMA PROGRAMI

38



9

Oda konforu için termostatla kazan kontrolü yanında ayrıca döşemenin belirli bir sıcaklıkta bulunmasınında kontrolü gerekebilir. Bu maksatla ısıtılan döşemenin varsa kaplama altına ve iki komşu boru ara noktasına isabet edecek şekilde hissedici eleman konur. Bu elemanın ayar noktası program tarafından 42°C olarak verilmiştir.

KAT NO:1/1 ODA NO:1/1 DEVRE NO:1/4

ISITMA KAPASiTELERi ve YÜKLER

Isima ile birim isitma kapasitesi,qr	= 47.7 [Kcal/h m²]
Tasinim ile birim isitma kapasitesi,qc	= 32.3 [Kcal/h m²]
Toplam birim isi kapasitesi,qy	= 80.0 [Kcal/h m²]
Isima/tasinim orani,qr/qy	= 59.6 [%]
Isitilan dosemenin birim yuzeyde isi kaybi	= 13.5 [Kcal/h m²]
Devrenin isitma kapasitesi,Qy	= 2000[Kcal/h]
Devrenin kazan yuku,BL	= 2338[Kcal/h]
Isitilan dosemenin isitma verimi,x	= 85.5 [%]

* ENTER tusuna basiniz

TUBiTAK MiSAG-12 PROJESi: DOSEMEDEN ISITMA PROGRAMI

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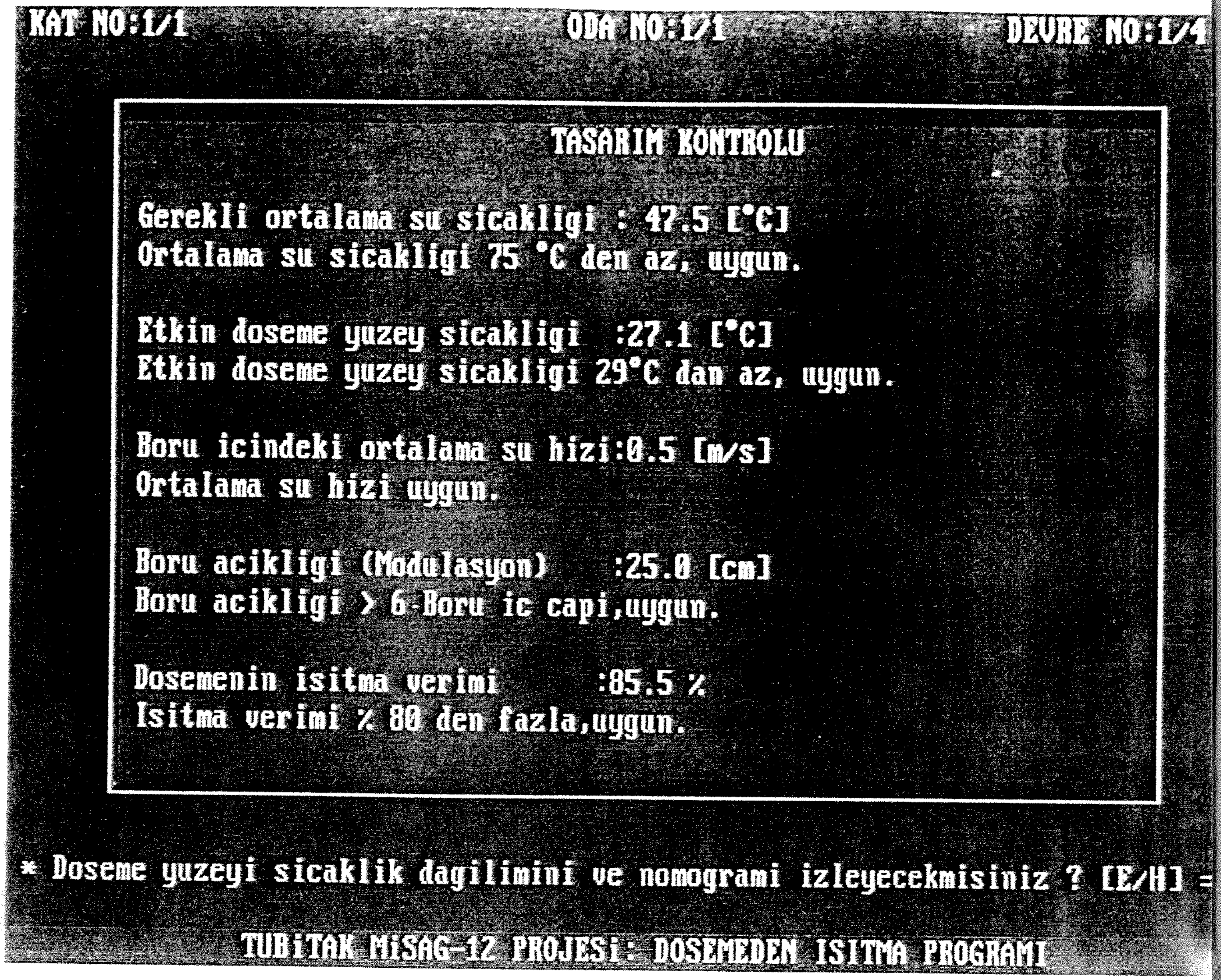
KAT NO:1/1 ODA NO:1/1 DEVRE NO:1/4

HIDROLIK

Borudaki su hizi,vs	= 0.5 [m/s]
Reynolds sayisi,Re	= 11184
Su-Boru isi tasinim katsayisi,α	= 593 [Kcal/h m²°C]
Boru surtunme katsayisi,τ	= 0.03430
Birim uzunlukta basinc kaybi,Dp	= 35.812 [mm.ss/m]
Devre icersindeki boru uzunlugu,L	= 110 [m]
Manifold ile devre arasindaki (gidis/gelis)	= 3 [m]
Devrenin toplam boru uzunlugu,Lt	= 113 [m]
Devredeki toplam basinc kaybi,Pt	= 3678.92 [mm.ss]
Devredeki su debisi	= 0.2364[m3/h]

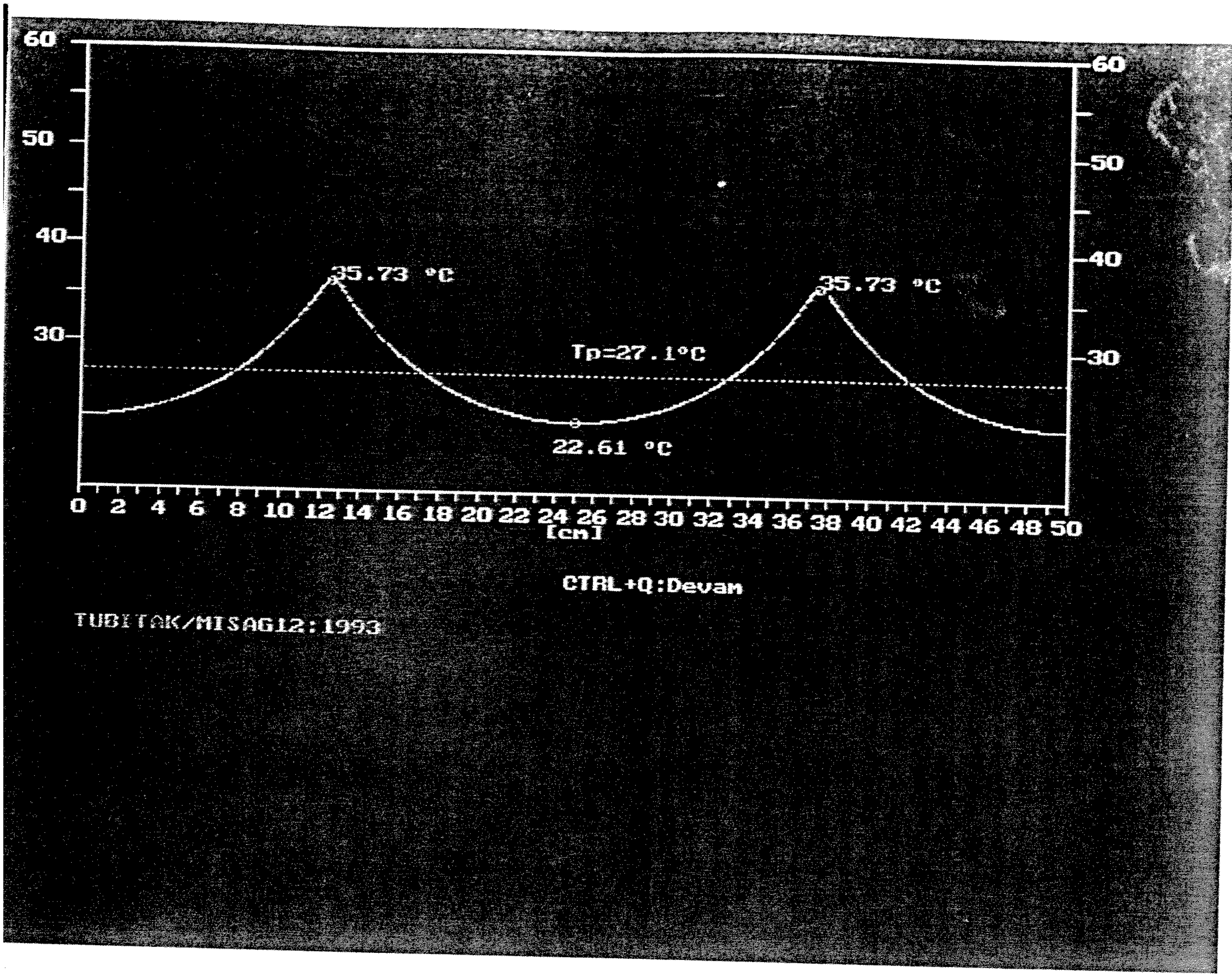
* ENTER tusuna basiniz

TUBiTAK MiSAG-12 PROJESi: DOSEMEDEN ISITMA PROGRAMI

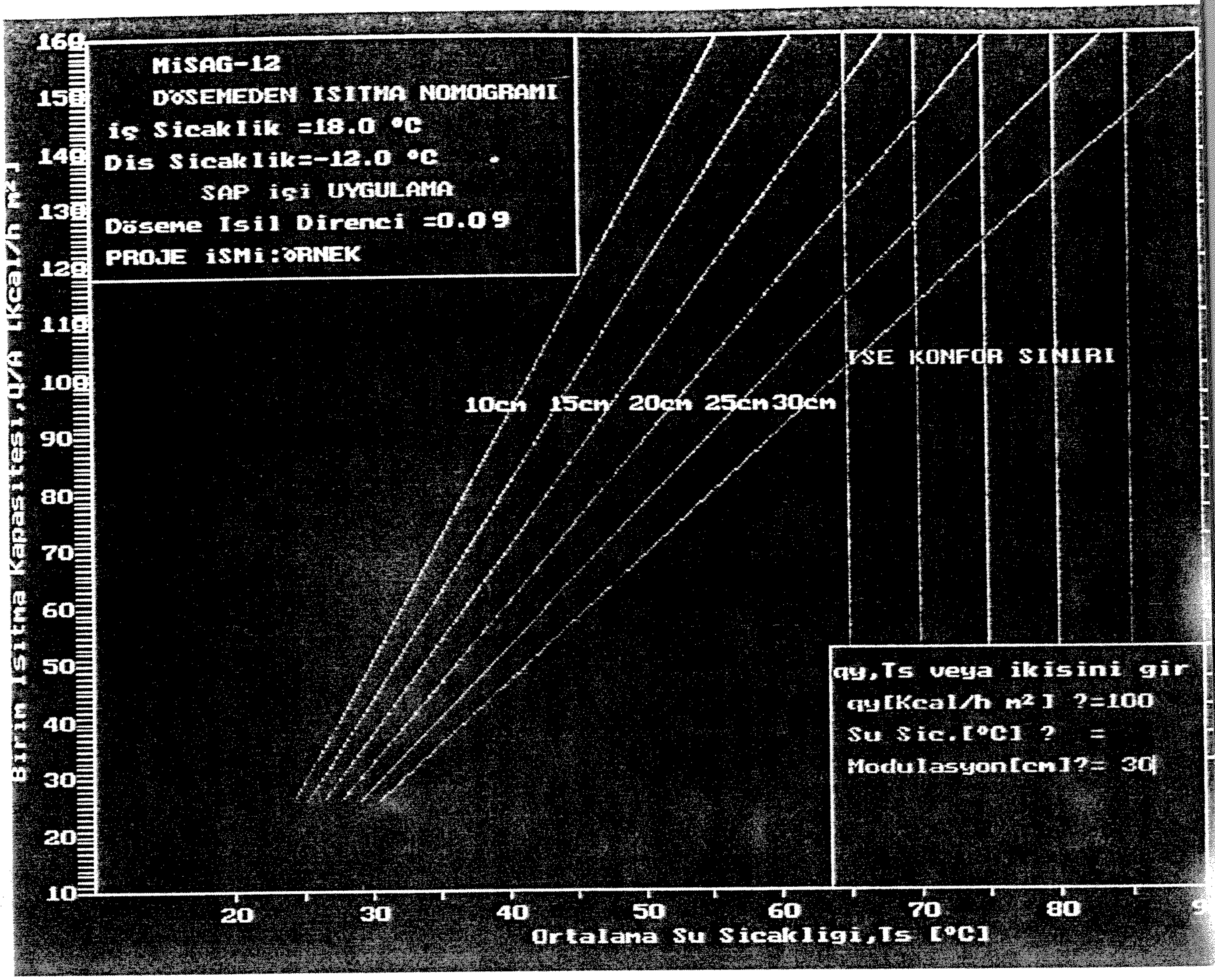


42

Bu ekranda tasarımın uygun sınırlar içersinde olup olmadığı kontrol edilmektedir.
Kullanıcı yüzeyde sıcaklık profilini ile tasarım nomogramını görmeyi E(vet)cevabı ile istemiştir.

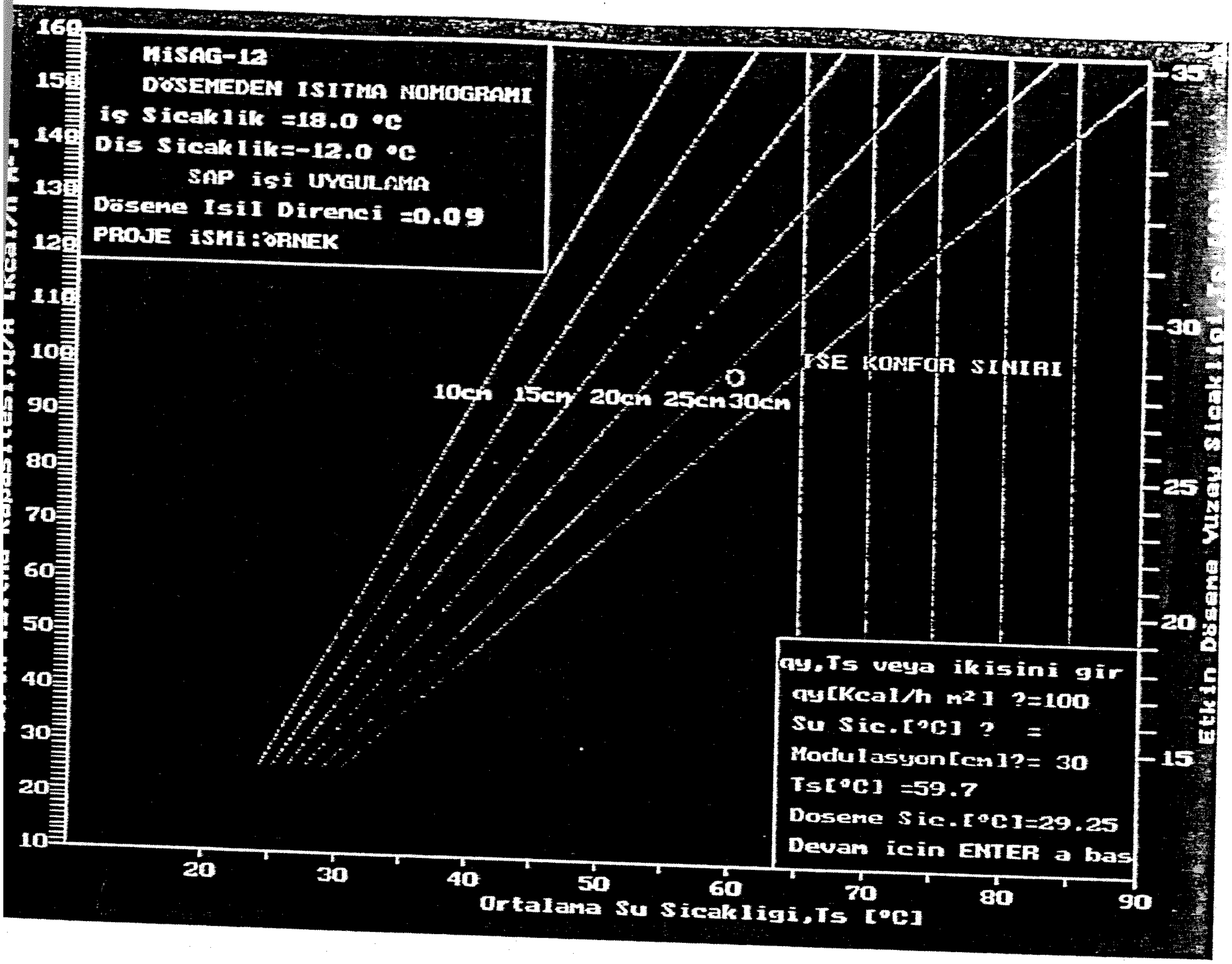


Bu ekranda yüzey sıcaklık profili görülmektedir.
CTR-Q ile bu ekrandan çıkılır.

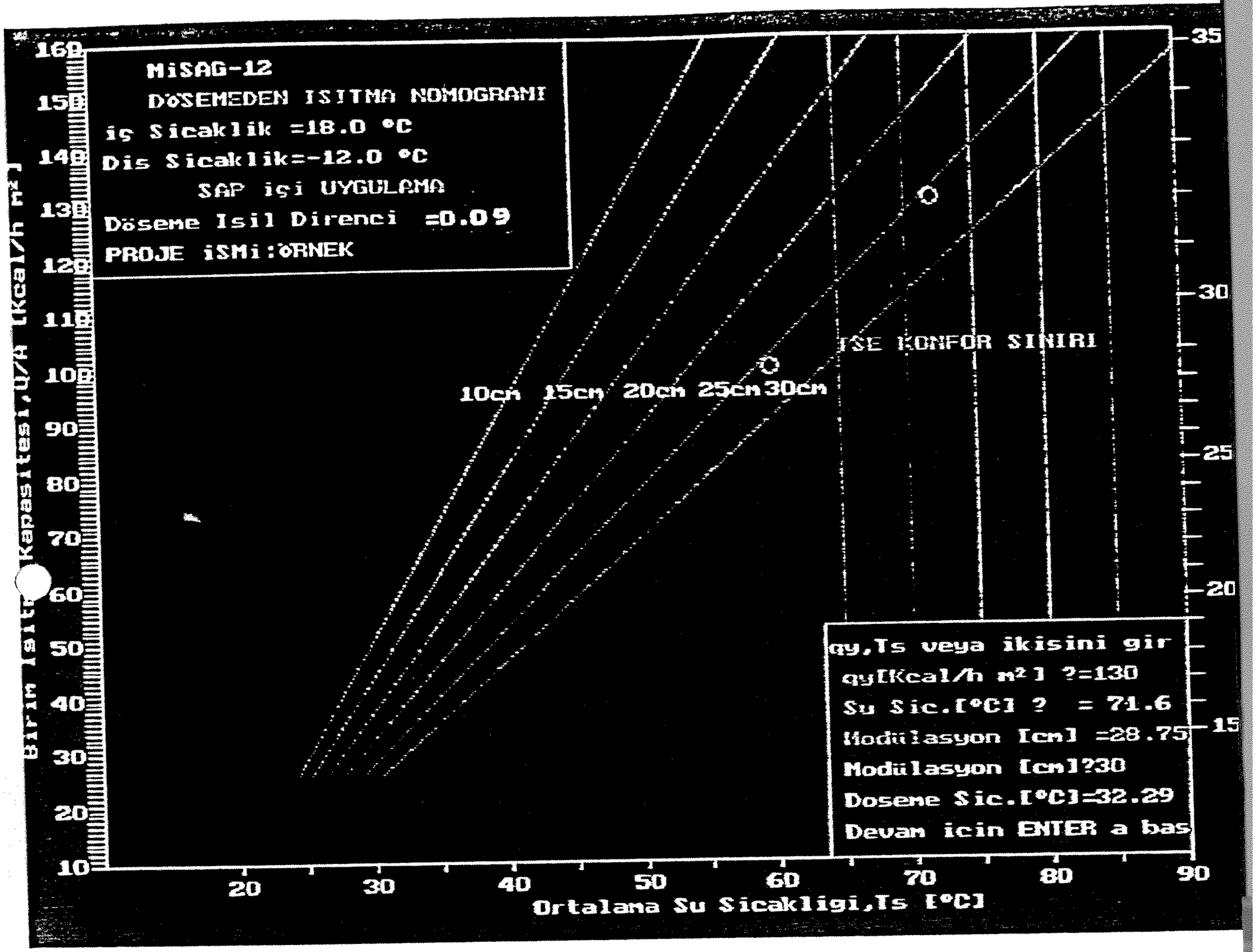


44 Bu ekran odaya özgü hazırlanmış olan tasarım nomgramını vermektedir.

TSE konfor sınırı döşemede en fazla 29°C olarak belirtilmiştir.Eğik çizgiler boru aralık çizgileridir. Solda birim ısıtma kapasitesi (yükü) ekseni,sağda döşeme sıcaklık ekseni,altta ise ortalama akışkan sıcaklığı ekseni bulunmaktadır.
NOT: Bilgisayar programı hesaplar için su özelliklerini kullanmakta,su sıcaklığı değiştikçe değerleri düzeltmektedir.
Tasarım statüsü sol üst köşede belirtilmiştir.
F5 tuşu ile sağ alt köşede beliren küçük pencerede,kullanıcı nomogram üzerinde boru aralığı,ortalama su sıcaklığı ve yük ile değişik senaryoları izleyebilir.
Burada yük 100 e , boru aralığı da 30 cm e çıkarılmıştır.
Gelecek sahifede görüldüğü gibi su sıcaklığı 59.7°C,döşeme sıcaklığı ise 29.25°C dir.
NOT:Bu senaryo bilgileri önceden bitirilen tasarım değerlerini etkilememektedir.
NOT:Bu nomogram sadece bu oda için hassas olup,başka odalar, konumlar veya ayrı tasarımlar için kullanılamaz.
Küçük pencereden CTR-Q ile çıkılır.

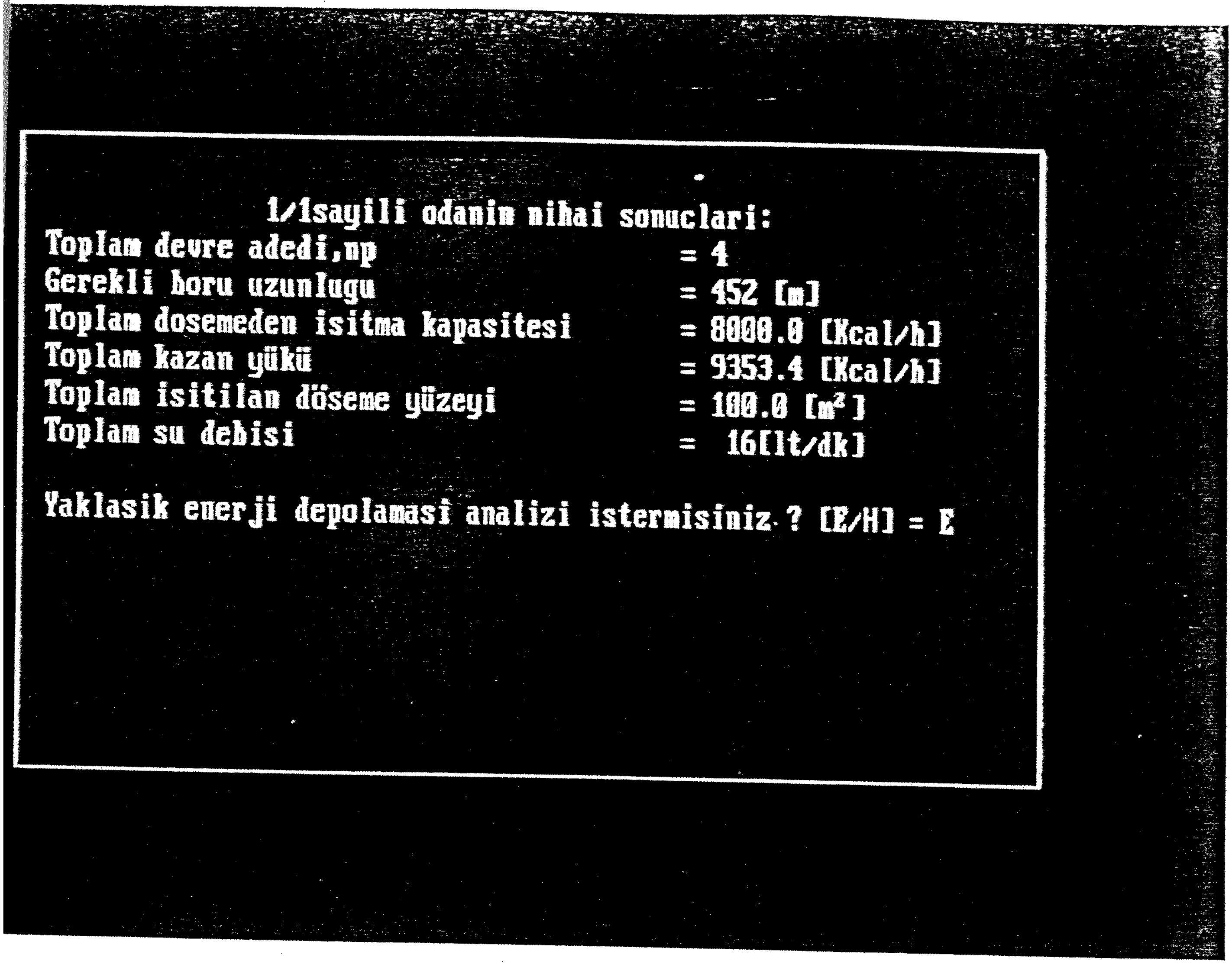


Bu ekranda kullanıcının girdiği yeni değerler ve sonuçları hem pencerede hem de ana ekranda yuvarlak nokta ile görülmektedir.



46

Bu ekranda ayrı bir senaryo incelenmiştir.

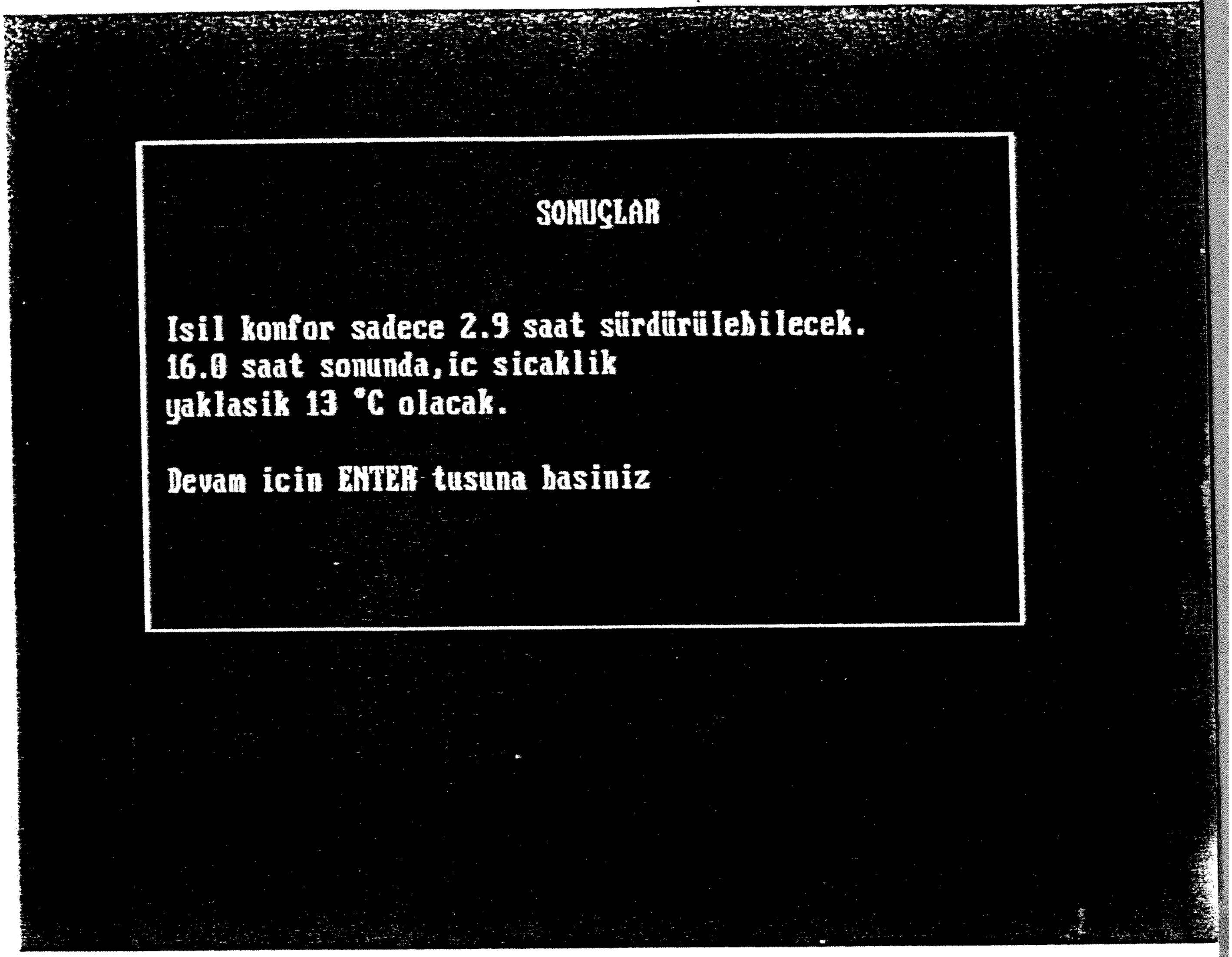


1/1sayili odanin nihai sonuclari:

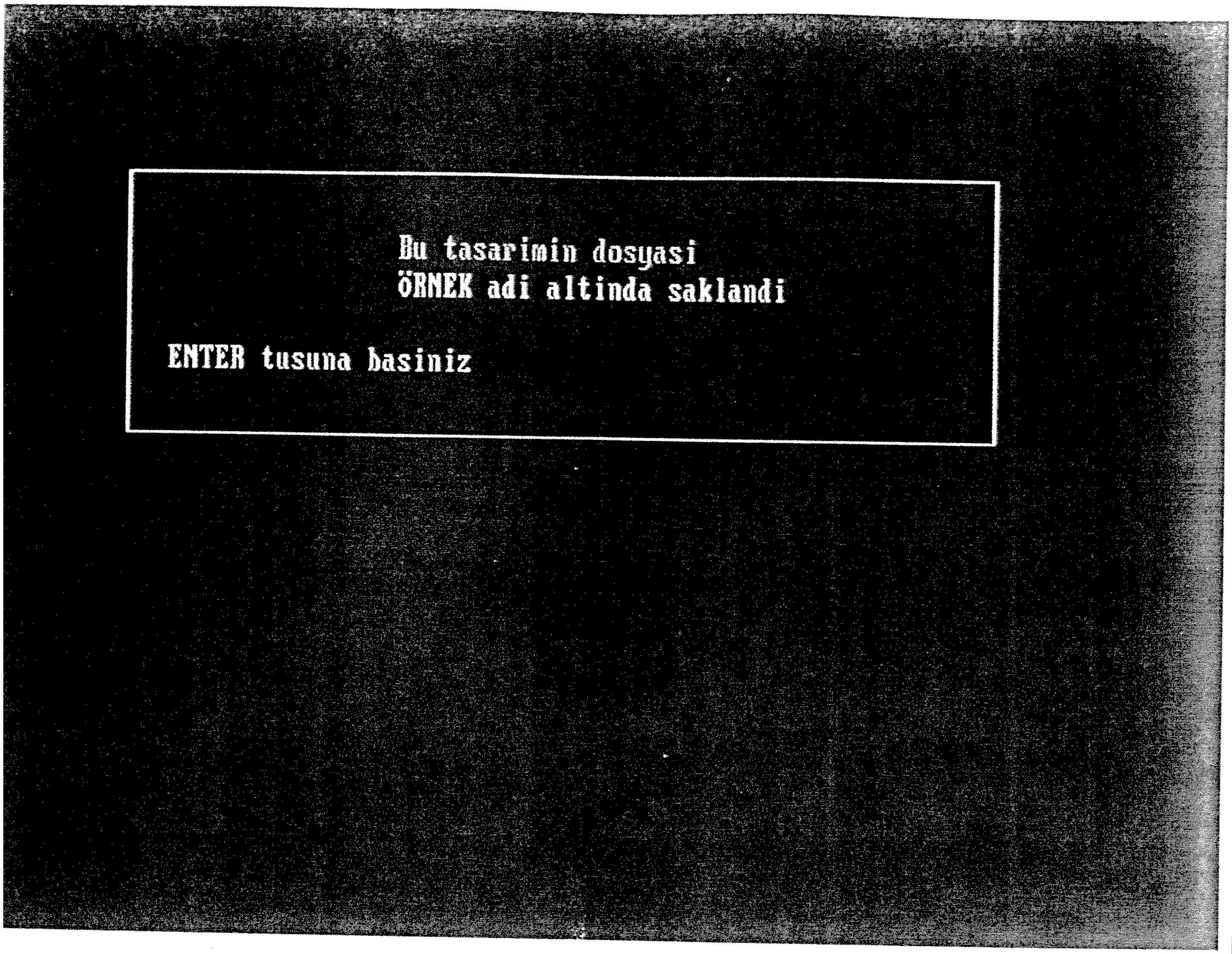
Toplam devre adedi,np	= 4
Gerekli boru uzunlugu	= 452 [m]
Toplam dosemeden isitma kapasitesi	= 8000.0 [Kcal/h]
Toplam kazan yükü	= 9353.4 [Kcal/h]
Toplam isitilan döseme yüzeyi	= 100.0 [m²]
Toplam su debisi	= 16[lt/dk]

Yaklasik enerji depolamasi analizi istermisiniz ? [E/H] = E

4 hidrolik devrede aynı olduğundan oda bilgileri hemen verilmiştir.



Uçak hangarı,depo gibi yerlerde ve eğer gece özel ucuz elektrik
48 tarifesi uygulanıyor ise elektrikli kazan veya elektrik motoru ile tahrikli ısı pompası kullanımı cazip olabilir. Isı pompası havadan suya,sudan suya (jeotermal veya toprak ısısı,proses ısısı gibi) olabilir. Bu durumda ucuz elektrik tarifesi saatlerinde döşemede ısı depolanır. Bu amaçla beton kalınlığı ve/veya altındaki toprak,taş ve benzeri tesviye katmanı kalın tutulur. Arka ve kenar yalıtımı ise en alta konur.
Pahalı tarife saatinde ise sistem durdurulur. İşte böyle bir tasarım için program YAKLAŞIK ISI DEPOLAMASI opsyonuna haizdir. Bu opsyona E(vet)cevabı verilerek 6 saat işletme,16 saat durdurma rejimi ara ekran sorusunda verilmiştir. Bu ekranda ise sonuçlar verilmektedir.
Eğer bu tasarımda böyle bir amaç güdülmüş olsa idi beton kalınlığnın daha fazla olması gerekirdi.



49 Yazıcı çıktısı almak için ve dosyanın saklanması için kullanıcının baştan girmiş olduğu dosya adı kullanılır.

Print ÖRNEK komutu ile ekteki yazıcı çıktısı alınmıştır.

```
Proje Dosya ADI          = ÖRNEK
Proje Sahibi             = MİSAG-12
Tarih                    =
Mahal                    = ANKARA, TR
Climatic Condition       = Soguk
Denizden olan yukseklik=  865 m
Dis sicaklik             = -12.0°C

    1 SAYILI ODA İÇİN ANA TASARIM DEGİSKENLERI

Odanin dis cephe katsayisi,c                  = 2.0
Iç tasarim sicakligi,Ta                       = 18.0 [°C]
Isitilmayan yüzeylerin ort. sicakligi,AUST    = 17   [°C]
Odanin döşemeden isitma yükü,Qt               =  8000[Kcal/h]
Odanin net döşeme ölçüleri                    = 15.0 m x 10.0 m
Toplam isitilan döşeme yüzeyi,Apr             =  100  [m²]
Odadaki hidrolik devre adedi,np               = 4
                                                [Butun devreler ayni]
Gidis/dönüs su sicaklik farki,DT              = 10.0 [°C]

-------------------------------------------------------------
************** ( 1) sayili devre sonuçlari ************
-------------------------------------------------------------

A-ÇEVRE KOSULLARI

Alttaki ortam                                   = Isitilmayan iç mahal
Alttaki ortam sicakligi,TI                      = 18   [°C]

B-DÖSEME DETAYI

Devrenin yüzey alani                          = 25.0 [m²]
Döseme kalinligi [boru ekseni üzerinde],T     = 2.0   [cm]
Döseme kalinligi [boru ekseni altinda],TJ     = 2.0   [cm]
Yalitim kalinligi,XI                          = 4.0   [cm]
Yüzeyde kaplama/örtü adedi                     = 1
Kaplama/örtü toplam isil direnci               = 0.08 [h m²°C/Kcal]
Isitilan dösemenin kanatçik verimi             = 49.8 [%]
Boru iç çapi                                   = Boru
Boru açikligi [Modülasyon]                     = 25.0  [cm]

C-SICAKLIKLAR:

Etkin döseme yüzey sicakligi,Tp               = 27.1 [°C]
Maksimum döseme yüzey sicakligi,Tb            = 35.7 [°C]
Minimum döseme yüzey sicakligi,Tm             = 22.6 [°C]
Hissedici noktasindaki sicaklik,Tprobe        = 43    [°C]
(sap betonu yüzeyinde ve boru ekseninde)
Boru dis yüzey sicakligi,Td                   = 44    [°C]
Ortalama su sicakligi,Ts                      = 48    [°C] +/- 1
Devreye su giris sicakligi,Tgiris             = 53    [°C]
```

SITMA KAPASiTELERi VE YÜKLER:

ma ile birim isitma kapasitesi,qr	= 47.7 [Kcal/h m²]
inim ile birim isitma kapasitesi,qc	= 32.3 [Kcal/h m²]
ma tasinim orani,qr/qy	= 59.6 [%]
lam birim isitma kapasitesi,qy	= 80.0 [Kcal/h m²]
renin isi kayiplari,qa	= 13.5 [Kcal/h m²]
renin toplam isitma kapasitesi,Qy	= 2000 [Kcal/h]
renin kazan yükü,BL	= 2338 [Kcal/h]
renin isitma verimi,x	= 85.5 [%]

iDROLiK :

renin manifold numarasi	= 1
udaki su hizi,vs	= 0.49 [m/s]
nolds sayisi,Re	= 11184
u-su tasinim katsayisi,α	= 2901 [Kcal/h m² °C]
u sürtünme katsayisi,τ	= 0.0343
im boru uzunlugunda basinc kaybi,Dp	= 35.8 [mm.ss/m]
re içersindeki boru uzunlugu,L	= 110 [m]
ifolda gidis/dönüs uzakligi	= 3 [m]
ifolddan itibaren toplam boru uzunlugu,Lt	= 113 [m]
lam basinç kaybi,Pt	= 4046.81 [mm.ss]
debisi	= 4 [lt/dk]

Yaklasik Enerji Depolamasi Hesabi :

uz tarife sirasinda döseme sicakligi	= 14.2 °C
uz tarife sirasinda isitma suresi	= 6.00 saat
semenin isitilmadigi süre	= 16.00 saat

polanan enerji isil konforu sadece2.9 saat
rdürebilir. 16.0 saat sonunda
sicaklik yaklasik -11 °C olacak.

1/1sayili oda için nihai sonuçlar:

oplam devre adedi	= 4
erekli toplam boru uzunlugu	= 452 [m]
oplam dosemeden isitma kapasitesi	= 8000 [Kcal/h]
oplam kazan yükü	= 9353 [Kcal/h]
oplam isitilan döseme yüzeyi	= 100 [m²]
oplam su debisi	= 16 [lt/dk]

Sayilar yuvarlatilmistir.)

EK 7

PANEL ISITMA ISI YÜKÜ PROGRAMI, ISI2

Bu ekte projede yardımcı araştırmacı Mahmut Uludağ'ın Yüksek Lisans Tezi olarak hazırladığı Isı programı tanıtılmaktadır. Bu programın iki versiyonu vardır. Birinci versiyon, Isı1 TS 2164 e göre, ikinci versiyon, Isı2 ise yeni standarta göre binaların ısı yükünü hesaplayabilmektedir.

7.1. Amaç

Bu projenin amacı doğrultusunda geliştirilen Yerden Istıtma Sistemlerinin Tasarım Esasları Türk Standardı, yerden ısıtma sistemlerinin tasarımında bilgisayar kullanımını nerdeyse zorunlu hale getirmektedir. Ülkemizde şimdiye kadar yerden ısıtma uygulanan binaların ısı kaybı hesaplamaları TS 2164'e göre, panellerin seçimide değişik kişi ve kuruluşlar tarafından hazırlanmış abaklarla yapılmaktaydı.

Isı2 programı, tasarımcıların kullanımı oldukça yaygınlaşan bilgisayarlardan faydalanabilmeleri ve hazırlanan standartın kolayca kabul görmesi amacıyla geliştirildi. Bu program girili bilgileri en akılcı bir şekilde kullanabilmesi ve aynı zamanda tasarım parametrelerinin değişmesine duyarlı olmasıyla binaların ısıl optimizasyonu içinde kullanılabilir bir araç oldu.

Programın girili bilgileri akıllıca kullanması ve paremetrelerin değişmesine duyarlı olması geliştirilen bina veri yapısı ile mümkün olmuştur. Bu yapı şematik olarak ek 7.1. de gösterilmektedir, ek 7.2. de ise bu veri yapısının prolog ile tanımlanması vardır.

7.2. Döşemeden Isıtmada Farklı Isı Yükü Hesaplama Metodu Kullanılmasının Nedenleri

1- Döşemeden ısıtma sistemlerinde iç mekan tasarım sıcaklığı T_a, eşdeğer konfor koşulu için standard değerlerden 2 ila 3 °C daha düşük seçilebilir. Bunun iki ana nedeni sırası ile iç konforun daha çok ışıma ısı transferi ile gerçekleşmesi ve iç mekandaki hava hareketlerinin daha zayıf oluşudur. Bu son faktör insan bedeninden taşınım yolu ile olan hissedilir ısı kayıplarının azalmasına katkıda bulunmaktadır. Bu faktör duvar ve özellikle pencerelerden olan ısı kayıplarının da bir miktar azalmasına yol açar.

2- Tasarım iç sıcaklığının (T_a') daha düşük seçilmesi ve iç hava hareketlerinin zayıf olması nedenleri ile ısı transferi ve infiltrasyon ısı kayıplarında önemli bir azalma söz konusudur. Ayrıca iç mekandaki hava sıcaklığı daha homojen bir dağılım sergiler. Bu ise tavandan olan ısı kayıplarını azalttığı gibi konfor koşullarını daha da iyileştirir. Eğer döşemeden ısıtma sistemi kesintili kullanılmıyor ise panelde, çevre duvarlarda ve tavanda belirli bir ısı depolaması söz konusudur. Bu nedenle ısı yükünün günlük pik değerinde azalma olacaktır. Böyle bir azalmanın söz konusu olduğu durumlarda pik yük indirim katsayısı, C kullanılır, bu katsayı günlük dış sıcaklık değişimine, yapı elemanlarının ısı iletim katsayılarına, panelin yapısına ve işletme rejimine bağlıdır.

7.3. Döşemeden Isıtmada Mahal Isı Yükü (Q_h) Hesabı

Döşemeden ısıtmada mahal ısı yükü aşağıdaki formül yardımıyla bulunur.

$$Q_h = C \cdot (Q_T + Q_L) \qquad \text{kcal/h} \qquad (1)$$

(C) Pik yük traşlama katsayısı olup iklime ve bina elemanlarının kütlesel yoğunluğuna bağlıdır.

(Q_T) Isı iletimi ve taşınımı ile meydana gelen ısı kaybı aşağıdaki formül ile hesaplanır.

$$Q_T = (1 + Z_H) \cdot Q_o \qquad \text{kcal/h} \qquad (2)$$

(Q_o) zamsız ısı kaybı, aşağıdaki eşitlikle hesaplanır.

$$Q_o = \sum_{i=1}^{n_y} k_i \cdot A_i \cdot \Delta T_i \qquad \text{kcal/h} \qquad (3)$$

Burada; n_y mahaldeki ısı kaybına maruz yüzey sayısıdır. (ΔT) sıcaklık farkı, hesabı yapılan mahallin eşdeğer konfor sıcaklığı ile dış hava veya komşu hacımlar sıcaklıkları arasındaki farktır.

(Q_L) Hava Sızıntısı ve Yenilenmesi ile Meydana Gelen Isı Kayıpları aşağıdaki formül yardımı ile hesaplanır:

$$Q_L = \sum_{A} (a.l) . R . H . Z_E . e_y . (T_a - T_d) \qquad \text{kcal/h} \qquad (4)$$

7.4. Isıtılmayan İç Yüzeyler Ortalama Sıcaklığı, T_u hesabı.

Tu ve Tp değerlerinin hesaplanması ve modül mesafelerinin bulunması için gerekli bilgiler şunlardır:

Q_h : düzeltilmiş hacim ısı ihtiyacı (kcal/h)
T_a : hacim tasarım iç sıcaklığı (°C)
T_s : panel ortalama su sıcaklığı (°C)
k_h : ısıtma borusu iletkenlik değeri (kcal/hm°C)
D_o : ısıtma borusu dış çapı (mm)
D_i : ısıtma borusu iç çapı (mm)
V_w : ısıtıcı suyun hızı (m/s)
T_i : i panelinde boru ekseninden panel yüzeyine kadar olan mesafe (mm)
Y_{a_i} : i paneli alt direnci
Y_{y_i} : i paneli üst direnci
T_{n_i} : i paneli alt sıcaklık
A_{p_i} : i paneli etkin alanı
T_u : hacim alan ağırlıklı ortalama iç yüzey sıcaklığı

T_u hacim alan ağırlıklı ortalama iç yüzey sıcaklığı aşağıdaki eşitlik yardımıyla bulunur.

$$T_u = \frac{\sum_{j=1}^{n} T_{u_j} A_j}{\sum_{j=1}^{n} A_j} \qquad (5)$$

eşitlikte:

n : hacimde ısı kaybeden iç yüzeylerin sayısı
A_j : j yüzeyi net alanı
T_{u_j} : j yüzeyi ortalama iç sıcaklığıdır

Bu denklemin tam doğru sonuç verebilmesi için iç duvarlarda dahil olmak üzere tüm duvarlar hasaba katılmalıdır.

İç yüzeyler için ısı taşınım denklemi yazılıp T_{u_j} için çözüldüğünde:

$$T_{u_j} = T_a - \frac{Q_j}{h A_j} \qquad (6)$$

bulunur. Burada Q_j j yüzeyinin ısı kaybı olup program ısı kaybı hesaplamalarında önceden bulmuştur ve bilgisayarın belleğinde saklamaktadır.

Eşitlikteki h değeri ise iç yüzeyler için ısı taşınım katsayısı olup, yerden ısıtmada hava hareketlerinin daha zayıf olmasından dolayı 5.25 kcal/h.m^2 olarak alınmaktadır.
Böylece eşitlik (6)

$$T_{u_j} = T_a - \frac{Q_j}{5.25\, A_j} \qquad (7)$$

olur.

7.5. Örnek Program Kodları

Bu bölümde program hakkında okuyucuyu bilgilendirmek için, doğrudan ısı kaybı hesaplamaları ile ilgili program kodu açıklama satırlarıyla tanıtılmaktadır.

7.5.1. Oda ısı kaybı hesabı.

```
oda_hesap(Code,room(Ref,Ze,SList),mahal(Mahal,Tin,T2),
losses(Cond,IZ,KZ,YZ,Zamli,inf(AL,R,H,INF),Total)):-!,
% Bina işletim şekli, durum katsayısı ve şehir dış sıcaklığını veri
tabanından çek.
      binabilgi(_,bina(_,_,OPMODE,_,city(_,Tout,_,_)),_,_,_,_,H),!,
% Odanını referans ettiği oda tipinden, odan adı ve tasarım sıcaklığını
öğren.
      ref_term(binadba,genel,Ref,mahal(Mahal,Tin,T2)),
% Her yüzeyin referans ettiği yüzey tipinden yüzeye ait bilgileri al.
```

```
    load(SList),!,
% Yüzey bazında ısı kaybı hesaplamalarını_yap.
    yuzey_hesabi(1,Tin,0,1,TOTALAREA,0,COND,0,AL,0,IK_AL,""),!,
% Oda yön zammını tesbit et.
    aspect_zam(YZ),!,
% Oda durum katsayısını (R) bul.
    r_katsayisi(IK_AL,AL,R),!,
% Kat zammını ve infilitrasyon için yükseklik zammını öğren.
    kat(CODE,_,KZ,_,E),!,
% Oda infiltrasyon ısı kaybını bul.
    INF=AL*R*H*E*(Tin-Tout)*ZE,
% Zamlı ısı kaybını hesapla.
    ZAMLI=COND*(100+IZ+YZ+KZ)/100,
% Toplam ısı kaybını hesapla.
    TOTAL=ZAMLI+INF.
```

7.5.2. Yüzey ısı kaybı hesabı.

```
yuzey_hesabi(LINE_NO,TR,PAREA,PTOTAREA,TOTAREA,PLOSS,TOTLOSS,
PREV_AL,TOTAL_AL,PREVIK,TOTIK,Pyon):-
% sıradaki yüzeyin bilgilerini hesaplamaları yapmak için değişkenlere aktar
ve sil.
    retract(satir(LINE_NO,Tip,YON,KAL,[L,H,N,K,T|AL],_,_,KEY,Is)),!,
% yüzey alanını hesapla.
    AREA=L*H*N,
% yüzey net alanını bul.
    cikan_alan(Tip,AREA,PAREA,NOWAREA,NEXTAREA,Pyon,Yon),!,
% yüzey ısı kaybı.
    LOSS=NOWAREA*(TR-T)*K,
% yüzey ısı kaybını oda ısı kaybına ekle.
    NEXTLOSS=PLOSS+LOSS,
% yüzeyin oda hava sızıntısı kaybına etkisi.
    toplam_al(Tip,AL,PREV_AL,NEXT_AL,N,PREVIK,NEXTIK),!,
% yüzey satırını yeni bulunan bilgilerle tekrar hafızaya yerleştir.
    assert(satir(LINE_NO,Tip,YON,KAL,[L,H,N,K,T|AL],NOWAREA,LOSS,KEY,Is)),
```

```
% bir sonraki yüzey no
      NEXT=LINE_NO+1,
% yüzey alanını oda yüzeyleri toplam alanına ekle
      NTOTAREA=PTOTAREA+NOWAREA,!,
% aktif yüzeyin sonuçlarının eklendiği yeni oda bilgileri ile
% bir sonraki yüzey için tekrar yüzey hesaplarını çağır.
yuzey_hesabi(NEXT,TR,NEXTAREA,NTOTAREA,TOTAREA,NEXTLOSS,TOTLOSS,
NEXT_AL,TOTAL_AL,NEXTIK,TOTIK,Yon).

% başka yüzey kalmadığında o anki değerleri oda toplamlarına eşitle.
yuzey_hesabi(_,_,_,TOTAREA,TOTAREA,TOTLOSS,TOTLOSS,
TOTAL_AL,TOTAL_AL,TOTIK,TOTIK,_).
```

7.5.3. Fuga değerleri hesabı.

```
% yüzey iç kapı ise fugasını iç kapı fugaları toplamına ekle.
toplam_al("I.K.",[A,L],PREV,PREV,N,PREVIK,NEXTIK):-!,
      NEXTIK=PREVIK+A*L*N,!.

% yüzey sızdırganlık değerine sahipse, fuga değerini oda fuga değerine ekle.
toplam_al(_,[A,L],P_AL,NEXT,N,IK,IK):-!,
NEXT=P_AL+N*A*L.

% yüzey kapı veya pencere değilse fuga etkisi yoktur.
toplam_al(_,_,PREV,PREV,_,IK,IK):-!.
```

7.5.4. Oda Yön zammı.

```
% Odada Kuzey, K. Doğu veya K. Batıya bakan bir yüzey varsa, yön zammı %5
dir.
aspect_zam(5):-
      satir(_,_,YON,_,_,_,_,_,_),
      frontchar(YON,'K',_),!.
```

% Odada Güney, G. Doğu veya G. Batıya bakan bir yüzey varsa, yön zammı %-5 dir.

```
aspect_zam(-5):-
      satir(_,_,YON,_,_,_,_,_,_),
      frontchar(YON,'G',_),!.
```

Diğer durumlarda yön zammı %0 dır.

```
aspect_zam(0).
```

7.5.5. Oda durum katsayısı (R).

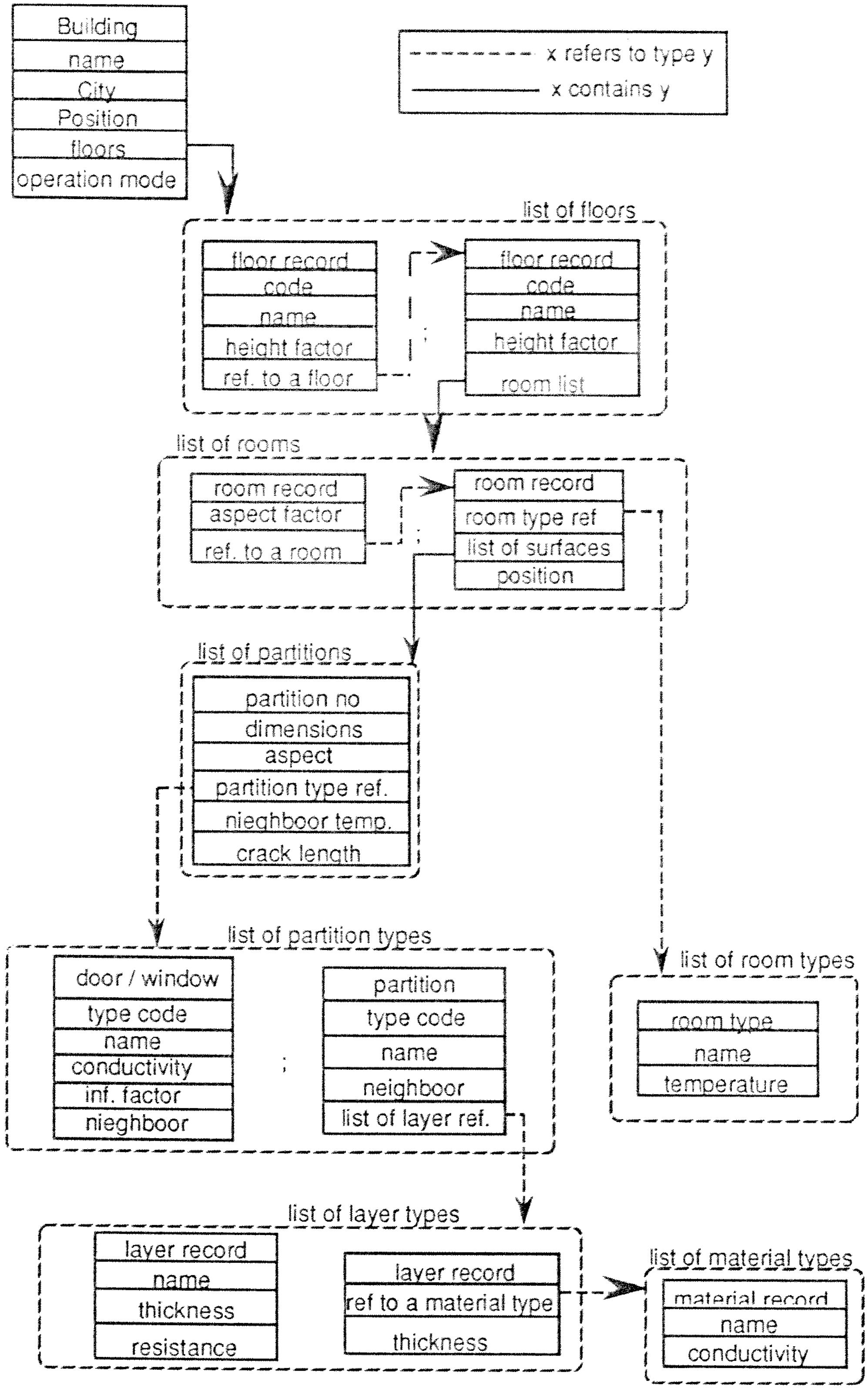
```
% iç kapı girilmediyse R yi 0.6 olarak seç.
r_katsayisi(0,_,0.6):-!.

% pencere fugalarının iç kapı fugalarına oranı 3 den azsa R=0.9
r_katsayisi(IK,AL,0.8):- AL/IK < 3,!.

% diğer durumlarda R=0.6
r_katsayisi(_,_,0.9).
```

7.5.6. Otamatik alan çıkartma kuralları

```
% Tavanlardan, döşemelerden ve panellerden alan çıkartma olmaz.
cikan_alan("Tav.",AREA,_,AREA,0,_,_):-!.
cikan_alan("Dos.",AREA,_,AREA,0,_,_):-!.
cikan_alan("Pan.",_,_,0,0,_,_):-!.

% İç ve dış duvarlar bir önceki yüzeyle aynı yönde ise, biriken alanı
% duvar alanından çıkar, değilse biriken alanı sıfırla.
cikan_alan("I.D.",AREA,PREV,NOW,0,Yon,Yon):- !,
NOW=AREA-PREV,!.
cikan_alan("I.D.",AREA,_,NOW,0,_,_):-!,NOW=AREA,!.
cikan_alan("D.D.",AREA,PREV,NOW,0,Yon,Yon):-!,
NOW=AREA-PREV,!.
```

```
cikan_alan("D.D.",AREA,_,NOW,0,_,_):-!, NOW=AREA,!.

% Kiriş ve kolonlardan alan çıkmaz.
cikan_alan("Kir.",AREA,_,AREA,NEXT,_,_):-!,NEXT=0.
cikan_alan("Kol.",AREA,_,AREA,NEXT,_,_):-!,NEXT=0.

% Yüzey duvar değil ve bir önceki yüzeyle aynı yönde ise
% alanını biriken alana ekle.
cikan_alan(_,AREA,PREV,AREA,NEXT,Y,Y):-!, NEXT=AREA+PREV.
cikan_alan(_,AREA,_,AREA,Area,_,_):-!.
```

Ek 7.1. Geliştirilen bina veri yapısının şematik gösterimi



Ek 7.2. Geliştirilen bina veri yapısının Turbo Prolog ile tanımlanması

```
domains

bina = bina(bina_adı, bina_durumu, işletim_şekli, şehir, kat_listesi)
      bina_durumu = bitişik; ayrık
      işletim_şekli = kesintili; kısa_kesintili; kesintisiz
      şehir = şehir(şehir_adı, dış_sıcaklık, rüzgar_durumu, bölge_no)
      rüzgar_durumu = rüzgarlı; rüzgarsız

kat = kat(kat_kodu, kat_adı, yükseklik_zammı, benzer_mi, oda_listesi)
      benzer_mi= benzer_kat(kat_kodu); benzer_degil(bt_selector)

oda = oda(oda_tipi_referansı, köşe_mi, yüzey_listesi);
    benzer_oda(oda_kodu,yön)
      oda_tipi = oda(oda_adı, iç_sıcaklık)
      köşe_durumu= köşe_oda; normal_oda

yüzey =      yüzey(yön, yüzey_tipi_referansı, boy, en, adet, komşu_sıcaklık,
      pencere_mi)
      pencere_mi = pencere(fuga_uzunluğu); pencere_değil
komşu_sıcaklık = dış_sıcaklık; yan_toprak; bitişik_ev; çatı_sıcaklığı;
alt_toprak; komşu_hacim(oda_tipi_referansı)

yüzey_tipi = duvar(duvar_adı, katman_listesi, taşınım_katsayısı,
taşınım_katsayısı);        pencere(pencere_adı, ısı_ilitkenliği,
infilitrasyon_katsayısı)

katman = katman(malzeme_referansı, kalınlığı)
malzeme = malzeme(malzeme_adı, ısı_iletim_katsayısı)

malzeme_adı, oda_adı, yüzey_adı, kat_adı, yön = string
ısı_iletim_katsayısı, katman_kalınlığı = real
oda_sıcaklığı, taşınım_katsayısı = integer
malzeme_referansı, yüzey_tipi_referansı, oda_tipi_referansı = ref

katman_listesi = katman*
yüzey_listesi = yüzey*
oda_listesi = oda*
kat_listesi = kat*
```

Ek 7.3. Yüzey tipleri muhtemel konumları

			h_i	h_o	T	
D.D.	Dış Duvar					
	i -	Dış hava temaslı	5	20	$T_{dış}$	
	ii -	Toprak temaslı	5	∞	T_{toprak}	
	iii -	Bitişik ev duvarı	5	∞	$T_{bitişik\text{-}ev}$	
İ.D.	İç Duvar					
	i -	Merdiven evi duvarı	5	5	$T_{komşu\text{-}hacim}$	
	ii -	Normal iç duvar	5	5	$T_{komşu\text{-}hacim}$	
Kir.	Kiriş					
	i -	Dış hava temaslı	5	20	$T_{dış}$	
	ii -	Toprak temaslı		5	∞	$T_{toprak\text{-}yan}$
	iii -	Bitişik ev temaslı		5	∞	$T_{bitişik\text{-}ev}$
Kol.	Kolon					
	i -	Dış hava temaslı	5	20	$T_{dış}$	
	ii -	Toprak temaslı	5	∞	$T_{toprak\text{-}yan}$	
	iii -	Bitişik ev temaslı		5	∞	$T_{bitişik\text{-}ev}$
Tav.	Tavan					
	i -	Üstü açık	5	20	$T_{dış}$	
	ii -	Çatı arasına bakar	5	7	$T_{çatı}$	
	iii -	Kat arası tavan	5	5	$T_{komşu\text{-}hacim}$	
Döş.	Döşeme					
	i -	Zemine oturan döşeme	3	∞	T_{toprak}	
	ii -	Açık geçit üstü	3	20	$T_{dış}$	
	iii -	Kat arası döşeme	3	3	$T_{komşu\text{-}hacim}$	

Ek 7.4 Örnek Çalışma:

Elektrik Mühendisleri Odası Ankara Şube binasının 3 ayrı durumda ısı kaybı hesabı yapılmıştır.

NO	ODA ADI	T °C	Qt1	Qt2	Qt3
		2. kat			
N201	MAKAM ODASI	22	2967	1941	1552
N202	MUHASEBE ODASI	22	3933	2611	2089
N203	TOPLANTI SALONU	20	3319	2231	1785
N204	MUTFAK	20	2057	1509	1207
N205	BEKLEME ODASI	20	909	667	533
N206	W.C.	18	15	11	9
N207	MERDİVEN	18	864	559	447
		1. kat			
N101	ÇALIŞMA ODASI	22	2505	1652	1322
N102	MÜHENDİSLER ODASI	22	6045	3991	3193
N103	SEKRETERYA	20	646	465	372
N104	ÇALIŞMA ODASI	22	1788	1279	1023
N105	W.C.	18	0	0	0
N106	MERDİVEN	18	747	470	376
		giriş katı			
Z01	DÜKKAN	20	9176	6995	5596
Z02	W.C.	18	0	0	0
Z03	MERDİVEN	18	793	574	459
Z04	MERDİVEN	18	421	284	228
		1. bodrum kat			
1B01	DEPO	20	2247	1530	1224
1B02	DEPO	20	4817	3486	2789
1B03	MERDİVEN	18	111	80	64
		2. bodrum kat			
2B01	KAZAN DAİRESİ	20	0	0	0
2B02	ARŞİV	15	114	76	60
2B03	OFİS	20	1292	878	703
2B04	TOPLANTI SALONU	20	1278	896	717
2B05	DEPO	20	1039	722	577
2B06	ANTRE	20	87	60	48
2B07	MERDİVEN	18	238	166	133
Total Heat load		:	47408	33143	26507
		%100	%70	%56	

Durum 1: TS 2164

Durum 2: Yeni standart + Hafif bina.

Durum 4: Yeni standart + Normal bina.

Kurala dayalı programlama

Prolog Japonların 6. nesil bilgisayarlarda (super computers) temel dil (kernel language) kabul ettikleri, klasik dillerdeki programlama metodlarından farklı olarak kurala dayalı programlamanın (rule based programming) esas alındığı bir dildir.

Turbo Prolog

Turbo prologda programlar 'domains', 'predicates', 'clauses' anahtar kelimeleriyle ayrılan üç bölümden oluşurlar:

1. (domains)
Problemle ilgili objelerin isim ve yapıları tanımlanır.

2. (predicates)
Objeler arasındaki ilişkilerin isim ve parametreleri tanımlanır.

3. (clauses)
Tanımlanan ilişkileri anlatan kurallar ve gerçeklerden oluşur: Bu kısım programın ana kısmı olup problemi anlatan mantıksal cümlelerin tamamıdır. Bu mantıksal ifadeler ya gerçekler "bugün yağmur yağıyor" şeklinde veya kurallar "eğer yağmur yağarken dışarı çıkmışsanız ve şemsiyeniz yoksa ıslanırsınız" şeklinde olur.

Prolog ile programlama iki adımla özetlenebilir

1. Objelerle ilgili gerçeklerin kuralların ve aralarındaki ilişkilerin tanımlanması.

2. Objelerle veya aralarındaki ilişkilerle ilgili soru sorulduğunda Prolog çıkarım mekanizmasının (inference engine) çözüm bulması. Bu mekanizma çözüm tarama (by backtracking) ve paremetre eşleme (unification) işlemlerinin belli bir uyum ile çalışmasından ibarettir. Prologda çözüm taraması aşşağıdan yukarıya doğru yapılır ve kullanıcının anlaması oldukça kolaydır.

Ek 7.6. Yüzey tiplerinin Prolog veri tabanında gösterilmesi

```
coeff(no_check,komsu,"I.D.",7,7,"ic duvar Hi=7 Hd=7").
coeff(no_check,tdis,"Kir.",7,20,"Dış hava temaslı Hi=7 Hd=20").
coeff(no_check,yantoprak,"Kir.",7,250,"Toprak temaslı Hi=7 1/Hd=0").
coeff(no_check,bitisikev,"Kir.",7,250,"Bitisik ev Hi=7 1/Hd=0").
coeff(no_check,tdis,"Kol.",7,20,"Dış hava temaslı Hi=7 Hd=20").
coeff(no_check,yantoprak,"Kol.",7,250,"Toprak temaslı Hi=7 1/Hd=0").
coeff(no_check,bitisikev,"Kol.",7,250,"Bitisik ev Hi=7 1/Hd=0").
coeff(check_r(0.47,0.7,0.92),tdis,"D.D.",7,20,"Dış hava temaslı Hi=7 Hd=20").
coeff(check_r(0.47,0.7,0.92),yantoprak,"D.D.",7,250,"Toprak temaslı ").
coeff(check_r(0.47,0.7,0.92),bitisikev,"D.D.",7,250,"Bitisik ev Hi=7 1/Hd=0").
coeff(check_r(1.5,2.4,3),tdis,"Tav.",7,20,"Üstü açık hi=7 hd=20").
coeff(no_check,komsu,"Tav.",7,7,"Ara kat hi=7 hd= 7").
coeff(check_r(0.65,0.93,1.5),tcati,"Tav.",7,7,"Cati alti hi=7 hd= 7").
coeff(no_check,komsu,"Dos.",5,5,"Ara kat döşeme hi=5 hd=5").
coeff(check_r(0.65,0.93,1.5),alttoprak,"Dos.",5,250,"Toprak temaslı ").
coeff(check_r(1.4,2,2.5),tdis,"Dos.",5,20,"Açık Geçit üstü hi=5 hd=20").
coeff(no_check,komsu,"Pan.",255,5,"Ara kat döşeme panel hi=7 hd=5").
coeff(no_check,alttoprak,"Pan.",255,250,"Toprak temaslı panel ").
coeff(no_check,tdis,"Pan.",255,20,"Açık Geçit üstü hi=7 hd=20").
```

Ek 7.7. ISI PROGRAMI KULLANIM KILAVUZU

Yeni bir bina için izlenecek işlem sırası şöyledir :

1. Programa girmeden önce yapılması gerekenler.
2. Bina için dosya açılması.
3. Yüzey Tipleri ısı iletkenlik değerleri hesabı;
   a. Malzemelerin girilmesi,
   b. Katmanların tanımlanması,
   c. K ısı iletkenlik değerleri hesabı.
4. Kapı-pencere tiplerinin girilmesi
5. Hacim tiplerinin girilmesi.
6. Katların tanımlanması.
7. Katlara odalarının girilmesi.
8. Odalara yüzeylerin girilmesi.
9. Isı kaybı raporlarının alınması;
   a. Kat ısı kaybı raporları,
   b. Bina özet ısı kaybı raporu,
   c. Bina genel raporu.

1. Programa Girmeden Önce Yapılması Gerekenler

- Binada kullanılan yüzey tiplerinin detayları incelenmeli ve kullanılan malzemelerin ısıl iletkenlik değerleri ve kullanım kalınlıkları listesi hazırlanmalıdır.
- Bina kapı ve pencerelerinin iletkenlik ve sızdırganlık değerleri öğrenilmelidir.
- Bina hacim tipleri ve sıcaklıkları öğrenilmelidir.
- Bina yönü, nizamı, uygulanacak ısıtma şekli ve binanın bulunduğu şehrin tasarım dış sıcaklığı, rüzgar durumu ve ait olduğu ısı bölgesi öğrenilmelidir.
- Katlar için kat kodu (Z, N1,N2,...) belirlenmeli ve katların infilitrasyon yükseklik zammı öğrenilmelidir.
- Benzer kat ve odalar tesbit edilmelidir.
- Odalar mimari proje üzerinde numaralandırılmalı, yön zammı haricinde benzer odaların hangi odaya benzediği proje üzerinde yazılmalıdır.

2. Bina için Dosya Açılması

"Bina" menüsünden "Bina Tanımla" satırını seçiniz. Program her bina için diskinizde bir dosya açar, bu amaçla önce çalışmak istediğiniz bina dosyasına bir isim vermenizi ister, program verdiğiniz isme ".bna" ekleyerek diskinizde bir dosya oluşturur. Program daha sonra binayı tanımlayıcı temel bilgileri şu sırayla sorar:

1. Binanıza bir isim vermenizi ister, "Alper Güzelgöz Binası" gibi.
2. Binanızın bulunduğu şehri sorar.
3. Şehrin dış sıcaklığını (°C).
4. Şehrin dahil olduğu ısı bölgesini (1., 2. veya 3. bölge).
5. Rüzgar durumunu (menü ile) (rüzgarlı bölge, normal bölge).
6. Bina durumunu (menü ile) (bitişik nizam, ayrık nizam).
7. Bina ısıtma sistemi şekli (menü ile)(kesintili, kısa kesintili, kesintisiz).

Verdiğiniz dosya ismine ".bna" eklenerek diskinizde bir dosya oluşturulur ve girdiğiniz binayı tanımlayıcı bilgiler diske yazılır.

3. Yüzey Tipleri ısı iletkenlik değerleri Hesabı

3.a. Malzemelerin Girilmesi

Önce binada kullanılan malzemelerin ısı iletkenlik değerleri girilir. Bu amaçla "Malzeme" menüsünden "Malzeme Gir" satırı seçilir. Burada program önce malzeme adını sonra iletkenlik değerini (kcal/h.m°C) sorar. Malzeme adı verilmeyene, veya ESC tuşuna basılana kadar program sorularına devam eder. Bu şekilde binanızdaki tüm malzemeleri programa aktarınız. Unuttuğunuz bir malzeme olursa veya herhangi bir nedenle girmediğiniz bir malzeme kalırsa daha sonrada bu girme işlemine geri dönüp devam edebilirsiniz.

3.b. Katmanların Tanımlanması

İkinci olarak bina yüzeylerinde kullanılan katmanlar tanımlanır. Bu amaçla "Katman" menüsünden "Katman Gir" satırını seçiniz. Burada program size bir menü ile binada girili malzemelerin listesini ve menünün ilk satırı olarak da "Hazır Katman" seçeneğini karşınıza getirir. Katman tanımlamak için iki seçeneğiniz vardır.

i - "Hazır Katman" seçilir; Bu durumda program size katmanınızın adını, kalınlığını ve direncini sorar.

ii - Malzeme listesinden bir malzeme seçilir; Bu durumda program sadece katman kalınlığını sorar, Katman adını ise malzemenin adına katman kalınlığını ekleyerek bulur. Seçtiğiniz malzemenin ısıl iletkenlik değeri daha sonra herhangi bir nedenle değiştirilirse program bunu algılar ve katman ısıl direnci değerini düzeltir.

3.c. K ısı iletkenlik değeri hesabı.

Bundan sonra yüzey tiplerinin tanımına sıra gelmiştir. Bu amaçla "Yüzey" menüsünden "Yüzey Hesabı" satırı seçilir. Burada program bir menü ile girmek istediğiniz yüzeyin tip kodunu seçmenizi ister. Program seçtiğiniz yüzey tipinin muhtemel konumlarını bir liste ile karşınıza çıkarır ve girmek istediğiniz yüzeyin konumunu seçmenizi ister.

Program yüzey K değerlerini hesaplarken, iç ve dış yüzey taşınım katsayılarını yüzeyin konumuna göre bulur, ayrıca yine ısı kaybı hesaplamalarında yüzey komşu sıcaklığı da burada girdiğiniz yüzey konumuna göre otomatik olarak bulunur.

Yüzey konumu seçildikten sonra program yüzeyinize bir isim vermenizi ister. Daha sonra da binada bulunan katmanların listesini bir menü ile karşınıza çıkarır. Burada yapılacak işlem, yüzeyi oluşturan katmanların içten dışa doğru sırayla seçilmesidir. Seçme işlemi seçilecek katmanın üzerine aşağı veya yukarı oklarla gitmek ve Enter tuşuna basmak suretiyle olur. Bütün katmanlar seçildikten sonra F10 ile menüden çıkılır. Bundan sonra program katman listesine göre yüzeyin ısıl direncini hesaplar, yüzeyin konumuna göre iç ve dış yüzey taşınım katsayılarını bulur ve yüzeyin ısı iletkenlik değerini hesaplar.

Yüzey tipi "Pan." yerden ısıtma panelleri için kullanılır. Tip olarak "Pan." seçildiğinde program önce tanımlanmak istenen panelin konumunu sorar sonrada bir isim vermenizi ister. "Boru hizası" satırınıda ekleyerek katman listesini karşınıza çıkarır. Bu listeden önce panelinizin en üst döşeme yüzeyinden başlayarak boru hizasına kadar olan katmanlarını seçiniz, sonra "Boru hizası" satırını seçiniz, daha sonrada boru hizasından döşeme alt yüzeyine kadar olan katmanları sırayla seçiniz, F10 ile çıkınız. Program seçtiğiniz katmanlara göre alt ve üst dirençleri hesaplar ve tanımladığınız panele ait bilgileri disk dosyanıza yazar.

4. Kapı-Pencere Tiplerinin Girilmesi

Yüzey menüsünden "K/P Gir" satırını seçiniz. Program size bir menüde girmek istediğiniz Kapı veya Pencere tipinin kodunu sorar. Seçiminizi Enter tuşu ile yapınız.

Tip kodunu seçtikten sonra program tanımlamak istediğiniz Kapı veya Pencere tipine bir isim vermenizi ister. Daha sonra, ısı iletkenlik değeri K (kcal / h.°C) ve sızdırganlık katsayısı a (m^3/m.h.) değerlerini sorar. Bu bilgiler girildikten sonra, program kapı veya pencere tipinizi diske yazar ve ana menüye geri döner.

Tablo 1. Kapı-Pencere Tip Kodları

T.P.	Tek Pencere
C.P.	Çift Pencere
CcP.	Çift camlı Pencere
I.K.	İç Kapı
I.P.	İç Pencere
D.K.	Dış Kapı
B.K.	Balkon Kapısı

5. Hacim Tiplerinin Girilmesi

"Hacim" menüsünden "Hacim Gir" satırını seçiniz. Program hacim tipi için önce isim sorar, sonra girilen hacim tipi için iç tasarım sıcaklığını sorar. Bir hacim tipi tanımlandığında diske yazılır ve program diğer hacim tipleri içinde soru sormaya devam eder. Program hacim tipi için isim verilmeyinceye veya ESC tuşuna basılıncaya kadar aynı soruları sormaya devam eder.

6. Katların Tanımlanması

"Katlar" menüsünden "Kat Gir" satırını seçiniz. Tanımlamak istediğiniz katın sırasıyla önce kodunu, sonra ismini, daha sonrada yükseklik zammını giriniz, bundan sonra program katın başka bir kata benzeyip benzemediğini sorar. Katınız başka bir kata bütünüyle veya kat yükseklik zammı haricinde benziyorsa benzer olduğu katın kodunu giriniz.

Not: Başka bir kata benzer tanımladığınız katların bilgilerini program doğrudan benzer olduğu kattan alır, bu nedenle böyle katların sonuçlarını görebilmeniz için benzer olduğu katın bilgilerini rapor istemeden önce girmiş olmanız gerekir.

7. Katlara Odaların Girilmesi

"Katlar" menüsünden "Odalar" satırını seçiniz. Burada program size çalışmak istediğiniz katı kat listesinden seçmenizi ister. Bundan sonra program seçilen katın odalarının listelendiği ve oda girme ve silme işlemlerinin yapılabildiği yeni bir ekran karşınıza çıkarır. Bu ekranda "ins" tuşu ile seçici satırın üzerinde olduğu odadan önce yerleştirilmek üzere yeni bir oda girilebilir. "ins" tuşunu seçtiğinizde, program girmek istediğiniz odanın tipini oda tipleri listesinden seçmenizi ister. Ayrıca listenin ilk satırı olarak "- Benzer Oda -" satırı gelir. Bu satır, gireceğiniz oda bütünüyle veya yön zammı haricinde başka bir odaya benziyorsa kullanılır. "Benzer Oda" satırı seçildiğinde program ek olarak benzenen odanın oda kodunu (N101, Z03 gibi) ve tanımladığınız odanın yön zammını sorar. Benzer olmayan odalarda oda yön zammını oda yüzeyleri yön durumlarına göre program bulur.

8. Odalara Isı Kaybı Yüzeylerinin Girilmesi

"Katlar" menüsünden "Odalar" satırını seçiniz. Program çalışmak istediğiniz katı kat listesinden seçmenizi ister. Bundan sonra seçilen katın odalarının listelendiği ve oda girme ve silme işlemlerinin yapılabildiği yeni bir ekran karşınıza çıkar. Bu ekranda "Enter" tuşu ile seçici satırın üzerinde olduğu odanın yüzeylerine ekleme silme yapabileceğiniz diğer bir ekrana geçilir.

Programın otomatik alan çıkartma özelliğini kullanabilmek için, pencere, kiriş, kolon gibi alanlar aynı yöne sahip dış duvardan önce girilmelidir. Bu kurala uyularak girilen aynı yöne sahip yüzeylerin arasına başka yüzey girilmedikçe, program pencere, kiriş gibi yüzeylerin alanlarını otomatik olarak aynı yöndeki dış duvardan çıkartır.

Yüzey girişi "ins" tuşu ile yapılır, "ins" tuşuna basıldığında program binada tanımlı yüzey tiplerin listesini bir menü ile karşınıza çıkarır. Girmek istediğiniz yüzeyin tipinin üzerine oklarla gidip Enter ile seçiniz. Bundan sonra program seçilen yüzey için bir menüde yön sorar, eğer yüzey I.K. veya I.D. ise otomatik olarak yön ihmal edilir (yani yön "--" alınır). Daha sonra program yüzeyin en ve boyunu sorar. Yüzeyiniz eğer Kapı, Pencere tipinde ise, program ek olarak fuga (pencere ve kapı açıklıkları) uzunluğunu sorar.

Yerden ısıtma panellerinin girilmeside bu ekranda yapılır. "ins" tuşu ile yüzey tipleri listesine girdiğinizde eğer tanımlanmış panel tipleriniz varsa bu listenin içinde onlarda gelir. Listeden panel tipi seçilirse program panelin en ve boyunu sorar. Bir odaya birden fazla panel girilebilir.

Yüzey komşu sıcaklığını program yüzey tip ve konumuna bakarak bulur. Yani

$T_{dış}$	;	dış ortam sıcaklığı, bina tanımından bulur.
$T_{komşu\text{-}hacim}$	;	bu durumda komşu hacmin tipini sorar.
$T_{bitişik\text{-}ev}$	;	veri tabanından bakar.
$T_{yan\text{-}toprak}$	;	veri tabanından bakar, f ($T_{dış}$).
T_{toprak}	;	veri tabanından bakar, f ($T_{dış}$).
$T_{çatı\text{-}arası}$	;	veri tabanından bakar, f ($T_{dış}$).

9. Isı kaybı raporlarının alınması;

9.a. Kat ısı kaybı raporları.

"Katlar" menüsünden "Isı kaybı raporu" satırını seçiniz. Program bir menü ile raporunu almak istediğiniz katı sorar. Kat seçildikten sonra raporu alınmak istenen ilk ve son odaların numaraları sorulur. Hazırlanan rapor ekrana kontrol amacıyla getirilir. Rapordan ESC ile çıkılırsa rapor yazıcıya gönderilmez, F10 ile çıkıldığında rapor yazıcıya gönderilmek isteniyor demektir. Program yazıcının hazır olup olmadığı konusunda uyarı verir.

9.b. Bina özet ısı kaybı raporu.

"Bina" menüsünden "Özet rapor" satırı seçilerek alınır, binanın tüm odalarının kod, ad, sıcaklık ve toplam ısı ihtiyacı bilgileri ile bina toplam ısı ihtiyacı bilgilerini içerir.

9.c. Bina genel raporu.

"Bina" menüsünden "Genel rapor" satırı seçilerek alınır, Bina tanımlayıcı bilgileri, malzeme listesi, katman listesi ve yüzey detayları bilgilerini içerir.

EK - 8
SONLU ELEMANLAR MODELLEMESİ (ANSYS 5.0) ÇALIŞMALARI

Biyolojik, jeolojik yada mekanik olsun doğada bulunan her olay fizik kurallarının yardımıyla matematiksel, diferansiyel, ya da integral denklemleriyle ifade edilebilir. Mekanik, ısıl, ya da aerodinamik yüklere maruz kalan bir basınçlı kaptaki gerilme dağılımının hesaplanması, denizde ya da atmosferde kirlilik konsantrasyonunun bulunması, kasırga ve rüzgarların mekaniğini anlamak ve önceden tahmin etmek için hava hareketlerinin simulasyonu birçok önemli pratik problemlerden sadece birkaçıdır. Bu tip problemlerle ilgili denklemlerin çıkarılması zor olmamakla beraber, bunların tam sonuç veren metodlarla çözülmesi oldukça zordur. Bu gibi durumlarda tam sonuca oldukça yakın değerler veren alternatif metodlar kullanılabilecek bir başka seçenektir. Bunlar arasında sonlu-farklılık metodu, Ritz ve Galerkin gibi varyasyonel metodlar ve sonlu-elemanlar metodu sayılabilir.

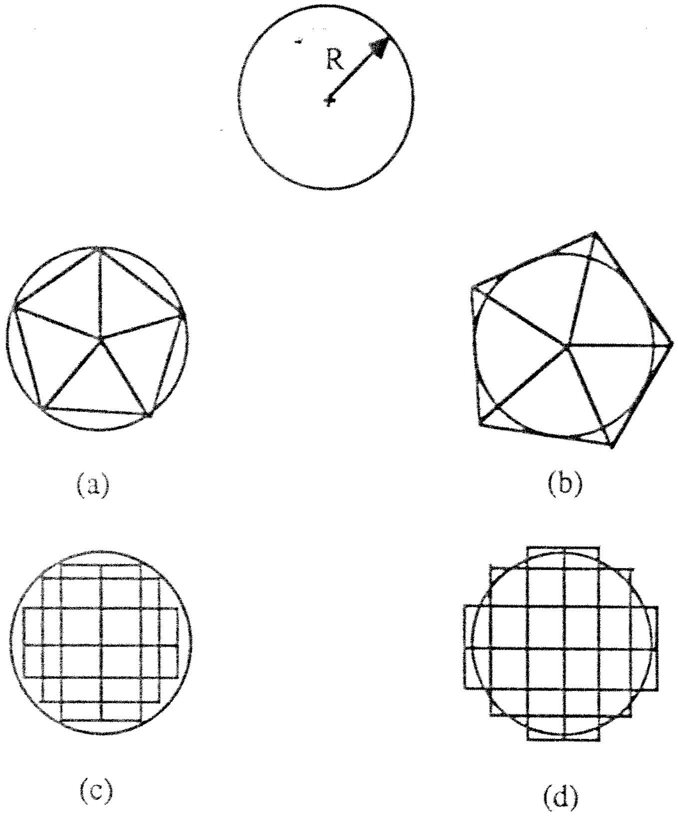
Diferansiyel denklemlerin sonlu-farklılık metoduyla çözümünde problem ayrı ayrı alanlara bölünerek herbirisi için ayrı ayrı denklemler elde edilir. Sınır şartları uygulanarak herbir denklem kullanılarak herbir alan için çözüm bulunur. Sonlu-farklılık metodu kullanımda kolay olmakla beraber, pek çok dezavantaja sahiptir. Bunlardan en önemlileri verdiği yaklaşık çözümün tam sonuca uzak olması, doğrusal olmayan sınırlara sınır şartlarının uygulanmasındaki zorluk, geometrik olarak karmaşık olan problemlerin çözümündeki zorluk, problemin düzgün ve kare olmayan alanlara bölünememesi gibi.

Diferansiyel denklemlerin varyasyonel metoduyla çözümünde yaklaşık çözümler veren ayrı ayrı fonksiyonların toplamıyla yaklaşık çözüm elde edilir. Bu metodun kullanımında da en büyük dezavantaj, genel ve karmaşık problemler için yaklaşık çözümler veren fonksiyonların oluşturulmasındaki zorluktur.

Yaklaşık çözümler veren fonksiyonların çıkarılmasında sistematik bir yol izlediği için sonlu-elemanlar metodu varyasyonel metodundaki zorlukları giderir. Sonlu-elemanlar metodunun diğer metodlara olan iki önemli üstünlüğü bu metodun tercih edilme nedenleridir: birincisi, geometrik olarak karmaşık alana sahip olan problemin basit geometriye sahip sonlu-elemanlar diye adlandırılan alt alanlara bölünebilmesi, ikincisi ise herbir sonlu eleman için yaklaşık çözümler veren fonksiyonların sürekli fonksiyon olacak şekilde matematiksel polinomların kombinasyonu olarak kolayca tanımlanabilmesi. Sonlu elemanlar metodu hakkında genel bir fikir edinebilmek için basit bir örnek verelim. Örneğin yarıçapı R olan bir dairenin alanını sonlu-elemanlar metoduyla aşağıdaki adımları takip ederek bulabiliriz.

Programdan çıkış için "Çıkış" menüsü seçilir, program çıkmadan önce binaya ait tüm bilgilerinizi diske kaydeder.

Aynı binayı tekrar yüklemek için programa girdiğinizde "Bina" menüsünden "Bina yükle" satırını seçiniz.



Şekil 8-1. Dairenin sonlu-elemanlara bölünmesi.

1. Problemin sonlu-elemanlara bölünmesi : Şekil 8-1 'de gösterildiği gibi (ya da başka bir tarzda) daire alt alanlara (sonlu-elemanlar) bölünür.
2. Eleman denklemleri : herbir elemanın denklemi çıkarılır. Bu problemde elemanın alanını veren fonksiyon o elemanın denklemidir.
3. Çözüm : Bu aşamada eleman denklemleri toplanıp sonuca ulaşılır. Örneğin herbir elemanın denklemini (alanını) a_e ile gösterirsek, toplam alan $A = \sum_{e=1}^{n} a_e$ olur.

   Burada n eleman sayısını gösterir.
4. Hata hesaplanması : 3. adımda sonlu-elemanlar metoduyla dairenin alanını (A) bulmuştuk. Ancak tam sonuç yani dairenin gerçek alanı πR^2 'dir. Bu iki sonuç arasındaki fark (πR^2 - A) hata payını verir.

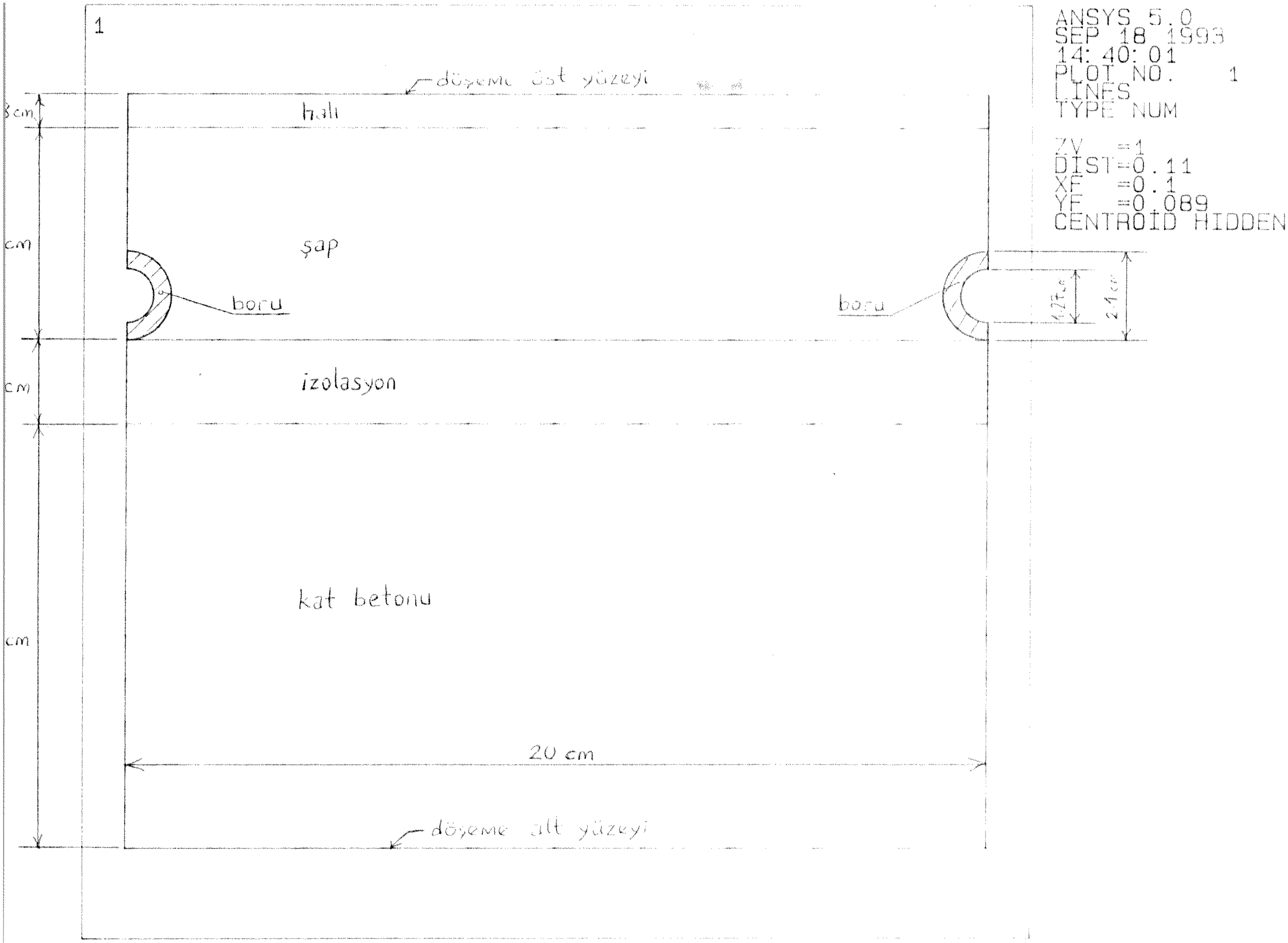
Şu herkesçe bilinen bir gerçek ki, problemler ve bunlarla ilgili eleman denklemleri yukarıdaki örnekten çok daha karmaşık ve zordur. Problemi daha fazla alt alana (sonlu-eleman) bölerek, kare ve üçgen eleman tiplerini bir arada kullanarak hata payını azaltmak ve tam sonuca oldukça yaklaşmak mümkündür.

Yukarıda anlatılan sonlu-elemanlar metoduyla ilgili temel kavramlardan da anlaşılacağı üzere, bir problemin çözümünde bu metodu kullanacak kişinin iki şeyi bilmesi gerekiyor: birincisi, problemi bir varyasyonel problem olarak nasıl formüle edeceğini, ikincisi, bu varyasyonel problemle ilgili matematiksel denklem sistemini nasıl tanımlayacağını bilmesi gerekiyor. Herhangi bir problemin sonlu-elemanlar modelini geliştirmek için yukarıda behsedilen her iki adımda büyük önem taşıyor. İşte bu aşamada ANSYS yazılım paketi bizi bu sorunlardan kurtarıyor. Problemleri sonlu-elemanlar metoduyla çözen ANSYS programı, bu metodla ilgili yukarıda bahsedilen ve diğer tüm işlemleri kendisi çözümleyip kullanıcıya büyük kolaylıklar sağlıyor. Kullanıcı bu program yardımıyla, ekranda (grafik olarak) problemine ait geometriyi ve sınır şartlarını giriyor ve çözümü ANSYS'e bırakıyor. Bu çalışmada statik, dinamik, ısı, akışkanlar mekaniği ve manyetik analize kadar pek çok alanda sonlu-elemanlar metoduyla problemlerin çözümünde kullanılan ANSYS 5.0 program paketi yardımıyla döşemeden ısıtılan bir odanın analizi yapıldı. Bu amaçla aşağıda anlatıldığı gibi döşeme ve oda ayrı ayrı modellenip gerekli sınır şartları için çözdürüldü.

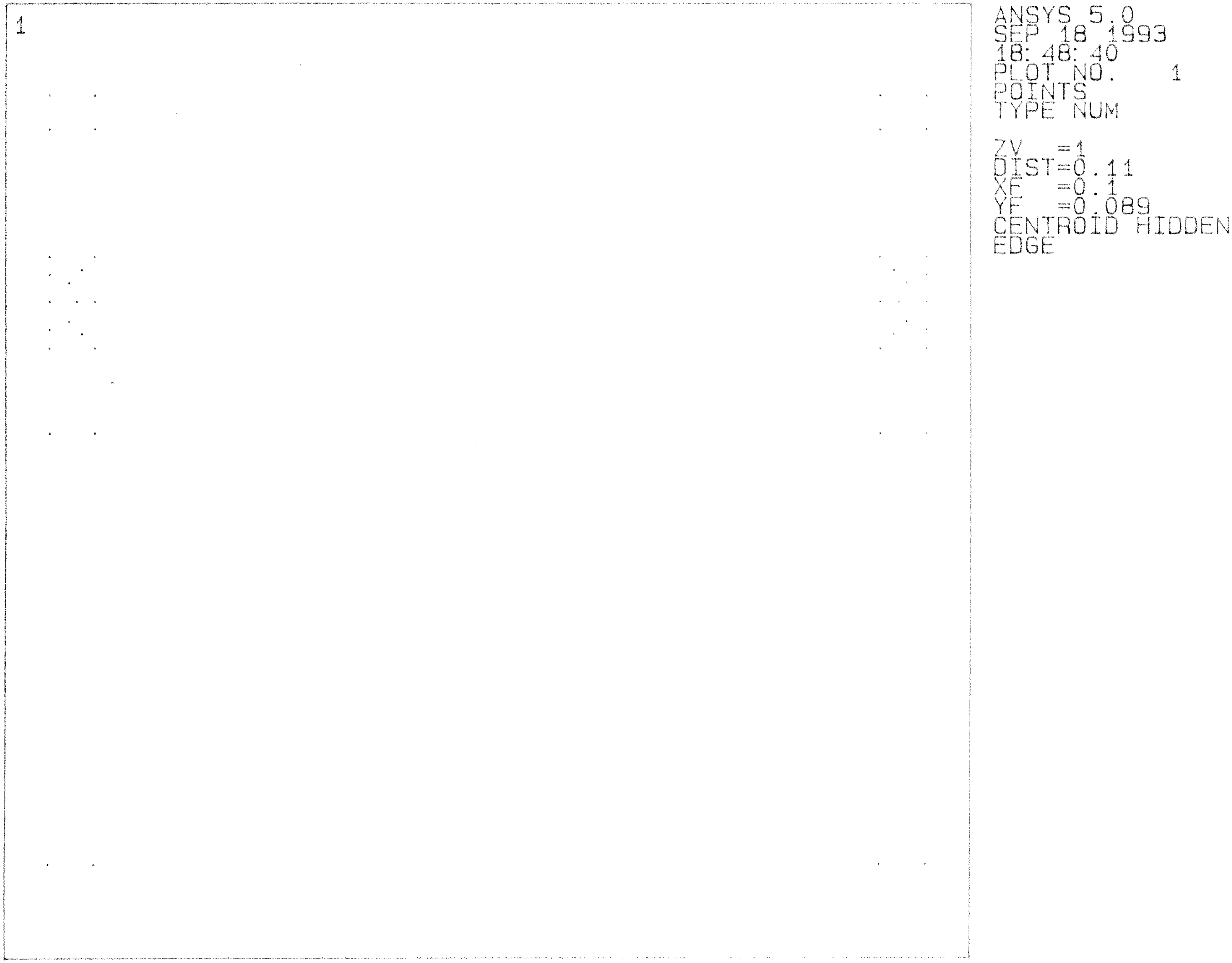
a - Döşeme Modeli

İncelenen odanın yer döşemesinden alınan simetrik bir kesit Şekil 8-2 'de gösterildiği gibi modellenip beş farklı boru aralığı için ANSYS 5.0 programıyla çözdürüldü. Bu çalışmalarda döşeme içindeki sıcaklık dağılımı, döşeme üst ve alt yüzeylerindeki sıcaklık profili, döşeme üst yüzeyinden ısıtılan mahale ve döşeme alt yüzeyinden alt kata olan ısı kayıpları hesaplanıp bunların grafik çıktıları rapora ilave edildi.

Döşeme ANSYS'de şu şekilde modellendi: Şekil 8-3 'deki gibi önce 1'den 24'e kadar noktalar oluşturuldu. Sonra bu noktalar arası çizgilerle birleştirildi (Şekil 8-4). Daha sonra 1'den 20'ye kadar alanlar tanımlandı (Şekil 8-5). Bu alanlara aşağıda anlatılan malzeme özellikleri tanımlandıktan sonra tüm model alt alanlara (sonlu-elemanlara) bölündü (Şekil 8-6). Şekil 8-7 'de gösterilen sınır şartları tanımlandıktan sonra problem ANSYS 5.0 programıyla çözdürüldü. Problemin çözümüyle ilgili çıktılar Şekil 8-8, 8-9, 8-10 'da gösterilmiştir. Bütün bu ANSYS çıktıları 20 cm boru aralığı ve 18 °C oda sıcaklığı için geçerli olup, tüm boru aralıkları ve farklı oda sıcaklıkları için yapılan analizlerin sonuçları Çizelge 8-1 'de verilmiştir.



Şekil 8-2. Döşeme modeli.


1
ANSYS 5.0
SEP 18 1993
18:48:40
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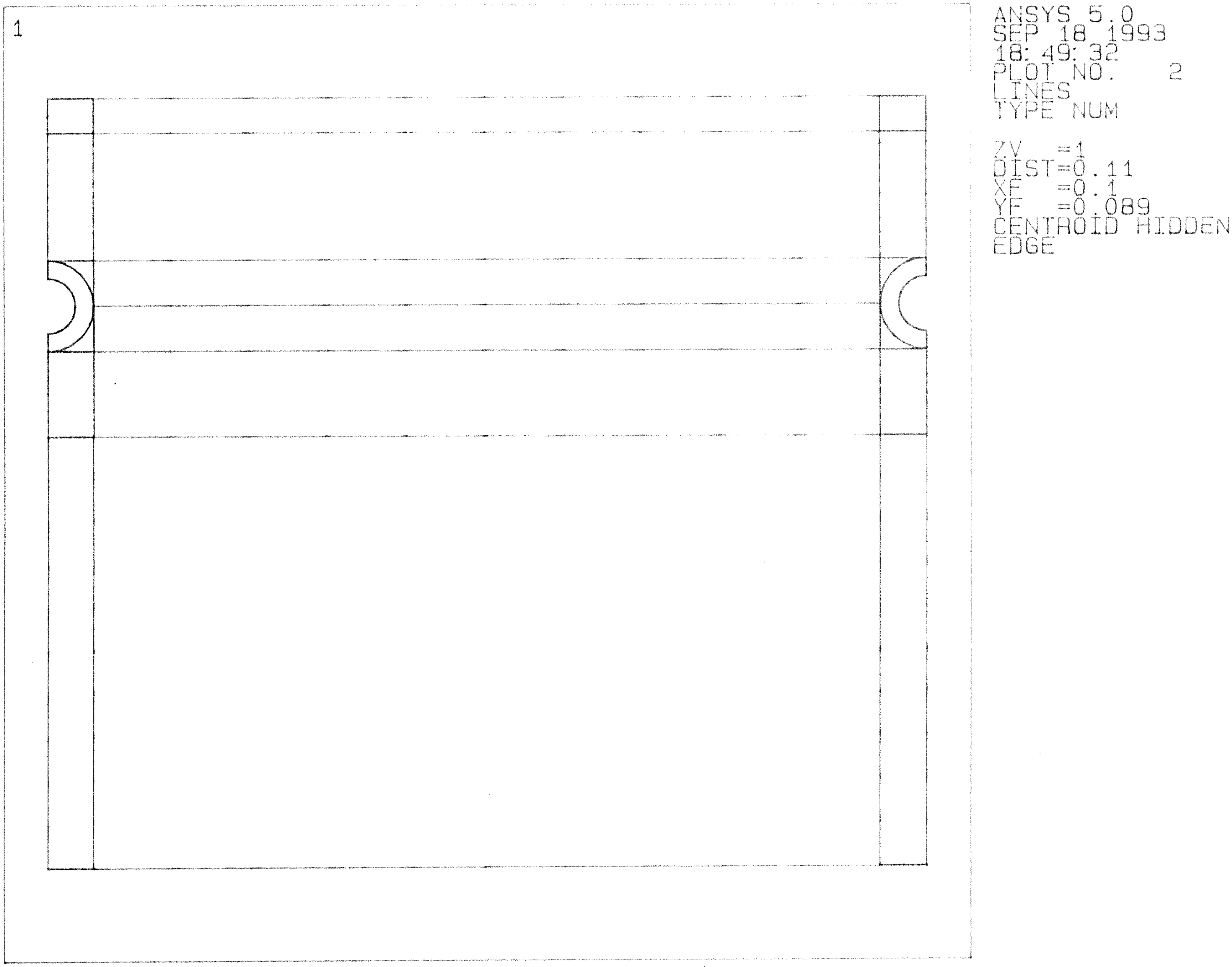
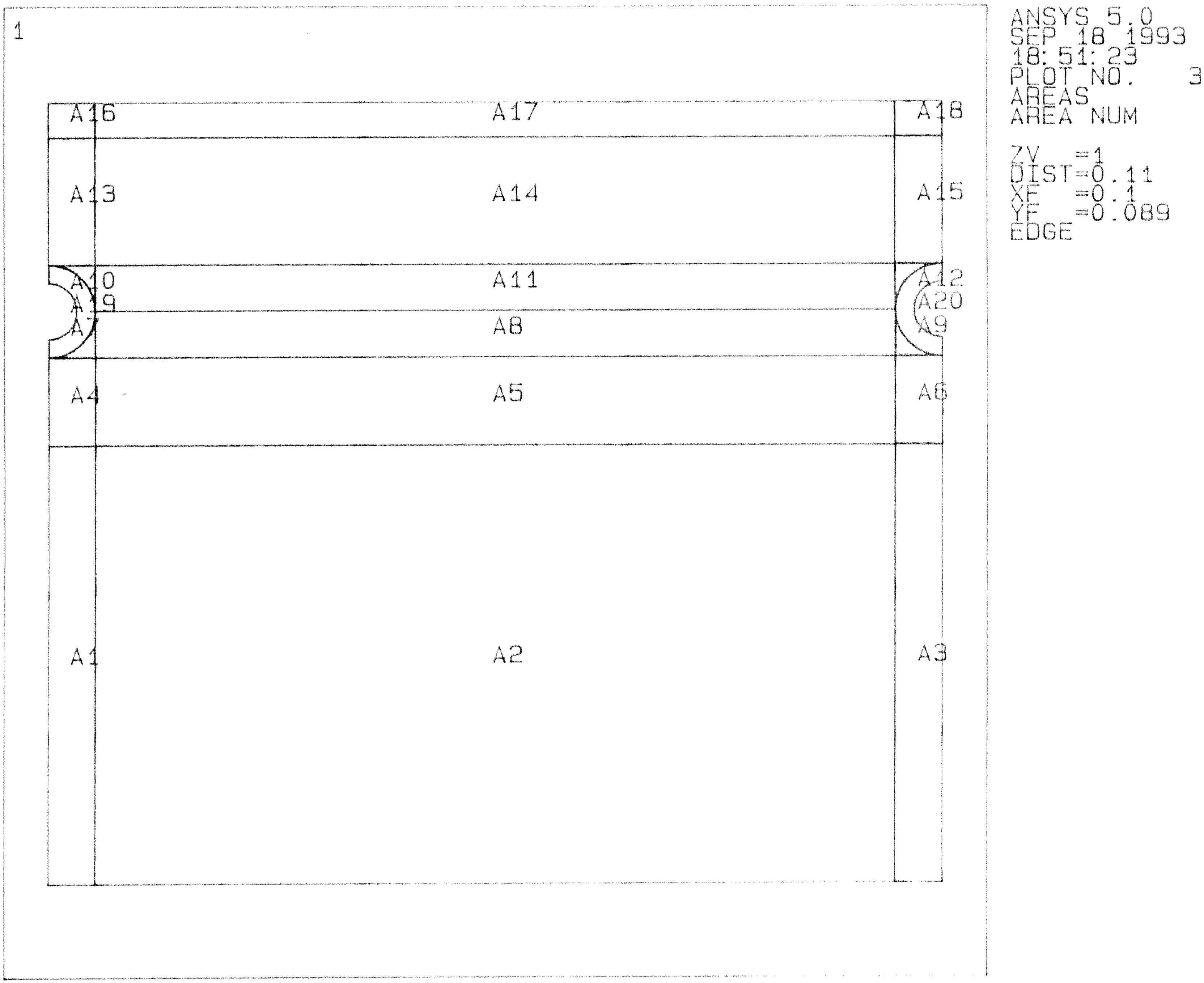
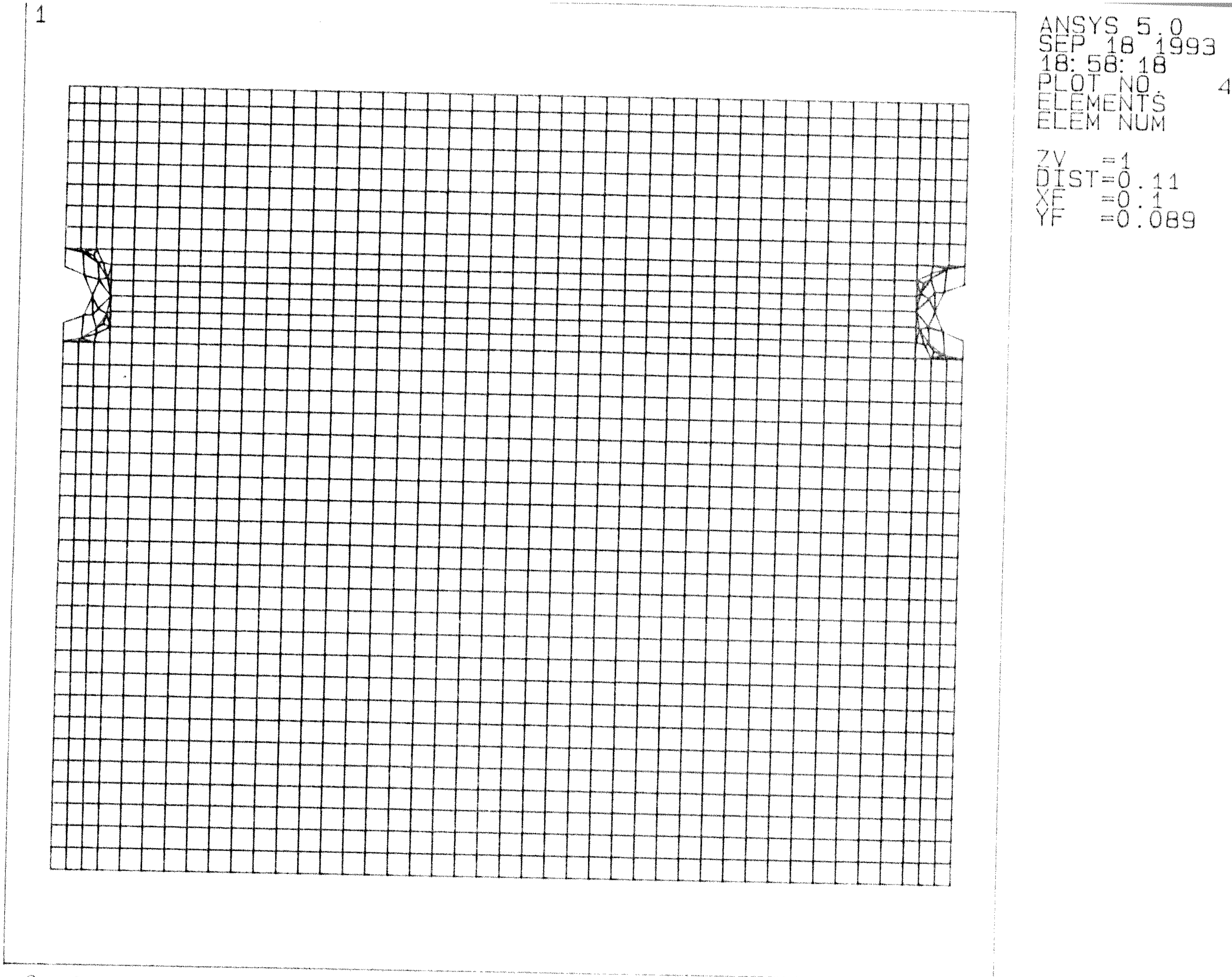
YF =0.089
CENTROID HIDDEN
EDGE



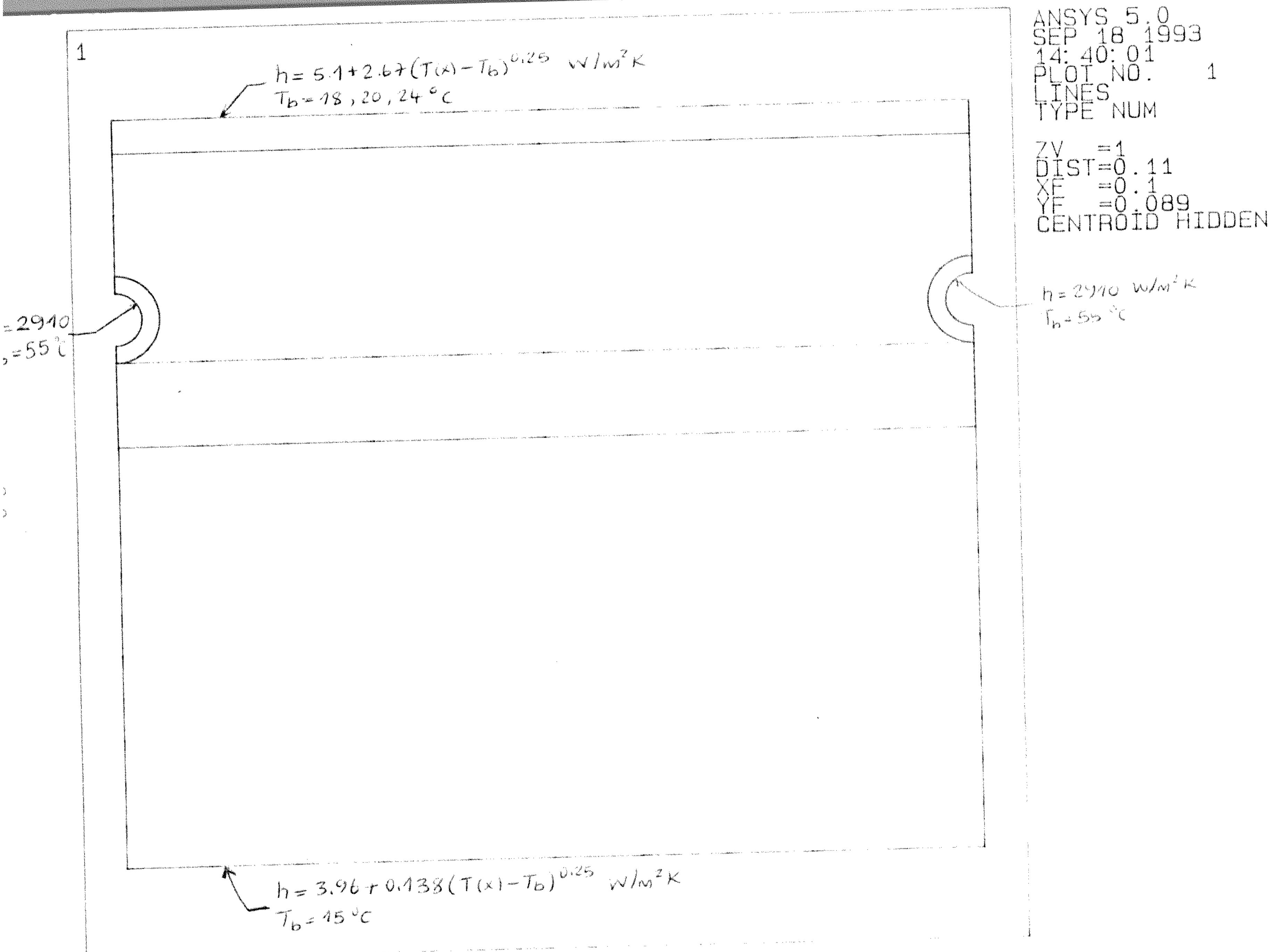
Şekil 8-4. Döşemenin modellenmesinde ikinci aşama (çizgiler).



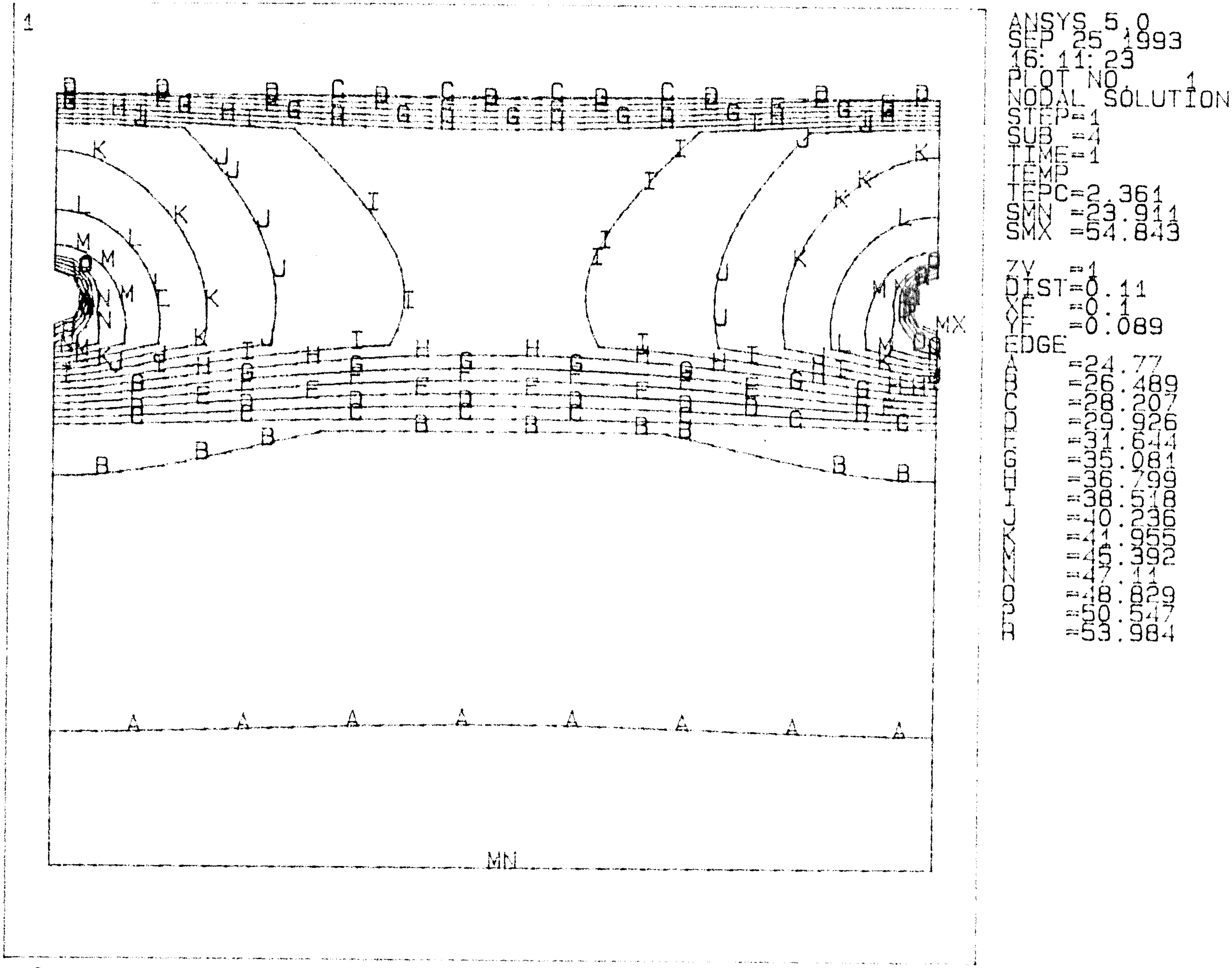
Şekil 8-5. Alanların tanımlanması.



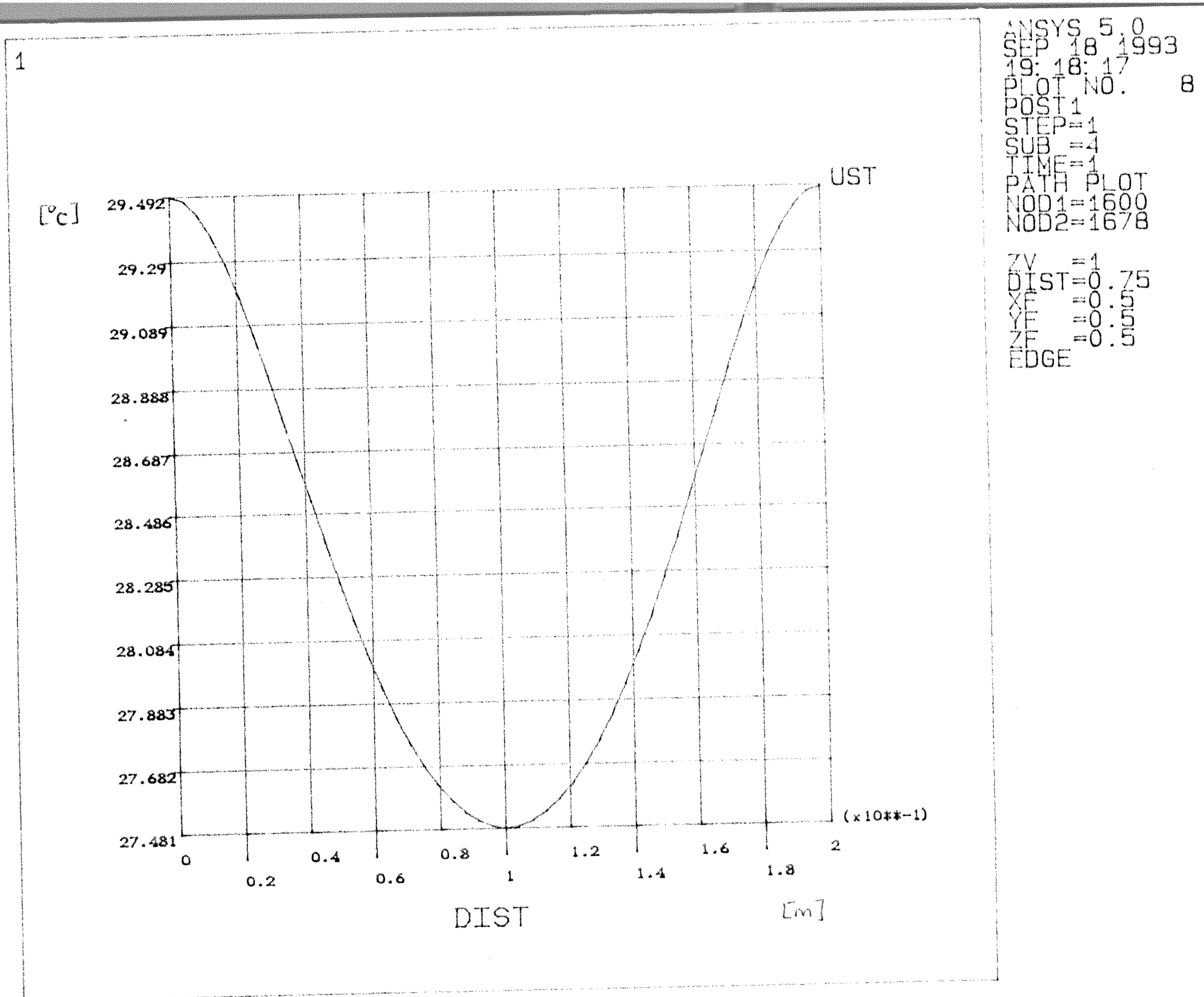
Şekil 8-6. Döşeme modelinin sonlu-elemanlara bölünmesi.



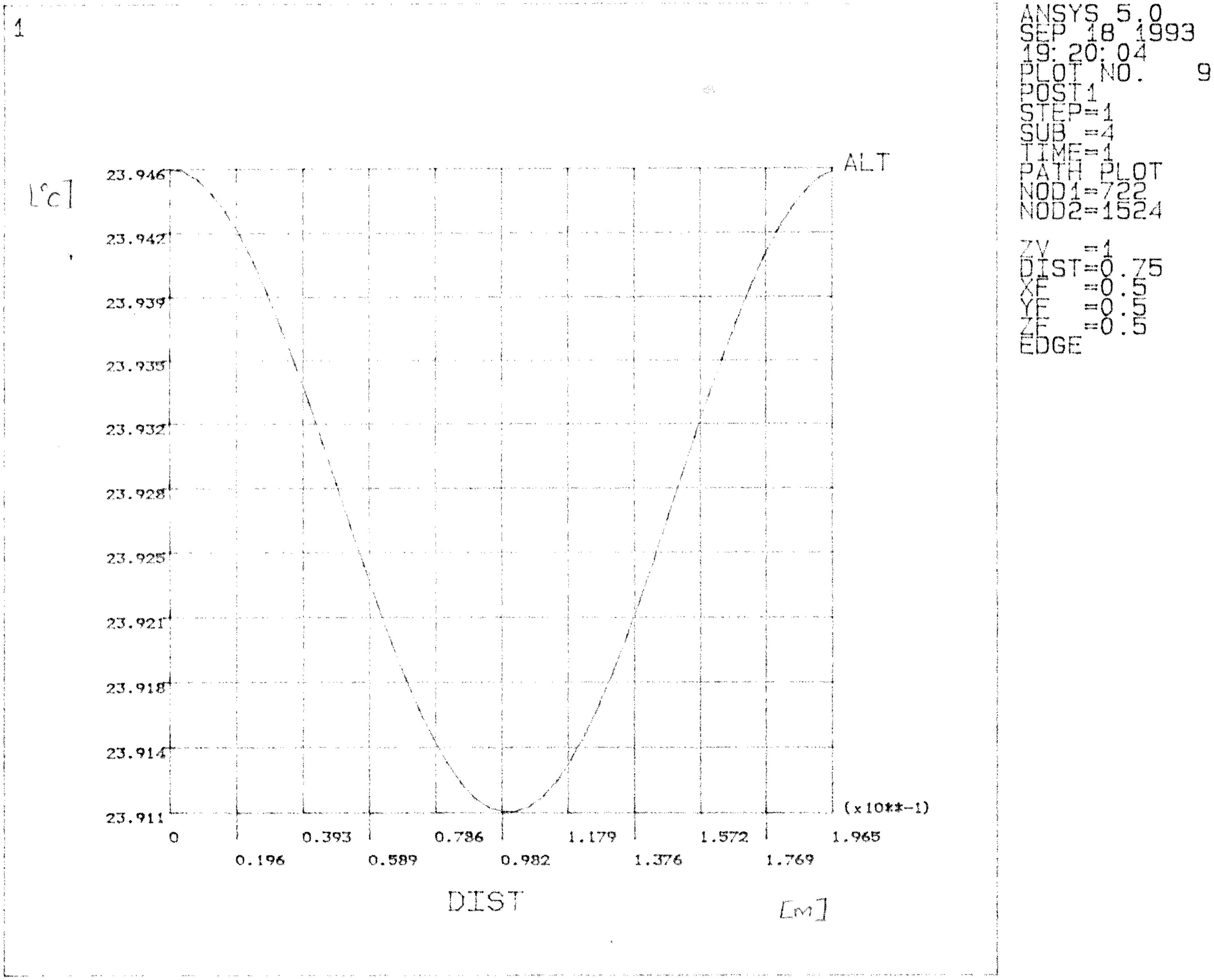
Şekil 8-7. Model üzerinde sınır şartlarının tanımlanması.



Şekil 8-8. Döşeme içindeki sıcaklık profilleri.



Şekil 8-9. Döşeme üst yüzeyinde sıcaklık profili.



Şekil 8-10. Döşeme alt yüzeyinde sıcaklık profili.

Çizelge 8-1. Farklı Döşeme Modelleri İçin Alınan ANSYS Sonuçları.

Oda Sıcaklığı / Boru Aralığı	q üst [W/m^2] üst oda 18 °C	q alt [W/m^2] alt oda 15 °C	q üst [W/m^2] üst oda 20 °C	q alt [W/m^2] alt oda 15 °C	q üst [W/m^2] üst oda 24 °C	q alt [W/m^2] alt oda 15 °C
10 cm	135.86	47.38	-	-	-	-
20 cm	103.85	37.55	98.15	38.21	-	-
25 cm	92.68	33.5	-	-	-	-
30 cm	83.12	30.1	-	-	65.43	32.9
35 cm	75.5	27.23	-	-	-	-

Bütün bu analizlerde kullanılan;

i) Malzeme özellikleri;

Halı için ısı iletim katsayısı	=	0.071	W/mK
Şap için ısı iletim katsayısı	=	1.5	W/mK
İzolasyon için ısı iletim katsayısı	=	0.05	W/mK
Kat betonu için ısı iletim katsayısı	=	1.4	W/mK
Boru için ısı iletim katsayısı	=	0.29	W/mK

ii) Sınır şartları;

Döşeme üst yüzeyinde : Konveksiyon katsayısı = $5.1 + 2.67\ (T_p(x) - T_b)^{0.25}$ W/m^{2}K
Ortam sıcaklığı = T_b = 18 °C, 20 °C, 24 °C

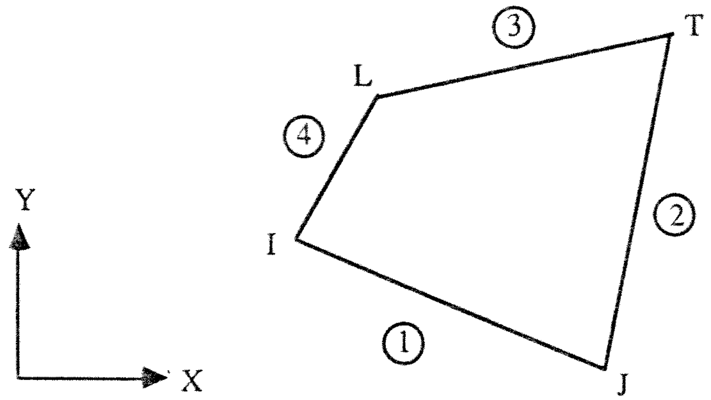
Döşeme alt yüzeyinde : Konveksiyon katsayısı = $3.96+0.138\ (T_p(x)-T_b)^{0.25}$ W/m^{2}K
Ortam sıcaklığı = T_b = 15 °C

Boru iç yüzeyinde : Konveksiyon katsayısı = 2910 W/m^{2}K
Su sıcaklığı = T_b = 55 °C

iii) Eleman tipleri;

Şekil 8-5'de gösterilen tüm alanlar PLANE55 tipi elemanla alt alanlara (sonlu-elemanlara) bölündü. Problem kondüksiyon ısı transferi olması sebebiyle bu amaca uygun olarak 2-boyutlu, 4 düğüm noktalı kondüksiyon eleman olan PLANE55 'in kullanılması tercih edildi. PLANE55 elemanı aşağıdaki özelliklere sahiptir;

PLANE55 elemanı 4 düğüm noktasına sahip olup, her bir düğüm noktasının bir serbestlik derecesi (sıcaklık) vardır. Eleman sıfır ya da negatif alana sahip olmamalı ve Şekil 8-11 'de gösterildiği gibi X-Y düzlemi içinde bulunmalı.



Şekil 8-11. PLANE55 2-Boyutlu Kondüksiyon Elemanı.

b - Döşemeden Isıtılan Mahal Modeli

Şekil 8-12 'de gösterildiği gibi 4x2.9 metre iç boyutlarında, sol tarafta bir pencere ve sağ tarafta bir kapıya sahip olan 2-boyutlu bir odanın ANSYS 5.0 programıyla analizi yapıldı. Oda analizi aşağıda anlatıldığı gibi iki aşamada gerçekleştirildi;

1) Radyasyon Analizi : Oda, içinde hava yokmuş gibi modellenip, döşeme yüzeyinden duvarlara, pencere ve kapıya olan radyasyon ısı transferi, duvarlar, pencere ve kapıdaki sıcaklık dağılımı, döşemeden odaya giren ısı yükü hesaplanıp bunların grafik çıktıları alındı.

Oda modellemesi ANSYS 5.0 'da şu şekilde yapıldı: Şekil 8-13 'deki gibi noktalar oluşturuldu. Sonra bu noktalar arası çizgilerle birleştirildi (Şekil 8-14). Daha sonra Şekil 8-15 'deki alanlar tanımlandı ve bu alanlara aşağıda anlatılan malzeme özellikleri tanımlandıktan sonra tüm model alt alanlara (sonlu-elemanlara) bölündü (Şekil 8-16). Şekil 8-17 'deki gibi sınır şartları tanımlandıktan sonra problem ANSYS 5.0 programıyla çözdürüldü. Odaya döşeme yüzeyinden giren ısı (radyasyon'a göre) 28.2 W/m^2 olup model üzerindeki sıcaklık dağılımı, cam ve kapı üzerindeki sıcaklık dağılımları Şekil 8-18, 8-19 ve 8-20 'de gösterilmiştir. Radyasyon analizinde kullanılan;

i) Malzeme özellikleri;

Dış duvarlar için ısı iletim katsayısı	= 0.7 W/mK
Kat arası beton için ısı iletim katsayısı	= 1.4 W/mK
İç duvarlar için ısı iletim katsayısı	= 1.5 W/mK
Cam için ısı iletim katsayısı	= 3 W/mK
Kapı için ısı iletim katsayısı	= 1.2 W/mK

ii) Sınır şartları;

Oda döşeme yüzeyinde	: Sabit sıcaklık = 26 °C
Dış duvarlar ve pencere	: Konveksiyon katsayısı = 20 W/m^{2}K Ortam sıcaklığı = - 12 °C
Üst kat döşeme yüzeyi	: Konveksiyon katsayısı = 5 W/m^{2}K Ortam sıcaklığı = 18 °C
İç oda duvar yüzeyi	: Konveksiyon katsayısı = 5 W/m^{2}K Ortam sıcaklığı = 15 °C
İç oda kapı yüzeyi	: Konveksiyon katsayısı = 5 W/m^{2}K Ortam sıcaklığı = 15 °C

iii) Eleman tipleri;

"Döşeme Modeli" konusunda anlatılan nedenlerden dolayı 2-boyutlu, 4 düğüm noktalı PLANE55 kondüksiyon elemanı duvarlarda, cam ve kapıda kullanılmıştır.

2) Konveksiyon Analizi : Radyasyon dahil edilmeyip oda ve içindeki hava modellenerek problemin ısıl ve akışkan analizi yapıldı. Bu analizde oda içindeki havanın doğal konveksiyonu çözdürülerek sıcaklık ve basınç dağılımları, oda içinde hava hareketleri ve döşemeden odaya giren ısı yükü çözdürülüp bunların grafik çıktıları alındı. Bilindiği gibi ANSYS genel amaçlı bir sonlu-elemanlar programı olup çok geniş bir alana hitap etmektedir. Ancak yukarıda adı geçen oda analizi ısıl ve akışkanlar dinamiği analizlerini birlikte ihtiva etmektedir. Bu amaca uygun olarak özellikle akışkanlar mekaniğinde etkili bir solver olan FLOTRAN sabrotini son zamanlarda ANSYS 5.0 'a entegre edilmiştir. Oda ANSYS 5.0 'da modellenip gerekli sınır şartları tanımlandıktan sonra ANSYS sabrotinleriyle değil FLOTRAN çözücüsüyle çözdürülmüş ve sonuçlar, grafik çıktılar tekrar ANSYS 5.0 'dan alınmıştır. Çünkü FLOTRAN yardımcı programı bu tip problemlerin modellenmesinde büyük kolaylıklar sağlıyor (daha az sonlu-eleman kullanımı gibi) ve problemin çok kısa zamanda (ANSYS 5.0 çözücü programlarına göre) çözdürülmesini sağlıyor. Oda modellemesi ANSYS 5.0 'da aynen "Radyasyon Analizi" konusunda anlatıldığı gibi yapıldı. Aradaki tek fark Şekil 8-21 ve 8-22 'de gösterildiği gibi oda içinde de alanların ve daha sonra sonlu-elemanların tanımlanmış olması. Şekil 8-17 'deki sınır şartlarına ilaveten duvarların, pencere ve kapının iç yüzeylerine sıfır hız (hava için) şartı tanımlandıktan sonra problem ANSYS 5.0 programı içinde ANSYS sabrotiniyle değil FLOTRAN sabrotiniyle çözdürüldü. Odaya döşeme yüzeyinden giren ısı (konveksiyon'a göre) 75.6 W/m^2 olup model üzerinde sıcaklık dağılımı, oda içinde basınç ve hız dağılımı ile hava hareketleri Şekil 8-23, 8-24, 8-25 ve 8-26 'da gösterilmiştir. Konveksiyon analizinde kullanılan;

i) Malzeme özellikleri;

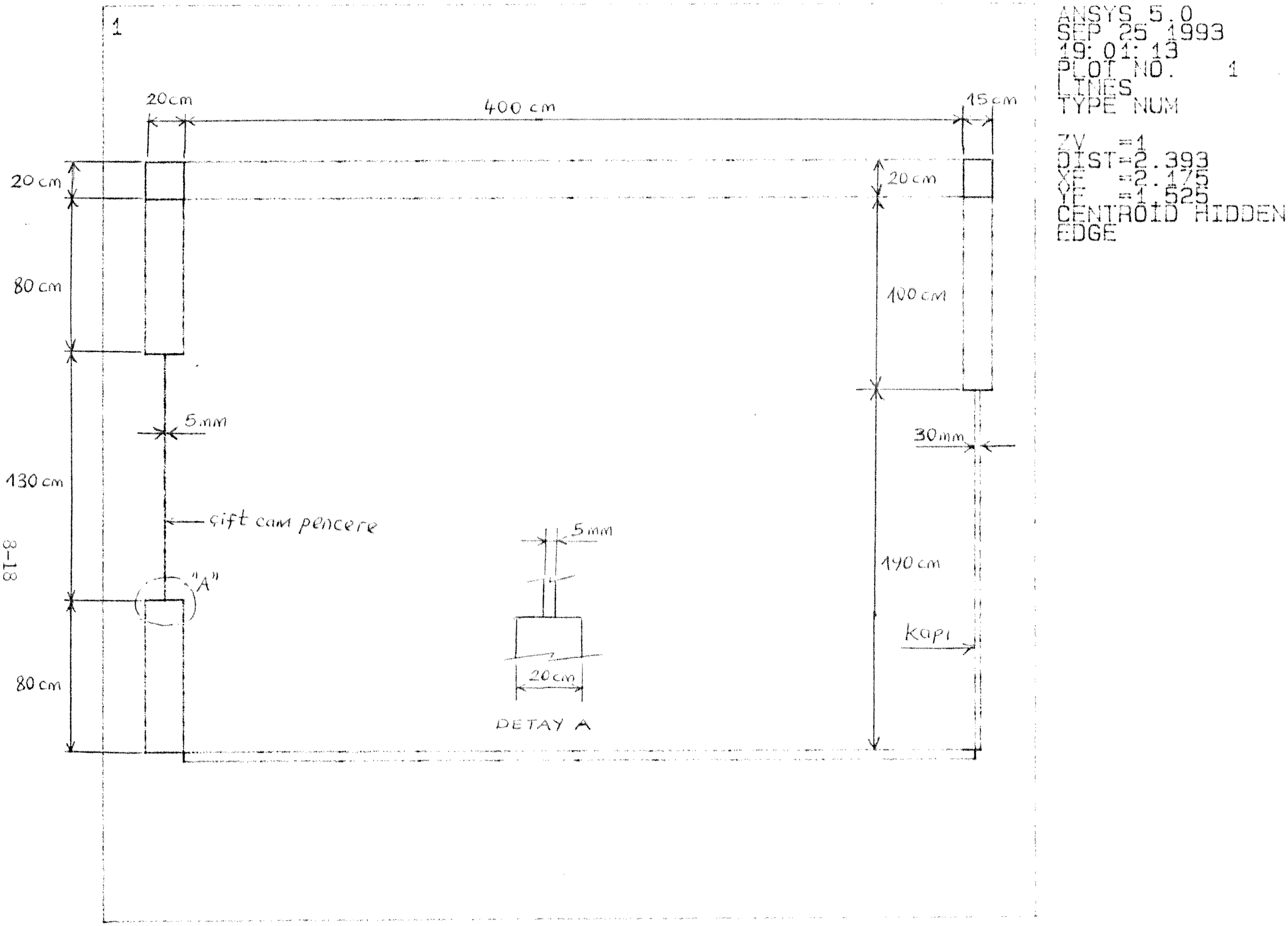
Dış duvarlar, kat arası beton, iç duvarlar, cam ve kapı malzeme özellikleri Radyasyon analizinde kullanılanlarla aynıdır. Bunlara ilaveten oda içindeki havanın sıcaklığa bağlı özellikleri FLOTRAN sabrotini içinde metrik sistem (MKS) tablosundan otomatik olarak okunmaktadır.

ii) Sınır şartları;

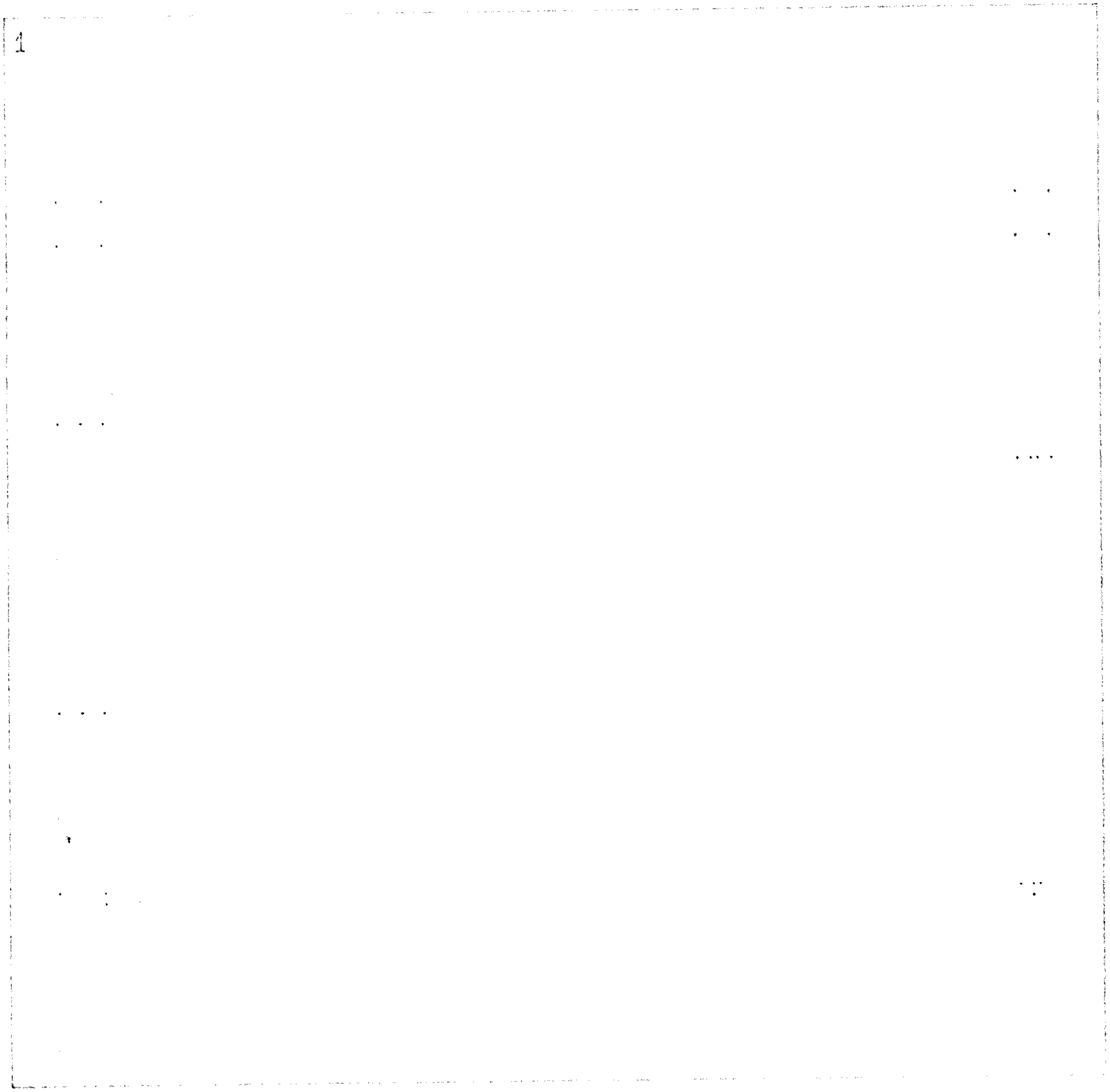
Radyasyon analizindeki sınır şartları burada da aynen geçerlidir. Bunlara ilaveten duvarların, pencere ve kapının odaya bakan iç yüzeylerine hava için sıfır hız sınır şartı verilmiştir.

iii) Eleman tipleri;

Döşeme modelinde ve radyasyon analizinde kullanılan PLANE55 elemanı burada da duvarlar, pencere ve kapı için kullanılmıştır. Buna ilaveten oda içindeki havanın modellenmesinde de bu eleman kullanılmış olup problemin çözümü sırasında FLOTRAN otomatik olarak düğüm noktalarındaki sıcaklık serbestlik derecesine basınç, hız, enerji gibi serbestlik derecelerini de ekleyerek oda içinde doğal konveksiyonu ve hava simulasyonunu hesaplamıştır.



Şekil 8-12. Döşemeden ısıtılan mahal modeli.



ANSYS 5.0
SEP 25 1993
19:01:50
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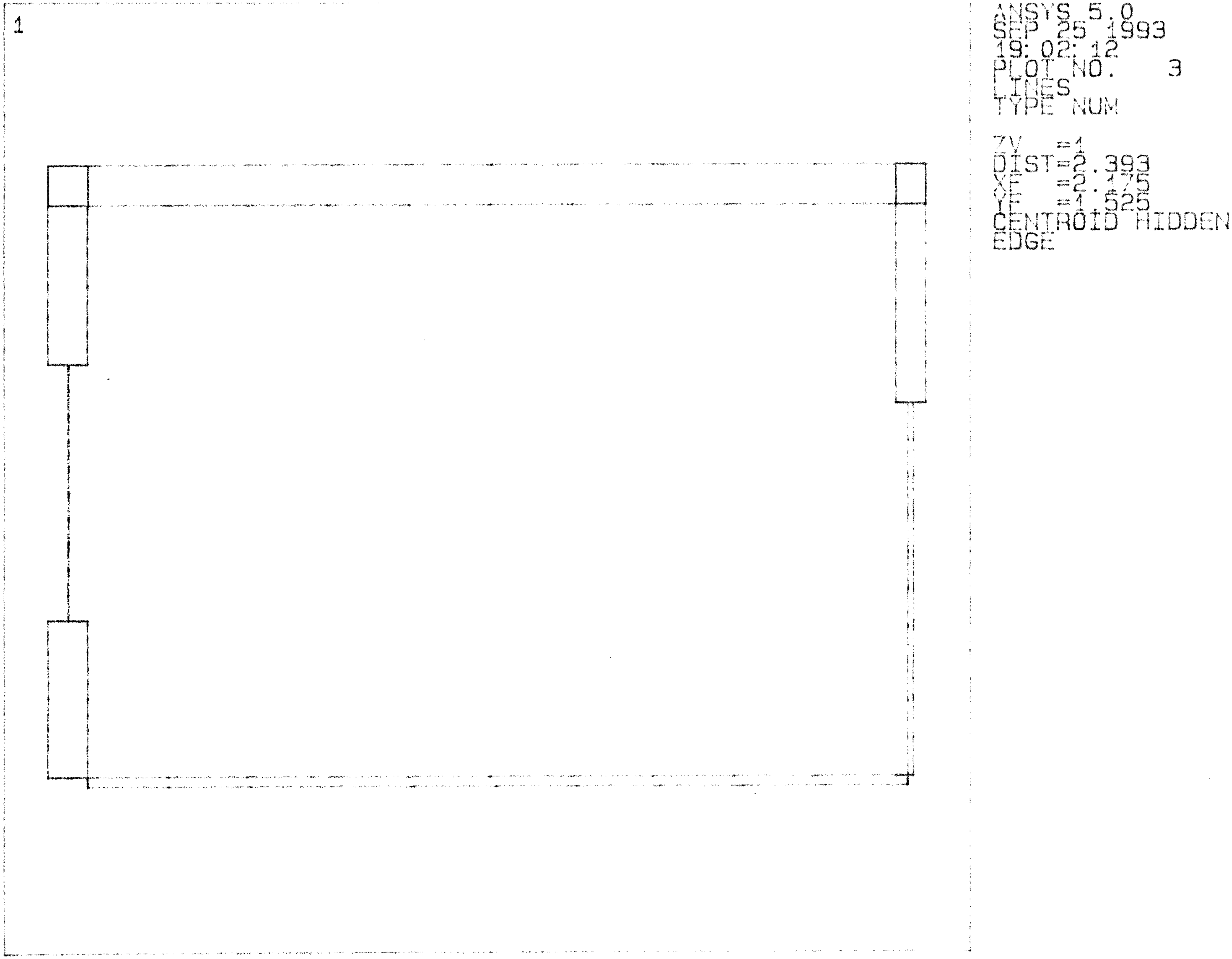
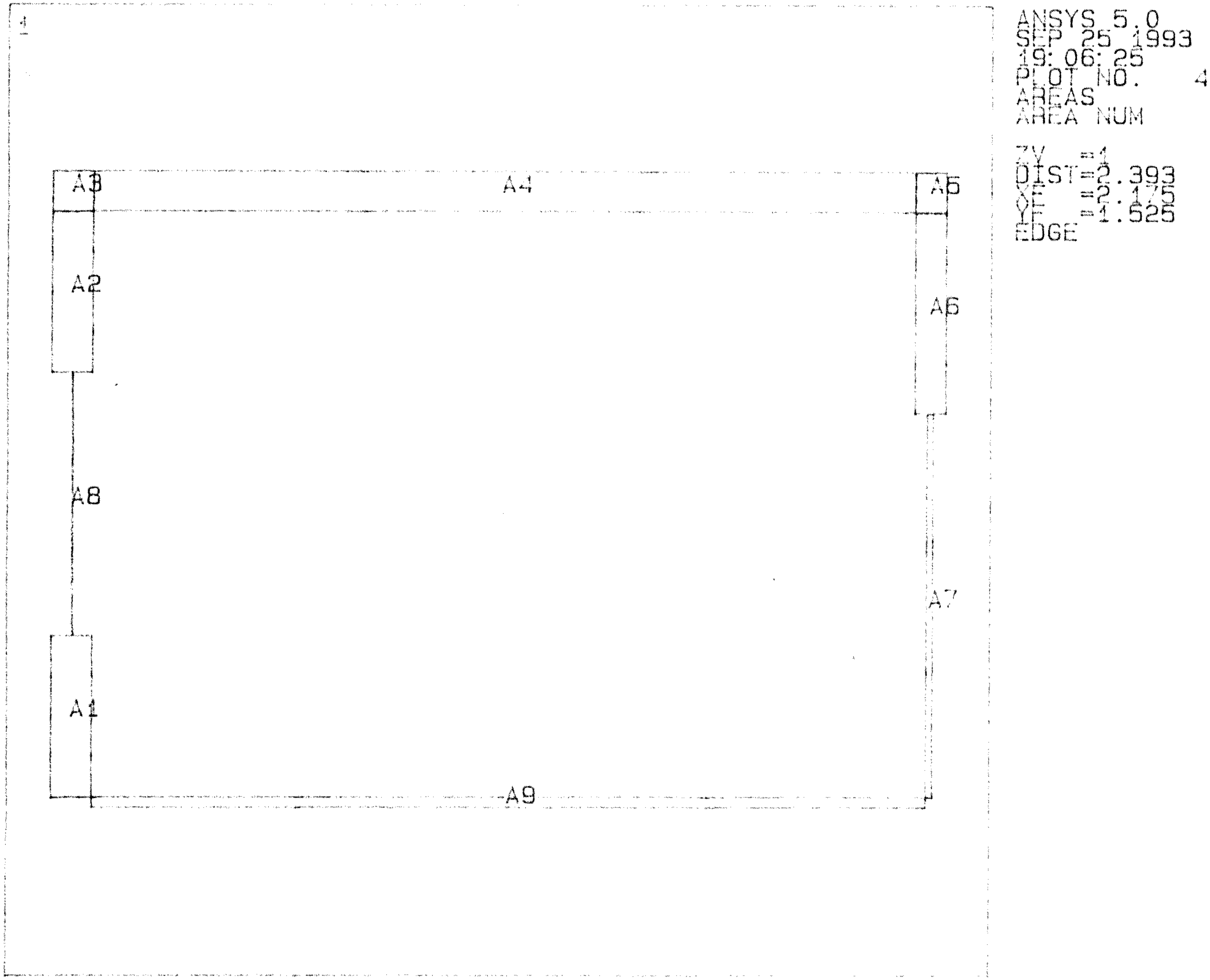
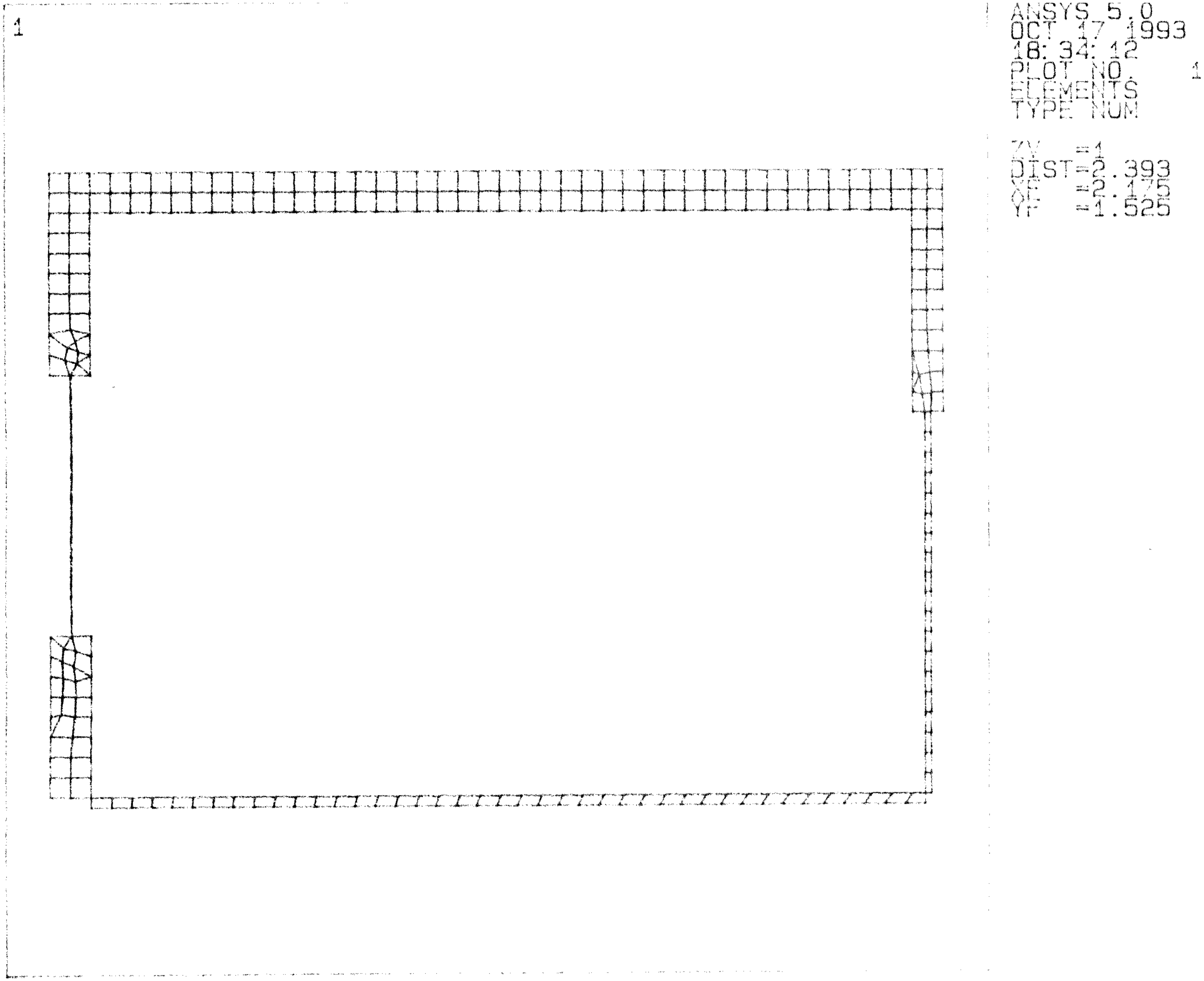
YF =1.525
CENTROID HIDDEN
EDGE



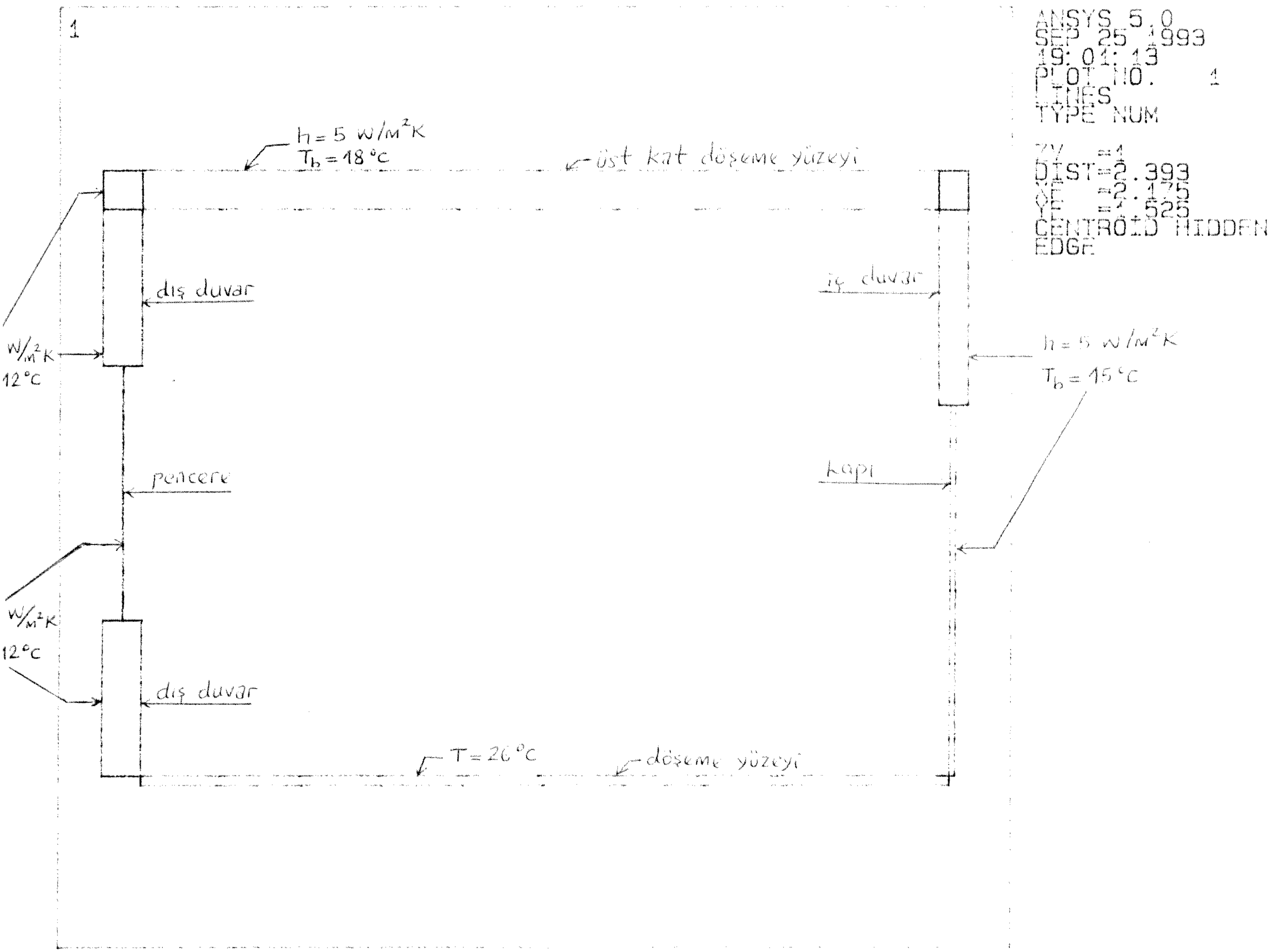
Şekil 8-14. Odanın modellenmesinde ikinci aşama (çizgiler).



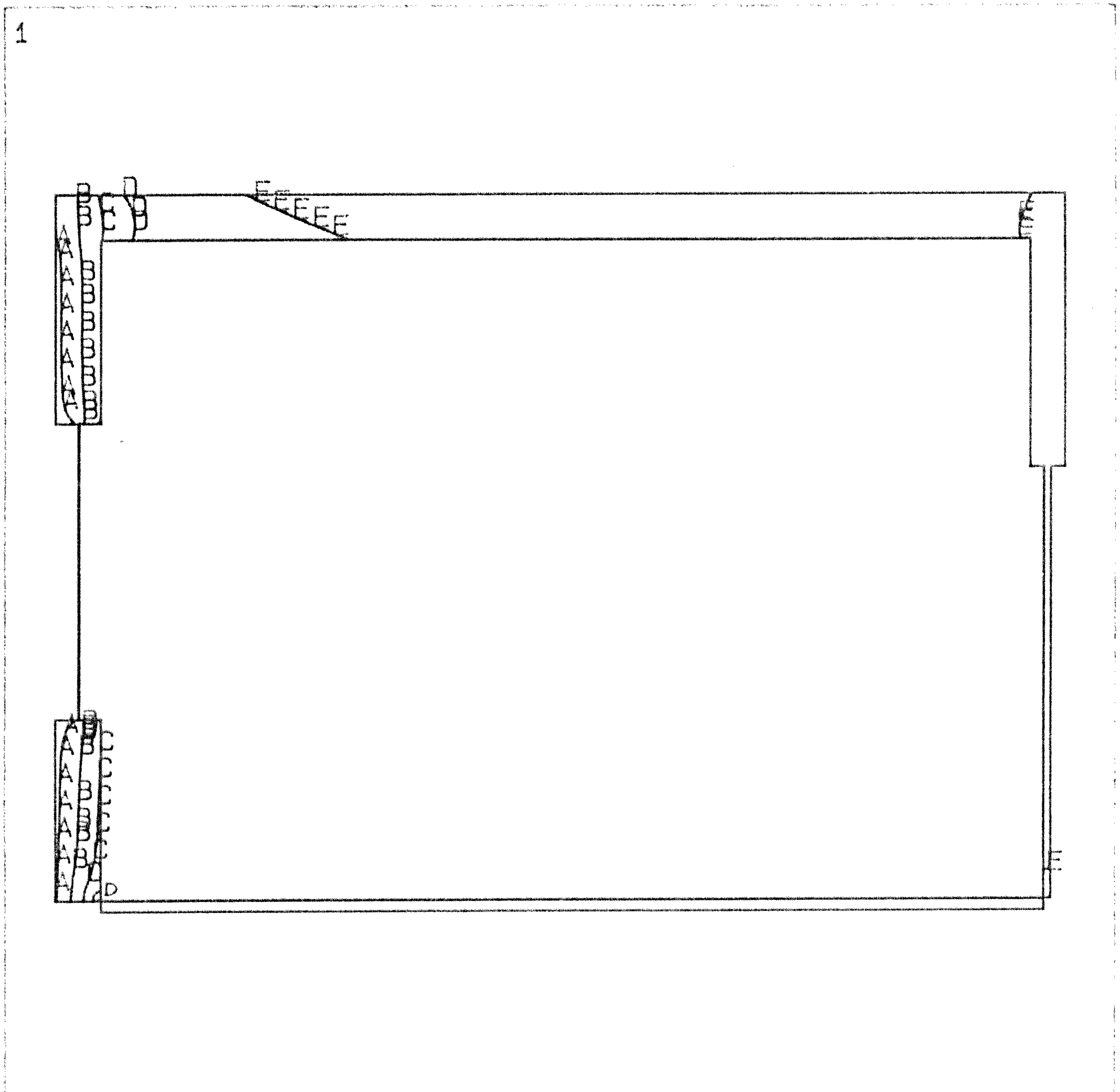
Şekil 8-15. Alanların tanımlanması.



Şekil 8-16. Oda modelinin sonlu-elemanlara bölünmesi.



Şekil 8-17. Model üzerinde sınır şartlarının tanımlanması.



ANSYS 5.0
OCT 19 1993
13:59:38
PLOT NO. 1
NODAL SOLUTION
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SUB =7
TIME=1
TEMP
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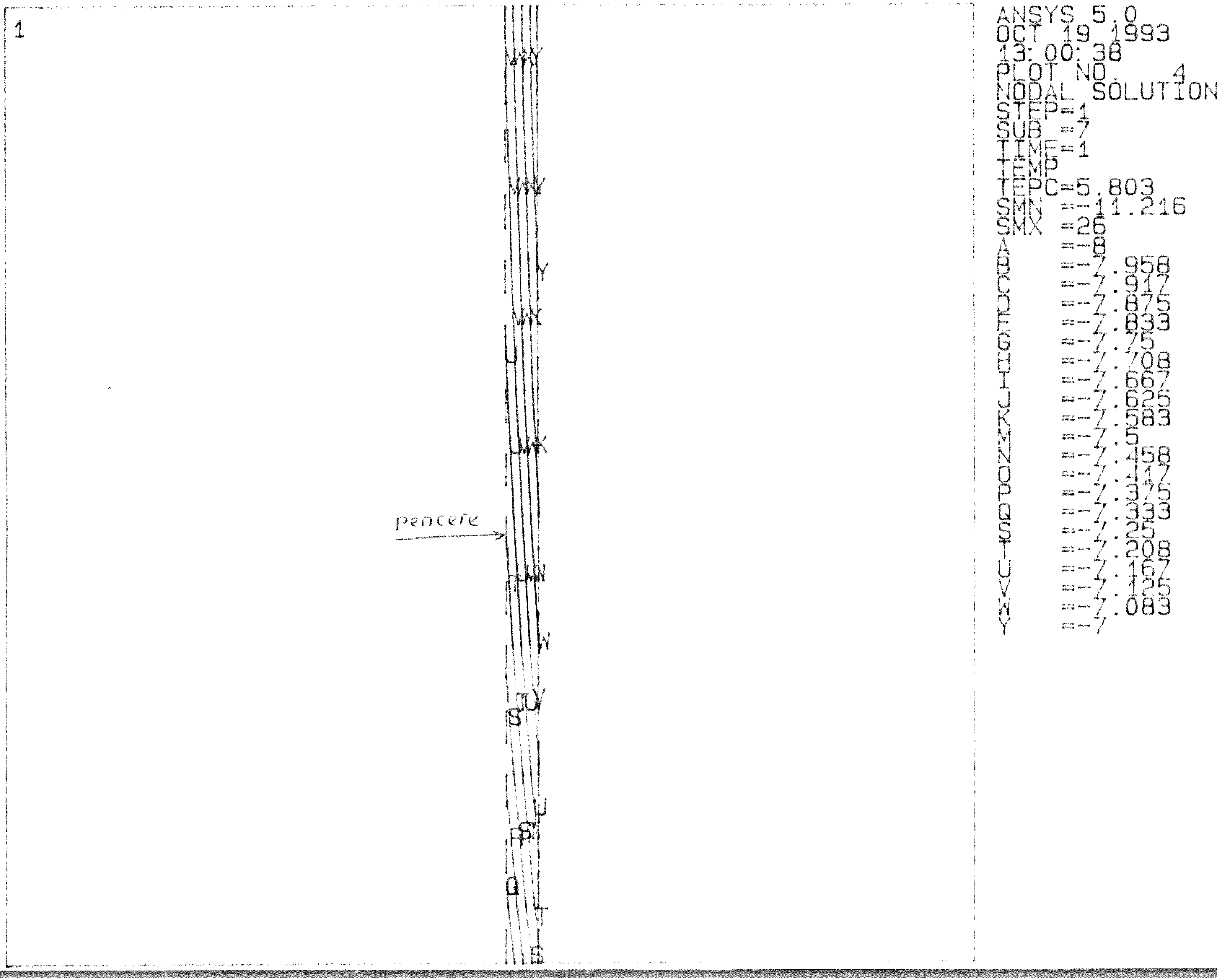
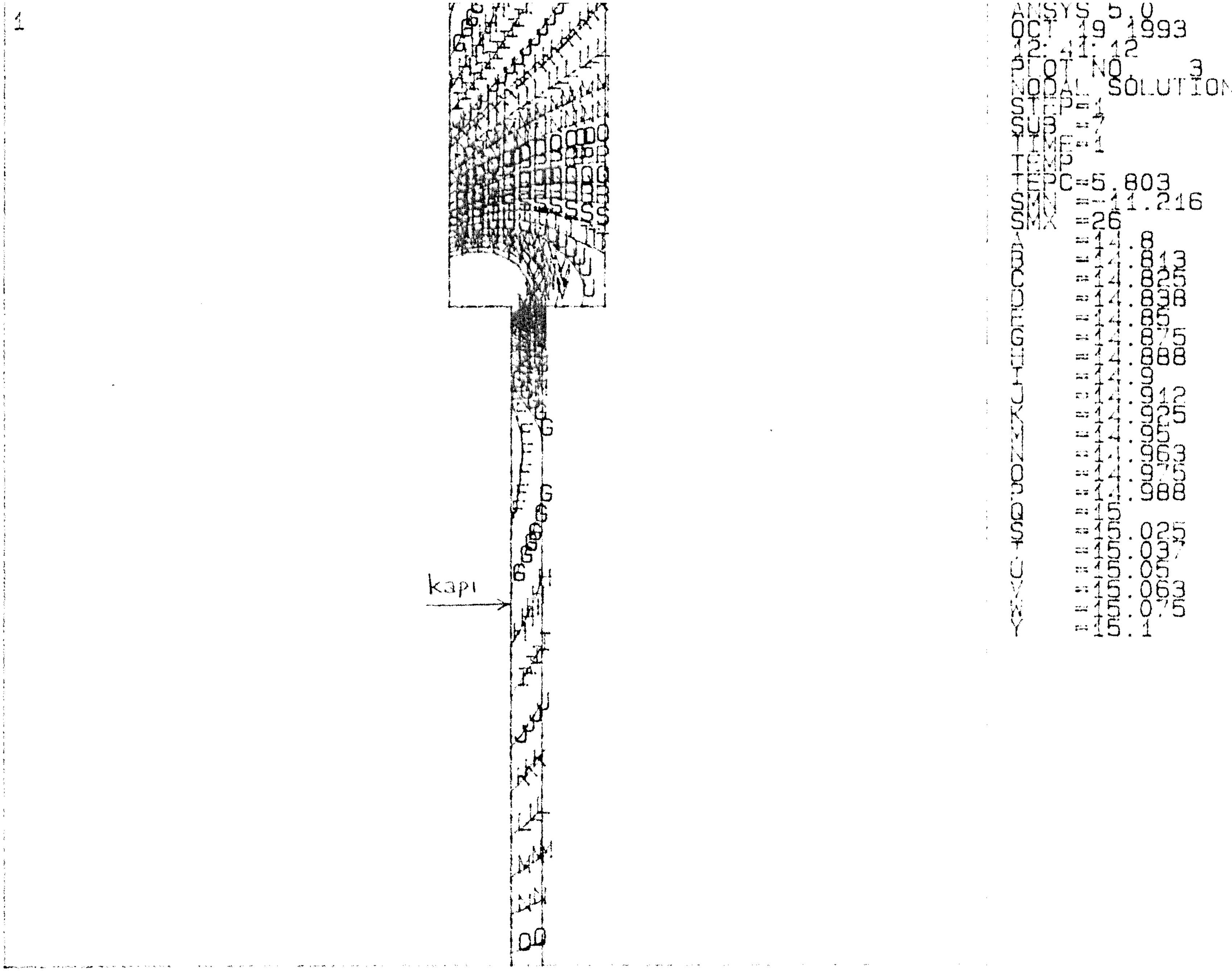
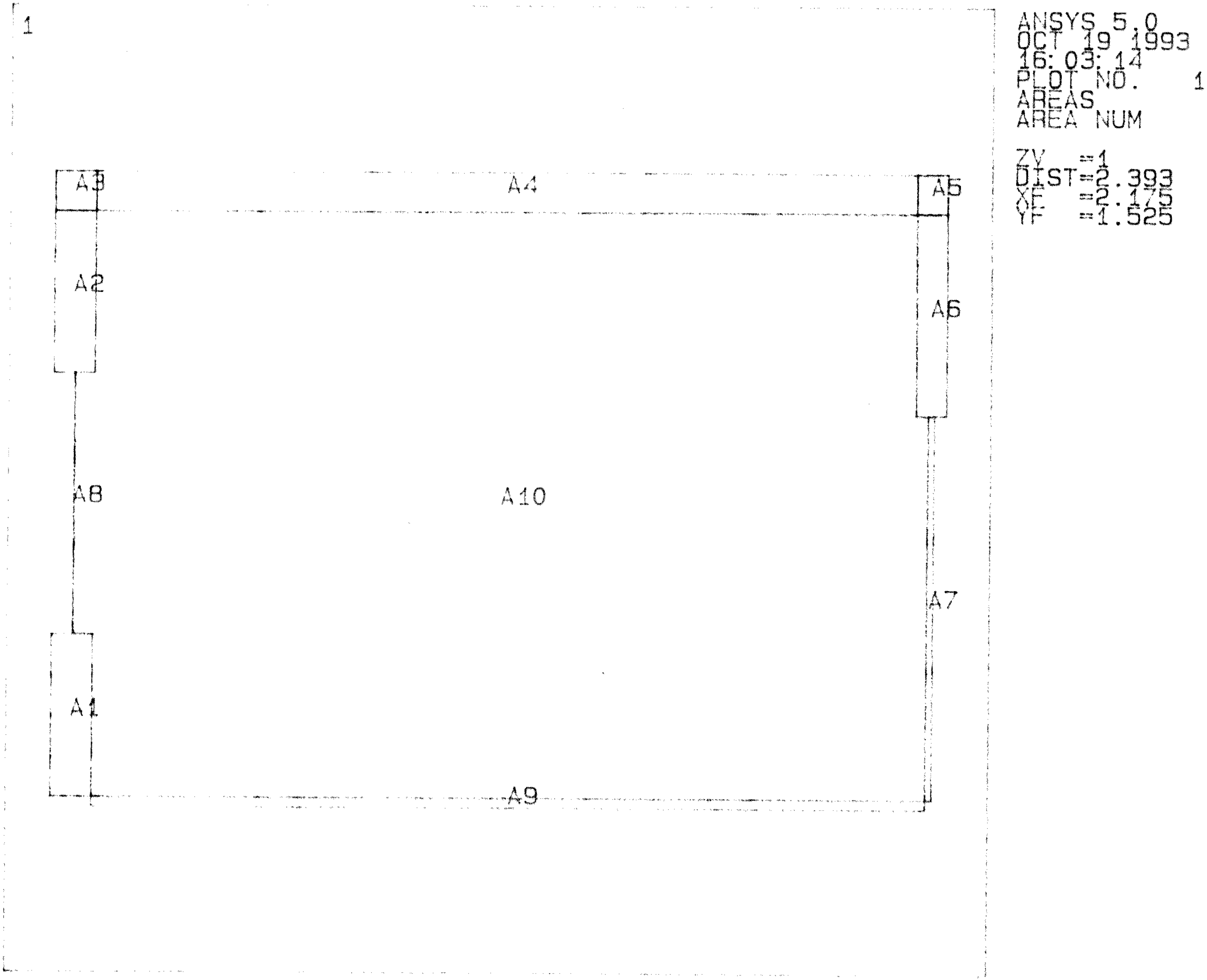
SMX =26
A =-8
B =-1.85
C =4.3
D =10.45
E =16.6


1
ANSYS 5.0
OCT 19 1993
13:00:38
PLOT NO. 4
NODAL SOLUTION
STEP=1
SUB =7
TIME=1
TEMP
TEPC=5.803
SMN =-11.216
SMX =26
A =-8
B =-7.958
C =-7.917
D =-7.875
E =-7.833
G =-7.75
H =-7.708
I =-7.667
J =-7.625
K =-7.583
M =-7.5
N =-7.458
O =-7.417
P =-7.375
Q =-7.333
S =-7.25
T =-7.208
U =-7.167
V =-7.125
W =-7.083
Y =-7
pencere



Şekil 8-20. Kapı içinde sıcaklık dağılımı.


1
ANSYS 5.0
OCT 19 1993
16:03:14
PLOT NO. 1
AREAS
AREA NUM
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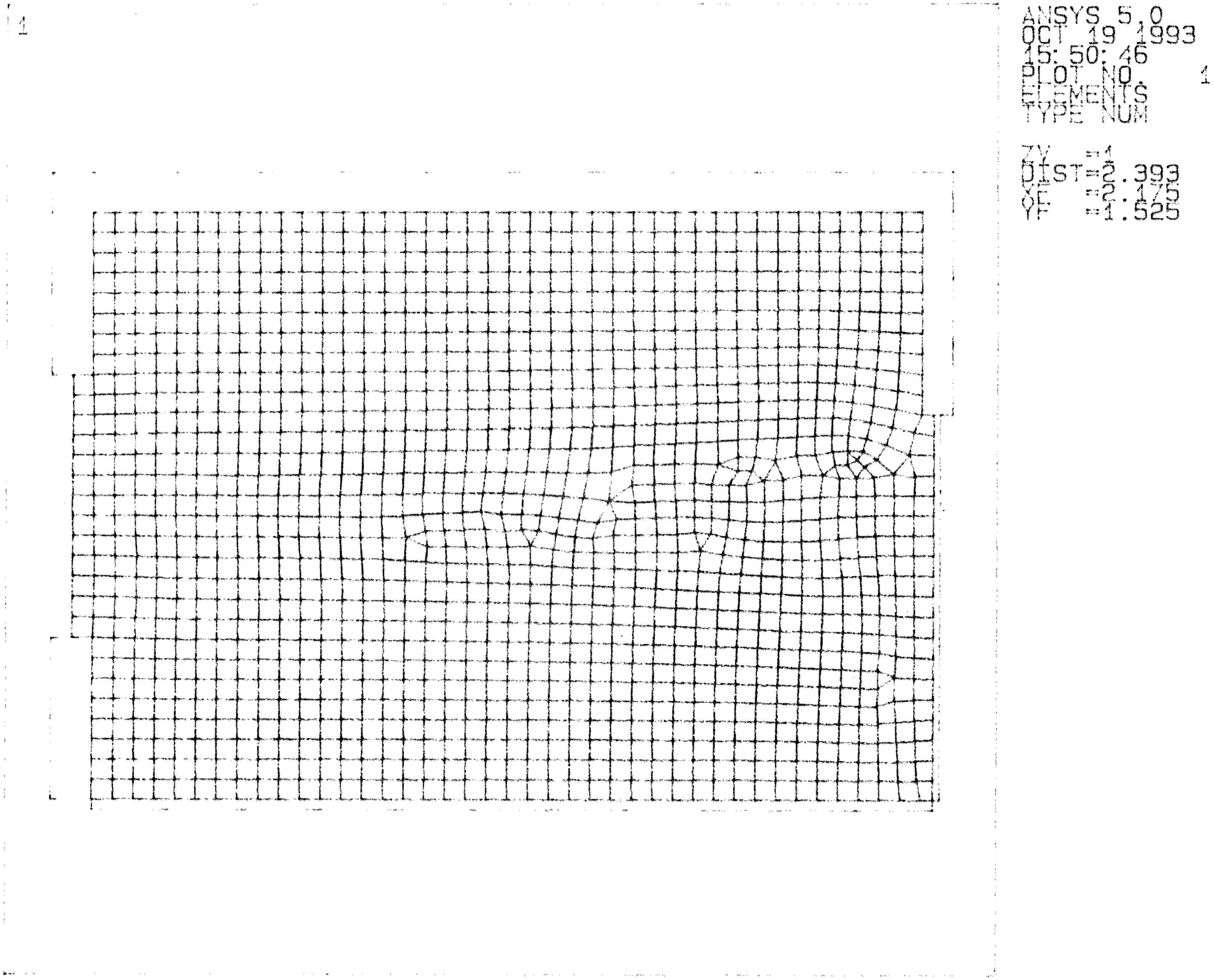
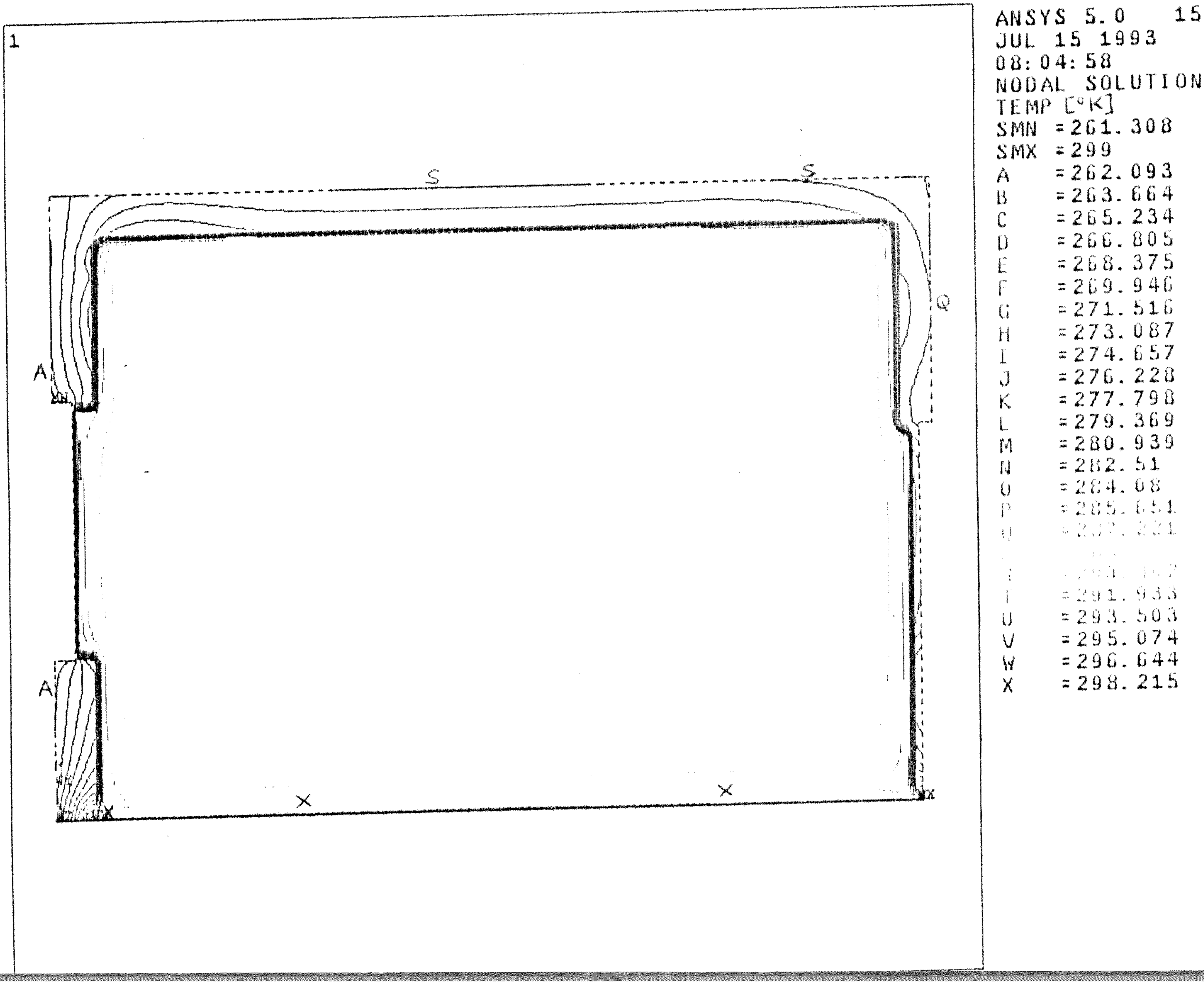
DIST=2.393
XF =2.175
YF =1.525
A3
A4
A5
A2
A6
A8
A10
A7
A1
A9



Şekil 8-22. Oda içinin sonlu-elemanlara bölünmesi.


1
ANSYS 5.0 15
JUL 15 1993
08:04:58
NODAL SOLUTION
TEMP [°K]
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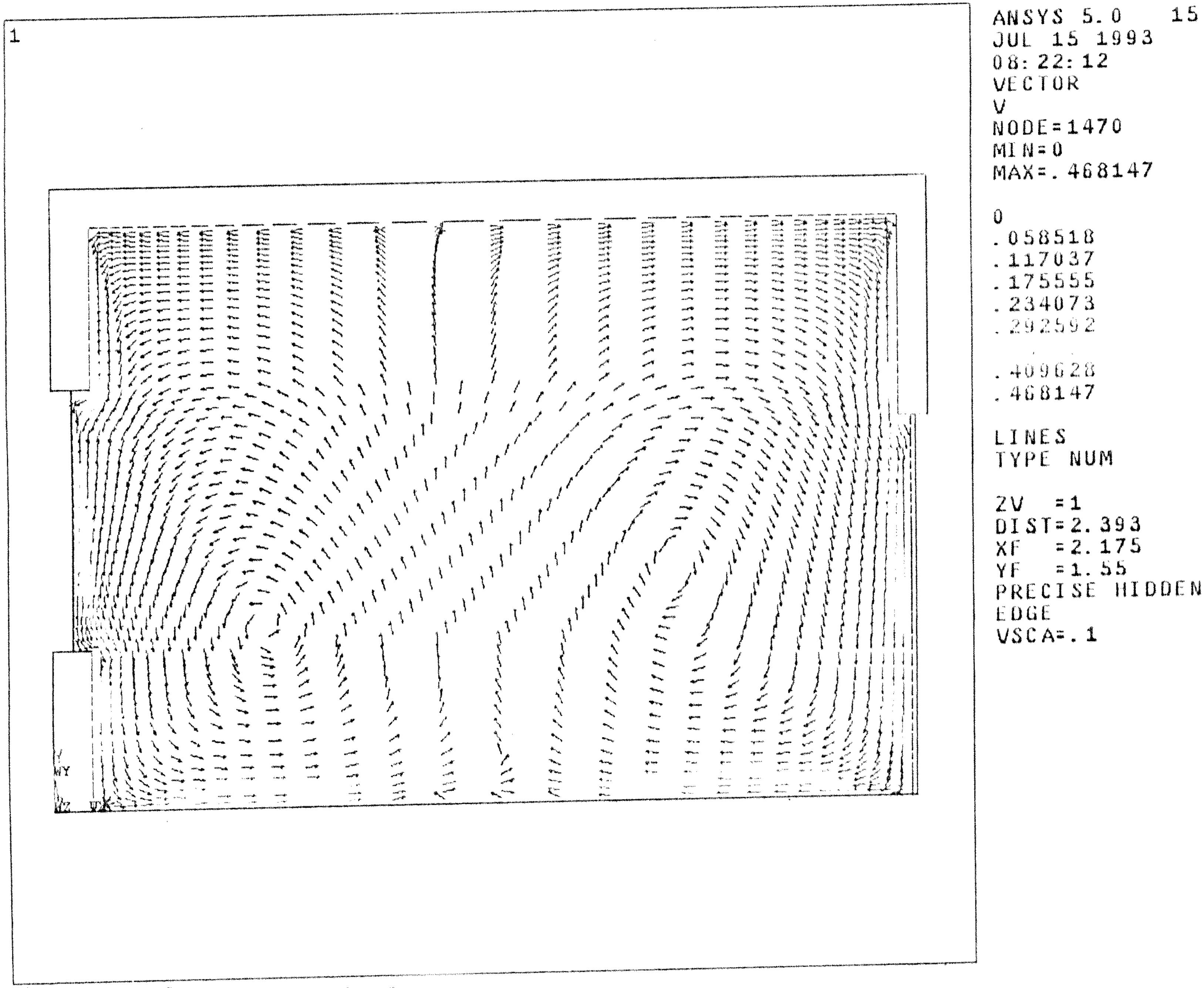
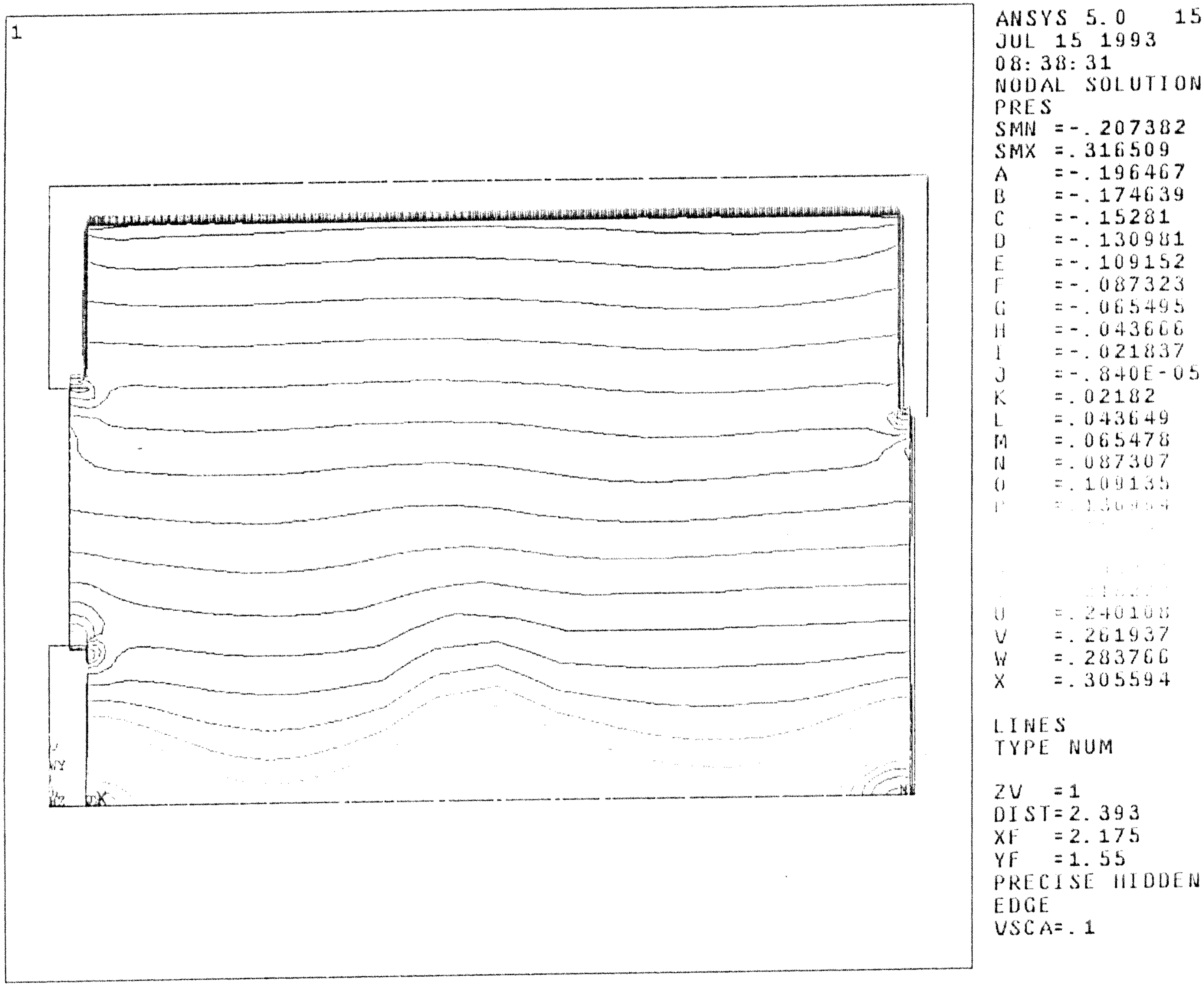
SMX =299
A =262.093
B =263.664
C =265.234
D =266.805
E =268.375
F =269.946
G =271.516
H =273.087
I =274.657
J =276.228
K =277.798
L =279.369
M =280.939
N =282.51
O =284.08
P =285.651
T =291.933
U =293.503
V =295.074
W =296.644
X =298.215
S
S
Q
A
A
X
X



Şekil 8-25. Oda içinde hız dağı



Şekil 8-24. Oda içinde basınç dağılımı.

THERMO-HYDRAULIC SIMULATION OF RADIANT FLOOR HEATED INDOOR SPACES

B. KILKIŞ, S. S. SAĞER, M. ULUDAĞ

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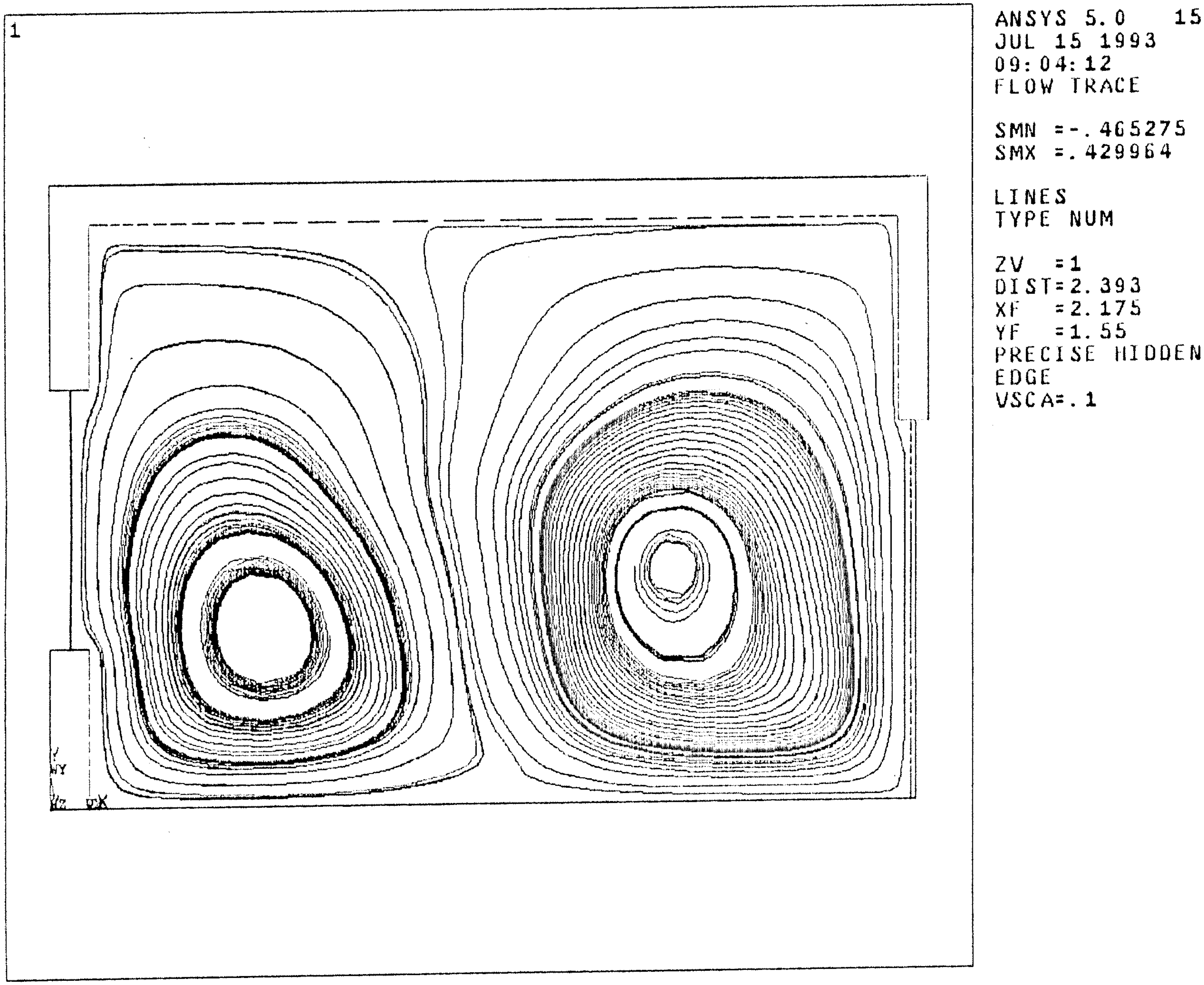
ABSTRACT

Radiant floor heating systems are highly energy efficient provided that a proper design and analysis is made. According to their thermophysical attributes, the heating loads may be substantially reduced while the human comfort is maintained even at a better level. One of the primary elements is the uniformity of temperature, velocity and pressure fields in a floor heated space. This uniformity reduces the infiltration heat losses and the indoor air temperature required for human comfort. The performance of the heated floor and the indoor space are highly interactive and an integrated solution is required. While these attributes are widely observed and some theoretical studies have been made, an integrated and thorough survey of the thermophysical behaviour of floor heated indoor spaces have not been accomplished yet. It is obvious that such a complete survey will substantiate the claims made for floor heating systems. In such a move the analytical algorithm given by Turkish standards for customizing the heating loads was substantiated by modeling typical indoor spaces. This analysis involved the use of ANSYS 5.0 and FLOTRAN (Computational Fluid Dynamics) package. Different boundary conditions for floor heating panels were imposed and indoor velocity, pressure and temperature fields were solved and processed for heat loss parameters. This paper gives the details of this application and summarizes the customized heat loss algorithm given by Turkish Standards.

This project was funded by Turkish Scientific and Technical Council under the grant MİSAG-12 and the analyses were carried out at METU/BİLTİR CAD-CAM Center.



Şekil 8-26. Oda içinde hava hareketleri.

ANSYS MODELING OF THE ROOM

The total heat output of a radiant panel has a convective (to heated air) and a radiative (to the surrounding surfaces) component:

$$q_Y = \underbrace{A \,.\, (T_p - AUST)}_{q_{yr}} + \underbrace{B \,.\, (T_p - T_a)}_{q_{yc}} \qquad (1)$$

q_Y is the total heat output intensity from the panel surface (W/m^2). Here A and B are the overall heat transfer coefficients for radiation and natural convection from the heated floor respectively. T_p is the effective surface temperature of the heated floor. T_a is the indoor air temperature, and AUST is the area weighted average temperature of the unheated surfaces surrounding the room. This type of distinction of the total heat transfer, enables to treat the problem in two nested main stages until AUST and q_Y converges, i.e.:

$$q_Y \,.\, A_p = Q_L \qquad \text{loop (ii)} \qquad (2)$$

$$(AUST)_R = (AUST)_C \qquad \text{loop (i)} \qquad (3)$$

Here A_p is the heated floor area (m^2).

Equation 2 depicts that the total heat output of the panel should be consistent with the heat losses from the room at steady state equilibrium conditions. Equation 3 depicts that the area weighted average inside surface temperature obtained by a radiation only analysis should agree with the convection only analysis. Figure 1 shows this iterative concept. In fact, the temperature distribution on the heated floor surface is not uniform. Depending upon the indoor conditions, heat diffusion in the floor, back and edge losses, and the mean fluid temperature in the piping, surface temperature is not constant. This necessitates the third nested loop where the input T_p (heat output equivalent effective surface temperature from Eqn. (1)) is verified with the actual T_p . However this loop is generally satisfied with one or two iterations. A typical surface temperature plot is shown in Figure 2.

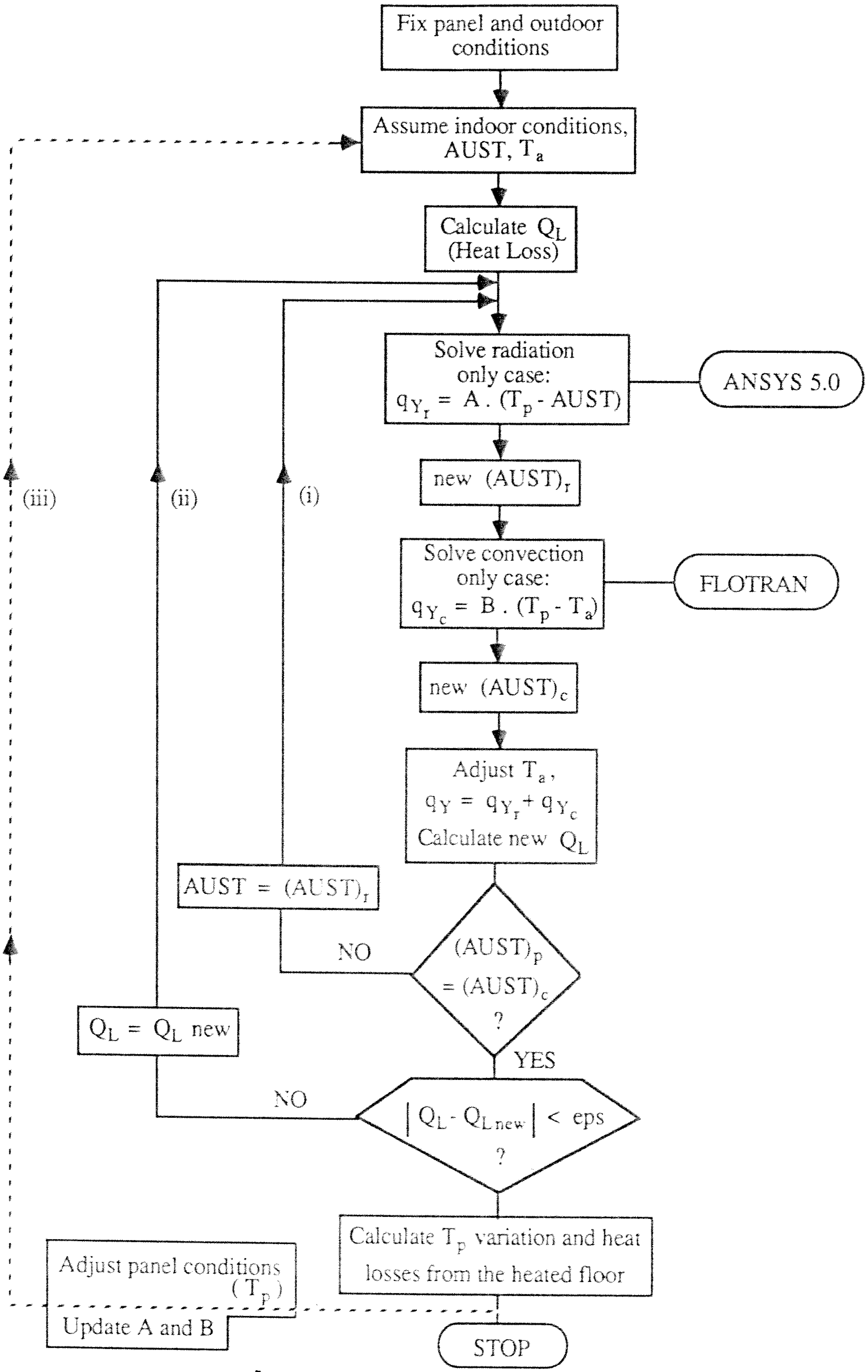


Figure 1. Iterative Concept for ANSYS Analysis of a Floor Heated Room.

SAMPLE SOLUTION

The room which was analyzed by ANSYS 5.0 has inner dimensions as 4 x 2.9 m. It has a window 1.3 m high located on the outer wall and door is 1.9 m height on the inner wall (Figure 3). The room was analyzed as two dimensional at two iterative steps as follows:

<u>i) Radiation Analysis :</u> At this step the room was modeled without including the air. Since the fluid medium is not important in radiation, inside of the room was considered as empty and the problem was solved by ANSYS 5.0 package to see the radiation effect of the heated slab on the walls, window and door. For this problem PLANE55 is the most suitable element type of ANSYS to model the solid parts (walls, window and door). PLANE55 is 2-Dim, 4-node conduction heat transfer element. After discretizing the walls, window and door with PLANE55 element, boundary conditions were defined (Figure 4). 28.2 W/m^2 heat input (due to radiation) from floor to room was calculated at the end of ANSYS solution. This is also used to update T_p . Screen outputs of the solution result are shown at Figures 5, 6 and 7. The constants used in the model for radiation analysis are as follows;

a) Material Properties;

Conduction heat transfer coefficient for outer walls	=	0.7 W/m K
Conduction heat transfer coefficient for ceiling	=	1.4 W/m K
Conduction heat transfer coefficient for inner walls	=	1.5 W/m K
Conduction heat transfer coefficient for window	=	3 W/m K
Conduction heat transfer coefficient for door	=	1.2 W/m K

b) Boundary Conditions;

- constant temperature (=26 °C) at floor surface
- convection heat transfer coefficient (20 W/m^2 K , T_{bulk} = -12 °C) on outer walls and window
- convection heat transfer coefficient (5 W/m^2 K , T_{bulk} = 18 °C) on floor surface of upper house
- convection heat transfer coefficient (5 W/m^2 K , T_{bulk} = 15 °C) on inner walls
- convection heat transfer coefficient (5 W/m^2 K , T_{bulk} = 15 °C) on door surface

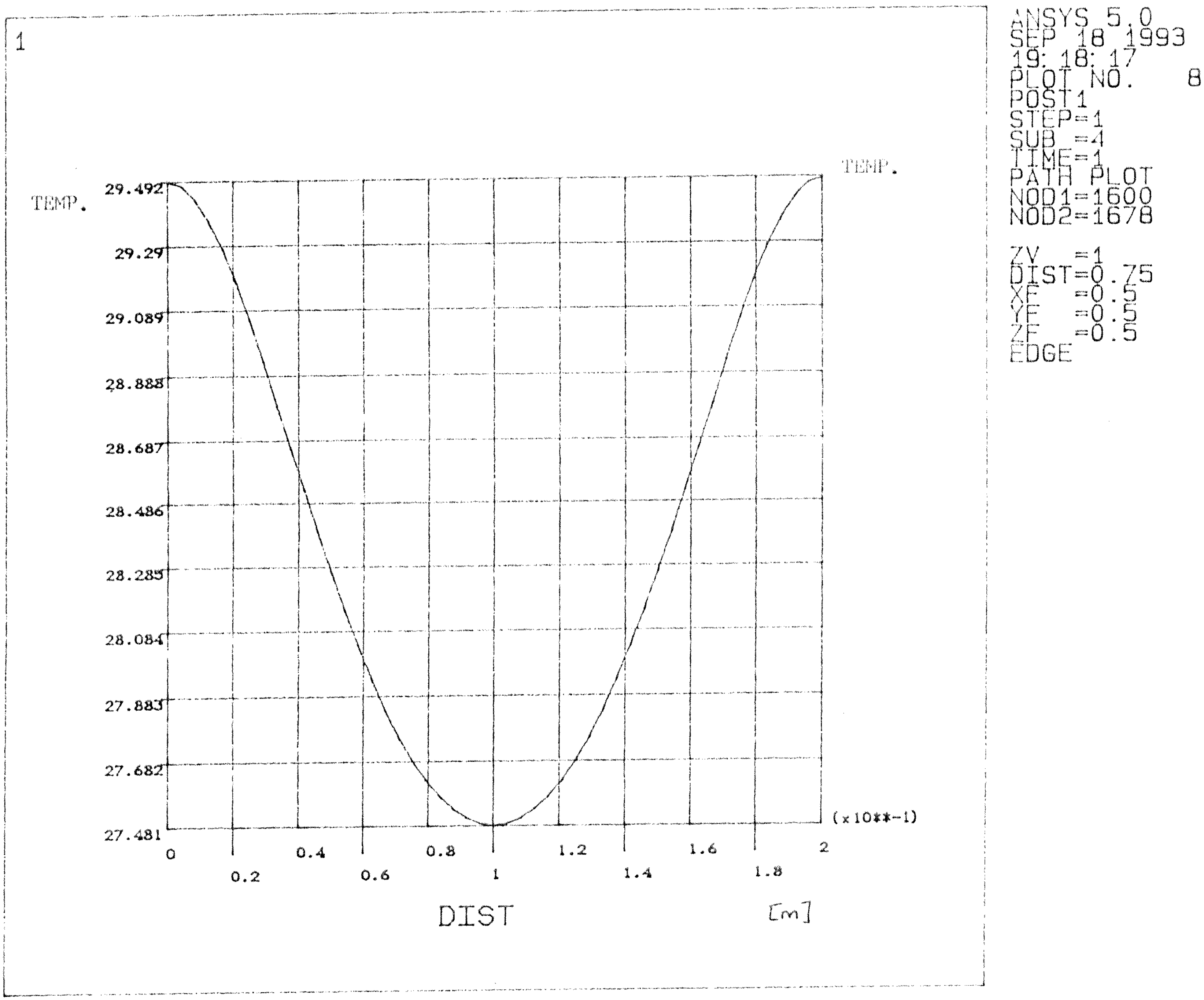


Figure 2. A Typical Surface Temperature Plot.

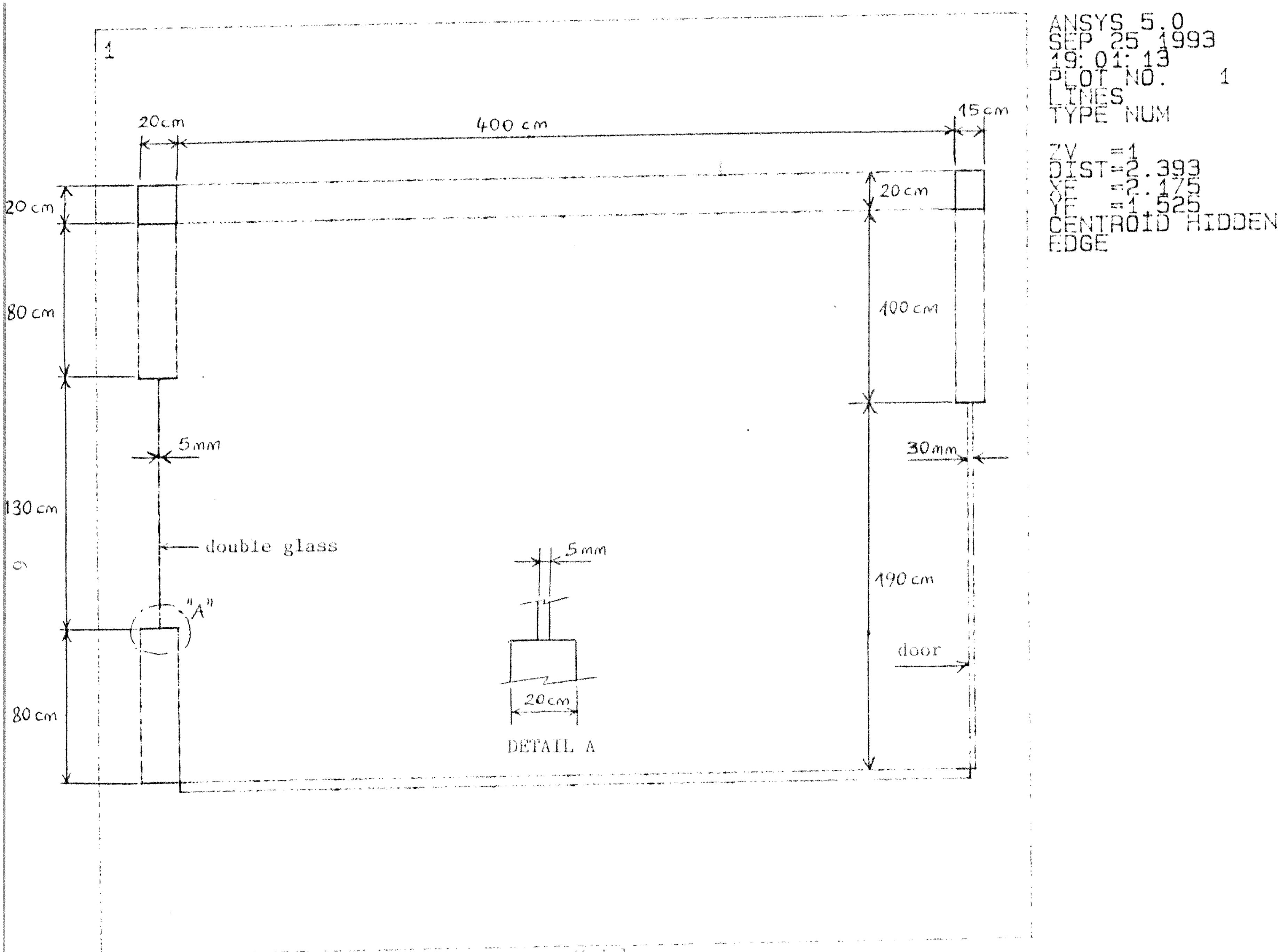


Figure 3. Sketch of the Floor Heated Room Model.

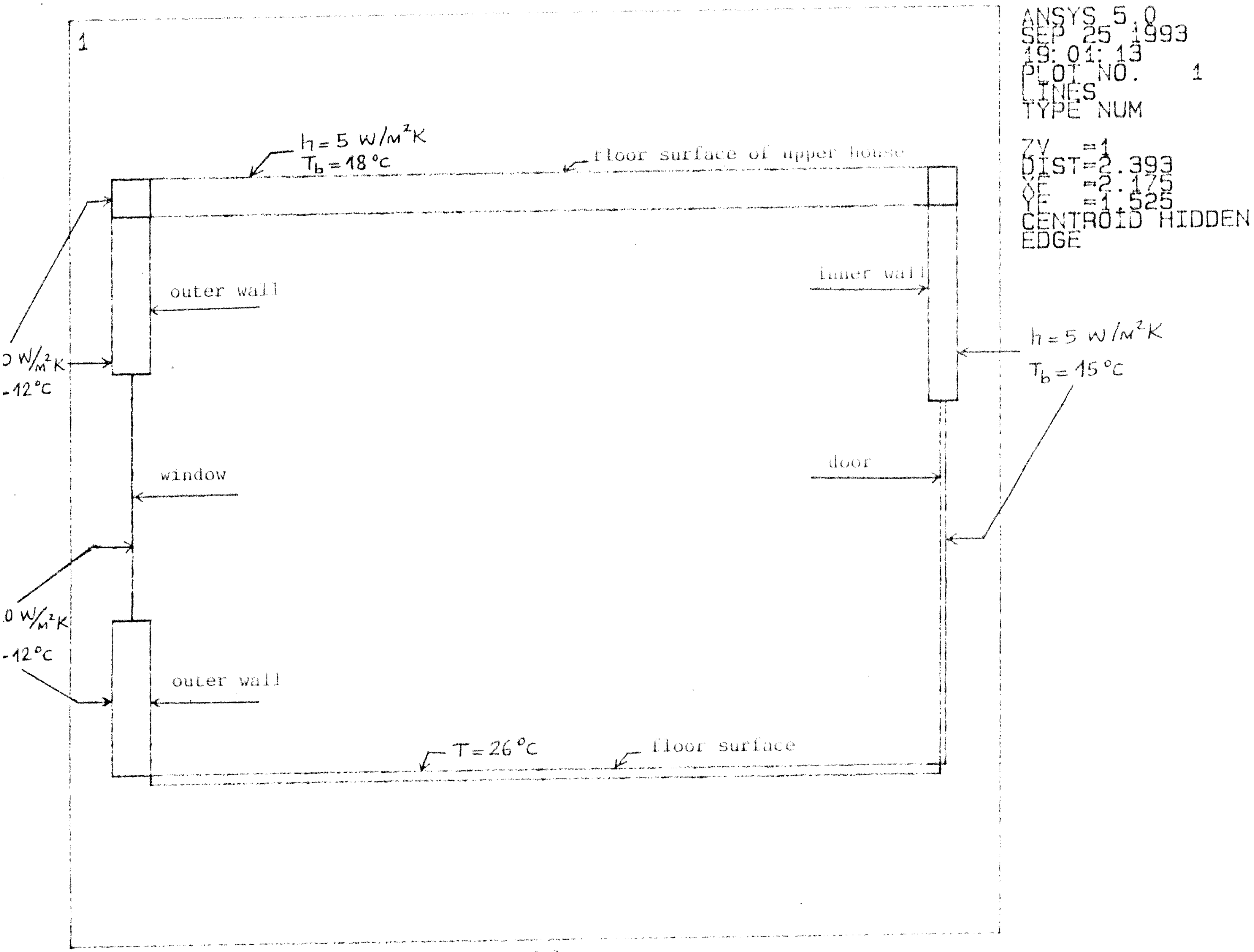


Figure 4. Boundary Conditions on the Model.

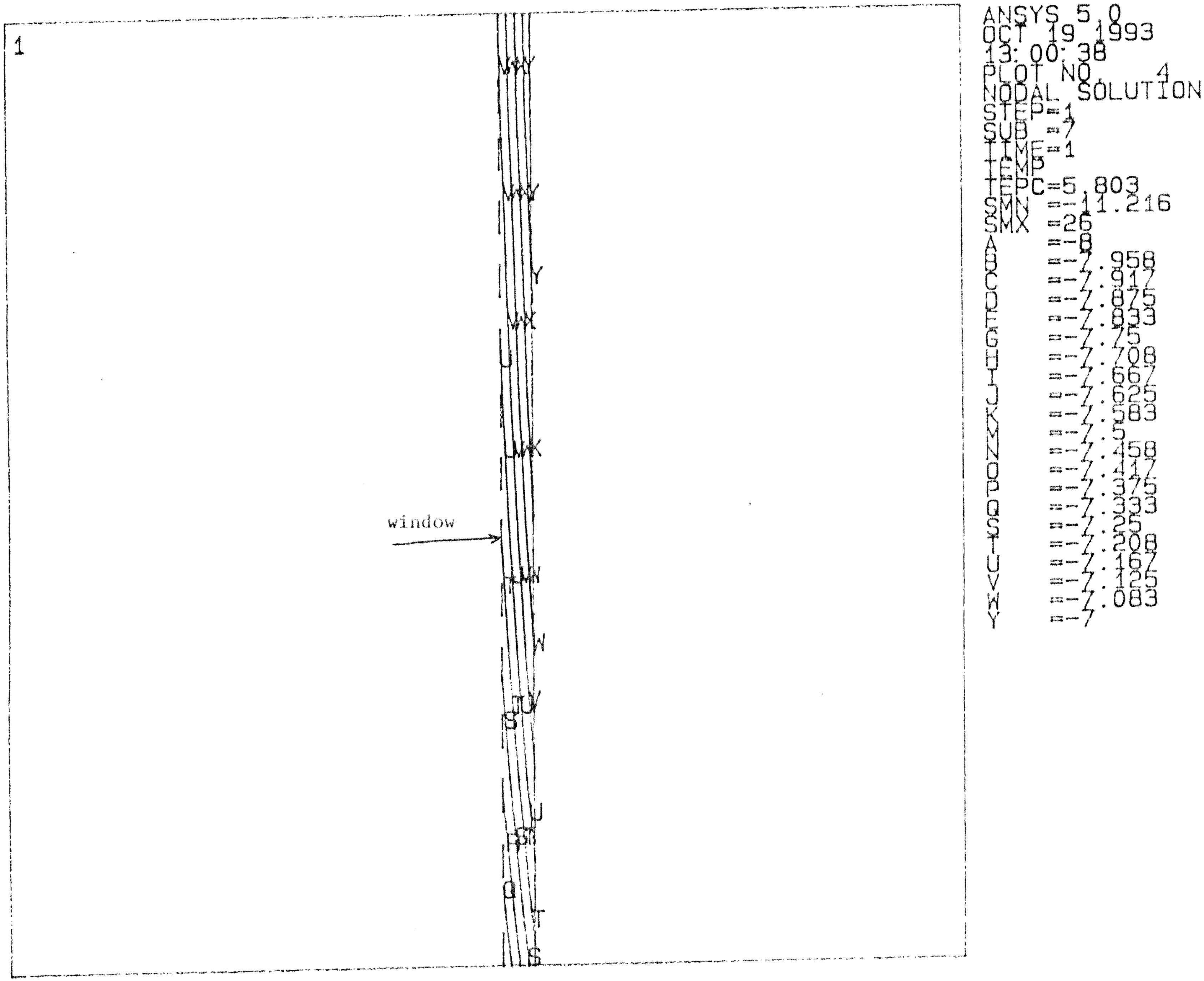


Figure 6. Temperature Distribution through Window.

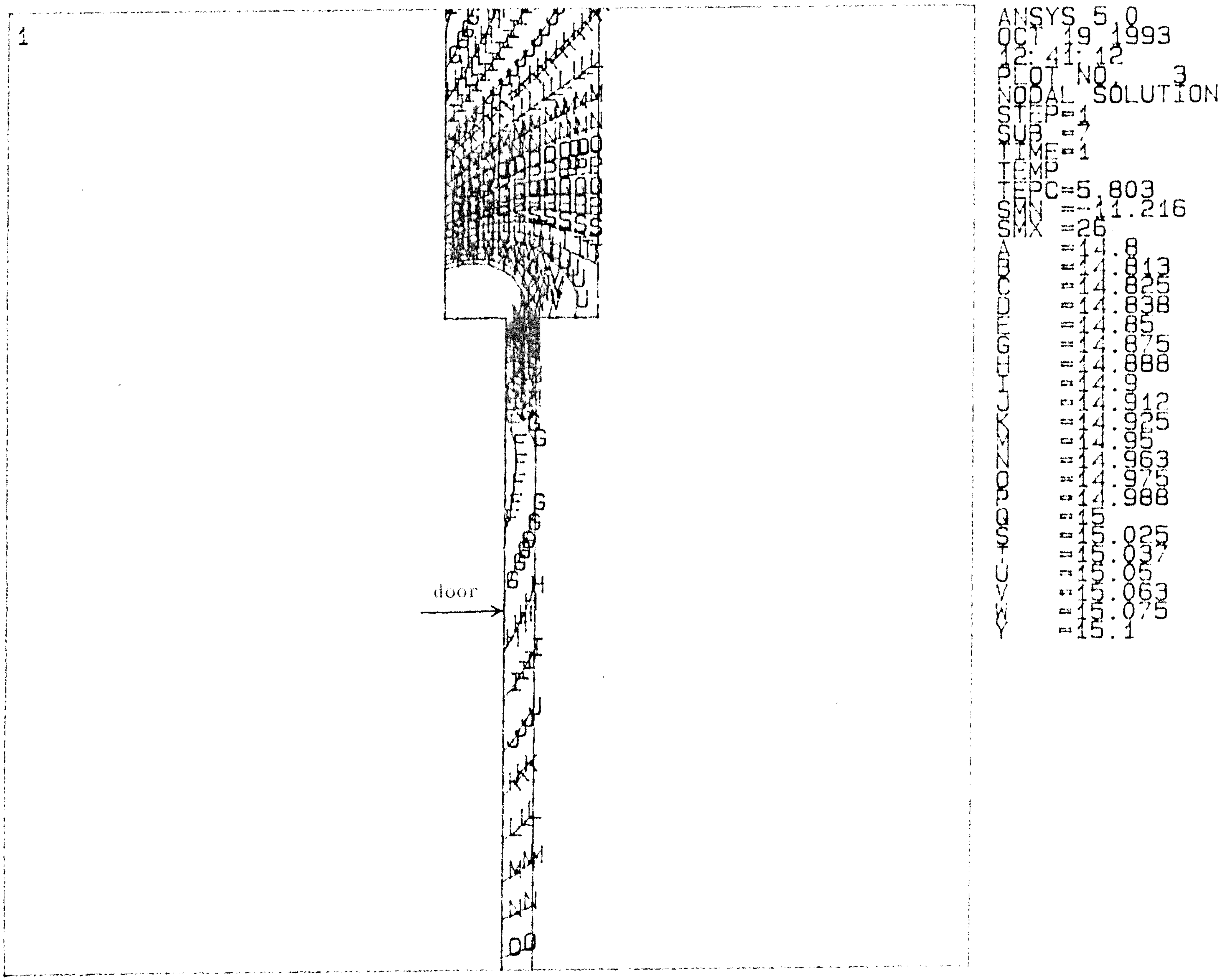
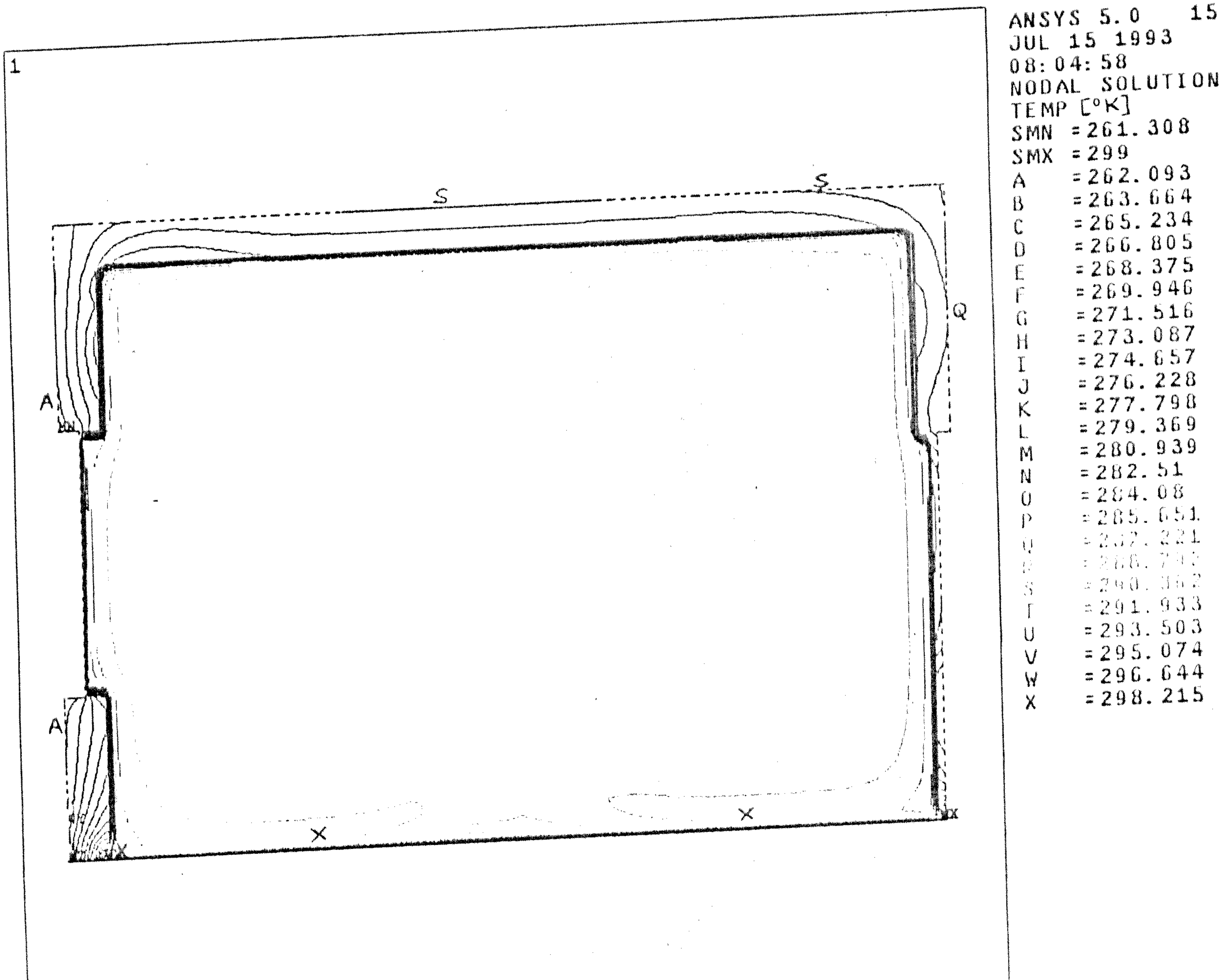


Figure 7. Temperature Distribution through Door.

ii) Convection Analysis : This time, radiation effect is excluded and the same room with air was modeled (Figure 3). For this purpose, natural convection of the air was solved by ANSYS to obtain temperature, pressure and velocity distributions on the air, and air circulation. While, ANSYS is a general purpose program package to perform analyses on wide range of area (static, dynamic, thermal, fluid, magnetic and their coupled analysis), the problem here is a fluid dynamics problem which includes both thermal and fluid flow effects. It is very difficult to solve such a kind of problem with regular ANSYS subroutines. For example, modeling of the room will be very difficult, since ANSYS requires at least one node inside the boundary layer thickness. This will increase the amount of nodes and elements. Very large disk space should be available on the computer, since size of the files created by ANSYS becomes very large (about 150 MBytes), and also computation time becomes very long (about 1 or 2 days). For this reason the FLOTRAN subroutine was employed to solve this natural convection problem. FLOTRAN brings the ease of modeling of the problem, since it does not want nodes in boundary layer thickness. It solves the problem in a rather short time (about 1 hour) with a relatively coarse mesh, with fewer nodes and elements. FLOTRAN is not a stand alone finite element program package but is a solver subroutine of ANSYS. The problem is modeled in ANSYS program, but it is solved using FLOTRAN solver subroutine. At the end, results are obtained again in ANSYS's post-processing section.

The walls, window, door and inner space of the room were discretized with PLANE55 element. In addition to boundary conditions defined in Figure 4, "no slip" condition for air at inner surfaces of walls, window and door was imposed. During the solution, in addition to temperature degree of freedom, FLOTRAN adds pressure, velocity and energy degrees of freedom to the nodes of PLANE55 element (at inner space of room). Thus ANSYS calculates air motion in the room with the help of FLOTRAN solver. 75.6 W/m^2 heat input (due to convection) from floor to room was calculated at the end of ANSYS solution. Screen outputs of the solution result are indicated at Figures 8, 9, 10 and 11.



1
ANSYS 5.0 15
JUL 15 1993
08:04:58
NODAL SOLUTION
TEMP [°K]
SMN =261.308
SMX =299
A =262.093
B =263.664
C =265.234
D =266.805
E =268.375
F =269.946
G =271.516
H =273.087
I =274.657
J =276.228
K =277.798
L =279.369
M =280.939
N =282.51
O =284.08
P =285.651
Q =287.221
R =288.792
S =290.362
T =291.933
U =293.503
V =295.074
W =296.644
X =298.215
S
S
Q
A
A
X
X

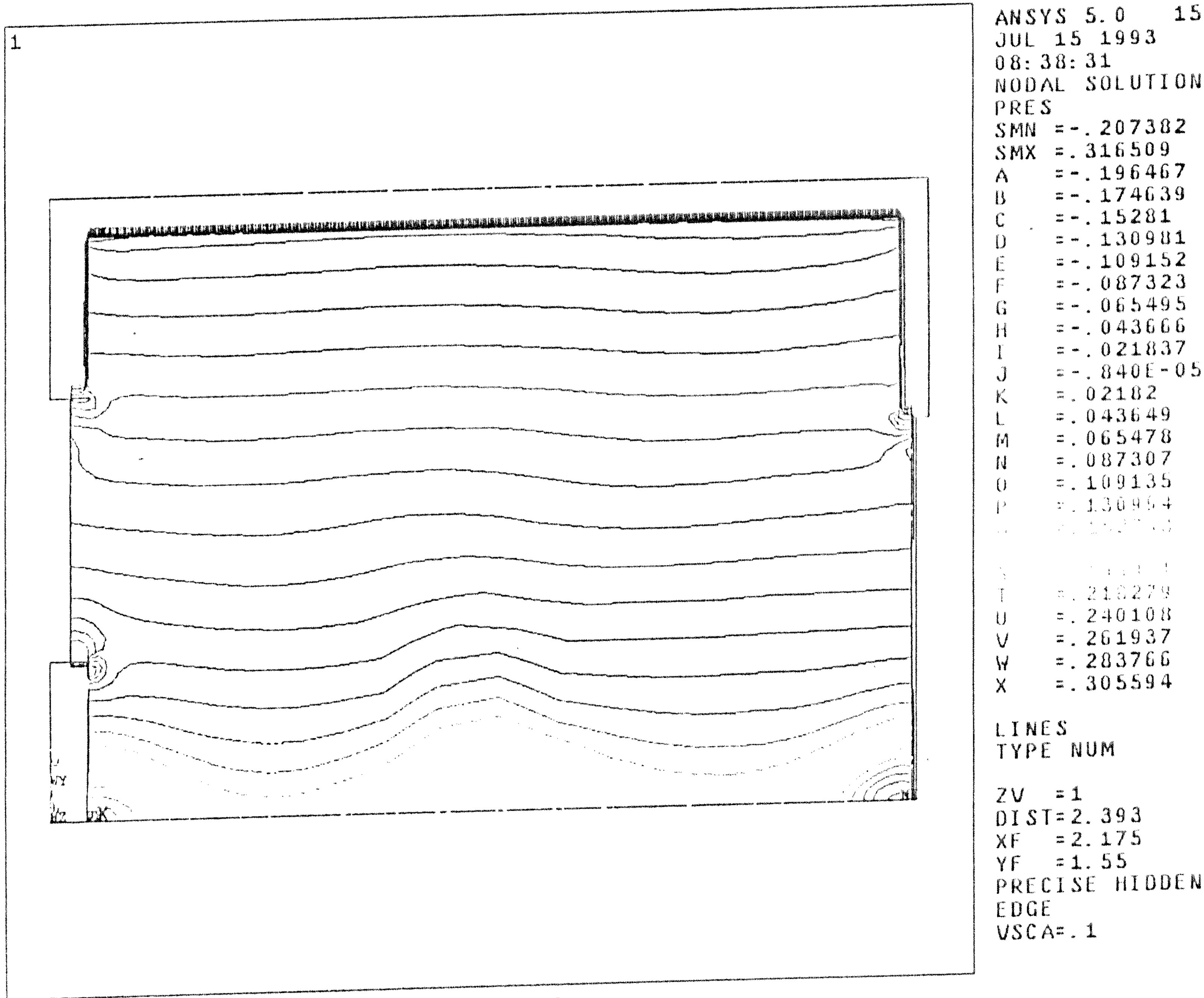


Figure 9. Pressure Distribution in Heated Space.

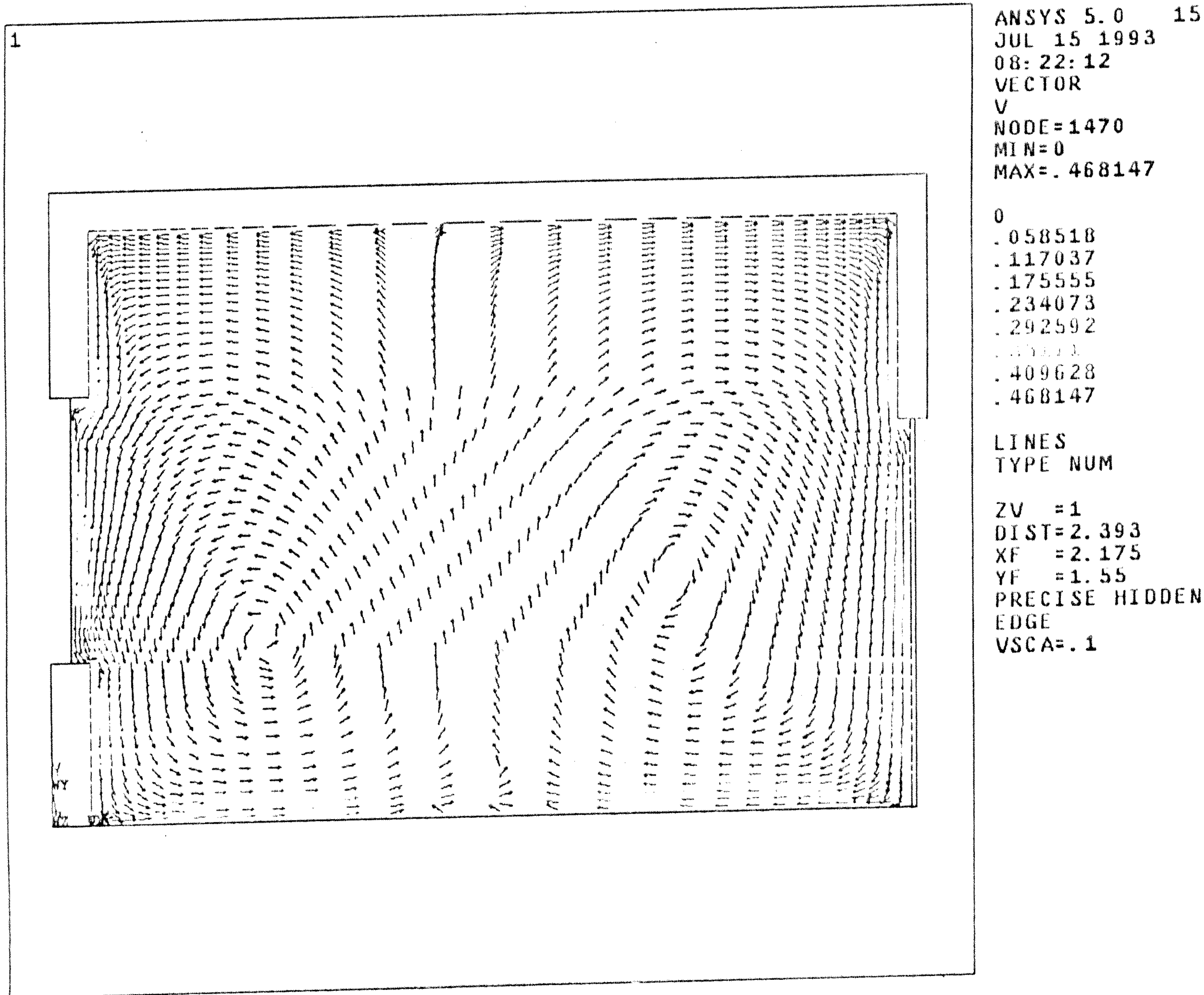


Figure 10. Velocity Distribution in Heated Space.

EK-9 PANEL SOĞUTMA TASARIM ALGORİTMASI ve BİLGİSAYAR PROGRAMI

Panel soğutma ile ilgili tasarım algoritmasında ana parametreler boru ölçüleri,aralığı ve malzemesi,panel ölçüleri,soğutma yükü ve iç sıcaklıktır.Bu algoritma ile ısıtma paneli tasarım algoritması arasındaki fark iki adettir:

1-Soğutma yükü eksi işaret ile gösterilir.
2-T_{min} ile T_{max} yer değiştirir.

Eksi işaret kullanımı ile bütün denklemler aynen kullanılır. Söz konusu algoritma kullanılarak bir bilgisayar programı geliştirilmiştir. Bu programın adı SOĞUTMA dır. Örnek bir çözüm ekte verilmiştir.
Bu örnekte odanın hissedilir soğutma yükü -4000 Kcal/h,toplam panel alanı ise 100 m^2 dir. Programa artı yük de girilse,otomatik olarak eksi işaret kazandırılır. Birim soğutma yükü-40 Kcal/h m^2 olmaktadır.Program yük ve panel yüzey alanına bakarak optimum hidrolik devre sayısını 9 olarak vermiş,kullanıcıda bunu kabul etmiştir. İç ve dış hesap sıcaklıkları girildikten sonra,oda konumu ekranda beliren tablodan girilir. (Bakınız TSE,1993-b Çizelge 8).
Daha sonra oda ölçüleri girilir. Mahal yüksekliği ve oda boyutu düzeltmelerini hesaplayan program,her bir panelin(toplam 9 adet) yüzey alanını doğruladıktan sonra,yüzey yayınım katsayısını da sorarak,AUST(Soğutulmayan yüzeylerin ortalama sıcaklığı) değerini hesaplar. Panel soğutma yükünü karşılayacak olan ortalama panel yüzey sıcaklığını program 17°Colarak hesaplamıştır.
Bu safhadan sonra panel malzemelerinin ısıl özellikleri ile kalınlıklar sıra ile girilir. Su sıcaklık farkı 5°C olarak girilmiştir. Panelin ısıl verimi,girilen yalıtıma da bağlı olarak hesaplandıktan sonra,boru ölçüleri,boyu ve ısı iletim katsayısı girilir. En son olarak boru aralığıda girildikten sonra,program gerekli ortalama su sıcaklığını hesaplar. Bu örnekte 15°C dır. Program daha sonra bu tasarıma ait dosya bilgilerini,girdileri ve sonuçları listeler ve istendiğinde,dosya adı ile print komutu verildiğinde kağıda döker. Tasarım çıktısı ekte verilmiştir.
Bu tasarımda T_{min} 16.3°C olarak bulunmuştur. Eğer bu değer oda içindeki kontrollu nem oranı (mekanik havalandarma var) veya kontrolsüz nem oranına(mekanik havalandırma yok) tekabül eden çiğ noktasından az ise,T_{min} de artış sağlamak üzere boru aralığı azaltılır. Bu mümkün değilse ya mekanik havalandırma ile nem kontrol altında tutulur veya,yeter ek yüzey tavanda mevcut ise panel toplam yüzey alanı arttırılır. Reynolds sayısı ise yeterli düzeyde değildir. Bu durumda su sıcaklık farkı azaltılır(5°C seçilmişti)ve/veya panel sayısı azaltılabileceği gibi boru aralığı azaltılarak paneldeki boru uzunluğu arttırılır.

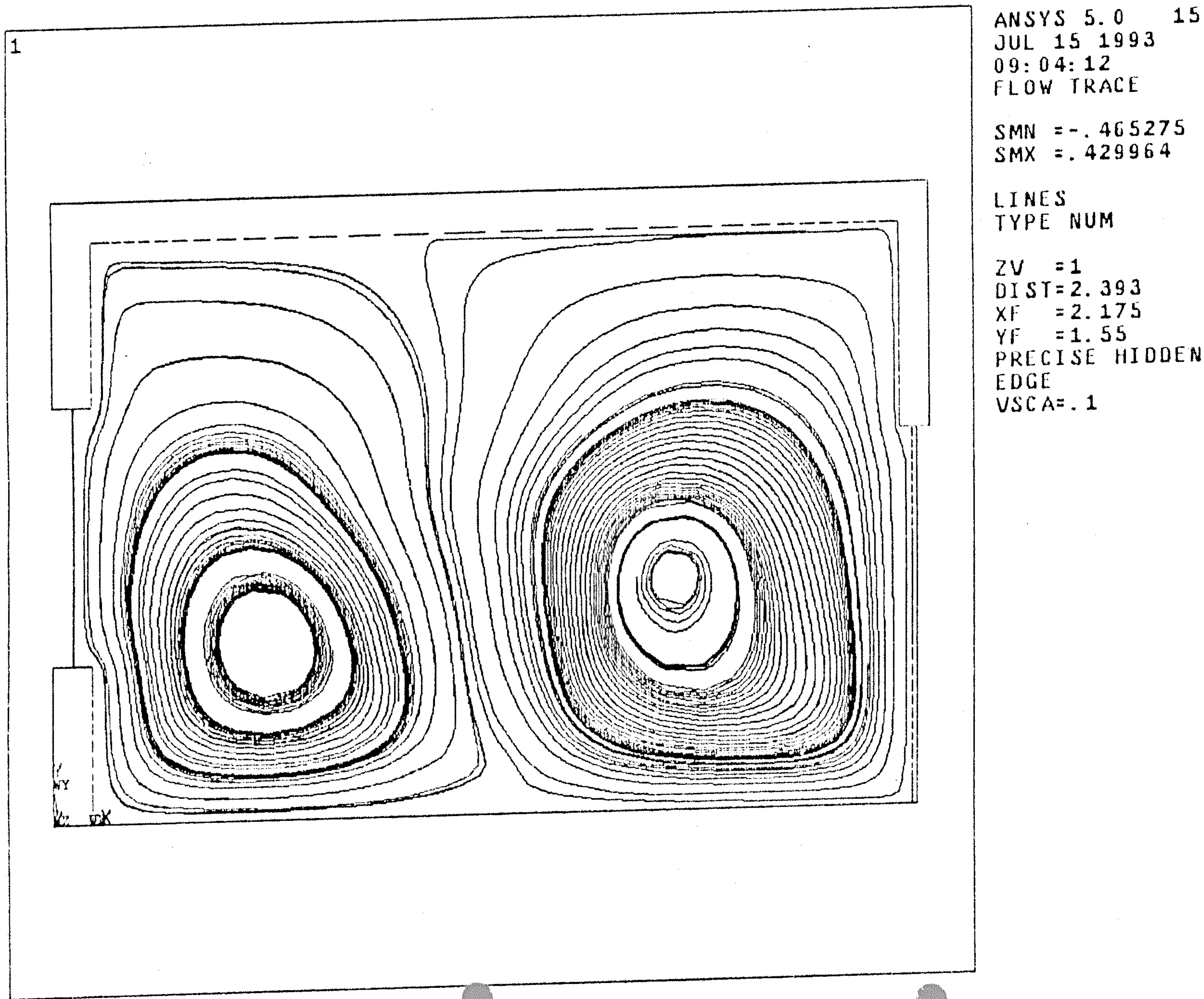


Figure 11. Air Motion in the Room.

Proje Adi   :SOGUTMA

Mahal   :ANKARA

Proje Sahibi:TÜBiTAK MiSAG-12

Tarih   :11/19/1993

Devam için ENTER tusuna basiniz

NOT: Degerlerinizi kesirli de girebilirsiniz (3/4 gibi)

* Odanin hissedilir sogutma yükü [Kcal/h] ? = -4000

* Odadaki toplam sogutma panel alani [m²]   = 100

Gerekli birim sogutma yükü [Kcal/h m²]   = -40.0 [Kcal/h m²]

Tavanda 9 adet panel (devre) kullanilmasi uygun.
* Tavanda kaç adet sogutma paneli kullanilacak ? = 9

* Yaz esdeger konfor sicakligi [°C] ? = 22

* Yaz için dis hesap sicakligi [°C] ? = 33

* Oda konum tipini,(c), gelecek tablodan giriniz :

Devam için ENTER Tusuna Basiniz

ODA KONUM TiPi, c					
Odanin bulundugu kat					
ARA KAT			ÇATI KATI		
iç oda	Dis Cepheli	Köse Oda	iç oda	Dis Cepheli	Köse Oda
c:					
0	1	2	1	2	3
Tavandan Sogutulan Bodrum Katlarinda					
0	c=c-0.5				

* Oda konum tipi, c ? = 2

* Oda dikdortgen (veya kare) mi [E/H] ? = E

* Odanin boyu [m] ? = 11

* Odanin eni [m] ? = 10

Tasinim isi transferi için oda büyüklügü düzeltmesi : 0.927

* Mahallin denizden olan yüksekligi [m] ? = 864

Tasinim isi transferi için yükseklik düzeltmesi : 0.950

Devam için ENTER tusuna basiniz

* Panel yüzey alani [m^2] = 12
* Panelin hissedilir sogutma yükü [Kcal/h] ? = -444

* Tavan yüzeyi yayinim katsayisi :

Yayinim Katsayisi, e (Siyah cisim için, e=1)

Malzeme	e
Parke (Parlak)	0.88
Parke (mat)	0.90
Sunta	0.90
Beton (Boyasiz)	0.88
Seramik Karo	0.87
Boya:	
Acik Renkler	0.75
Koyu Renkler	0.90

* Yayinim katsayisi ? = 0.75

Sogutulmayan yüzeylerin ortalama sicakligi = 23.2 °C

Panel yüzeyi etkin sicakligi = 17.1 °C

Devam için ENTER tusuna basiniz

* Bu asamada bazi verileri degistirmek istermisiniz ? [E/H] = H

* Simdi tasarim için gerekli ölçüler girilecek

* Boru merkezi/tavan yüzeyi itibari ile sap kalinligi [cm] = 1
* Toplam sap kalinligi [cm] = 2

* Sap malzemesinin k degeri [Kcal/h m°C] ? =
<Beton icin= 1.20 >= 1.2

* Isi yalitim malzemesi kalinligi [cm] ? = 3
* Isi yalitim malzemesi k degeri [Kcal/h m°C] ? =
<Strapor icin=0.038 > = 0.038

* Sap kat betonu altindami ? [E/H] = E

* Kat betonu kalinligi [cm] = 10
* Kat betonu k degeri [Kcal/h m°C] = 1.2
* Yukaridaki mahal sogutuluyormu ? [E/H] = H

* Sistemdeki Gidis/Dönüs akiskan sicaklik farki [°C]

[Genellikle 5°C] kullanilir = 5

* Tavan yüzeyinde kaç ayri kaplama var ? = 0

* Tavan çevresi yeterli isi yalitimina sahipmi ? [E/H] = E

Isil verim,x = 87.6 [%]

Panelin sogutucu (Chiller) yükü = -506.8 [Kcal/h]

Devam için ENTER tusuna basiniz

Boru seçimini yapiniz :

* Standard boru ic çaplari :

1 - 3/8 parmak	3 - 5/8 parmak
2 - 1/2 parmak	4 - 3/4 parmak

Kendi ölçülerinizi girmek istermisiniz ? [E/H] = E

* Boru iç çapi [mm] = 13
* Boru dis çapi [mm]= 17

* Boru malzemesi k katsayisi [kcal/h m°C] = 0.24

* Stokta mevcut standard boru uzunlugunu [m] ? =125

```
* Boru merkezleri arasindaki mesafe, M  [cm] ?    :
                  ( 10 cm ≤ M ≤ 30 cm ) = 15

Boru acikligi, 2W = 13.3 [cm]

Gerekli ortalama akiskan sicakligi = 15.0 [°C]

Devam için ENTER tusuna basiniz
```

```
 ANA TASARIM GiRDiLERI:
iç Sicaklik      = 22.0
Dis Sicaklik     = 33.0
Sogutma yükü     = -4000.0
Oda alani        = 100.00
Oda ölçüleri     = 11.00  x  10.00
Sogutma paneli adedi =   9


 Devam icin ENTER tusuna basiniz:
```

```
 * Odaya Ait Sonuç Bilgiler :
  --------------------------

* Toplam Panel Alani          = 108.0 [m²]
* Toplam Sogutma Kapasitesi = -3996.0[Kcal/h]
* Odanin Sogutucu Yükü      = -4560.9[Kcal/h]

 ENTER tusuna basiniz.
```

Bilgisayar Yardimi ile Tasarim Programi

```
e Adi      :SOGUTMA
l          :ANKARA
e Sahibi   :TÜBiTAK,MiSAG-12
h          :19/11/1993

GiRDiLER

ger konfor oda sicakligi,Ta          = 22.0    [°C]
Için dis hesap sicakligi,Td          = 33.0    [°C]
in toplam hissedilir sogutma yükü= -4000.0 [Kcal/h]
ndan Sogutma yapilabilecek alan    = 100.0   [m²]
l sogutma kapasitesi                 = -444.0    [Kcal/h]
Ölçüleri                             = 11.00 x 10.00[m]
tma panel adedi               ,np =    9    [ adet ]

RIM SONUÇLARI:

l numarasi                                    =  1
l yüzey alani                                 =12.0 [m²]
 merkezi üzerindeki sap kalinligi             =1.0  [cm]
am sap kalinligi                              =2.0  [cm]
yalitimi kalinligi                            =3.0  [cm]
s/Dönüs akiskan sicaklik farki                =5.0  [°C]
 iç capi                                      =13.00 [mm]
 dis çapi                                     =17.00 [mm]
 malzemesi isi iletim katsayisi               =0.24  [Kcal/h m°C]
 merkezleri arasi mesafe                   ,M =250.0 [ mm ]
lin kanatçik verimi                           =33.86 [ % ]

SICAKLIKLAR:
. yüzeyi etkin sicakligi                 ,Tp = 17.1 °C
num  tavan yüzey sicakligi             ,Tmin = 16.3 °C
num  tavan yüzey sicakligi             ,Tmax = 21.4°C
 dis yüzey sicakligi                     ,Tb = 16.3 °C
:ulmayan yüzeyler ortalama sicakligi,Tu = 23.2 °C
.ama akiskan sicakligi (pik yukte),Ts    = 14.2 °C  +/- 1
.tma Kapasitesi ve Yükler:
  yolu ile sogutma kapasitesi           ,qr = -23.9 [Kcal/h m²]
.im yolu ile birim sogutma kap.         ,qc = -14.6 [Kcal/h m²]
.m birim sogutma kapasitesi             ,qy = -37.0 [Kcal/h m²]
.  orani                             ,qr/qy = 64.6  [%]
.im orani                            ,qc/qy = 39.4  [%]
 verim                                   ,x = 87.6  [%]
 in isi kazanci (% olarak)           ,(1-x) = 12.4  [%]
 in sogutucu yükü                            = -506.8 [Kcal/h]

rolik Bilgiler:

an debisi                                 ,V = 1.7 [l/dk]
aki ortalama akiskan hizi               ,vs = 0.2  [m/sn]
Reynolds sayisi                         ,Re = 2603
an/Boru isi tasinim katsayisi            ,α = 1058 [Kcal/h m²°C]
sürtünme katsayisi                       ,τ = 0.0474
birim basinç kaybi (Düzeltilmis)         ,R = 9.20 [mm.ss/m]
m boru uzunlugu                          ,L = 52.8 [m]
```

ɔplam basinç kaybi (Düzeltilmis), ,H = 485.6 [mm.ss.]

daya Ait Sonuç Bilgiler :

Odadaki Toplam Panel Yüzeyi Alani =108.0 [m²]
Toplam Tavandan Sogutma Kapasitesi=-3996.0[Kcal/h]
Odanin Sogutucu Yükü =-4560.9[Kcal/h]

EK-10. KAR VE BUZ ERİTME TASARIM ALGORİMASI

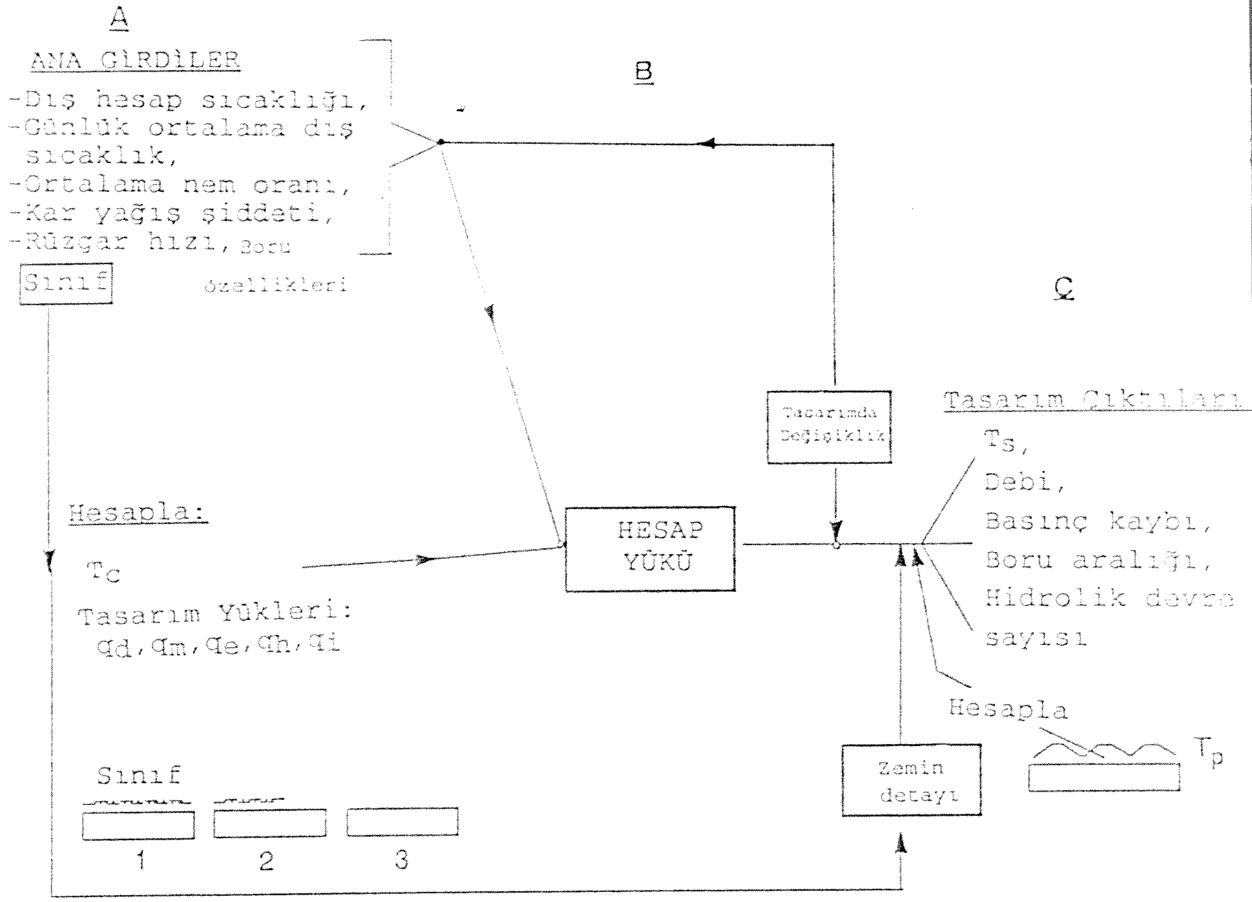
Giriş:

Panel ısıtma ve soğutma konusunda proje kapsamında yapılan teorik ve sayısal çalışmalar ile ASHRAE kuruluşunun 6.1 sayılı teknik komitesi ile olan ilişkiler çerçevesinde kar ve buz eritme konusuna da eğinilmiştir. Bu konu Ülkemizde ne teorik ne de uygulama seviyesinde yeterince bilinmemektedir. Ufak çaplı sistemler dışında uygulama da yoktur. Ankara Otelinin yaya giris yol rampası ile Abant da inşa edilen yeni bir otel açık hava havuz kenarı ve yolunda bu tür uygulamalar var isede henüz yeterli bir hesap algoritması geliştirilmemıtir.

Yurt dışında ise büyük ölçekli uygulamalar,örneğin Zürih hava alanı,ABD de New York Lincoln tüneli giriş ve çıkış viyadüklerinde kar ve buz eritme sistemleri yol zemini hidrolik olarak ısıtılarak gerçekleştirilmektedir. Elektrikli ısıtma sistemleri de mevcuttur. Uluslararası düzeyde tek kayda değer yayın ASHRAE El kitabında yer alan bir bölümdür. Kılkış bu bölümün yetersiz olduğunu öne sürerek yeniden ele alınmasını önermiş (Bakınız Ek-5,yayın no.4) iki ayrı uluslararası makalesi ASHRAE Transactions Cilt 1200 de basılmak üzere kabul edilmiştir. (Ek 5,yayın no.12 ve 13).

Aslen kar eritme sistemi döşemeden ısıtma sistemi ile çok benzerdir. Kar eritmede iç bir hacmin ısıtılması yerine dış bir yüzeyin ısıtılarak kar ve buzdan korunması amaçlanmaktadır. Bu nedenle döşeme içi ısı transfer modeli ayni kalmakta fakat,panel yükü olarak ısıtma yükü yerine kar veya buz eritme yükü hesaplanmaktadır. Bunun ötesinde çalışma esnasında yüzeyde kar ile örtülü kısım olup olmaması,var ise ne kadar bir bölümünün kar ile örtülü olduğu,yüzeydeki ısı kayıplarını ve kanatçık verimini etkiler. Yüzey tasarım sıcaklığı buna göre seçilmelidir. Ancak,ısı kayıplarıda yüzey sıcaklığna bağlıdır. Bu etkileşim problemi daha karmaşık bir duruma getirir. ASHRAE 6.1 teknik komitesi ile ilişkiler çerçevesinde, ASHRAE Handbook: HVAC Applications,kısım 45 in yeniden düzenlenmesi,konunun proje ürünü olarak Ülkemizde tanıtılarak yeterli bir hesap algoritmasına ve bilgi birikimine kavuşturulması amaçları ile kanatçık modeli kar ve buz eritme sistemlerine adapte edilmiş ayrıca Ülkemiz Meteorolojik kar yağışı rasat verilerinin türlerine uygun bir kar ve buz eritme yükleri hesap algoritması geliştirilmiştir.
Bu algoritma 10.3 sayılı bölümde verilmiştir.

Etkileşimli bir bilgisayar yardımı ile tasarım programı ise geliştirilerek program disketi ekte sunulmuştur. Tanıtımı ve kullanıcı kılavuzu,örnek bir uygulama ile birlikte 10.3 sayılı bölümde verilmiştir. Genel akış şeması Şekil 10.1 de verilmiştir.



Şekil 10.1. Kar Eritme Bilgisayar Programı Genel Akışı.

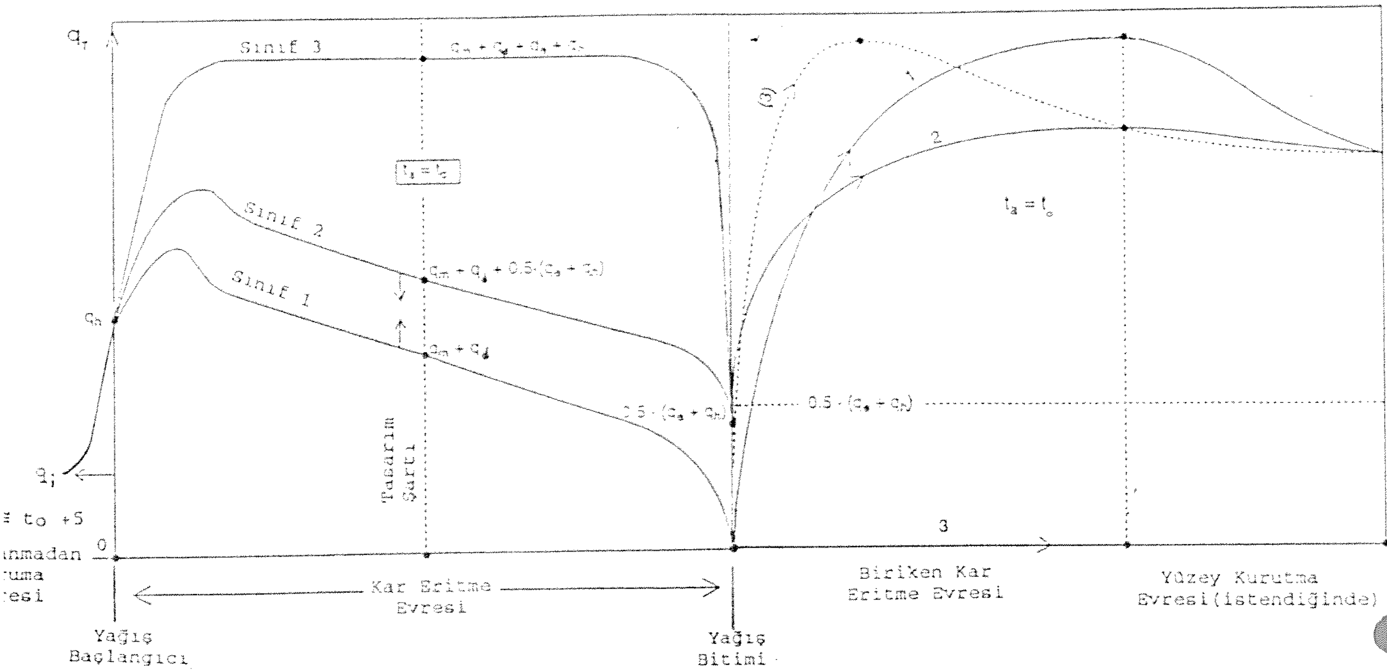
10.1. Analitik Yaklaşım:

Projede iki adet çözüm geliştirilmiştir. Bunlar:

1-Kar ve Buz yükü hesap algoritması,
2-Isıtılan zemin kanatçık modeli. dir.

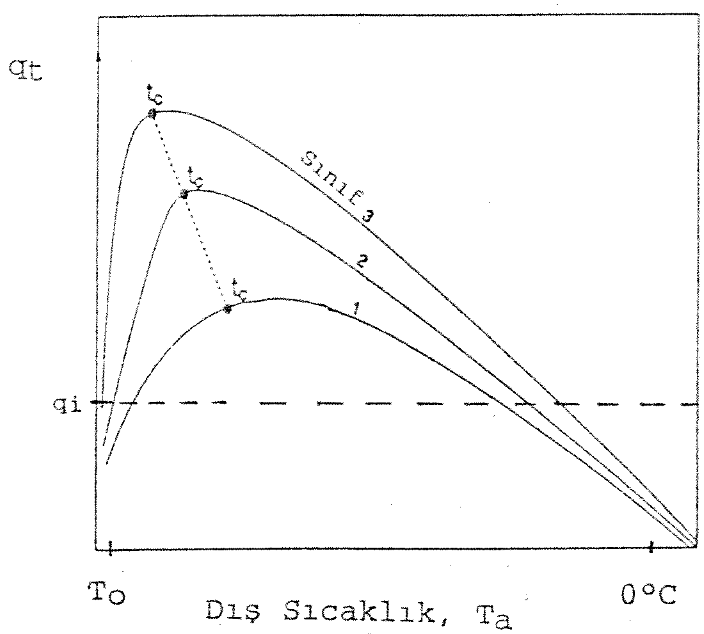
10.1.1 Kar ve Buz Eritme Yükleri

Kılkış geliştirdiği algoritmayı Ek-5 de verilen 12 no.lu yayınında, izah etmiş, ayrıca 10.3 bölümünde özetlenmiştir. Bu algoritma ile sadece kar yağışı esnasındaki performans ve yükler modellenmemiş, yüzeyde biriken karın yağış sonrasında eritilmesi işlemi de modellenmiştir. Zira, bazı durumlarda bu yük kar yağışı esnasındaki yükten fazla olabilmektedir. Bu durum Şekil 10.2 de gösterilmiştir.



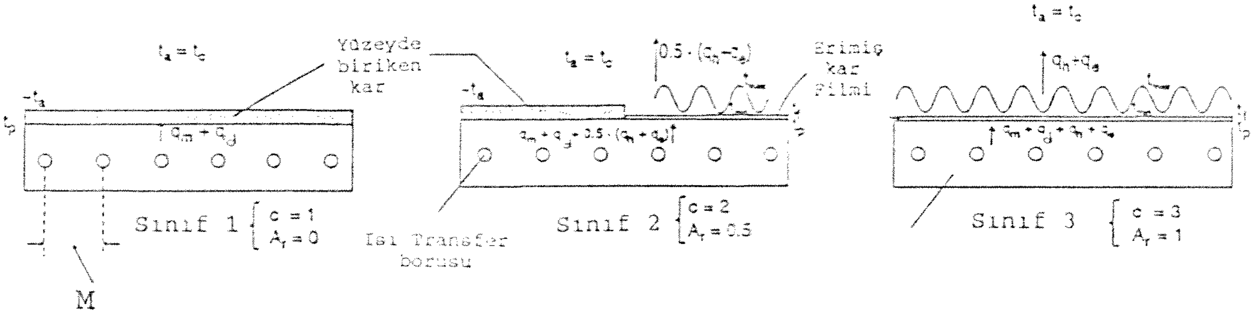
Şekil 10.2. Kar ve Buz Eritme Yüklerinin Evrelere Göre Değişimi.

Genellikle kar yağışı esnasında hava sıcaklığı dış hesap sıcaklığına inmez. Kar yağış esnasındaki hesap değerinin ayrı hesaplanması gerektiğini öne süren Kılkış,yeni bir yöntem geliştirmiştir. Bu yöntemde T_C nin performans sınıfına da bağlı olduğu öne sürülmektedir (Şekil 10.3).



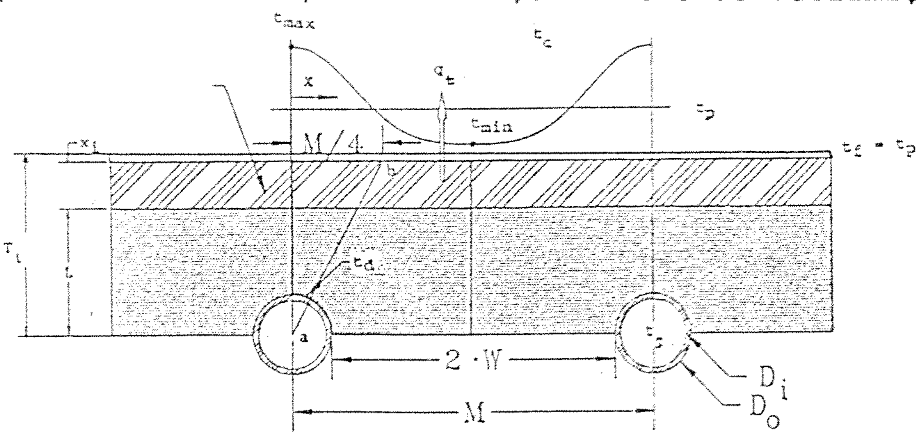
Şekil 10.3. Kar Yağışı Esnasında Dış Sıcaklık Hesap Değeri.

Kar eritme performans sınıfına (1,2,3) bağlı olarak kar eritme yüzeyindeki şartlar Şekil 10.4 de gösterilmiştir. Sınıf-1 de yüzey yağış esnasında tamamen kar ile örtülüdür. Sınıf-3 de ise yüzeydeki kar tamamen erimiş vaziyettedir. Ancak henüz tam olarak kurumamıştır. Yüzeyde ince bir su filmi olduğu kabul edilmektedir.



Şekil 10.4. Performans Sınıflarına Göre Yüzey Şartları.

Sınıf-1 de yüzey kar ile örtülü olduğundan kanatçık verimi 1 alınabilir. Sınıf-3 de ise kanatçığın davranışı döşeme panelininki gibidir ve yüzeyde bir sıcaklık profili mevcuttur. Adapte edilen kanatçık modeli Şekil 10.5 de verilmiştir.



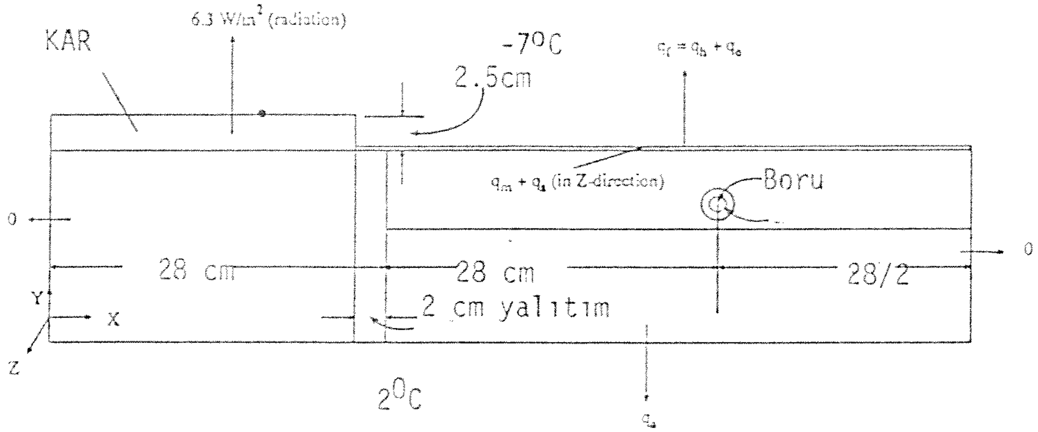
Şekil 10.5. Kar Eritme Paneli Fin Modeli.

Bu model sayesinde T_{min} değeride kolaylıkla hesap edilerek, boru aralığının fazla olduğu durumlarda yüzeyde buzlanma olup olmayacağı tahmin edilebilir. Geliştirilen bilgisayar modeli

$T_{min} > 0°C$ şartını yerine getirerek bu sorunu çözer.

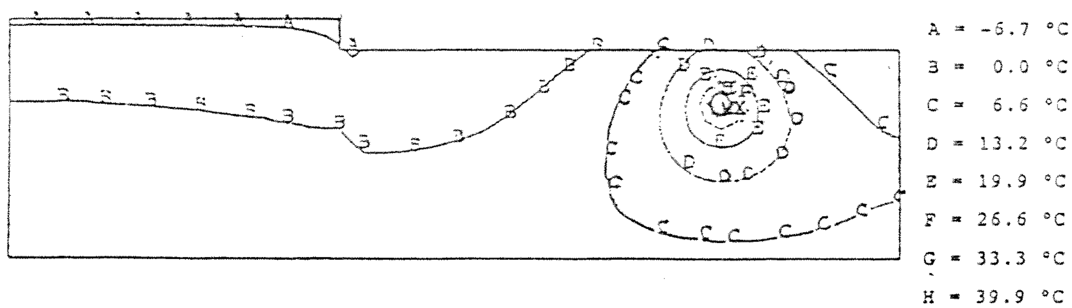
10.2. Sayısal Yaklaşım:

Kar eritme paneli ANSYS program pakedi kullanılarak modellenmiş ve analitik çözümler mukayese edilmiştir. Örnek bir çözümün anahatları Şekil 10.6,10.7 ve 10.8 de gösterilmiş,detayları ise Ek-5,14 no.lu yayında verilmiştir.

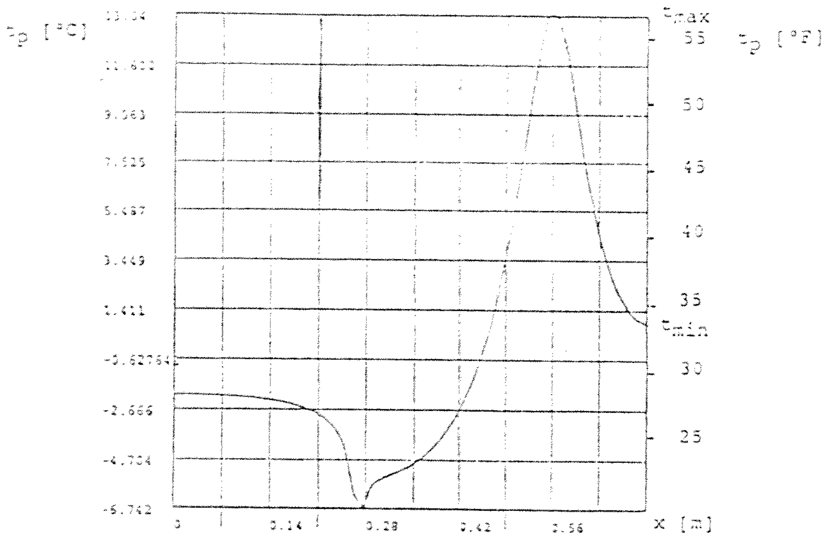


Şekil 10.6. Kar Eritme Paneli İdealizasyonu.

Duyulabilir ısı ve kar eritme ısısı yüzeyi terk etmediklerinden kanatçık veriminde etkili değildirler. ANSYS modelinde bu yükler (Z) yönünde yani panel düzlemine göre panel içersinde kalır (kar tarafından yutulur).



Şekil 10.7. Kar Eritme Paneli Isı Transferi Sonlu Elemanlar Çözümü.



Şekil 10.8. Kar Eritme Paneli Yüzeyinde Sıcaklık Dağılımı.

10.3. TÜBİTAK MİSAG 12 PROJESİ BİLGİSAYARLA KAR ERİTME TASARIM PROGRAMI KULLANICI KILAVUZU

Prof.Dr.İbrahim B.Kılkış

Giriş

Kışın kar ve buza tolere edilemeyen yüzeyler vardır. Örneğin uçak pistlerinde,çatı üzeri helikopter alanlarında,yol kavşaklarında,tretuarlarda can ve mal güvenliği için kar ve buzun belli bir performans seviyesinde eritilmesi gerekir.
Bu performans seviyesi ASHRAE [1] tarafından üç sınıfta tanımlanmıştır:

Sınıf 1: Kar yağışı sürerken kar yüzeyde birikir. Yağıştan sonra zemin ısıtılmaya devam olunarak kar eritilr. En ucuz yöntem bu performans sınıfında sağlanır. Bunun nedeni kar yağarken,yüzeyin kar ile örtülmesine müsaade edilmesi sonucu atmosfere olan taşınım,ışıma ve buharlaşma kayıplarının önlenmesidir. Yağış sırasında kar sadece erime noktasına getirilir. Kar yağışını takiben hava çoğunlukla soğuduğundan kapasite tayininde dikkat edilmelidir. Aksi halde,kar erimez ve buz şeklinde yüzeyde birikerek daha da tehlikeli bir durum arzedebilir. Bu sınıf daha çok yaya yolu,garaj,otopark girişi gibi yerlerde tatbik edilir.

Sınıf 2: Kar yağışı sırasında yüzeyin yaklaşık % 50 si eritilir. Bu sınıf genellikle daha kritik yerlerde,örneğin depo giriş/çıkış rampalarında,havaalanlarında,apron ve kavşaklarda tatbik edilir.

Sınıf 3: Daha kar yağmaya başlar başlamaz zeminde karın erimesi isteniyor ise bu sistem seçilir. Örneğin çatıdaki bir helikopter platformunda en ufak bir kaymaya müsaade edilemez. Bu taktirde Sınıf 3 seçilir.

Kar veya buz eritme işlemi yüzey altındaki zeminin,örneğin betonun veya dolgu malzemesinin içersine yerleştirilen özel sıcak su borusu veya elektrik direnç telleri vasıtası ile yürütülür. Sistem ısı yükünün makul bir seviyede tutulabilmesi için kar sadece eritilir,tamamı buharlaştırılmaz. Bu nedenle yeterli ve uygun bir drenaj sistemininde tasarımlanması ve tatbiki gerekir. Aksi taktirde,eriyen kar kar eritmesi tatbik edilmeyen bir yerde birikerek donabilir.

Sistem manuel veya gerekirse otomatik kontrollü olarak işletilebilinir. İyi bir performans sergilenebilmesi için sistem daha kar yağmaya başlamadan bir iki saat önce devreye alınmalıdır. Böyle bir usul yerine özellikle sınıf ve 2 ve 3 tatbikinde yüzeyi dondan da koruma maksadı ile sistem sürekli fakat düşük bir kapasitede devrede tutulur. Bunun için gerekli ısıtma yüküne Buz Eritme yükü adı verilir.

Teori:

Tasarım için şu aşamalar takip edilir:

1-Kar eritme Sınıfının tesbiti:

Tatbikatın ne kadar kritik olduğu göz önünde tutularak karar verilir.

2- Kar eritme ve dondan koruma/buz eritme yüklerinin hesabı:

Bu yüklerin bulunabilmesi için dış hesap sıcaklığı,kar yağışı esnasındaki dış sıcaklık,yüzeydeki rüzgar hızı ve o yöredeki saatlik kar yağışı hesap miktarının bilinmesi gerekir. Türkiye de kar yağış rasatları saatlik değer yerine günlük toplam değer olarak verilir. Hesaplar için gerekli (s) yani cm su cinsinden bir saat içindeki kar yağışı hesap miktarı Kılkış [2] tarafından verilen aşağıdaki denklem ile kar eritme performans sayısı ve dış sıcaklığa bağlı olarak bulunabilir:

$$s = G/24 * C * Y/1000 \qquad [1]$$

Burada, G o yöredeki günlük kar yağış miktarı [cm/24 saat],C kar eritme performans sınıf sayısı [1,2 veya 3], Y ise kar yağışı esnasında dış sıcaklığa (T_C) bağlı olarak değişen kar yoğunluğudur. Bu ilişki Adlam [3] in verileri kullanılarak Kılkış [2] tarafından:

$$Y = 41.7 + 0.96 * (1.8*T_C + 32) + 0.043 * (1.8*T_C + 32)^2 \qquad [2]$$

şeklinde verilmiştir.

Genellikle kar yağışı esnasında hava sıcaklığı hesap değerine (T_O) inmez. Kar yağışının yoğunluğu ile bu sıcaklık arasında bir ilişki olduğunu öne süren Kılkış [2],performans sayısının gerektirdiği emniyet seviyesine de bağlı olarak ısı yükünün bulunması için düzeltilmiş bu dış hesap sıcaklığını şu şekilde vermiştir:

$$T_C = T_O + (1 - T_O)/(1.3 + 2.4 * Ar) \qquad [3]$$

A_r,yağış esnasında kar eriyen yüzeyin bütün yüzeye olan nisbetidir:

Performans tarifine bağlı olarak:

Sınıf-1 de Ar : 0 (bütün yüzey kar ile örtülü),
Sınıf-2 de Ar : 0.5 (yüzeyin sadece %50 si kar ile örtülü),
Sınıf-3 de Ar : 1 (bütün yüzey açık).

Diğer hesap girdileri örnek proje hesabında ve bilgisayar kullanım seansında gösterilmiştir.

Kar eritme yükü hesabı takibeden bölümde verilmiş olup buz eritme yükü ise:

$$q_i = (0.83 * V_r + 16) * (T_o + 5) \qquad [4]$$

Burada son terimin mutlak değeri alınır.

KAYNAKÇA

[1]-Snow Melting, <u>ASHRAE Handbook:</u>HVAC Applications, Chap.45, Atlanta, GA, 1991.

[2]-Kılkış, İ.B., Design of Embedded Snow Melting Systems Part.1 Heat Requirements : An Overall Assessment and Recommendatios <u>ASHRAE Transactions</u>, (100), paper no: 3778, 1994.

[3]-Adlam, T.N. Snow Melting, The Industrial Press, New York, 1950.

[4]-Kılkış, İ.B., Design of Embedded Snow Melting Systems Part.2 Heat Transfer in the Slab: A Simplified Model, <u>ASHRAE Transactions</u>, (100), paper no: 3779, 1994.

Kış aylarında genel kullanıma açık yollarda, rampalarda, kaldırımlarda, havaalanı pistleri, köprülerde ve park yerlerinde, ulaşım emniyeti, sürekliliği ve genel emniyet açılarından yağan karın daha birikip donmadan eritilmesi için en uygun sistem zemin altından ısıtma yapmaktır. Prensip itibarı ile yerden ısıtma uygulamasının aynısı olup diğer sistemlere göre çok ucuz ve emniyetlidir. Amerika'da Boston kentinde şehirlerarası otoyol üzerinde köprü üstleri (toplam 20.000 m^2), kavşaklar bu sistemle kardan ve buzdan korunmaktadır İsviçre'de ise Zürih havaalanı pist ve apron yolları bu sistemle kışın kar ve buzdan korunmaktadır. Kar eritme sistemi ısıtma kapasitesinin tesbitinde 4 ana unsur önemlidir

a- Kar yağış hızı, b- Dış sıcaklık, c- Rüzgar ortalama hızı, d- Nem oranı.

En hızlı kar yağışı genellikle ılık hava sıcaklıklarına rastlar. Bu nedenle kar yağışı için tasarım hava sıcaklığı aşağıdaki formülden bulunur:

$$T_c = T_o + \frac{(1 - T_o)}{1{,}3 + 2{,}4 \times A_r} \qquad (5)$$

Burada, T_o dış hesap sıcaklığıdır.

Böyle bir sistem daha kar yağışı başlarken çalışmaya başlatılmalıdır. Bu nedenle gece olabilecek yağışlar için elektromekanik bir uyarı düzeni de kurulabilir. Isıl ataletin azaltılması amacı ile müsaade edilen zemin basıncını karşılamak kaydı ile borular yüzeye olduğunca yakın konulmalıdır. Aksi taktirde özellikle soğuk günlerde kar yağmasa da sistemin devrede kalması gerekebilir. Yüzey yeterince sıcak ise kar yağar yağmaz, önce + 0^{o}C'ın üzerine çıkarak erir, eriyen su ise buharlaşır. Bu nedenle hem ısı hem kütle transferi söz konusudur.

Gerekli q_T yükü:

$$q_T = q_d + q_m + A_r \cdot (q_e + q_h) \qquad (6)$$

Burada

q_d = Duyulabilir ısı (kcal/h m^2)

q_m = Kar eritme ısısı (kcal/h m^2)

q_e = Buharlaşma ısısı (kcal/h m^2)

q_h = Atmosfere ışıma ve taşınım ile ısı kaybı (kcal/h m^2)

Burada A_r, kar yağarken eritilebilen yüzeyin eritilmesi istenen toplam yüzeye olan oranıdır. Kar yağmaya başlar başlamaz eritme başlayacak ise A_r, 1 alınır. Aksi taktirde 0.5 alınır. Kar yağdığı sürece karın yüzeyde birikmesine müsaade edilecek ise A_r 0 alınır.

$$q_d = 5.0 \cdot s \cdot (0^oC - T_c) \quad (7)$$

Burada s, saatteki cm-su cinsinden kar yağışıdır.

$$q_m = 794 \cdot s \quad (8)$$

$$q_e = 2632 \cdot (0.0125 \cdot V_r + 0.055) \cdot (0.185 - 0.0029 \cdot P_v) \quad (9)$$

Burada; V_r = Zemin üzerindeki rüzgar hızı (m/s), P_v = nemli havanın buhar basıncı (mm.ss)dır.

Yüzeyde oluşan ışıma ve taşınım ısı kaybı genel ısı transferi denklemleri ile tariflenebilirse de yüzey sıcaklığı yerine erimiş kar filminin ortalama sıcaklığı T_f kullanılmaktadır. Yaklaşık 2 mm su kalınlığı öngörüldüğünde bu sıcaklık yaklaşık olarak bulunabilirse de pratikte + 1°C alınması kafidir.

$$q_h \cong 55.6 \cdot (0.0125 \cdot V_r + 0.055) \cdot (1^oC - T_c) \quad (10)$$

Eğer T_c 1 °C'dan fazla ise sadece q_m ve q_e değerleri gözönünde tutulur.

Sıcak su devresinde ise antifriz katkılı su dolaştırılır. Ortalama sıvı sıcaklığı, M = 150 mm ve 40 mm toplam şap kalınlığı için aşağıdaki denklem aracılığı ile bulunabilir:

$$\overline{T}_s \cong 0.035 \cdot q_s + 15^\circ C \tag{11}$$

Yukarıdaki ifade yerine,daha hassas neticeler için KAR ERiTME Programı kullanı

Sistem yükü q_s ise % 30kayıp düşünülerek, q_T değeri 1.3ile çarpılarak bulunabilir. Su sıcaklığı giriş-çıkış farkı 10°C alınır. Eriyen kar sularının etkin drenajı sağlanmalıdır. Aksi taktirde biriken su sistem performansını olumsuz yönde etkiler. Donmaya karşı Etilen (Propilen) - glikol kullanılabildiği gibi su yerine hafif yağ da kullanılabilir. Bu durumda sistem -40°C'a kadar korunmuş olur. Etilen-glikol kullanıldığında ise bu koşul için hacımsal olarak % 51.2 Etilen-glikol kullanılmalıdır. Antifriz veya hafif yağın zorunlu olarak kullanılmasının en büyük dezavantajı borulardaki sürtünme kayıplarını ve gerekli pompa kapasitesini arttırmasıdır.

A yüzeyine haiz bir modülasyondaki gerekli pompa debisi:

$$\dot{V} = \frac{q_s \cdot A}{10^\circ C \cdot \gamma \cdot C_p \cdot 3600} \quad [m^3/s] \tag{12}$$

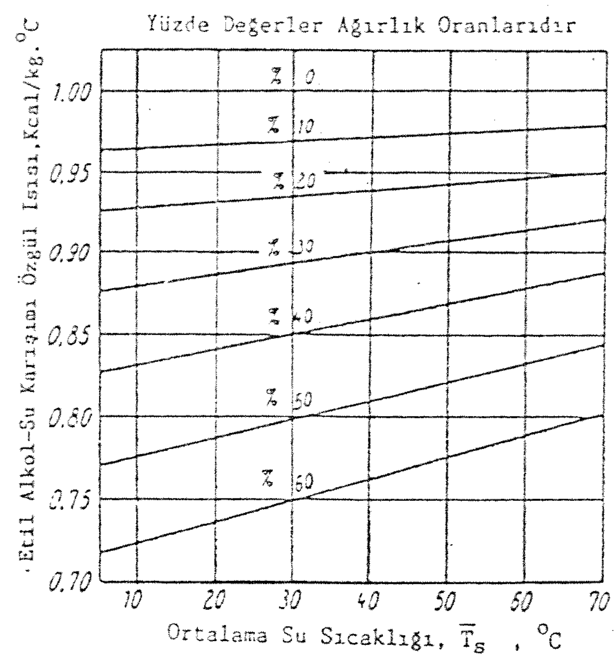
Burada γ ve C_p antifriz karışımının özgül ağırlığı ve özgül ısıları olup sırası ile Çizelge 1 ve Şekil 1 'den okunmalıdır.

Çizelge 1 . Antifriz Karışımının Özgül Ağırlığı, γ

($\overline{T}_s \cong 45°C$)

Karışım maddesi	Donmaya karşı korunma (°C)	Karışım oranı (Hacımsal) (%)	γ (kg/m^3)
Etilen-Glikol	- 18°C	31.4 %	1030
	- 29°C	42.7 %	1044
	- 40°C	51.2 %	1054
Hafif yağ(1) (10 numara)	- 40°C	100 %	956

(1) $C_p \cong 0.47$ kcal/kg



Şekil 1 . Etilen-Glikol, Su Karışımının Özgül Isısı

ÖRNEK

Ankara'da yeni yapılacak bir havaalanı için pistin 500'er metrelik iniş ve kalkışta kritik iki ucunda yerden ısıtma ile kar eritmesi yapılacaktır. Rüzgar yönüne göre her zaman sadece pistin bir tek yönü kullanıldığı için sistem ısıtma yükü, pistin sadece bir tek ucu için seçilecektir. Yerden ısıtma, pist eninin 25 metresi için düşünülmektedir. Kritik bir uygulama olması nedeni ve ekonomi gözönünde tutularak $A_r = 0.5$ seçilmiştir.

Ankara için tasarım dış sıcaklığı -12^oC'dır. Ankara'da en hızlı rüzgar bahar aylarında estiği için bu değer hesapta gözönünde tutulmayarak Ocak ayı en hızlı rüzgar ortalaması alınmıştır. Rasat yüksekliği 10 m olan bu değer, toprak hizası için düzeltildiğinde:

$V_r = 12$ m/s

Ocak ayı nisbi nem ortalaması olan % 78 değeri kullanılarak nemli havanın buhar basıncı bulunur (34 mm.ss). Ocak ayında tesbit edilen en şiddetli yağış hızı 1.0 cm-su/h'dir (yaklaşık 15cm kar). Bu değerler kullanılarak:

$T_c = -6.8^oC = -7^oC$ alındı.

$q_d = 35$ kcal/m^{2}h

$q_m = 794$ kcal/m^{2}h

$q_e = 58$ kcal/m^{2}h

$q_h = 91$ kcal/m^{2}h

$A_r = 0.5$ alınarak;

$$q_T = 35 + 794 + 0.5\,(58 + 91) = 904 \text{kcal/m}^2\text{h}$$

Eğer pistin ısıtılan mahallerinde kar yağışı esnasında hiç kar birikimi istenmiyor ise:

$$q_T = 35 + 794 + 1\,(58 + 91) = 978 \text{ kcal/m}^2\text{h}$$

İlk değerlere (A_r = 0.5) itibar olunarak,

$$q_s = 904 \times 1.3 = 1175 kcal/m^2h$$

$$Q_s = 500 \times 25 \ m^2 \times q_s \cong 15 \cdot 10^6 \ kcal/h$$

$$\overline{T}_s \cong 56^oC \qquad \text{(Denklem 11)}$$

Sistemde - 40°C'a kadar emniyetli 51.2 % etilen-glikol kullanılacaktır. Bu durumda:

$$\gamma = 1054 \ ; \ C_p \cong 0.83$$

Pistin her ucundaki modülasyon sayısı N ; (M = 150 mm için)

$$N = \left(\frac{500 \times 25}{150/1000} \times 1.1\right) \cdot \frac{1}{120} = 763 \ ; 765 \text{ alındı.}$$

Burada standard boru uzunluğu 120 m.dir.

Her modülasyondaki sistem yükü: $15 \cdot 10^6 / 765 = 19608$ kcal/h

Dolayısı ile her modülasyondaki debi =

$$\frac{19608}{10^oC \cdot 1054 \cdot 0.83 \cdot 3600} = 6 \times 10^{-4} m^3/sn.$$

Toplam sistem yükü ve emniyet payı ile 5 adet 150 m²'lik kazan seçildi. Her bir kazan 153 modülasyonu besleyecektir.

Her modülasyondaki boru uzunluğu (M = 150 mm) 120 m olmaktadır.

Bilgisayar Programı ile Örnek Uygulama

Kılkış [4],ASHRAE [1] el kitabında yetersiz olan sistem tasarımına ilişkin esasları TÜBİTAK MİSAG-12 projesi çerçevesinde geliştirerek etkileşimli bir bilgisayar programı hazırlamıştır. Bu bilgisayar programı ile aşağıdaki tasarım parametreleri değiştirilerek ısıtılan döşeme için optimum çözümü elde etmek mümkündür:

- Boru merkezleri arası mesafe,
- Boru malzemesi ve ölçüleri,
- Standard boru uzunluğu,sistemdeki hidrolik devre sayısı,
- Isıtılan zemin malzemesi ve derinlik ölçüleri,
- Kenar ve arka ısı yalıtımı,
- Zemin üzerindeki kaplamalar (örneğin beton üzeri asfalt gibi)
- Devredeki akışkan gidiş/dönüş sıcaklı farkı.

Kar ve buz eritme yükleri ise önceki bölümde verilen esaslar dahilinde bilgisayar programı tarafından etkileşimli olarak hesaplanır.

Ankara,Esenboğa hava alanında kar eritme sistemi için Sınıf-2 performansında bir tasarım seansı düzenlenerek,ekran görüntüleri ve bilgisayar çıktısı sırası ile verilmiştir:

İlk ekran görüntüsünde proje ana bilgileri girilir. Bunlar sırası ile Proje Adı,Mahal,Proje Sahibi ve Proje Tarihleridir. Esenboğa için TS 2164 den Ankara ili değeri olan -12°C dış hesap sıcaklığı olarak girilmiştir. Ekrana gelen Kar Eritme Performans Katsayısı ile ilgili açıklamaları takiben performans katsayısı Sınıf-2 olarak girilmiştir. 3 sayılı denklem aracılığı ile program,kar yağışı sırasındaki dış tasarım sıcaklığını -7°C olarak vermiş,kullanıcı da bu değeri aynen girmiştir. Don mevsiminde günlük ortalama dış sıcaklık için (T_o + 5°C) ifadesi ile bulunan -7°C default değeri girilmiştir. Bu ifade genellikle emniyetli bir buz eritme yükü tesbiti için yeterlidir. Daha hassas veriler,Devlet Meteoroloji İşleri Gn.Md.nün Rasat Bültenlerinde "Ortalama Düşük sıcaklık satırından okunabilir. Örneğin Ankara için bu değer -3.5°C dır. Örnek bir tablo ekte verilmiştir. -7°C olan T_c değerine tekabül eden kar yoğunluğu ise 76.5 kg/m^3 dür.

Rasat bültenlerinden,saatlik kar yağış miktarının doğrudan bulunması mümkün değildir. Bu amaçla,eğer günlük toplam kar yağışı verilmiş ise 1 numaralı denklem kullanılır. Buna mukabil,Ankara için bültende saat 7 14 ve 21 de sırası ile Ocak ayında su eşdeğeri olarak 1.66,1.06,ve 1.1 cm su/saat değerleri verilmiştir. Günlük ortalaması ise 1.27 cm su/saat dir. Ayrıca Ankara da en yüksek kar örtüsü kalınlığı Ocak ayında 33 cm olarak verilmiştir. Bu miktarın bir günde ve 4 saat içinde yağdığı varsayımı ile kar yağışı tasarım değeri Sınıf-2 performansı için şu şekilde bulunur:

$$s = 33/4 * 2 * 76.5/1000$$

$$= 1.26 \text{ cm su/h.}$$

Bu iki değer birbirine oldukça yakındır.
Örnek uygulama girdisi olarak 1 cm (10mm) kullanılmıştır. Ekran menüsünde hafif,orta,kuvvetli kar yağışı kategorileri kullanılmayarak 4 seçeneği ile 10 mm değeri girilmiştir. Rüzgar şiddeti ise yer seviyesinde düzeltilmiş değer olan 12 km/h alınmıştır. Bu düzeltme,ASHRAE Fundamentals El kitabı,bölüm 14 deki esaslar dahilinde yapılır:

$$V_r = V' * ((H+2)/10)^a * A_o \qquad [13]$$

Burada V' meteoroloji rüzgar rasat değeri, H kar eritme yüzeyinin yer seviyesinden olan yüksekliği, a ve A_o ise çevre şartlarına bağımlı düzeltme katsayılarıdır.

Çizelge.10-1. Rüzgar Rasat Değeri Düzeltme Katsayıları.

	KATSAYILAR	
ÇEVRE ŞARTLARI	a	Ao
Açık saha	0.15	1.0
Korumalı	0.28	0.6
Şehir içi	0.40	0.35

Ankarada günlük ortalama rüzgar hızı rasat değeri V',Ocak ayı için 3.3 m/sn (11.5 km/saat) dir. Açık saha katsayıları kullanıarak Vr degeri 10 km/saat bulunmakla birlikte,bu tasarım seansında daha emniyetli bir değer olarak 11.5 değerini yuvarlatarak 12 km/saat olarak girmiştir.

Havanın nisbi nemi Ocak ayı için 0.78 olarak Meteoroloji rasatlarından okunmuştur.

Bu değerler bilgisayara girildikten sonra,program Sınıf-2 için kar eritme yükünü 903.4 Kcal/h m^2,buz eritme yükünü ise 196.5 Kcal/h m^2 olarak bulmuştur. El ile yapılan önceki hesap örneğinde kar eritme yükü 904 Kcal/h m^2 olarak bulunmştu. Kullanıcı ekrandan bu değerleri aynen veya değiştirerek girebilir.

Program daha sonra toplam yüzey alanını ve kar eritmesi için kullanılan zeminin detay bilgilerini sorar. Burada döşeme olarak tanımlanan,beton.sıkıştırılmış çakıl,toprak gibi ısı transfer borularının muhafaza edildiği kesittir.

Yüzeyde sıcaklık farkını azaltmak için akışkan gidiş-dönüş sıcaklık farkının en çok 10°C alınması tavsiye edilir.
Yüzeyde ayrıca bir veya daha fazla kaplama katmanı var ise sırası ile bunların kalınlığı ve ısı iletim katsayıları girilir.
Bu örnekte 2 cm kalınlığında asfalt kaplaması bulunmaktadır.
Program çevre ve alt ısı yalıtımı verilerine dayanarak sisteme arzolunan ısının % 91 lik bölümünün yüzeye ulaştığını geriye kalan bölümün ise çevreye kaybolduğunu belirtmektedir. Bu nedenle,gerekli performansın gerçekleşmesi için kazan yükü bu verime göre düzeltilmiştir.

Daha sonra ısı transfer borusunun boyut ve ısı iletim katsayısı girilmekte,ayrıca standard boru boyu bildirilmektedir. Kullanıcı boru merkezleri arası mesafeyi 15 cm olarak girmiştir.Bu boru boyuna,aralığa ve toplam yüzey alanına göre hidrolik devre sayısı default olarak 764 olarak hesaplanmış,kullanıcı bunu 765 olarak girmiştir.

Program tüm verileri kullanarak gerekli ortalama sıvı sıcaklığını 51.7°C olarak bulmuştur. Bu değerin el ile yapılan hesaba göre az oluşunun nedeni Denklem 11 in yaklaşık ve emniyetli bir ifade olmasıdır. Bilgisayar programı ise girilen bilgileri detaylı olarak değerlendirmektedir. Örneğin boru malzemesinin k değeri daha düşük seçilse idi T_S değeri artacaktı.
Kullanıcı bu ve ekrandaki diğer sonuçlar ışığı altında ara geri dönüşlerini kullanarak tasarım değerlerinde değişiklik yapabilir. Bu suretle optimum çözüme ulaşılır.

Program ekran çıkısı olarak detaylı veya özet bilgi opsyonlarına sahiptir. Detaylı bilgi opsyonunda üç türlü bilgi mevcuttur. Bunlar sırası ile,sıcaklıklar,kapasite ve yükler,ve her bir paralel devre için hidrolik hesap neticeleridir. Hidrolik hesaplar %50 Propilen Glikol/Su karışımı ısı transfer akışkanı için yapılmaktadır.

Cizelge 10-3 de ekran çıktıları, Çizelge 10-4 de ise yazıcı çıktısı verilmiştir.

PROJE ANA BiLGiLERi

PROJE ADI = KARERiTME

MAHAL = Esenboga,ANKARA

PROJE SAHiBi= TÜBiTAK

TARiH = 9/11/1993

ENTER TUŞUNA BASINIZ

* DIŞ HESAP SICAKLIGI ? [°C] = -12

* KAR ERiTME PERFORMANS KATSAYISI [ASHRAE] ? =

1-SINIF 1: YagIş anInda kar yüzeyi tamamen örter.
Yüzeyde biriken kar bilahare eritilir.

UYARI = Genellikle kar yagIşIndan sonra
sogur ve yüzeydeki kar donabilir.

2-SINIF 2: Kar yagarken toplam yüzeyin %50 'si eritilir.

3-SINIF 3: Kar yagarken yüzeyin tamamI eritilir.

* KAR ERiTME PERFORMANS KATSAYISI ? [1,2,3] = 2

BEYAZ RAKAMLAR DEFAULT DEGERLERiDiR

* KAR YAGIŞI SIRASINDA DIŞ TASARIM SICAKLIGI ? [°C] = -7

* DON MEVSiMiNDE GÜNLÜK ORTALAMA DIŞ SICAKLIK ? [°C] = -7

* SAATLiK ORTALAMA KAR YAGIŞ SiDDETi KATEGORiSi :

1-HAFiF KAR YAGIŞI örnegin ≤ 2 mm/saat [Su eşdegeri]
2-ORTA ŞiDDET örnegin ≤ 4 mm/saat [Su eşdegeri]
3-KUVVETLi KAR YAGIŞI örnegin ≤ 6 mm/saat [Su eşdegeri]
4-DiGER

* KATEGORi ? [1,2,3,4] = 4
* Su eşdegeri saatlik kar yagIsI ? [mm su/saat] = 10

RÜZGAR ŞiDDETi :

* 1- HAFiF Şiddeti [< 8 km/saat]
* 2- ORTA Şiddet) [8 ≤ HIZ ≤25 km/saat]
* 3- KUVVETLi [>25 km/saat])
* 4- DiGER

* RÜZGAR ŞiDDETi [1,2,3,4] = 4

* RüZGAR HIZI [km/saat] = 12

* HAVA NiSBi NEM ORANI ? = 0.78

* Kar Eritme IsI YÜKÜ = 903.4 [Kcal/h m²]

* Buz Eritme IsI YÜKÜ = 196.5 [Kcal/h m²]

ENTER TUŞUNA BASINIZ

* KAR eritme yükünün dökümünü istermisiniz ? [E/H] = E

*	Hissedilir yük	= 35.0 [Kcal/h m²]
*	Kar eritme yükü	= 796.8 [Kcal/h m²]
*	Buharlasma kaybI	= 28.8 [Kcal/h m²]
*	Yüzey IsI KaybI	= 43.3 [Kcal/h m²]

* BAZI Girdileri Degiştirecekmisiniz ? [E/H] = H

* Kar eritme sisteminin toplam yüzey alanI [m²] = 12500

KAR ERiTME DÖŞEMESi DETAYLARINI GiRiNiZ =

* Boru merkezi itibarI ile üst beton kalInlIgI [cm] = 4
* Boru merkezi itibarI ile alt beton kalInlIgI [cm] = 4

* Beton IsI iletim katsayIsI [Kcal/h m°C] = 1.2

* YalItIm malzemesinin kalInlIgI [cm] = 0

IsI transfer sIvIsI gidiş/dönüş sIcaklIk farkI [°C] = 10

* Yüzeyde kaç adet kaplama var ? = 1

* Her kaplamanIn kalInlIk [cm] ve k katsayIsI girilecek :

* Kaplama 1:

* Kaplama kalInlIgI [cm] : 2
* k katsayIsI [Kcal/h m°C] : 0.86

* Beton,zemin içinde mi ve/veya etrafI YALITIMLI MI ? [E/H] = E

KAR ERiTME ISIL VERiMi = 91.1 [%]

SiSTEMiN KAZAN YüKü = 12395165 [Kcal/h]

ENTER TUŞUNA BASINIZ

Çizelge 10-2. Ankara için Meteoroloji Rasatları

ANKARA

Rasat Süresi : 1926 - 1970
Yükseklik (H) : 894 m.

Enlem (φ) : 39° 57′ N.
Boylam (λ) : 32° 53′ E.

eteorolojik Elemanlar	Rasat süresi	I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XII	Yıllık
ortalama aktüel basınç (mb)	45	912.9	911.2	911.2	910.3	911.3	[illegible]		[illegible]	913.5	915.4	915.6	914.2	912.4
en yüksek	45	931.1	928.5	925.7	924.5	922.6	921.4	[illegible]	926.5	925.0	927.4	934.1	936.5	936.5
en düşük	45	883.4	884.0	882.6	892.7	892.2	[illegible]	[illegible]	[illegible]	897.8	895.1	875.9	869.0	882.[illegible]
07⁰⁰ deki ortalama sıcaklık (°C)	45	-2.2	-1.7	1.3	7.3	12.9	16.6		17.5	13.4	8.0	3.7	0.3	8.0
14⁰⁰	45	2.9	4.3	9.5	15.5	20.5	24.9	28.5	28.0	24.6	19.0	12.5	5.5	16.5
21⁰⁰	45	-0.3	0.5	4.8	10.5	15.1	19.[illegible]	22.5	22.7	17.5	12.2	6.9	2.2	11.2
ortalama sıcaklık °C	45	0.3	1.0	4.7	11.2	16.1	20.[illegible]	23.[illegible]	23.3	18.4	12.9	7.7	2.5	11.9
ort. sıcak. ≥ 5.0 °C old. ort. gün sayısı	15	6.5	7.3	17.3	27.[illegible]	31.0	30.0	31.0	31.0	30.0	30.4	23.1	12.5	278.[illegible]
≥ 10.0 °C	15	0.2	0.6	5.3	18.5	29.7	29.[illegible]	31.0	31.0	29.5	23.4	10.5	1.6	211.5
ortalama yüksek sıcaklık °C	45	4.1	5.6	10.3	17.4	22.4	26.5	30.1	30.3	25.7	19.9	13.5	6.5	17.7
düşük °C	45	-3.5	-3.0	-0.0	4.3	9.4	12.3	15.2	15.4	11.2	6.6	2.8	-0.8	5.9
en yüksek sıcaklık günü	45	31.951	23.953	31.952	23.923	31.935	23.942	[illegible]	1.954	9.952	3.952	2.926	2.956	1.8.954
en yüksek sıcaklık °C	45	16.4	20.4	23.5	31.[illegible]	34.4	36.4	[illegible]	40.0	35.7	33.3	25.3	20.4	40.0
en yük. sıc. ≥ 30.0 °C old. ort. gün sayısı	45	.	.	.	0.2	1.2	5.7	[illegible]	17.4	5.0	0.3	.	.	45.9
≥ 25.0 °C	45	.	.	0.2	2.[illegible]	9.2	20.1	28.9	29.2	17.9	4.6	0.0	.	112.4
≥ 20.0 °C	45	.	0.0	2.1	9.9	22.0	28.5	30.9	30.9	26.5	16.5	2.6	0.0	169.8
≤ −0.1 °C	45	6.0	4.7	0.7	.	.	.	.	.	.	.	0.2	3.0	14.7
günlük en yüksek sıcaklık farkı °C	45	17.9	22.8	21.2	24.3	21.6	21.1	22.3	22.6	23.7	23.8	20.2	18.4	24.3
en düşük sıcaklık günü	45	5.942	7.932	5.929	10.929	14.931	9.958	11.955	21.949	29.931	24.950	29.948	31.941	5.1.962
en düşük sıcaklık °C	45	-24.9	-24.2	-16.3	-7.2	-0.2	3.6	4.5	5.5	-1.5	-5.3	-17.5	-24.2	-24.9
en düş. sıc. ≤ − 0.1 °C old. ort. gün sayısı	45	22.5	18.7	14.5	3.4	0.0	.	.	.	0.0	1.7	7.6	16.2	84.7
≤ − 3.0 °C	45	14.9	12.5	8.0	0.9	.	.	.	.	.	0.2	3.4	9.2	49.1
≤ − 5.0 °C	45	11.0	9.2	4.6	0.2	.	.	.	.	.	0.0	1.6	6.1	32.5
≤ − 10.0 °C	45	3.9	3.1	1.0	.	.	.	.	.	.	.	0.2	1.5	9.7
≤ − 15.0 °C	45	0.9	0.9	0.1	.	.	.	.	.	.	.	0.0	0.4	2.2
≤ − 20.0 °C	45	0.3	0.1	.	.	.	.	.	.	.	.	.	0.0	0.4
≥ 20.0 °C	45	.	.	.	.	.	0.2	0.7	1.3	0.1	.	.	.	2.3
≥ 15.0 °C	45	.	.	0.0	0.1	0.7	5.4	17.3	18.7	3.5	0.1	.	.	45.9
≥ 10.0 °C	45	.	.	0.4	2.5	13.6	24.8	30.2	30.0	20.0	5.5	0.7	.	125.2
≥ 5.0 °C	45	0.8	0.8	4.4	15.9	28.7	29.8	31.0	31.0	28.9	21.3	9.6	3.2	205.2
toprak üstü düşük sıcaklık ortalaması °C	35	-4.0	-3.6	-0.7	4.1	8.1	11.2	13.9	14.1	10.0	5.6	1.9	-1.1	5.0
toprak üstü en düşük sıcaklığı °C	33	-27.5	-22.4	-16.6	-8.[illegible]	-1.6	2.1	4.5	2.0	0.0	-6.1	-17.6	-18.0	-27.5
tpr. üs. düş. sıc. ≤ − 0.1 °C old. ort. gün sayısı	14	20.9	20.1	14.1	4.2	0.4	.	.	.	.	2.8	9.2	15.8	87.4
≤ − 3.0 °C	14	14.2	13.4	8.3	1.1	.	.	.	.	.	0.9	4.2	8.2	50.4
≤ − 5.0 °C	14	11.2	10.0	4.6	0.4	.	.	.	.	.	0.1	2.5	5.1	34.1
≤ −10.0 °C	14	4.1	4.4	0.4	.	.	.	.	.	.	.	0.3	0.3	9.4
ortalama buhar basıncı (mb)	45	5.0	5.1	5.8	7.2	9.9	11.2	11.4	10.8	9.2	8.1	7.3	6.0	8.1
07⁰⁰ deki ortalama nisbî nem %	45	85	83	80	73	71	64	57	57	63	73	84	86	73
14⁰⁰	45	69	64	50	41	40	34	25	26	30	39	51	68	45
21⁰⁰	45	79	78	67	58	60	52	43	40	46	58	73	82	61
ortalama nisbî nem %	45	78	75	66	57	57	50	42	40	46	56	70	77	60
en düşük nisbî nem %	45	19	19	5	6	11	5	3	3	4	9	17	25	3
07⁰⁰ deki ort. bulutluluk (0 — 10)	45	8.0	7.4	6.3	5.6	4.8	2.9	1.5	1.5	2.4	4.2	5.8	7.4	4.8
14⁰⁰ (0 — 10)	45	7.3	7.4	6.9	6.7	6.5	5.2	3.5	2.8	3.5	4.7	6.0	7.5	5.6
21⁰⁰ (0 — 10)	45	6.2	5.9	5.1	4.5	4.4	3.2	1.7	1.1	1.9	3.0	4.2	6.3	4.0
ortalama bulutluluk (0 — 10)	45	7.0	6.9	6.1	5.6	5.3	3.8	2.2	1.8	2.6	4.0	5.4	7.1	4.8
ortalama açık günler sayısı (0.0 — 1.9)	45	3.1	2.2	3.9	3.9	4.7	8.1	16.3	19.3	15.7	10.3	5.8	2.6	95.8
bulutlu (2.0 — 8.0)	45	13.2	13.7	16.7	18.7	21.4	20.3	14.2	11.4	12.8	16.9	16.3	12.9	188.4
kapalı (8.1 — 10.0)	45	14.8	12.4	10.4	7.4	4.9	1.6	0.5	0.3	1.5	3.8	7.9	15.5	81.0

ISI TRANSFER BORUSU ÖZELLiKLERi :

Boru iç çapI [mm] = 22

Boru dIs çapI [mm] = 29

Boru IsI iletim katsayIsI [Kcal/h m °C] = 0.29

ÖngÖrÜlen boru aralIgI ? [cm] = 15

Standard boru uzunlugu ? [m] = 120

Sistemdeki Hidrolik Devre SayIsI ? = 765

Ortalama AKISKAN SICAKLIGI, Ts = 51.7 [°C]

ENTER Tuşuna BASINIZ

ANA TASARIM PARAMETRELERi

KAR ERiTME PERFORMANS KATSAYISI[ASHRAE]= 2
DIŞ HESAP SICAKLIGI = -12.0 [°C]
KAR YAGISINDA DIŞ HESAP SICAKLIGI = -7.0 [°C]
KAR ERiTME YÜKÜ = 11292729.2 [Kcal/h]
KAR ERiTiLEN TOPLAM YÜZEY ALANI = 12500.0 [m²]
BorularIn yüzeyden derinligi = 4.0 [cm]
YalItIm KalInlIgI = 0.0 [cm]
Devrede sIcaklIk Düşüşü = 10.0
Boru iÇ ÇapI = 22.0 [mm]
KanatcIk Verimi = 60.8 [%]
Yüzeydeki kaplama adedi = 1
Boru aralIgI [modül] = 15.0 [cm]

ENTER tuşuna BASINIZ:

BU TASARIMIN DOSYA ADI = KARERiTME

ÇIKTI TiPiNi GiRiNiZ =
1 - DETAYLI ÇIKTI
2 - ÖZET ÇIKTI

ÇIktI tipi = 1

ANKARA

Rasat Süresi : 1926 - 1970
Yükseklik (H) : 894 m.

Enlem (φ) : 39° 57′ N.
Boylam (λ) : 32° 53′ E.

Meteorolojik Elemanlar	Rasat süresi	AYLAR I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XII	Yıllık
Ortalama buharlaşma (mm) (Wild)	33	23.2	31.8	66.5	113.9	129.6	163.1	220.8	227.6	154.7	97.4	50.6	28.3	1307.[illegible]
Günlük en çok buharlaşma (mm) (Wild)	33	9.8	12.4	15.0	24.5	17.2	17.2	16.2	19.2	15.0	10.7	14.6	7.8	24.5
Saat 07^{00} deki ortalama yağış miktarı (mm)	45	16.5	15.0	14.6	12.5	14.6	5.2	4.4	2.1	9.3	7.8	11.6	20.0	133.0
» 14^{00} » » » »	45	10.6	11.0	9.5	10.2	12.5	7.0	2.7	0.7	3.1	5.9	8.1	12.0	93.1
» 21^{00} » » » »	45	11.0	11.6	13.1	13.5	23.4	17.2	6.1	6.2	5.4	8.2	8.8	15.4	140.1
Ortalama yağış miktarı (mm)	45	38.1	37.7	37.4	35.7	50.6	30.3	13.2	8.5	13.2	21.7	28.5	47.1	367.0
Günlük en çok yağış miktarı (mm)	45	37.7	28.2	28.2	28.8	37.5	57.5	47.3	47.3	40.5	30.1	27.9	69.5	69.5
Yağış ≥ 0.1 mm old. günler sayısı	45	13.2	12.0	10.3	10.1	11.9	8.6	3.3	1.9	4.2	6.2	7.8	12.8	102.3
» ≥ 10.0 » » » »	45	0.7	0.8	0.8	0.8	1.4	0.5	0.4	0.2	[illegible]	0.5	0.8	1.1	8.9
» ≥ 50.0 » » » »	6	.	.	.	.	.	.	.	.	.	.	.	.	.
Ortalama kar yağışlı günler sayısı	15	4.8	4.6	1.5	0.3	.	.	.	.	.	.	0.7	1.9	13.9
» karla örtülü günler sayısı	45	8.0	7.1	2.0	0.2	.	.	.	.	.	.	0.5	3.6	21.5
En yüksek kar örtüsü kalınlığı (cm)	45	33	32	24	12	.	.	.	.	.	.	15	25	33
Ortalama sisli günler sayısı	45	5.6	3.7	2.9	1.8	0.4	0.3	0.2	0.1	0.4	1.4	3.9	5.1	25.7
» dolulu » »	45	0.2	0.3	0.7	0.9	1.3	0.7	0.2	0.2	0.1	0.2	0.2	0.1	4.8
» kırağılı » »	45	9.2	8.3	7.1	1.9	.	.	.	.	0.1	1.3	6.2	8.9	43.1
» orajlı » »	45	0.0	0.1	0.6	2.3	5.7	5.1	2.2	1.3	1.2	1.1	0.4	0.0	20.0
Saat 07^{00} deki ort. rüzgâr hızı (m/sec)	42	2.6	2.8	2.8	2.5	2.3	2.3	2.9	3.0	2.5	2.4	2.4	2.6	2.6
» 14^{00} » » » »	42	3.7	4.6	5.0	5.3	4.6	4.2	4.1	4.0	3.8	3.6	3.3	3.4	4.1
» 21^{00} » » » »	42	3.0	3.0	3.0	2.9	2.5	3.0	4.0	3.8	3.2	2.5	2.5	3.0	3.0
Ortalama rüzgâr hızı (m/sec)	42	3.2	3.4	3.6	3.6	3.1	3.2	3.6	3.6	3.1	2.8	2.7	2.9	3.2
En hızlı rüzgâr yönü ve hızı (m/sec)	42	SSE 32.1	S 28.3	W 37.5	S.SW31.5	W 35.0	WSW23.2	SW 28.6	SW 33.0	W 33.8	SW 29.2	S 39.4	SW 32.4	S 39.4
Ort. fırtınalı gün. sayı. (17.2 ≥ m/sec)	22	1.2	1.1	1.5	1.9	1.0	1.0	0.7	0.3	0.1	0.3	0.6	0.9	10.7
Ort. kuv. rüzg. gün. sayı. (10.8 — 17.1 m/sec)	15	5.2	5.5	8.5	9.4	10.1	8.9	10.6	8.6	5.6	3.5	2.8	4.9	83.7
N rüzg. ort. hızı ve esme sayısı toplamı	22	2.8-339	3.0-362	2.9-306	2.7-222	2.6-245	2.9-331	3.6-337	3.6-338	2.8-314	2.6-331	2.3-316	2.7-316	2.9-3757
NNE » » » » »	22	3.4-246	3.8-197	4.0-216	3.6-210	3.0-186	3.8-210	4.2-348	4.2-338	3.6-258	3.3-263	3.1-181	3.5-187	3.7-2840
NE » » » » »	22	3.0-397	3.0-333	3.3-365	2.8-346	2.7 330	2.8-325	3.5-511	3.4-501	3.0-527	2.8-561	2.6-478	2.9-428	3.0-5102
ENE » » » » »	22	2.6-39	2.8-28	3.4-48	3.0-38	3.0-48	2.5-45	4.2-60	3.5-59	3.4-48	2.5-37	2.5-38	3.2-54	3.2-542
E » » » » »	22	1.7-40	2.5-23	2.7-30	2.2-54	2.4-65	2.3-61	2.8-39	2.6-64	2.5-52	2.4-52	2.1-39	2.5-58	2.4-577
ESE » » » » »	22	4.0-15	2.4-13	2.6-24	3.1-18	3.1-18	3.9-10	2.7-12	1.7-13	2.3-15	3.2-6	2.9-4	3.0-17	2.8-165
SE » » » » »	22	3.2-27	3.8-16	3.1-37	2.9-53	1.9-45	2.5-40	2.5-40	2.5-36	1.9-25	2.5-27	3.3-29	3.6-45	2.8-420
SSE » » » » »	22	4.9-38	4.1-24	4.4-24	4.3-27	3.3-29	3.7-7	2.7-16	2.2-17	3.0-19	3.5-15	2.3-14	5.2-20	3.8-250
S » » » » »	22	3.8-72	4.3-64	4.1-80	3.8-75	3.7-77	2.5-70	2.6-45	2.9-33	2.5-67	2.1-43	3.5-65	4.7-68	3.5-759
SSW » » » » »	22	3.5-64	3.6-71	4.1-79	4.0-67	4.2-68	3.6-36	2.7-31	2.9-41	2.6-39	2.7-56	3.2-80	3.9-82	3.6-714
SW » » » » »	22	2.8-247	3.1-217	3.7-241	3.7-249	3.2-251	3.5-237	3.2-118	2.9-143	2.8-177	3.0-195	2.3-216	2.7-253	3.1-2544
WSW » » » » »	22	3.5-107	4.0-109	3.9-108	4.4-134	3.9-121	3.6-75	3.4-56	3.5-61	3.5-66	3.2-75	3.0-98	2.6-71	3.6-1081
W » » » » »	22	2.8-99	3.4-130	3.2-102	3.8-122	3.9-127	3.1-104	3.0-90	2.8-92	2.9-104	2.8-132	2.5-127	2.4-101	3.1-1330
WNW » » » » »	22	2.7-13	3.9-23	3.1-32	4.9-43	3.2-35	3.6-32	3.1-21	3.0-28	2.6-15	2.9-30	3.6-20	2.3-12	3.4-304
NW » » » » »	22	2.8-71	3.1-72	3.4-123	3.1-110	2.8-111	3.2-136	3.4-105	4.4-81	2.8-80	2.4-77	2.4-78	2.2-93	3.1-1137
NNW » » » » »	22	2.0-35	2.2-33	3.2-77	5.3-32	3.0-50	3.1-68	3.4-81	3.9-59	3.1-49	2.7-31	2.4-67	2.3-49	3.0-631
Ortalama toprak sıcaklığı °C (5 cm)	37	0.9	2.1	6.2	13.0	19.1	24.0	27.7	27.7	22.1	14.5	7.9	2.8	14.0
En düşük » » °C (5 cm)	37	-15.8	-15.0	-10.3	-0.2	5.9	10.1	13.5	11.1	5.9	-0.8	-7.6	-10.0	-15.8
Ortalama » » °C (10 cm)	14	1.7	2.3	7.0	13.4	19.0	23.4	26.9	27.4	21.8	14.5	8.1	3.9	14.1
En düşük » » °C (10 cm)	14	-7.2	-7.3	-3.8	2.7	7.6	11.0	14.0	14.3	8.0	3.4	-0.8	-2.1	-7.3
Ortalama » » °C (20 cm)	14	2.2	2.4	6.8	12.9	18.3	22.6	26.1	26.8	21.6	14.6	8.6	4.4	14.0
En düşük » » °C (20 cm)	14	-4.4	-4.4	-0.3	4.5	9.0	12.2	16.5	18.8	11.9	4.4	0.7	-0.6	-4.4
Ortalama » » °C (50 cm)	43	4.3	4.2	6.6	11.8	16.8	21.0	24.4	25.4	22.2	16.8	11.6	6.8	14.3
En düşük » » °C (50 cm)	43	-1.4	-1.9	0.7	5.2	10.3	14.7	19.1	20.9	16.3	10.6	3.6	1.4	-1.9
Ortalama » » °C (100 cm)	11	8.2	6.6	7.8	10.7	14.5	18.0	20.9	22.8	21.6	18.1	14.6	10.9	14.6
En düşük » » °C (100 cm)	11	5.3	3.7	4.9	7.9	11.0	15.3	17.4	20.8	19.3	15.3	11.6	7.6	3.7

Çizelge 10-3. E K R A N Ç I K T I L A R I

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******************* DETAYLI ÇIKTI ********************
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 A - MUHTELiF SICAKLIK DEGERLERi:

 Yüzey Ortalama SIcaklIgI                    ,Tp = 4.5 °C
 Maksimum Yüzey SIcaklIgI                    ,Tb = 9.8 °C
 Minimum Yüzey SIcaklIgI                   ,Tmin = 0.2 °C
 Boru DIs Yüzey SIcaklIgI                    ,Td = 28.7 °C
 Ortalama SIvI SIcaklIgI                     ,Ts = 51.7 °C +/- 1
 Ölçüm noktasInda ayar sIcaklIgI                 = 1.0 °C


  ENTER TUŞUNA BASINIZ


  B-KAPASiTE VE YÜKLER:

  Atmosfere olan IsI kayIplarI                ,ql = 43.3  [Kcal/h m²]
  m² de Kar Eritme ISI YÜKÜ                   ,qy = 903.4 [Kcal/h m²]
  m² de Buz EriTme ISI YÜKÜ                ,qidle = 196.5 [Kcal/h m²]
  Kar eritme IsIl verimi                        ,x = 91.11 [%]
  AşagIya olan IsI kaybI yüzdesi            ,(1-x) =  8.89 [%]
  Toplam Kazan yükü                           ,BL = 12395165.3 [Kcal/h]

  ENTER TUŞUNA BASINIZ

  C-HiDROLiK HESAPLARI:

  IsI Transfer akIşkan debisi                 ,V  = 0.5 [litre/sn.]
  BorudakI ortalama akIşkan hIzI              ,Vs = 1.3 [m/sn]
  Boru Reynolds sayIsI                        ,Re = 53895
  AkIşkan/Boru IsI taşInIm katsayIsI          ,α  = 5491.4 [Kcal/h m² °C]
  Boru sürtünme katsayIsI                     ,Γ  = 0.02558
  Birim BasInç KaybI                          ,Dp = 121.185 [mm.ss./m]
  Toplam boru uzunlugu                        ,L  = 120 [m]
  Devredeki toplam basInç kaybI               ,Pt = 14521.10 [mm.ss]

  NOT =  Sonuçlar %50 Propilen Glikol/Su  içindir.

  ENTER TUŞUNA BASINIZ


 * PROGRAMIN BAŞINA DÖNMEK iSTERMiSiNiZ ? [E/H] = H


   F3 TUŞUNA BASINIZ.
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E ADI    = KARERiTME
l        = Esenboga,ANKARA
e Sahibi= TüBiTAK MiSAG-12
h        = 9/11/1993

                    ANA TASARIM PARAMETRELERi
ERiTME PERFORMANS KATSAYISI [ASHRAE]     =  2
LiK KAR YAGISI                           = 10.0     [mm.su/h]
AR HESAP HIZI                            = 12.0     [km/h]
HESAP SICAKLIGI                          = -12.0    [°C]
YAGISINDA DIS HESAP SICAKLIGI            = -7.0     [°C]
ERiTME TOPLAM ISI YÜKÜ                   = 11292729[Kcal/h]
ERiTiLEN TOPLAM YÜZEY ALANI              = 12500    [m²]
LARIN YÜZEYDEN DERiNLiGi                 = 4.0      [cm]
TIM KALINLIGI                            = 0.0      [cm]
KAN DEVRESiNDE SICAKLIK DÜSÜSÜ           = 10.0     [°C]
 iÇ ÇAPI                                 = 22.0     [mm]
 DIS ÇAPI                                = 29.0     [mm]
 ISI iLETiM KATSAYISI                    = 0.29     [Kcal/h m °C]
 MERKEZLERi ARASI MESAFE [modül]         = 15.0     [cm]
TÇIK VERiMi                              = 61       [%]
YDEKi KAPLAMA ADEDi                      = 1

RIM SONUCLARI:

Muhtelif SIcaklIklar:

lama yüzey sIcaklIgI                 ,Tp = 4.5    [°C]
imum yüzey sIcaklIgI                 ,Tb = 9.8   [°C]
mum yüzey sIcaklIgI                ,Tmin = 0.2    [°C]
 dIs yüzey sIcaklIgI                 ,Td = 28.7   [°C]
lama sIvI sIcaklIgI                  ,Ts = 51.7   [°C] +/- 1
m noktasInda ayar sIcaklIgI              = 1.0    [°C]

pasite ve Yükler:

e  atmosfere olan IsI kayIplarI      ,ql = 43.3  [Kcal/h m²]
e toplam IsI ihtiyacI                ,qy = 903.4 [Kcal/h m²]
e buz eritme IsI ihtiyacI         ,qidle = 196.5 [Kcal/h m²]
eritme IsIl verimi                    ,x = 91.1  [%]
Iya olan IsI kaybI yüzdesi        ,(1-x) = 8.9   [%]
am kazan yükü                        ,BL = 12395165 [Kcal/h]

er Bir Devre için Hidrolik HesaplarI:

olik Devre  SayIsI                   ,np =  765
Transfer akIskan debisi              ,V  = 0.5    [litre/sn.]
daki ortalama akIskan hIzI           ,Vs = 1.3    [m/sn]
 Reynolds sayIsI                     ,Re = 53895
kan/Boru IsI tasInIm katsayIsI       ,α  = 5491   [Kcal/h m² °C]
 sürtünme katsayIsI                  ,Γ  = 0.02558
m BasInç KaybI                       ,Dp = 121    [mm.ss./m]
edeki boru uzunlugu                  ,L  = 120    [m]
edeki toplam basInç kaybI            ,Pt = 14521 [mm.ss]

= Sonuçlar %50  Propilen Glikol/Su  karIsImI içindir.
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BLİYOGRAFİK BİLGİ FORMU	
Proje No: MİSAG-12	2- Rapor Tarihi: EKİM 1993

Projenin Başlangıç ve Bitiş Tarihleri: 1.8.1991 - 1.9.1993

Projenin Adı: PANEL ISITMA SİSTEMLERİNİN MODELLENMESİ VE BİR STANDARD TASARIM ALGORİTMASININ GELİŞTİRİLMESİ

Proje Yürütücüsü ve Yardımcı Araştırıcılar: Prof. Dr. Birol KILKIŞ
Y.Müh. Meriç SAPÇI, Ar. Gör. Selçuk SAĞER, Ar. Gör. Mahmut ULUDAĞ

Projenin Yürütüldüğü Kuruluş ve Adresi: ODTÜ, Makina Müh. Bölümü, 06531 ANKARA
Deneyler ve yazılım geliştirme (kısmen): 3131 West Chestnut Expwy, Springfield, MO 65802, U.S.A.

Destekleyen Kuruluş(ların) Adı ve Adresi:
HEATWAY, 3131 West Chestnut Expwy, Springfield, MO 65802, U.S.A.

Öz (Abstract):

Yapılarda panel ısıtma, yeni, yenilenebilir ve atık enerji kaynaklarının, daha yaygın, etkin ve verimli kullanılmasına büyük ölçüde imkan sağlayan bir sistemdir. Panel ısıtmaya özgü termo-fiziksel konfor koşulları sayesinde ısıtma yüklerinde de önemli ölçüde bir azalma ve enerji tasarrufu söz konusudur. Bu nedenlerle panel ısıtma, ilk yatırım maliyetlerinden işletme maliyetlerine ve enerji tüketimine uzanan ekonomik perspektif içersinde, ulusal ve evrensel enerji tasarrufu ile çevre sorunlarının hafifletilmesinde giderek önemli bir seçenek halini almaktadır. Bilinen bütün bu avantajlara ve uygulamalardaki olumlu gözlemlere karşın, uluslararası platform da dahil olmak üzere Ülkemizde panel ısıtmaya özgün tasarım yöntem ve yaklaşımları yeterince incelenmemiş ve sistem tasarımı ve analizi belirli bir sistematiğe kavuşturulmamış idi. Bu çalışmada uluslararası düzeyde ve bilimsel bir yaklaşımla, bir analitik model ve bilgisayar yardımı ile tasarıma yönelik bir algoritma hazırlanmıştır. Bu çalışmalar ASHRAE ve ISO kuruluşlarına da yansıtılmıştır. Ayrıca iki adet zorunlu TSE standardı hazırlanmış, bu kapsamda, panel ısıtmaya özgü ısı yükü hesap yöntemi de geliştirilmiştir. Ayrıca panel soğutma konusunda da çalışmalar yapılmıştır.

Anahtar Kelimeler: Panel ısıtma ve soğutma tasarımı, Panel ısıtma yükü, Isı, Alternatif enerji kaynakları ile ısınma.

Proje ile ilgili Yayın/Tebliğlerle ilgili Bilgiler

Yayın listesi ekte olup toplam 26 adet yayın yapılmıştır.

Bilim Dalı: Makina Mühendisliği

Doçentlik B. Dalı Kodu: 625 ISIC Kodu:

Uzmanlık Alanı Kodu: 625.02.07

Dağıtım (*) : ❒ Sınırlı ❒ Sınırsız

Raporun Gizlilik Durumu : ❒ Gizli ❒ Gizli Değil

(*) Projenizin Sonuç Raporunun ulaştırılmasını istediğiniz kurum ve kuruluşları ayrıca belirtiniz

TÜBİTAK MİSAG-12 PROJESİ KAPSAMINDA HAZIRLANAN YAYINLAR LİSTESİ

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Kılkış B.,A Tentative List of Contributions to ASHRAE HANDBOOK-HVAC Applications,Chap.45 on Snow Melting,Technical Report to TC 6.1,12 p.,ASHRAE WAM 1992,Anaheim,CA.

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14. TEŞEKKÜR

Bu projenin kapsamı ve amaçları ne kadar geniş ve çok olursa olsun, aslında 2 yıl gibi kısa bir sürede bütün çalışmaları bitirip, sonuçları toparlamak doğrusu tahminlerin üzerinde özveri, çalışma ve feragat gerektirdi ama bir ekip olarak hem ulusal hem de uluslararası düzeyde çalışmaları yürütmek olumlu ürünler ortaya çıkartmak ve proje amaçlarını aşmak hepimize kıvanç verdi. Bu sonuca hep birlikte ulaşıldı. Özellikle proje çalışanları olarak Meriç Sapçı'ya, Selçuk Sağer'e, Mahmut Uludağ'a, Nuri Çarkoğlu'na ve Sayın Gülseren Beyaz'a teşekkür ediyorum. TSE 'de Abdullah Yenen ve Şükrü Er, standartların zamanında çıkarılabilmesi için her türlü destek ve gayreti gösterdiler. ODTÜ Biltir'de ise çalışmalarımızda her türlü kolaylığı gösteren Prof. Dr. Bilgin Kaftanoğlu ve Prof. Dr. Y. Samim Ünlüsoy başta olmak üzere tüm çalışanlara teşekkür ediyorum. ASHRAE Teknik Komite 6.5 ve 6.4 üyeleri ilk başta beni bir Türk araştırmacı olarak ve açıkcası mevcut yayınlarını kıyasıya eleştirmemden ötürü hem yadırgadılar hem de canları sıkıldı. Ama şimdi güzel sonuçlara ulaşıldıkça sanırım beni ve biz Türkleri biraz daha aslında epeyi seviyor ve taktir ediyorlardır. Sağduyulu davranışlarını ve işbirliğine olan inançlarını kutluyorum. HEATWAY kuruluşunda Mike Chiles başta olmak üzere Mike Coley ve Scooter Dipper'ın katkıları, yakın alaka ve destekleri, projenin sadece maddi yönden değil, manevi yönden de kuvvet ve hız kazanmasını sağladı.

TÜBİTAK camiası olarak MİSAG biriminin çok yakın anlayış ve desteği, her an bizlere yardımcı olmaları, tüm işlemlerin anında neticelendirmesi bütün övgülerin üzerindedir. Özellikle Nergiz Bağcan'a teşekkür ediyorum. Sayın Selçuk Batualp, bizlere hiç bir zaman yakın ilgi ve desteğini esirgemedi. Kendisine müteşekkirim. TÜBİTAK'a getirdiği ve getireceğinden emin olduğum dinamizm ve yenilikler için kendisini kutlarım.

Belki de sorunlarımızı, diğer birimler yetki ve sorumlulukları çerçevesinde zaten çözdükleri için diğer üst kademelerde yeterli izlenimim olmadı. Aynı ilgi ve taktiri gördüğümü şu anda söylemem sanırım erken.

İki yıl süre ile bitmeyen bir özveri ile benim sabahlara kadar çalışmama sabır gösterip, tahammül eden sevgili eşime, kızıma ve mini mini oğluma bu raporu ve projeyi ithaf ediyorum.

Prof. Dr. İbrahim B. Kılkış, 1993