EXERGY ANALYSIS OF A SOLAR ASSISTED ABSORPTION HEAT PUMP FOR FLOOR HEATING SYSTEM

A THESIS SUBMITTED TO
THE GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES
OF
THE MIDDLE EAST TECHNICAL UNIVERSITY

BY

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IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF
MASTER OF SCIENCE
IN
THE DEPARTMENT OF MECHANICAL ENGINEERING

JANUARY 2004

Approval of the Graduate School of Natural and Applied Sciences

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ABSTRACT

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Jan 2004 ,188 pages

Solar assisted single-stage absorption heat pump (AHP) was used to supply energy to a floor-heating system by using the exergy methods. An existing duplex-house,in Ankara, with a heating load of 25.5 kW was analysed. Heating loads of the spaces in the building were calculated and a floor heating panel was modelled for each space leading to the capacity of the AHP before it was designed. Solar energy was delivered to the evaporator and high temperature heat input delivered to the genarator are met by auxiliary units operating with natural gas.The solar energy gained by flat-plate collectors was circulated through AHP.The anaysis performed according to the storage tank temperature reference value if the water temperature leaving the storage tank exceeds a predetermined value it is directly circulated through the floor heating system.

Exergue analysis were carried out with Mathcad program. Exergy analysis showed that irreversibility have an impact on absorption system performance.This study indicated which components in the system need to be improved thermally.A design procedure has been applied to a water-lithium-bromide absorption heat pump cycle and an optimisation procedure that consists of determinig the enthalpy, entropy ,exergy, temperature, mass flow rate in each component and coeficient of performance and exergetic coefficient of performance has been performed and tabulated.

Keywords : Solar energy, Flat-plate collector, absorption heat pump, floor heating, exergy

ÖZ

DÖŞEMEDEN ISITMA SİSTEMLERİ İÇİN GÜNEŞ DESTEKLİ SOĞURMALI ISI POMPASININ EKSERJİ ANALİZİ

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Ocak 2004 ,188 sayfa

Güneş enerjisi destekli tek kademeli soğurmalı ısı pompası, ekserji yöntemine göre, döşemeden ısıtma sistemine enerji sağlamakta kullanıldı. Ankara'da , ısıtma yükü 25,5 kW olan çift katlı evin analizi yapılmıştır.Binadaki mahallerin ısıtma yükleri ve her mahal için döşemeden ısıtma paneli modellenmiş ,bunun sonucunda da soğurmalı ısı pompasının kapasitesi belirlenmiştir.Evaporatör güneş enerjisi ile beslenirken jeneratöre yüksek sıcaklık girişi doğal gaz ile çalışan yardımcı birimler ile karşılanmaktadır.Kollektörün aldığı enerji depolama tankından geçirilerek ısı pompasına beslenmektedir.Analizler depolama tankının referans sıcaklığına göre yapıldı.Depolama tankından beslenen suyun sıcaklıgı önceden belirlenmiş sıcaklık değerini aştığında doğrudan döşemeden ısıtmaya verilmektedir.

Ekserji analizleri Mathcad programından yararlanılarak yapılmıştır.Exergy analizi, tersinmezliğin soğurma sistem veriminde etkisi olduğunu gösterdi.Bu çalışma ısıl yönden hangi bileşenlerin düzenlenmesi gerektiğini gösterir.Tasarım yöntemi su-lityum bromit ısı pompası çevirimine uygulanmıştır ve her bir bileşendeki entalpi, entropy, exergy ,sıcaklık, debi, enerji ve ekserji verimleri gerçekleştirildi ve tablolaştırıldı.

Anahtar kelimeler : Güneş enerjisi , Düz Plakalı Kollektör , Soğurmalı Isı Pompası , Döşemeden Isıtma , Ekserji

ACKNOWLEGEMENTS

I would like to thank my supervisor ,Assoc .Prof. Dr. Cemil YAMALI for his continuous guidance,help and criticism throughout this study.

I offer sincere appreciation to my family for their help and endurance to me,understanding and support.

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a	Infiltration rate coefficient, m^3/h-m
A_c	Collector area, m^2
A_{net}	Net area of a partition, m^2
A_p	Floor heating area, m^2
C_{pw}	Specific heat of water, kJ/kg-K
C	Peak load shaving factor
C	Heat capacity, kW/K
COP	Coefficient of performance
CR	circulation ratio
D_i	Inside pipe diameter, m
D_o	Outside pipe diameter, m
E	Probable hourly heating load, kW
E_{conv}	Probable hourly heating load with a conventional system, kW
ECOP	Exergetic coefficient of the AHP
F_1	Panel efficiency factor
FNP	Fraction of non-purchased energy
F_R	Collector heat removal factor
FWT	Fraction of working time length of the AHP
H	Enthalpy of the H_2O-LiBr solution, kJ/kg
h_f	Convection heat transfer coefficient on the pipe inside surface, W/ m^2.K
h_n	Convection heat transfer coefficient at the downside space surface, W/m^2K
$h_{p,i}$	Convection heat transfer coefficient at the interior surface of a

	Partition, kcal/h.m^2°C or kW/m^2. K
$h_{p,o}$	Convection heat transfer coefficient at the exterior surface of a partition, kcal/h.m^2°C or kW/m^2.K
h_{ps}	Convection heat transfer coefficient for the floor heating panel surface, W/m^2.K
H	Building site coefficient, kcal/m^3-°C
I	Solar insolation, kW/m^2
k_{eq}	Equivalent thermal conductivity of the floor materials abovethe pipe center, W/m.K
k_p	Thermal conductivity of the structural materials in the Partition, kcal/h-m.°C or kW/m.K
l	Crack length of window or door,
L_p	Thickness of a partition, m
m_{col}	Mass flow rate of the water circulating through the collector, kg/s
m_f	Total mass flow rate of the panel, kg/s
m_L	Mass flow rate of the load stream, kg/s
M_s	Storage tank mass, kg
N	Number of collector
n_l	Number of structural material layers in a partition
n_p	Number of partitions in a space
q_f	Heat delivered to a space per unit area of the panel, W/m^2
$q_{f,des}$	Design heating load of a space per unit area of the panel, W/m^2
$q_{f.max}$	Maximum value of heat flux which can be delivered to a space, W/m^2
q_s	Rate of energy which should be supplied by the heating system to the heating water circulating through the floor heating panel per unit area of the panel, W/m^2
Q_{ahp}	Sum of the capacities of absorber and condenser, which should

	be delivered to the floor heating system, kW
$Q_{aux,g}$	Rate of auxiliary energy delivered by AUX_1 to the generator,
$Q_{aux, go}$	Rate of auxiliary consumed by AUX_1, kW
$Q_{aux,s}$	Rate of auxiliary energy delivered by AUX_2 to the water leaving the storage tank, kW
$Q_{aux,so}$	Rate of auxiliary consumed by AUX_2, kW
$Q_{aux,t}$	Total auxiliary energy supplied to the system, kW
$Q_{aux,to}$	Rate of total auxiliary energy spent by the auxiliary heaters, kW
Q_a	Heating effect rate of the absorber, kW
Q_c	Heating effect rate of the condenser, kW
Q_e	Cooling effect rate of the evaporator, kW
Q_g	Generator heat input rate, kW
Q_h	Total heat loss of a space, kcal/h or kW
Q_l	Infiltration heat loss of a space, kcal/h or kW
Q_o	Gross transmission heat loss of a space, kcal/h or kW
Q_{st}	Energy stored in or removed from the storage tank, kWh
Q_t	Total transmission heat loss of a space, kcal/h or kW
Q_u	Rate of heat addition by the collectors (useful energy), in kW
P	Refrigerant pressure, kPa
P_{sat}	Vapor pressure of the water arising from the LiBr-water solution, kPa
R	Room coefficient
R_a	Thermal resistance between the pipe center and the downside space,m^2K/W
R_y	Thermal resistance between the pipe center and the surface of the floor heatingpanel,$m^2.K/W$
S	Pipe spacing of the floor heating panel, m
SPC	Solar performance coefficient

t_y	Total thickness of the floor materials above the pipe center,m
T	Solution temperature of the H_2O-LiBr pair, °C
T_a	Inside air design temperature, °C
T_{ab}	Absorber temperature, °C
T_b	Mean temperature along the centers of the tubing, °C
T_c	Condenser temperature, °C
T_{dp}	Dew point temperature of the H_2O-LiBr solution, °C
T_e	Evaporator temperature, °C
T_F	Temperature at the source for AUX_2 , °C
Tg	Generator temperature, °C
Tn	Downside space temperature, °C
T_f	Mean temperature of the heating water, °C
T_{fe}	Temperature of water at the exit of AUX_2 , °C
T_{f0}	Temperature of water at the exit of colector , ° C
$T_{f,i}$	Inlet water temperature of the floor heating panel, °C
$T_{f,o}$	Outlet water temperature of the floor heating panel, °C
T_L	Temperature of the load stream returning to storage tank, °C
T_0	Ambient air temperature, °C
Tp	Floor heating panel effective surface temperature. °C
$T_{p.max}$	Maximum value of floor surface temperature, °C
Tr	Refrigerant temperature, °C
T_s	Storage tank temperature, °C
$T_{s,f}$	Storage tank temperature at the end of each day, °C
$T_{s,i}$	Storage tank temperature at the beginning of each day, °C
$T_{s\ ref}$	Storage tank reference temperature for AUX2, °C
$T_{so,ref}$	Storage tank reference temperature at the beginning of a 24 hour period for AUX_3, °C
ΔT_L	Temperature difference around the load, K
ΔT_P	Overall temperature difference across a partition, °C
$(UA)_m$	Overall heat transfer coefficient-area product of the mixture heat exchanger, kW/K

$(UA)_r$	Overall heat transfer coefficient-area product of the refrigerant heat exchanger, kW/K
$(UA)_S$	Storage tank overall heat-loss coefficient area product kW/K
U_L	Collector overall heat-loss coefficient, $kW/m^2.K$
Up	Overall heat transfer coefficient of a partition, kcal/h-m^2.°C or $kW/m^2.K$
w	Panel thickness, m
Wp	Solution pump power, kW
x	Mass fraction of LiBr in the solution
X	Heating efficiency of the floor heating panel
X	Solution concentration (weight percent) of LiBr, %
y	Mass fraction of H_2O in the solution
Z_E	Corner room coefficient
Z_h	Allowance for room aspect
Z_d	Combined allowance
ε_m	Effectiveness of the mixture heat exchanger
ε_r	Effectiveness of the refrigerant heat exchanger
ε_y	Height correction factor
η_e	Collector efficiency
η_h	Auxiliary heater efficiency
$\tau\alpha$	Collector effective transmittance-absorptance product
$\eta_{II,sub}$	Exergetic efficiency of collector-evaporator subsystem
η_{floor}	Exergetic efficiency of the floor heating panel
$\eta_{overall}$	Exergetic Efficiency of the overall system
ψ_{aux2}	Exergy gained by AUX_2,kW
ψ_e	Exergy gained by the evaporator,kW
ψ_f	Exergy delivered to the water flowing through the panel pipes, kW

ψ_{i}	Exergy of the Sun incident on collector surface,kW
ψ_{rec}	Exergy Exergy recovered flow the flowing water in panel, kW
ψ_{u}	Useful exergy delivered to the load, kW
ψ_{a}	Exergy of heat delivered to the water in absorber at a constant temperature T_a, kW
ψ_{c}	Exergy of heat delivered to the water in condenser at a constant temperature T_c, kW
ψ_{eva}	Exergy of heat delivered to the water in evaporator at a constant temperature T_e, kW
ψ_{g}	Exergy of heat delivered to the water in generator at a constant temperature T_g, kW
$\psi_{1..18}$	Exergy at state points 1 to 18, kW
$\Delta\psi_{abs}$	Exergy loss in the absorber, kW
$\Delta\psi_{c}$	Exergy loss in the condenser, kW
$\Delta\psi_{eva}$	Exergy loss in the evaporator, kW
$\Delta\psi_{gen}$	Exergy loss in the generator, kW
$\Delta\psi_{pp}$	Exergy loss in the pump, kW
$\Delta\psi_{rhe}$	Exergy loss in the refrigerant heat exchanger, kW
$\Delta\psi_{she}$	Exergy loss in the solution heat exchanger, kW
$\Delta\psi_{tot}$	Exergy loss in total AHP system, kW

CHAPTER 1

INTRODUCTION

Oil crises in the past years made more obvious the dependency of economies on fossil fuels [24]. As fossil fuels such as coal, oil are being depleted, the need for using renewable sources is becoming more and more urgent. Renewable energy sources could provide a solution to the problem, as they are inexhaustible and have less adverse impacts on the environment than fossil fuels. Yet, renewable energy sources technology has not reached a high standard at which it can be considered competitive to fossil fuels.

Absorption heat pump (AHP) systems provide an alternative to today's dominating vapor-compression machines and other conventional systems absorption AHP has the option to reduce the fossil fuel energy demand by taking advantage of the possibility of being fired by renewable energy.

Solar energy technology is one of the most discussed subjects in applied heat transfer research today. Understandably, the interest in solar energy is motivated by the global energy challenge, by the quest for new energy resources and for new ways of thinking about energy. The application of solar power for cooling products for preservation purposes, as well as air conditioning in buildings, is a field with high potential in the coming years and as the technological setbacks are overcome, it may become a highly profitable sector.

1.1. Solar-Assisted Absorption Heat Pump Systems

1.1.1 Heat Pump Working on a Vapour-Compression Cycle vs. Absorption Heat Pump

A device that transfer heat from a low temperature medium to a high temperature one is the heat pump. The objective of a heat pump is to maintain a heated space at a high temperature. [24].

A basic absorption heat pump, as shown in Figure 1.1, whereas, is a type of “heat-driven heat pump”, which through reversible absorption processes, uses the thermodynamic availability of a "high-temperature heat input" to extract heat from a low temperature source and upgrade its temperature [31].

Refrigeration and air-conditioning by a mechanical vapour-compression system may be an efficient method. However, the energy input is shaft work (to a compressor normally driven by electricity), which is high-grade energy and therefore expensive. Furthermore, relatively large amount of work is needed, since during compression the "vapour" undergoes a large reduction in specific volume.

In an absorption heat pump, on the other side, a liquid (refrigerant-absorbent pair) is compressed instead of a vapour. Hence, an AHP raises the pressure of the working fluid without significantly changing its volume. Therefore, the amount of work needed is greatly reduced such that it’s often neglected in the cycle analysis [13].

In an absorption cycle, the input energy demand is shifted to a heat source by replacing the compressor of a vapour-compression cycle by several components, which are namely the absorber, solution pump, generator, solution heat exchanger, pressure reducing valve and. if needed, rectifier. This low-grade input energy is therefore thermal energy such as waste heat, geothermal energy or solar energy. However, this heat input is usually many times greater than the work input of the vapour-compression cycle.So, if the provided heat input is sufficiently cheap, absorption machines may be economically attractive, since in most cases they require only an insignificant amount of electrical energy to operate.

1.1.2 History of Absorption Heat Pump System

During the last three decades there has been “renewed” interest on AHP’s for efficient heating and cooling of buildings [33], [23] .The word renewed is used, because basically absorption system is an old technology with a large number of models available on the market for capacities from ten kilowatts to tens of megawatts. Absorption heat pumps, first developed in the 19 century, were indeed the first refrigeration machines [21]. These were ammonia- water systems fired by gas or kerosene. With the emergence of reliable electric motors and rather cheap electricity, they gave way to vapour-compression machines. Mainly due to their being very inefficient with a very low coefficient of performance (COP). Development of the more efficient water-lithium bromide systems has changed the point of view for absorption machine [15], and the increasing cost of electric power besides the particular features of the absorption cycle mentioned before, have made them attractive for both residential and industrial applications [21].

1.1.3 The use of Solar Energy in Absorption Heat Pump Systems

The use of solar energy in order to energise the absorption machine, either totally or partially is one outstanding option to reduce hydrocarbon combustion. However, even in this favourable case, the solar input has to be backed by a conventional furnace for the sake of reliability and continuity. Therefore, the environmental impact caused by the use of solar assisted heating and cooling is correlated directly to the solar fraction that can be attained in the total system [33]. Turkey, with the abundancy of this renewable energy source, should be one of the main countries to be interested in research and development of solar-assisted absorption heat pump systems and their utilisation in the future.

1.1.4. Advantages and Disadvantages of Absorption Heat Pump Systems

Besides reducing the primary energy consumption considerably and being almost pollution-free, absorption heat pump systems have the following advantages.

The possibility of performing heating and cooling of a building by the same cycle which allows the AHP to be used throughout the year, renders the system more attractive and cost effective (compared to conventional systems), particularly in places where electricity is expensive or in short supply [22], [23].

Absorption systems virtually have no moving parts, since they are based on a reversible thermal process. Therefore, they perform a silent operation and the possibility of being subject to wear and requiring maintenance is low. This also means high reliability and long service life. Moreover, liquid carry-over from the evaporator causes no problem [34], [15].

An AHP using solar energy is especially attractive if used throughout the year for both heating and cooling, since such a system using the free energy of the sun would have a considerably high-energy conservation potential. In Turkey, about 45% of the total energy consumption are occurring in the residential sector for domestic water and space heating, and space cooling [13]. Hence, there is a significant potential for solar energy applications. In addition, solar energy has the advantages of being non-depletable, non-polluting, clean, free and reliable (on space).

Furthermore, in cooling mode, cooling loads and availability of solar radiation are approximately in phase, since maximum amount of solar energy is available when cooling demand is about its peak value [43].

AHP systems have their disadvantages, too. They are bulky systems with large equipment size, having high initial costs. This is due to the large area and consequently the high cost of solar collectors, besides the cost of the auxiliary equipment. Finally, they are complex and have rather low efficiency.

It should also be reminded that solar energy, alone, also has its disadvantages such as being intermittent, dispersed (not concentrated) and non-dependable (on earth).

1.1.5. Energy , Exergy flow and Operation Principle of the Absorption Cycle

The first law of thermodynamic states that the cyclic integral of heat transfer is equal to the cyclic integral of work transfer. Since the first law does not distinguish between heat and work, it cannot indicate that one form of energy can or cannot be completely converted to another. As the energy is concerned, the effect of irreversible processes, such as fluid pressure drop is not shown. [5]

The exergy method (sometimes known as "availability method"), known as the second law property, measures the useful heat that can be obtained from a substance (fluid) or amount of work needed to complete a process. Unlike energy, exergy is not conserved; anaysis of exergy losses provides information as to where the real efficiencies in a system lie. According to the first law, all forms of energy can be convertible to one another. However, the second law puts limitations on energy conversion For example, work can be completely converted to heat, but it is always impossible to convert all available heat to work. In the first law, heat and work can be defined as energy, which is conserved, whereas in the second law, heat and work interactions are better understood when defined in terms of exergy which always decreases during real irreversible processes. The magnitude of loss in exergy is a measure of irreversibility.

In an absorption cycle, the strong solution (rich in refrigerant) is pumped from the absorber to the generator, as its pressure is raised to the high-side pressure, passing through the solution heat exchanger, where its temperature is upgraded. By the high-temperature heat input to the generator (Qg) refrigerant vapour is driven out of the liquid solution, which then becomes a weak solution (poor in refrigerant).

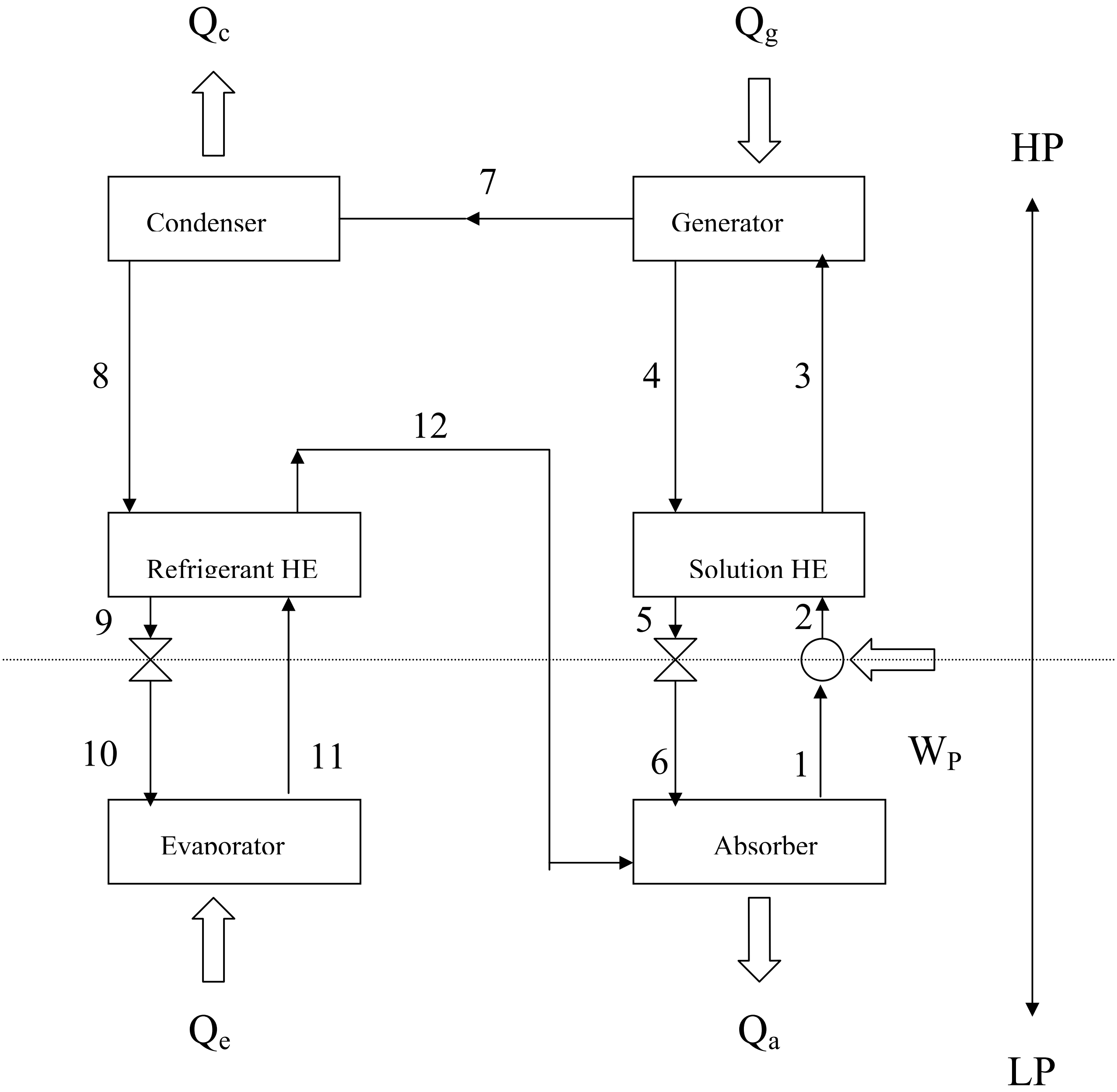

Figure 1.1. Energy Flow in Absorption Heat Pump

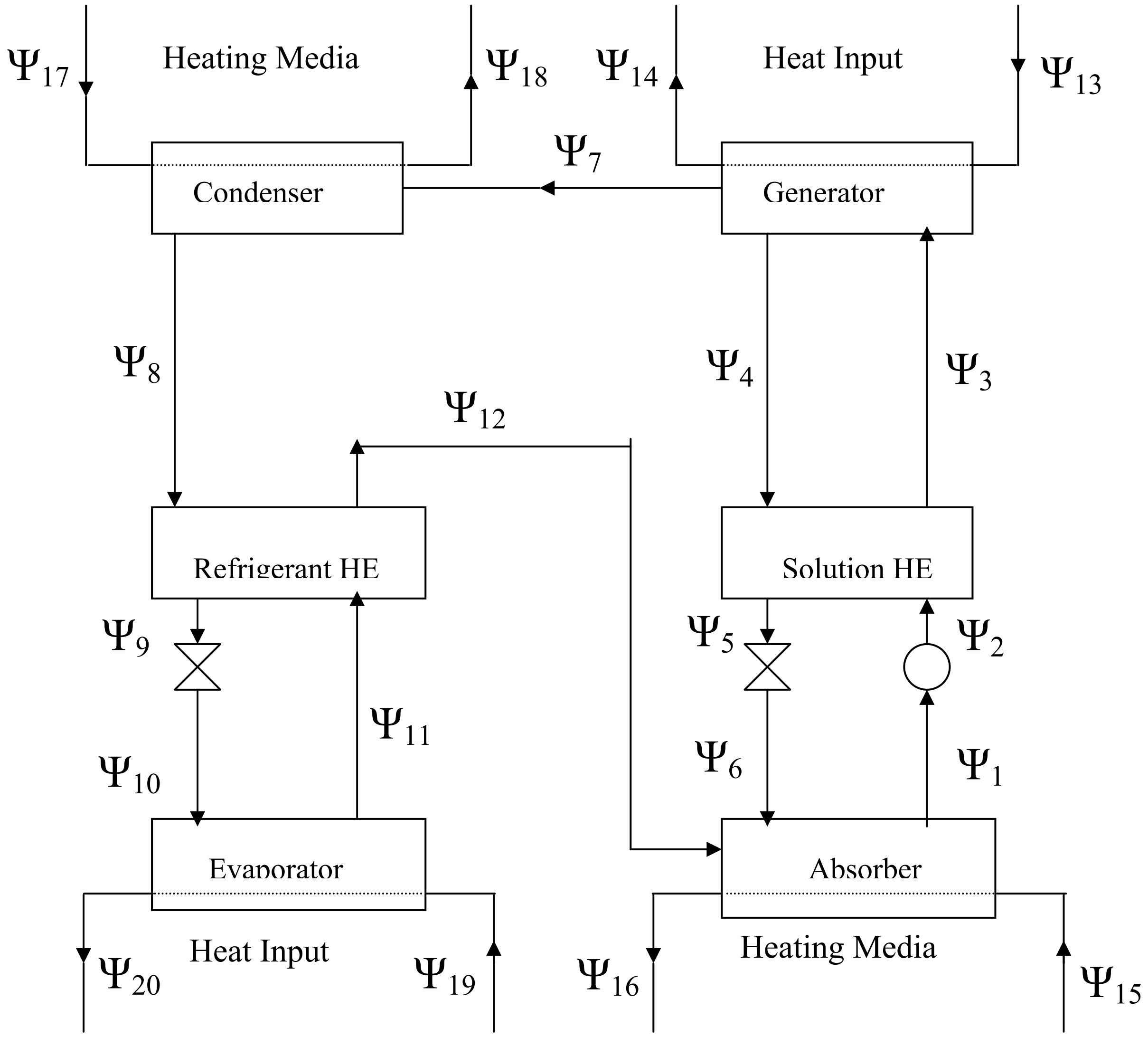

Figure 1.2. Exergy Flow in Absorption Heat Pump

Refrigerant vapour is then condensed in the condenser with the rejection of heat (Qc). Before it is throttled to the low-side pressure via the expansion valve, and evaporated by the low-temperature heat addition (Qe), as it flows through the evaporator.

Refrigerant vapour leaving the evaporator picks up further heat in the refrigerant heat exchanger from the liquid refrigerant leaving the condenser. Then it enters the absorber where it joins the weak solution coming from the generator, through the solution heat exchanger and the pressure-reducing valve. In the absorber, an exothermic reaction takes place such that as heat (Qa), is released, refrigerant vapour is absorbed by the weak solution, forming a liquid solution stronger in refrigerant and completing the cycle. Absorption occurs due to the mixing tendency of miscible substances and because of the affinity between absorbent and refrigerant molecules.

The refrigerant heat exchanger enhances the COP of the cycle, since in heating mode absorber capacity Q_a is increased due to state 12 entering the absorber at a higher temperature. The solution heat exchanger, on the other side, also improves the performance of the system, since heat input to the generator is reduced.

In addition to performance improvements, heat exchangers have some practical advantages such as: cavitation which may occur in the throttling device is prevented by cooling the refrigerant leaving the condenser, thereby reducing its vapour pressure.

In Figure 1.2, exergy flow balance is shown. The system considered consists of an internal and an external system. The external system represents the connection between the internal system and surrounding. The internal system contains evaporator, generator, absorber, condenser, solution heat exchanger, refrigerant heat exchanger, pump and expansion valves. The external system consist of solar collector, auxiliary heating and floor heating panel. Solar collector carries exergy (solar exergy) to the evaporator leading to the change of the exergy content of the circulating refrigerant. The exergy change in absorber and condenser results from the flow of exergy to the heating media. (Heating panel). [5]

1.1.6. Refrigerant-Absorbent Pair

Perhaps the major question in the design of solar-assisted absorption cycles is the choice of the working fluids (combination of refrigerant and absorbent), since the effective performance of an AHP depends on this pair. When choosing the components of this binary mixture, one has to consider their thermodynamic and environmental properties. Desirable characteristics for the refrigerant-absorbent pair are [27], [35]:

- Absence of a solid phase absorbent.
- High degree of chemical stability for long-term operations, by themselves and as a pair.
- Low corrosive effect, non-toxic, non-flammable.
- A refrigerant more volatile than the absorbent and a large temperature difference (at least about 200 K) between the boiling points, to facilitate easy separation of the refrigerant from the absorbent in the generator, thereby eliminating the need for a rectifier.
- Refrigerant having a moderate condensing pressure, and an evaporating pressure above atmospheric to prevent air leakage to the system.
- A refrigerant with a latent heats as high as possible, in order to minimise the refrigerant and strong solution mass flow rates, hence the pump power and the physical dimensions of the system.
- Low viscosity.

Two of the refrigerant alternatives, which have been used in refrigeration applications in the past, are R21 and R22. These have been suggested to be used in absorption heat pumps by forming pairs with alcohol derivatives such as DMF (di-methylformamide), NMP (N-methylpyrrolidinone), DMA (di-methylacetamide). DEGDME (di-ethyleneglycol dimethylether), TEGDME (tetra-ethyleneglycol dimethylether) and DBF (di-ethylformamide). However, R21 and R22 are HCFC (hydrochlorofluorocarbon) compounds, which cause ozone-depletion and global warming hence face a gradual phase-out. (Their production is to be halted completely by 2015 throughout the European Union [10].) Moreover, there have been doubts about the use of R21 due to its toxicity, and R22 is a high-pressure compound which has the disadvantage that the power required to drive solution pump is rather large due to the high pressure difference.

Another refrigerant alternative is TFE (trifluoroethanol) which can form pairs with TEGDME or NMP. These pairs have been proposed as alternatives to the classical pairs in the last decade or so, but TFE has the drawbacks of toxicity and inflammability. Also, TFE has a quite low latent heat (449 kj/kg at 0°C compared to 2502 kj/kg for water) [9].

Pairs NH_3 - H_2O (ammonia-water) and H_20-LiBr (water-lithium bromide) are the oldest combinations that have been tried in the absorption systems for air-conditioning, and they have found widespread application and extensive commercial use. Of these two pairs, NH_3 - H_2O meets more of the desired characteristics listed above, having very satisfactory thermodynamic properties. It also has the features of being possible to be used in systems with evaporating temperatures below 0°C and having no risk of air leakage, However, this working pair has the main disadvantage of requiring rectification (due to the volatility of water), which causes the COP of the cycle to be low, besides making the system more bulky and complex. NH_3 - H_2O pair exhibits a wide solution field with the potential of a high temperature increase, but again a considerable drop in COP occurs with increasing temperature increase. Furthermore, it requires high operating pressures, as the condensing pressure becomes very high (hence, that of the generator, not to evaporate the water) at the high condensing temperatures required. This leads to large pressure differences (between the high-pressure and low-pressure sides) in the system, which increases the solution pump power. Also, they are difficult to operate with flat-plate collectors due to the high generator temperatures required (in the range of 95 to 170 °C). Finally, ammonia is a toxic and flammable fluid with reliability problems, which makes it unattractive for residential air-conditioning applications.

H_20-LiBr pair, on the other side, has the problem of crystallisation (tending to form solids) in the absorber and generator, at moderate LiBr concentrations crystallisation occurs in cycles with high absorber temperature, since a high absorbent concentration of the working fluid is required to maintain low vapour pressure in the absorber (hence in the evaporator). It should also be noted that, pure LiBr is a solid, but when it is mixed with sufficient H_2O, homogeneous liquid solutions are formed. However, in order to overcome this problem, “anti- crystallisation” additives with high boiling points (ethylene glycol, ethanolamine, 1,3-propanediol) and absorbent salts ($ZnBr_2$, $ZnBr_2$-LiCl and $ZnCl_2$-$CaBr_2$), combined with LiBr have been widely used and developed by different researchers in the last decade [27], [32]. Moreover, it may be preferable to operate an AHP at relatively low flow ratios to prevent crystallisation in the absorber [27].

Another disadvantage of the H_2O-LiBr pair is its limitation to relatively high evaporating temperatures (about 5°C), since the refrigerant is water. Also, for this pair, generator temperatures are normally in the range of 70 to 95°C, hence, there is a small temperature range over which an unpressurized water storage tank can operate, reminding that most of flat-plate collectors operate around 100°C. Furthermore, high chance of air leakage to the system is present, since operating pressures are quite low. However, proper equipment design can effectively overcome these disadvantages.

H_2O-LiBr pair, whereas, does not need any rectifying equipment due to the non-volatility and very high boiling point of LiBr (normally 1265°C). This pair is also non-toxic and odourless, which means high safety. Moreover, low-cost, simple handling and relatively high latent heat of water (2502 kj/kg at 0°C compared to 1262 kj/kg for ammonia) favours it as a refrigerant. Another advantage of the H_20-LiBr pair is that, pressures and pressure differences in the cycle are low, therefore the pump power requirement also becomes low.

As discussed above, desired properties of a refrigerant-absorbent pair are so various, that the choice of this pair depends on the particular application. In this study, H_2O-LiBr pair was selected as the working fluid, owing to its unique features and advantages.

1.2. Floor Heating Systems

Floor heating panel systems find wide application in many types of buildings around the world, offering various advantages, especially from thermodynamics and energy conservation points of view, compared to conventional methods of heating.

A basic advantage of floor heating systems is that thermal comfort is obtained with lower temperatures of the circulating fluid (in the range of 45 to 60oC), owing to the large heat transfer surfaces and direct contact of people's feet with the heat source [4]. These factors may also decrease the indoor air temperatures for an equivalent human thermal comfort down to about 16-18°C in floor heating, whereas the same value for radiator heating case is about 20-22°C. As a consequence of lower circulating fluid temperatures, of course, savings in energy consumption, hence lower pollution levels besides lower initial and operating costs are obtained. In addition, less calcification occurs due to the lower rates of evaporation of the circulating water. Moreover, floor-heating systems enable the utilisation of low temperature heat sources such as solar energy, geothermal energy and waste heat for space heating [25].

In a room heated from floor, inside wall temperatures are higher in (usually 16-18°C, about 2°C higher than conventional systems), which yields higher radiation heat transfer rates. About 60-70% of total heating takes place with radiation and 30-40% with convection, where a typical adult body also tends to lose heat in a similar proportion. In contrast, for conventional systems, the share between radiation and convection is around 20% to 80%, respectively [25].

A unique feature of floor heating systems is their being a "self-regulating system". This means that the system is much more sensitive to the changes in temperature, since the temperature differences between the floor (~26-29°C), wall inside, and surface. (~16-18°C) and inside air (~16-20°C) are relatively small (as compared to a system with radiators). For instance, due to solar radiation when the heat gain in a room-facing south increases around noon, circulating hot water would pass to the colder rooms, releasing less energy in this room [25].

An. interesting characteristic of floor heating systems is that heat storage within structural elements of the building (thermal inertia characteristic) may decrease the peak load, since the occupants are not affected from sudden temperature drops and extremely cold days [4]. Also, the large floor thermal capacitance minimises the influence of source temperature fluctuations. For the same reasons to heat the inside air will take more time, however the building will cool down more slowly (maybe 1-2 days). Hence, this may be an advantage if there is a sudden shortage of the heat source [25].

From the architectural point of view it is very attractive since no radiators or similar heating devices and pipes are placed against walls.

Also, strapor foam and nylon canvas laid inside the panel brings an additional insulation for heat losses and noise [14].

As proved in theory and practice, floor-heating systems provide savings up to 25% in initial costs and 40% in operating costs. Moreover, if the same thermoplastic pipes are used for heating and cooling, big savings in investment costs are attained besides less corrosion and clogging problem.

Floor heating systems have their drawbacks, too. For buildings, which are not well insulated, maximum heat flux is limited by the reason that the maximum floor temperature should be below a certain value (usually 29°C).

Furthermore, since the difference between the inlet and outlet temperatures of the water circulating through the floor heating panel is generally taken as 10°C, mass flow rates are higher, which causes larger circulating pump power.

Pipes used in floor heating systems are usually made of polyethylene crosslink (PE-X), which is a kind of thermoplastic material. These pipes can withstand temperatures up to 95°C and a pressure of 6 bar (at 95°C), where their melting point is 400°C. Also, they can be installed cold and they have a very smooth inside surface which has resistance to clogging and decaying [25]

CHAPTER 2

HEATING LOAD CALCULATION

In Turkey, heating load calculation of buildings is performed in accordance with Turkish Standard TS 2164. However, this standard takes a central heating system with heating elements of radiators as the basis. Therefore, calculating heating load in floor heating systems differs from this standard at a number of points.

In floor heating systems thermal comfort is achieved with lower inside air temperatures since this temperature remains almost constant in the upper direction, unlike conventional systems with radiators where the warmer air is accumulated close to the ceiling. Lower space temperatures reduce infiltration heat loss besides transmission heat loss. Inside air design temperatures in floor heating systems and in conventional heating systems of residences are given in Table 2.1 [25].

Furthermore, due to the thermal inertia (heat storage within the structural elements) of the building, design ambient temperatures are taken higher, since the occupants are not affected from sudden temperature drops and extreme cold days.

Lower inside air temperatures and thermal inertia characteristic in floor heating systems also affect some other parameters while determining heat loss, which will be discussed next. Hence, design-heating load is not based

Total heat loss (Q_H) of a space is the sum of transmission heat loss (Q_T) and infiltration heat loss (Q_L).

$$Q_H = Q_T + Q_L \tag{2.1}$$

Table 2.1 inside air design temperatures of residences.

Space name	**Design inside air temperature, T_a [°C]**	
	Floor heating system	Conventional heating system
Living room	18	22
Bedroom	17	20
Hall, Entrance	17	18
Kitchen, WC	17	18
Bathroom	20	26

2.1. Transmission Heat Loss

Transmission heat loss is the sum of conduction and convection heat losses through the partitions (walls, ceilings, floors, windows and doors) of a space.

Gross transmission heat loss (Q_o) (in kcal/h) of a space is given by

$$Q_o := \sum_{i=1}^{n_p} U_{p_i} \cdot A_{net_i} \cdot \Delta T_{P_i} \tag{2.2}$$

Where n_p is the number of partitions in the space, A_{net} (in m^2) is the net area of the partition, ΔT_P (in °C) is the overall temperature difference across the partition, and U_p (in kcal/hm^2-°C) is the overall heat transfer coefficient of the partition given as

$$U_p := \left[\left(\frac{1}{h_{p_i}} \right) + \left[\sum_{i=1}^{n_1} \left(\frac{L_P}{K_p} \right)_i \right] + \left(\frac{1}{h_{p_o}} \right) \right] \tag{2.3}$$

where n_1 is the number of structural material layers in the partition, $h_{p,i}$ and $h_{p,0}$ are the convection heat transfer coefficients at the interior surface and exterior surface of the partition (kcal/hm^2.°C), respectively. L_p is the thickness (m) and k_p is the thermal conductivity (kcal/h m^3°C) of the structural materials in the partition.

A number of allowance factors and coefficients are introduced after finding Qo such that the total transmission heat loss in floor heating systems is calculated as

$$Q_T = Q_O \,.\, (1 + Z_H) \,.\, C \tag{2.4}$$

and that in conventional heating systems is given as

$$Q_T = Q_O \,.\, (1 + Z_H + Z_D) \tag{2.5}$$

Where Z_H is the allowance for room aspect to account for the incident solar radiation, Z_D is the combined allowance to consider the effect of interruption of heating system and the cold enclosing surfaces, and C is the peak load shaving factor. Z_D is determined according to the operation length of the heating system, where for systems, which are continuously operated but operate with reduced capacity at night (which is the case for most residences) it is taken as 0.07 C depends on climate and heat storage capability of the building elements. Z_H and C are given in Table 2.2 [25] and Table 2.3 [47], respectively.

Table 2.2 Allowance for room aspect, Z_H

Aspect	S	SW	W	NW	N	NE	E	SE
Z_H	-0.05	-0.05	0	+0.05	+0.05	+0.05	0	-0.05

Table 2.3 Peak load shaving factor, C

	C		
Climate	Construction Type*		
	Light	Regular	Heavy
Seaside	1.0	0.88	0.80
Moderate	1.0	0.85	0.78
Cold, Mountain	1.0	0.80	0.75
* Light construction type corresponds to a building mass less than 600 kg/m^2, and heavy construction type corresponds to a building mass greater than 1400 kg/m^2.			

2.2. Infiltration Heat Loss

Infiltration heat loss is the heat loss caused by the leakage of outside air into the heated space due to the pressure difference, and in floor heating systems it is given as [25]

$$Q_L = (a.l).R.H.Z_E.\varepsilon_y\,.\,\Delta T_p \qquad (2.6)$$

Where a is the infiltration rate coefficient (in m^3/h.m), 1 is the crack length of window or door (in m), R is the room coefficient, H is the building site coefficient (in kcal/m^3.°C), Z_E is the corner room coefficient and ε_y is the height correction factor.

In conventional heating systems ε_y is not taken into account therefore the infiltration heat loss is found by the following relation [34]

$$Q_L = (a.l).R.H.Z_E.\Delta T_p \qquad (2.7)$$

Z_E is taken as 1.2 for rooms with windows on both walls, and 1.0 for rooms having window on a single wall. For spaces having windows and doors with regular dimensions, R is taken as 0.8 in floor heating systems and 0.9 in conventional heating systems, whereas these values reduce to 0.6 and 0.7, respectively, for spaces having large windows and a single interior door [25].

Infiltration rate coefficient in floor heating systems and in conventional heating systems are given in Table 2.4 [24]. Crack length is found by multiplying the height of the window or door in question with the constant given in Table 2.5 [33]. Building site coefficient H, which takes into consideration the wind condition at the location of the building is, presented in Table 2.7 [25], and height correction factor ε_y is given in Table 2.6 [25].

Design criteria and data for the heating load calculations of the sample building are listed in Appendix A.1. Structural elements of the sample building, together with their calculated overall heat transfer coefficients are presented in Appendix A.2. Heating load calculations of the sample building for the floor heating system and for a conventional system (with respect to TS 2164) are given in Appendix A3

Table 2.4 Infiltration rate coefficient, a

Material	**Type**	**a [m^3/h.m]**	
		Floor heating system	Conventional heating system
Wood or plastic	Single window Composite window Double window	2.5 2.0 1.5	3.0 2.5 2.0
Metal	Single window Composite window Double window	1.2 1.2 1.0	1.5 1.5 1.2
	Interior doors - with threshold Interior doors - without threshold	12.0 35.0	15.0 40.0
	Exterior doors	2.5	2.5

Table 2.5. Constant for calculation of crack length, ϖ.

Type of window or door	Height of window or door [m]	$\varpi = I/F$ ϖ
Windows with number of cases	0.50	7.20
	0.63	6.20
	0.75	5.30
	0.88	4.90
	1.00	4.50
	1.25	4.10
	1.50	3.70
	2.00	3.30
	2.50	3.00
Doors with double case	2.50	3.30
Doors with single case	2.10	2.60

Table 2.6 Height correction factor ε_y.

Height [m]	0	5	10	15	20	25	30	35	40	45	50	55
ε_y	1.0	1.0	1.0	1.2	1.4	1.5	1.6	1.7	1.9	2.0	2.0	2.1

Table 2.7 Building site coefficient H.

		H [kcal/m^3-°C]			
Wind condition	Siting of building	Floor heating system		Conventional heating system	
		Terrace	Detached	Terrace	Detached
Moderate	Shielded Exposed Extremely exposed	0.20 0.35 0.50	0.32 0.50 0.70	0.24 0.41 0.60	0.34 0.58 0.84
Windy	Shielded Exposed Extremely exposed	0.35 0.50 0.68	0.50 0.70 0.93	0.41 0.60 0:82	0.58 0.84 1.13

CHAPTER 3

SYSTEM DESCRIPTION

The AHP system is illustrated in Figure 3.1. Heat rejected by the absorber and condenser is transferred to the water circulating through the floor-heating panel. As concluded by ileri [22], plate type collectors are found to be insufficient for supplying high-temperature heat input to the generator in winter. Hence, for this reason and in order to eliminate the possibility of freezing in the evaporator (since the refrigerant is water), solar collectors provide low-temperature heat input to the evaporator, while the generator is heated by a heater (AUX_1).

For the effective operation of the AHP unit, temperature of the water provided from the storage tank should be over a certain limit. Hence, an auxiliary heater (AUX_2) is placed after the storage tank for raising the temperature of the water leaving the tank to the predetermined minimum value, denoted as $T_{s,ref}$. This way auxiliary heating is not wasted in directly rising the temperature of the whole tank each time interval. However, an auxiliary heater (AUX_3) is used for raising the temperature of the water in the storage tank to a predetermined value, $T_{so \cdot ref}$, at the beginning of a 24-hour period in certain months, for optimum operation of the system.

When the storage tank temperature exceeds a predetermined value,no auxiliary heating is used,the water is directly circulated to the absorption heat pump.

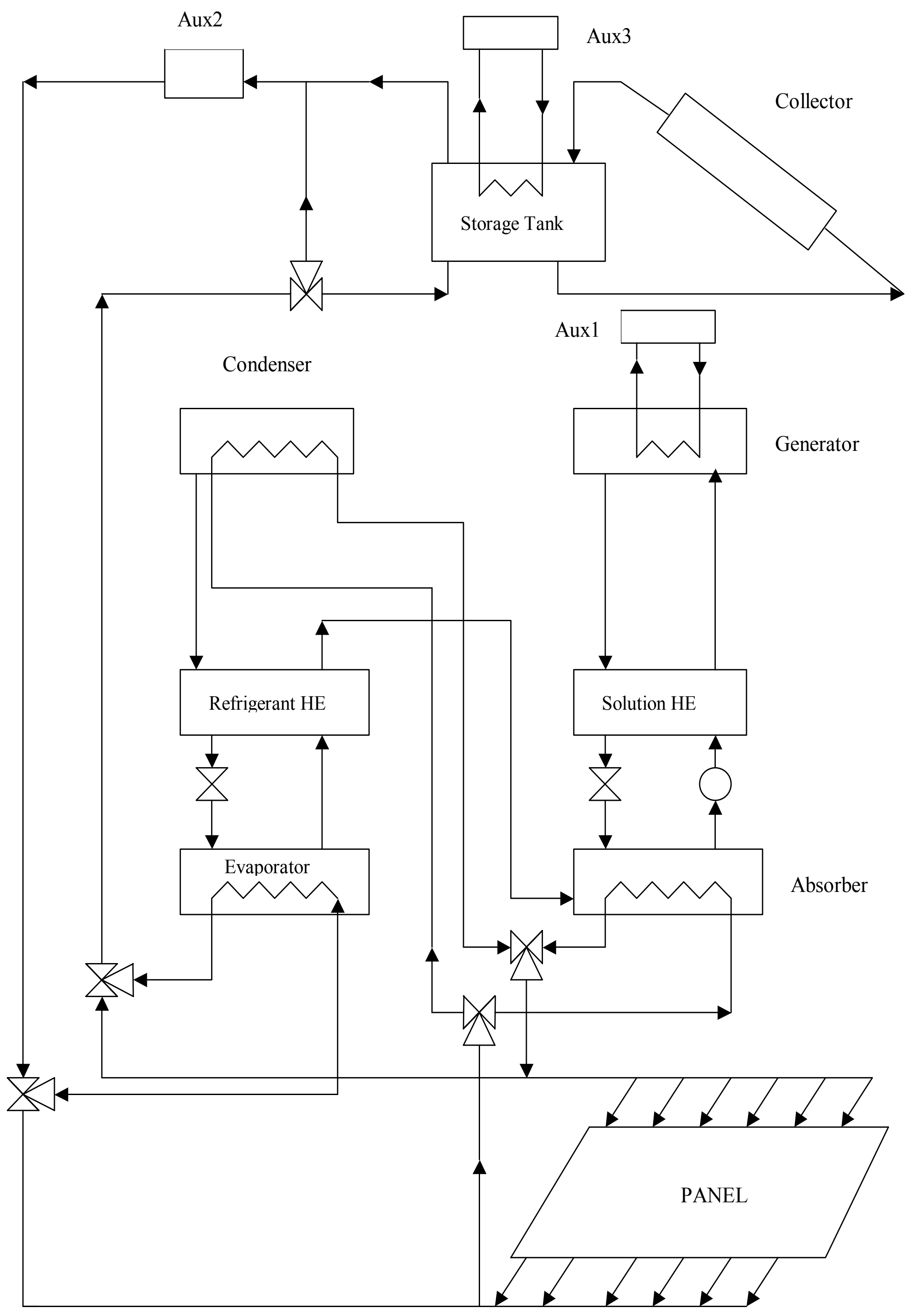

Figure 3.1 System Description

3.1.Modelling of the System Components

3.1.1 The Exergy of Sunlight

Unlike diffuse blackbody radiation,whose exergy content can be calculated using equation as [49]

$$\psi\left(T,T_o\right) := \sigma \cdot T^4 \cdot \left[1 - \frac{4}{3}\cdot\left(\frac{T_o}{T}\right) + \frac{1}{3}\cdot\left[\left(\frac{T_o}{T}\right)^4\right]\right] \qquad (3.1)$$

Where

$$\psi\left(T,T_o\right)$$

is the exergy of blackbody radiation emitted per unit area and per unit time ,the radiation we receive from the sun has a directional character that can be affected by the day's atmospheric condition.Starting from photon energy and entropy calculations of the type outlined , Zemanski [49] ,Parrot [37] determined the effect of the sun's cone angle on the exergy content of solar radiation.In the nomenclature of this chapter ,Parrots result is

$$\psi_i\left(T,T_0\right) := Q_i \cdot \left[1 - \frac{4}{3}\cdot\left(\frac{T_0}{T_s}\right)\cdot\left(1 - \cos\delta\right)^{\frac{1}{4}} + \frac{1}{3}\cdot\left(\frac{T_0}{T_s}\right)^4\right] \qquad (3.2)$$

where δ is the half-angle of the cone subtended by the sun's disc and this angle has the value of 0.0047 in a clear day,the other quantities

T_s :the equivalent temperature of the sun as a blackbody (5780 K)

T_o: the ambient temperature(300 K)

Q_i:solarbeam radiation on earth,bounded above by the solar constant 1353 W/m2

;on a sunny day (1000 W/m^2)

The above calculations demonstrates that the heat flux received from the sun is exergy-rich .It makes sense to regard the exergy–rich sun as a high temperature fuel.Therefore by writing

$$\psi_i := Q_i \cdot \left(1 - \frac{T_o}{T_x}\right) \tag{3.3}$$

T_x is defines as the equivalent temperature of the sun as an exergy source .So T_x can be set to 5780 K.

It is easy to simplify that the exergy content of the sun incoming on the collector surface can be reduced to

$$\psi_i = 0.948\, Q_i \tag{3.4}$$

where Q_i can be calculated with respect to the solar insolation value and collector area. That is;

$$Q_i := I \cdot A_c \tag{3.5}$$

3.1.2. Solar Collectors

Solar collectors were modelled in the manner proposed by Duffie and Beckman [16]: Rate of heat addition by the collectors (useful energy) Q_u (in kW) is given by:

$Q_u = \eta_c \cdot A_c \cdot I$ (3.6)

Where the collector efficiency η_c defined as the fraction of solar energy gained by the water circulating through the collector is expressed as:

$$\eta_c := F_R(\tau\alpha) - F_R \cdot (U_L) \cdot \frac{(T_s - T_o)}{I}$$

(3.7)

In the preceding equations, A_c (m^2) is the collector area; I (kW/m^2) is the solar insolation; F_R (dimensionless) is the collector heat removal factor; ($\tau\alpha$) (dimensionless) is the collector effective transmittance absorptance product; U_L (kW/m^2K) is the collector overall heat-loss coefficient, and T_s and T_0 (in °C) are the collector inlet temperature (coming from the storage tank) and the ambient air temperature, respectively.

It was concluded by Ileri [22] that an AHP system such as presented in this study could save primary energy both in winter and summer, only if collectors with high performance were used. Hence, evacuated flat plate collectors with selective surface were used in the simulation. $F_R(\tau\alpha)$ and F_RU_L were taken as 0.71 and 0.00244 kW/m^2K, respectively [4]. Solar insolation data are taken from the "Solar Radiation Handbook of Turkey" [44] for a surface tilted 60° from the horizontal and with collector orientation (azimuth angle) 0° south, in Ankara, to give the maximum solar insolation values in winter months for Ankara.

3.1.3. Storage Tank

Change in the storage tank temperature in a time period is calculated from [16]:

$$T_{snew} := T_{sold} + \frac{\Delta t}{M_s \cdot c_{pws}} \cdot \left[Q_u - FWT \cdot Q_e - UA_s \cdot \left(T_{sold} - T_o \right) \right] \tag{3.8}$$

It is proved by Duffie and Beckman [16] that the above equation gives successful results for time intervals of one hour (Δt = 3600s), where the rate of heat removal by the load Q_L (kW) which is Q_e in this case, is expressed as:

$$Q_L = m_L c_{pw,L}(T_{s,old} - T_L) \tag{3.9}$$

FWT, on the other hand, is the fraction of working time length of the AHP unit for a certain time period, defined as the ratio of the probable hourly heating load, E (in kW) to the design value of the rate of energy which should be supplied by the AHP unit to the floor heating system. Q_{ahp} (kW).

$$FWT := \frac{E}{Q_{ahp}} \tag{3.10}$$

In the above equations, T_s and T_0 (°C) are the tank mean storage temperature for the time period and the ambient air temperature, respectively; M_s (kg) is the storage tank mass; $c_{pw,s}$ and $c_{pw,\,L}$ (kJ/kgK) are the specific heats of water in the tank and the load stream, respectively; $(UA)_S$ (in kW/K) is the storage tank overall heat-loss coefficient and area product; m_L(in kg/s) is the mass flow rate of the load stream, and T_L (°C) is the temperature of the load stream returning to the storage tank.

After the storage tank temperature has been determined, the outlet temperature of the collector can be obtained by the following equation

$$T_{fo} := \left[\left[\frac{Q_u}{\left(m_{col} \cdot C_{pwcol} \right)} \right] \right] + \left(T_s \right) \tag{3.11}$$

and then useful exergy delivered to the water can be found as follow:

$$\psi_u := m_{col} \cdot C_{pwcol} \cdot \left(T_{fo} - T_s - T_0 \cdot \ln\left(\frac{T_{fo}}{T_s} \right) \right) \tag{3.12}$$

In the above equations, mcol(kg/s) and C_{pwcol}(kJ/kgK) are the mass flow rate and specific heat of water circulating through the collector, respectively.

It was concluded by Ileri [22] and Ergül [19] that the effect of the storage tank mass on the performance parameters decreases when the mass exceeds a certain value, and the recommendation was to use a storage tank capacity of about 50-75 kg per m^2 of collector area, for minimum cost of solar energy.

3.1.4. Auxiliary Units

Auxiliary units in the system are assumed to be operating with natural gas and to have thermal efficiencies of 0.80 to account for heat losses in the pipes. Rate of auxiliary energy (in kW) delivered to the water leaving the tank, by AUX_2 to raise its temperature to $T_{s,ref}$, if needed, or to $T_{f,i}$ in cases when it is directly delivered to the floor heating system, is given by:

$$Q_{aux,so} = m_L c_{pw,L} (T_{s,ref} - T_s).FWT \tag{3.13}$$

while the rate of auxiliary energy spent by the same auxiliary heater is:

$$Q_{aux,s} = Q_{aux,so} / \eta_h \tag{3.14}$$

m_L is found by using the following relation:

$$m_L := \frac{Q_L}{\Delta T_L \cdot c_{pw}} \tag{3.15}$$

where ΔT_L (K) is the temperature difference of the water exiting and entering to the evaporator

Exergetic efficiency of the auxiliary heating can be calculated by the following equation as[42]:

$$\eta_{IIb} := \frac{(\psi_{aux2})}{\left[Q_{aux2} \cdot \left(1 - \frac{T_0}{T_F}\right)\right]} \tag{3.16}$$

where ψ_{aux2} is the exergy delivered to the water circulating through the cycle at a constant temperature as ,

$$\psi_{aux2} := m_L \cdot C_{pwL} \cdot \left[\left(T_{fe} - T_s\right) - T_{02} \cdot \ln\left(\frac{T_{fe}}{T_s}\right)\right] \tag{3.17}$$

In the above equations T_{fe}(K) is the temperature of the load stream flowing to the evaporator,T_s(K) is the storage tank temperature entering to the auxiliary heater and T_F(K) is the temperature of the heat source (boiler) assumed to be constant.m_L ,C_{pwL} and T_o are the mass flow rate ,specific heat of the load stream and ambient temperature respectively.

Rate of auxiliary heat input (kW) delivered to the generator is:

$$Q_{aux,go}=Q_g \,.\, FWT \tag{3.18}$$

and the rate of auxiliary energy (in kW) spent by AUX_1 is:

$$Q_{aux,g} = Q_{aux,go} / \eta_h \tag{3.19}$$

where Q_g is the rate of required heat input to the generator calculated from the thermodynamic analysis of the AHP.

Hence, the total auxiliary energy supplied to the system becomes:

$$Q_{aux,to} = Q_{aux,s\,o} + Q_{aux,go} \tag{3.20}$$

and the rate of total auxiliary energy spent by the auxiliary' heaters is:

$$Q_{aux,t} = Q_{aux,s} + Q_{aux,g} \tag{3.21}$$

It should also be considered that, in practice the system will work all over the heating season and use all the stored energy in the storage tank and if needed, energy will be supplied by the auxiliary heaters. However, in this study, one day each month was simulated each starting with a specified storage, tank temperature. Therefore, energy stored in the storage tank at the end of each day (in kWh) should be taken into consideration in analysing the sytem performance, which is given as

$$Q_{sto} = M_S \, . \, C_{pw,s} \, . \, (T_{s,f} - T_{s,i}) / 3600 \tag{3.22}$$

where $T_{S,f}$ (in °C) is the storage tank temperature at the end of each day, while $T_{s,i}$ is the value at the beginning of each day if energy is stored in the storage tank.

Energy (in kWh) supplied to the storage tank by AUX_3 to raise its temperature to $T_{so,ref}$, if needed, is also calculated by using equation (3.22), where $Q_{st,o}$ becomes a negative value. However, in this case, $T_{s,i}$ is taken as $T_{so,ref}$ and the energy (in kWh) spent by the same auxiliary heater is found from

$$Q_{st} = Q_{,st,o} / \eta_h \tag{3.23}$$

Mathcad program written for the simulation of the modeling of the system components is presented in Appendix C.

3.2. Thermal Behaviour of the Floor Heating Panel

3.2.1 Heat Transfer at the Panel Surface

A floor heating panel, such as shown in Figure 3.2, transfers heat to a space by radiation and convection. For practical purposes. DIN 4725 proposes a single function for the sum of radiant and convective heat transfer intensities by assuming the area-weighted average temperature of the unheated surfaces is equal to the inside air temperature [47]

$$q_f=8.92.(T_p-T_a)^{1.1} \tag{3.24}$$

where q_f the heat delivered to a space per unit area of the panel (in W/m2), Tp (in °C) is the panel effective surface temperature, and Ta (in °C) is the inside air design temperature.

Antonopoulos has presented a one-dimensional analytical solution for the heat transfer analysis of ceiling cooling and floor heating panels, and has compared it with a two-dimensional solution using finite-difference method [3], [4]. It was stated that the results of the two methods were in very good agreement, especially for thin panels, and use of the analytical solution, which is very simple to apply, was recommended for practical applications. The analytical solution was performed in the directions normal and parallel to the tubes of the panel as shown in Figure 3.3 and Figure 3.4. For the purpose of the analysis, use was made of the theory of flat-plate solar collectors (presented by Duffie and Beckman [16]), which have some similarities to floor heating panels.

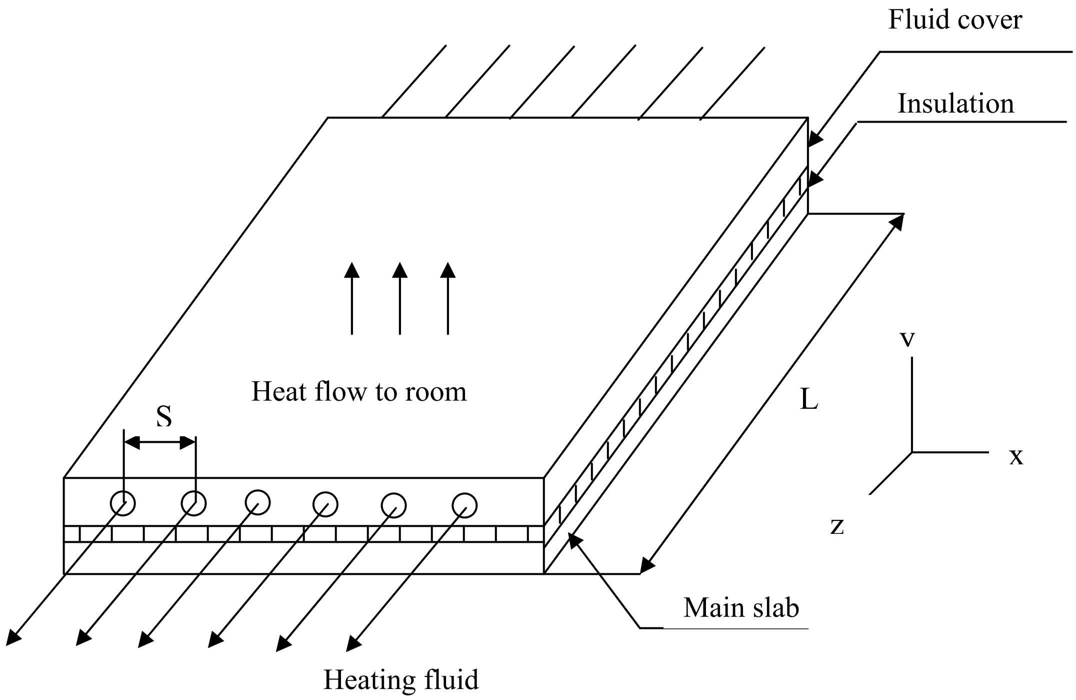

Figure 3.2 Floor heating panel

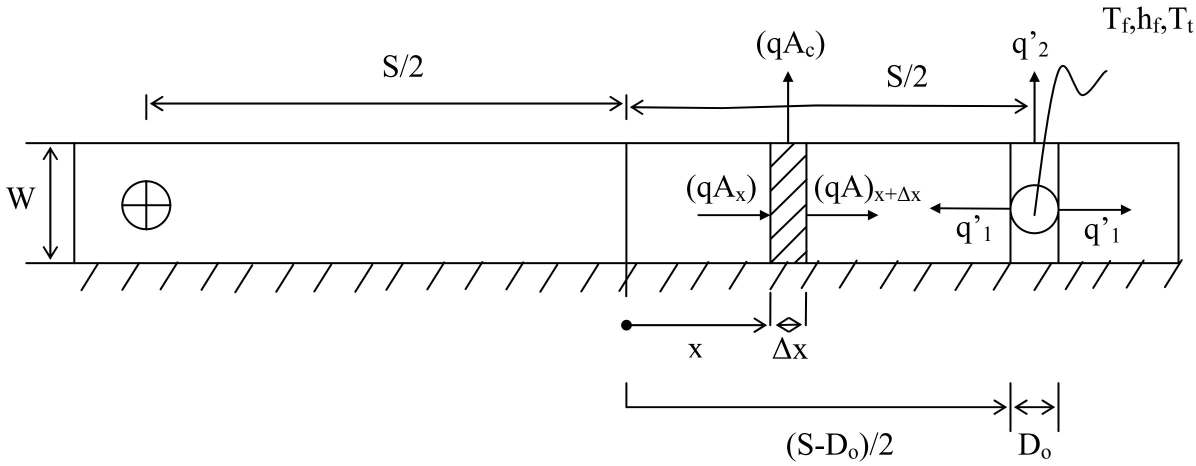

Figure 3.3 Energy balance on an element Δx of the floor heating panel

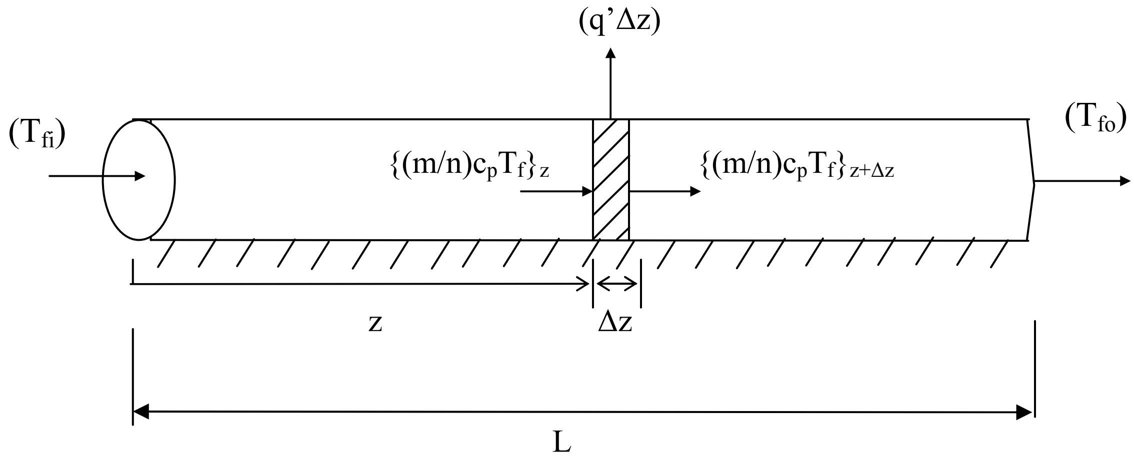

Figure 3.4 Energy balance on an element Δz of the floor heating panel

Where in Figure 3.4 , m and n are the total mass flow rates and the number of parallel tubes ,respectively.

The analytical solution gives another relation for q_f as

$$q_f = F_1 \cdot h_{ps} \cdot (T_f - T_a) \tag{3.25}$$

where

$$F_1 := \left[\frac{S}{D_o + (S - D_o) \cdot F(S)} + \frac{h_{ps} \cdot S}{\pi \cdot D_i \cdot h_i} \right]^{-1} \tag{3.26}$$

$$F(S) := \frac{\tanh\left(m \cdot \frac{S - D_o}{2}\right)}{m \cdot \frac{S - D_o}{2}} \tag{3.27}$$

$$m = [h_{ps} / k_{eq} \, . \, t_y]^{1/2} \tag{3.28}$$

In the preceding equations, F_1 (dimensionless) is the panel efficiency factor, h_{ps} (W/m^2K) is the convection heat transfer coefficient for the panel surface, T_f (°C) is the mean temperature of the heating water, S and t_y (m) represent the pipe spacing (distance between the axes of two adjacent tubes) and the total thickness of the floor materials above the pipe center, respectively. Standard pipe spacing values are presented in Table 3.1 [25]. D_0 and D_i (m) are the outside and inside pipe diameters, respectively, h_f (in W/m^2K) is the convection heat transfer coefficient on the pipe inside surface, and k_{eq} (in W/mK) is the equivalent thermal conductivity of the floor materials above the pipe center. k_{eq} is defined as

$$k_{eq} := \frac{t_y}{R_y} \tag{3.29}$$

where R_y is the thermal resistance between the pipe center and the surface of the panel (m^2K/W) found by

$$R_y := \sum_{i=1}^{n_1} \frac{t_i}{k_i} \tag{3.30}$$

n_1 is the number of structural material layers above the pipe center.

Table 3.1 Standart pipe spacing values.

S [m]	0.05	0.10	0.12	0.15	0.20	0.25	0.30	0.35

3.2.2. Heating Efficiency

Depending on the thermal resistance between the pipe centre and the downside space, some amount of heat supplied to the floor heating panel flows to the downside space. Heating efficiency is given as [47]

$$X := \frac{1}{1 + \frac{R_y \cdot (T_b - T_n)}{R_a \cdot (T_b - T_p)}} \tag{3.31}$$

Where R_a is the thermal resistance between the pipe centre and the downside space (m^2K/W), T_b (°C) is the mean temperature along the centres of the tubing, and T_n (°C) is the downside space temperature. T_b and R_a are calculated by the following relations

$$T_b = T_p + q_f \,.\, R_y \tag{3.32}$$

$$R_a := \left(\frac{1}{h_n}\right) + \left(\sum_{i=1}^{n_1} \frac{t_i}{k_i}\right) \tag{3.33}$$

Where h_n ($W/m^2.K$) is the convection heat transfer coefficient for the downside space surface.

R_y, R_a, k_{eq} and t_y values of the floors of the sample building are listed in Appendix A.1.

3. 2.3. Heat Supplied by the Heating System

Rate of energy which should be supplied by the heating system (absorption heat pump in this study) to the heating water circulating through the floor heating panel per unit area of the panel, q_s (W/m^2) is found by

$$q_s := \frac{q_f}{X} \tag{3.34}$$

3.2.4. Exergy Analysis of Heating Panel

A rough exergy analysis has been made assuming mean plate temperature. The exergetic efficiency of the panel is computed with the following equation as,

$$n_{floor} := \left(\frac{\psi_{rec}}{\psi_f}\right) \tag{3.35}$$

where in this equation ψ_f is the exergy supplied to the water flowing through the pipes and ψ_{rec} is the exergy recovered from the flowing water exergy. ψ_f , ψ_{rec} are given as:

$$\psi_f := m_f \cdot C_{pf} \cdot \left[\left(T_{in} - T_{out}\right) - T_o \cdot \ln\left(\frac{T_{in}}{T_{out}}\right)\right] \tag{3.36}$$

and

$$\psi_{rec} := Q_s \cdot \left(1 - \frac{T_o}{T_p}\right) \tag{3.37}$$

In the above equations Q_s is the heat supplied to the space ,T_{in},T_{out},T_p and T_o are the temperatures of the water entering to the panel,exiting from the panel,mean plate temperature and ambient air temperatures,respectively.

3.2.5. Procedure of Floor Heating Panel Modelling

- Maximum value of heat flux: which can be delivered to a space $q_{f,max}$ (W/m^2) is found by considering that this value is limited by the maximum value of floor surface temperature $T_{p \cdot max}$ (°C), which is given as 29°C in ASHRAE [25]

$$q_{f,max} = 8.92\ (T_{p \cdot max} - T_a)^{1.1} \tag{3.38}$$

- The appropriate value of pipe spacing S, which ensures that all heat offered by the AHP may be transferred to the space, under specified mean fluid temperature and inside air design temperature is calculated by solving for S in the equation

$$q_{f,des} = F_1(S).h_{ps}.(T_f - T_a) \tag{3.39}$$

Where $q_{f,des}$ is the design heating load of the space per unit area of the panel (in W/m^2).

$$q_{fdes} := \frac{Q_H}{A_P} \tag{3.40}$$

- A standard pipe spacing value is selected from Table 3.1. This value should be the first standard value smaller than the value calculated in the previous

- step. It should be taken into account that, as S is decreased, heat flux increases moderately, but heat flux per unit length of the heating pipes decreases, hence (output) efficiency decreases. Thus, high S values should be preferred as much as possible, especially from an initial cost point of view, However, if the pipes are too loosely placed, temperature differences occur within the floor. On the other hand, if the pipes are too densely placed, $T_{p,max}$ may be exceeded which would cause uncomfort.

- Heat flux delivered to the room with the selected value of pipe spacing is calculated from equation (3.25). This value should not exceed $q_{f,max}$

- X is calculated from equation (3.31), after finding T_p as below

$$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}} \tag{3.41}$$

and T_b via equation (3.32).

- Finally, q_s is found from equation (3.34), and heat which should be supplied by the heating system to the heating water. Q_s (in W) is calculated

$$Q_s = q_s \,.\, A_p \tag{3.42}$$

Mathcad program outputs applied to the above procedure for each room of the sample building are given Appendix B.2. While the tabulated results are presented in Appendix B.1.

3.3. Thermodynamic Property Data for H_2O-LiBr Solution and Pure H_2O

The dew point temperature of the H_2O-LiBr solution is given by Patterson and Perez-Bianco [37] as:

$$T_{dp} := \sum_{i=0}^{5} \sum_{j=0}^{2} C_{i,j} \cdot X^{i} \cdot T^{j} \tag{3.43}$$

Where T is the solution temperature given in °C, T_{dp} is in °C and X is the solution concentration (weight percent) of LiBr in %. Coefficients C are presented in Table 3.2.

Table 3.2 Coefficients for equation (3.41).

C_{00}=1.313448E-1	C_{01} = 9.967944E-1	C_{02}= 1.978788E-5
C_{10}= 1.820914E-1	C_{11} = 1.778069E-3	C_{12}= -l. 77948 JE-5
C_{20} = -5.177356E-2	C_{21} = -2.215597E-4	C_{22} = 2.002427E-6
C_{30} = 2.827426E-3	C_{31} = 5.913618E-6	C_{32} = -7.667546E-8
C_{40} = -6.38054 1E-5	C_{41} = -7. 3.0855 6E-8	C_{42}= 1.201525E-9
C_{50} = 4.340498E-7	C_{51} == 2.788472E-10	C_{52} = -6.641716E-12

Vapour pressure of the water arising from the LiBr-water solution is calculated as shown by McNeely [32]:

$$P_{sat} := 10^{\left(k_0 + \frac{k_1}{T_{dp}} + \frac{k_2}{T_{dp}^{2}}\right)} \cdot (6.89474) \tag{3.44}$$

Where T_{dp} is the absolute dew point temperature in °C and P_{sat} is in kPa.

Due to the non-volatility of LiBr and the large difference between the boiling points of H_2O (100°C) and LiBr (1265°C), absorbent (LiBr) pressure is neglected and the total pressure of the solution is taken as the refrigerant (H_20) pressure, at the related state points.

Equations (3.43) and (3.44) are applicable in the LiBr concentration range of 0% to 70%, dew point temperature range of -20°C to 110°C and solution temperature range of 4.4°C to 182.2°C.

Lei and Bunn [27] suggested that equilibrium conditions for the H_2O-LiBr could be found by calculating the refrigerant temperature and refrigerant pressure from the solution temperature and solution concentration with the following equations:

$$T_r = (T-b)/a \tag{3.45}$$

and

$$\log_{10}P = 7.050 - \left(\frac{1603.5406}{T_r + 273.15}\right) - \left[\frac{104095.51}{(T_r + 273.15)^2}\right] \tag{3.46}$$

where T_r is the refrigerant temperature in °C, T is the solution temperature in °C, and P is the refrigerant pressure in kPa.

a and b are given as functions of X, LiBr solution concentration (weight percent) in %, with the following relations:

$$a = -2.008 + (0.17)\text{-}\ X\text{-}\ (3.133E\text{-}3)\text{-}X^2 + (1.977E\text{-}5)\text{-}X^3 \tag{3.47}$$

$$b = 178.404 - (10.734)X + (0.208) \cdot X^2 - (1.147E\text{-}3) \cdot X^3 \qquad (3.48)$$

Equations (3.43) and (3.44) are valid in the following ranges:

$45 \leq X \leq 700\%$

$-18 \leq T_r \leq 110°C$

$16 \leq T_S \leq 166°C$

Enthalpy of the H_2O-LiBr solution is calculated with the relation suggested by Patterson and Perez-Bianco [38]:

$$h := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X^i \cdot T^j \qquad (3.49)$$

where h is in kJ/kg and the coefficients A are tabulated in Table 3.3.

Region of the temperature-concentration space covered by the above fit is:

$0 < T < 180°C$

$45 < X < 70\%$

Table 3.3 Coefficients for equation (3.49).

A_{00}= 1.134125EO	A_{01}= 4.1294891EO	A_{02} = 5.743693E-4
A_{10} = -4.800450E-l	A_{11} = -7.643903 E-2	A_{12}= 5.87092 IE-5
A_{20} = -2.161438E-3	A_{21} = 2.589577E-3	A_{22} = -7.3753 19E-6
A_{30}= 2.336235E-4	A_{31} == -9.500522E-5	A_{32} = 3.277592E-7
A_{40} = -1.188679E-5	A_{41} = 1.708026E-6	A_{42} = -6.062304E-9
A_{50}= 2.291532E-7	A_{51}= -1.102363E-8	A_{52} = 3.901897E-11

Alizadeh [1] has shown that the specific heat of a H_2O-LiBr solution can be approximately taken as:

$$c_p = -3.09.X + 4.18 \tag{3.50}$$

Where c_p is in kJ/kgK.

The following form is used to calculate the enthalpy of the saturated water vapour and superheated water vapour [32]:

$$h_v = [[(0.00274)\text{-}(\ 1\ .8\text{-}T+32) - 0.989805)]\text{-}(P/6.89474) + (0.44942X1.\ 8\text{-}T+32) + 1060.8] \bullet 2.326 \tag{3.51}$$

Where h_v is in kJ/kg, T is in °C. and P is in kpa.

In the simulation, specific heat values for water and steam are taken as 4.18 kJ/kgK and 2.0 kJ/kgK, respectively, for practical purposes.

Entropy value of the solution is computed by the correlation given by Feuerecker [20]

$$s := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X^{j} \cdot T^{i} \tag{3.52}$$

where s is in kj/kgK and the coefficients B are tabulated in Table 3.4 . Region of the temperature and concentration are

$0 < T < 180°C$

$45 < X < 70\%$

Entropy of the saturated water vapour and superheated water vapour are also calculated by the same equation. (3.52)

Table 3.4 Coefficients for equation (3.52).

$B_{0,0} := 5.127558 \cdot 10^{-1}$	$B_{0,1} := -1.3939 \cdot 10^{-2}$	$B_{0,2} := 2.92414 \cdot 10^{-5}$
$B_{0,3} := 9.03 \cdot 10^{-7}$	$B_{1,0} := 1.227 \cdot 10^{-2}$	$B_{1,1} := -9.156 \cdot 10^{-5}$
$B_{1,2} := 1.821 \cdot 10^{-8}$	$B_{1,3} := -7.99 \cdot 10^{-10}$	$B_{2,0} := -1.36489 \cdot 10^{-5}$
$B_{2,1} := 1.0689 \cdot 10^{-7}$	$B_{2,2} := -1.381 \cdot 10^{-9}$	$B_{2,3} := 1.529 \cdot 10^{-11}$
$B_{3,0} := 1.021 \cdot 10^{-8}$	$B_{3,1} := 0$	$B_{3,2} := 0$
$B_{3,3} := 0$		

3.4. Assumptions for the AHP System

Following assumptions are made in the analysis of the solar-assisted absorption heat pump system:

a) Analysis is carried out under steady state conditions.
b) Solution is saturated at the exit of the generator.
c) Liquid refrigerant at the exit of the condenser and the refrigerant vapour leaving the evaporator are saturated.
d) Absorber and condenser operate at the same temperature,
e) For the mixture heat exchanger, low-temperature end approach is assumed as 10°C, for the design (primary) simulation.
f) The reference enthalpy (h_o) and entropy (s_o) used for calculating the exergy of the working fluid are the entalphy and entropy of water at an environment temperature (T_o) of $25\,^{o}C$

g) Heat exchange between the system and surroundings, other than that prescribed by heat transfer at the generator, evaporator, condenser and absorber does not occur.

Temperature assumptions and data used in the simulation are presented in Table 3.5.

Table 3.5. Assumptions and data used in the design simulation.

Simulation Parameter	Assumption / Data
Generator Temperature, T_g [°C]	82°C by auxiliary heating
Absorber Temperature, T_{ab} [°C]	Equal to floor heating panel design inlet temperature
Condenser Temperature, T_c [°C]	Equal to absorber temperature
Evaporator Temperature, T_e [°C]	28 °C by the energy supplied from Sun
Solar Insolation I [W/m^2]	Hourly, for 60° inclination angle and 0° south orientation [36]
Ambient Air Temperature, T_o[^{o}C]	Hourly ambient air temperature of Ankara for 21st day of the month (1988) [19]

3.5. Simulation Algorithm of the AHP

3.5.1. Design Simulation Algorithm of the AHP

Temperatures of the state points are (Figure 1.1):

$$T_1 = T_2 = T_8 = T_{ab} = T_c \tag{3.53}$$

$$T_4 = T_7 = T_g \tag{3.54}$$

$$T_{11} = T_{10} = T_e \tag{3.55}$$

$$T_5 - T_2 = 10°C \tag{3.56}$$

(For the mixture heat exchanger, low-temperature end approach is assumed as 10°C, for the primary simulation.)

Low-side and high-side pressures of the cycle are calculated via equation (3.44), at the evaporator and condenser temperature, respectively. Since it is assumed that states 8 and 11 are saturated and that no LiBr exists at these state points, solution temperature is taken as equal to the dew point temperature at the mentioned states.

Concentration of the solution at the exit of the absorber, X_1, is computed from equation (3.46) at T_1 and P_L. Similarly, concentration of the solution at the exit of the generator, X_4, is found by the same equation, at T_4 and P_H

Since the solution from the generator enters the absorber through the mixture heat exchanger,

$$x_4 = x_5 = x_6 = x_w \tag{3.57}$$

Similarly,

$$x_1 = x_2 = x_3 = x_r \tag{3.58}$$

Moreover, since it is assumed that only pure refrigerant (water vapour) is driven out from the solution in the generator,

$$x_7 = x_8 = x_9 = x_{10} = x_{11} = x_{12} = 0 \tag{3.59}$$

Mass balance for the absorber gives:

$$m_1 = m_6 + m_{12} \tag{3.60}$$

$$m_1 y_1 = m_6 y_6 + m_{12} \tag{3.61}$$

where y is the weight fraction of H_2O in the mixture.

Rearranging the above equations yields circulation ratio (CR) (defined in the "performance parameters" section) as:

$$CR = \frac{m_1}{m_{12}} = \frac{1 - y_6}{y_1 - y_6} \tag{3.62}$$

or in terms of x, since y = 1-x :

$$CR = \frac{m_1}{m_{12}} = \frac{x_6}{x_6 - x_1} \tag{3.63}$$

h_1, h_2 and h_4 are calculated from equation (3.49), knowing the temperature and the concentration at these states. h_8 is also found from the same equation, since at state 8, H_2O is assumed to be saturated liquid with LiBr concentration of zero.

Saturated vapour enthalpy of H_2O, State (h_{11}) is calculated at T_{11} and P_L,. Superheated vapour enthalpy of H_2O, State (h_7) is calculated at T_7 and P_H, by the same equation (3.51).

s_1 ,s_2 , s_4 and s_8 are calculated with the equation 3.52

Treating the refrigerant heat exchanger as counterflow type, which according to Ergül [19], is the preferred flow configuration in absorption systems, T_9 is given as:

$$T_9 := T_8 - \frac{C_{minr}}{C_{hr}} \cdot \varepsilon_r \cdot (T_8 - T_{11}) \tag{3.64}$$

It should be noted that for the temperature ranges considered, $C_{pc,r}$ (specific heat of water vapour) is in the range of 1.9 to 2.1 kJ/kg. K and $c_{ph \cdot r}$ (specific heat of liquid water) is about 4.18 kJ/kg. K, hence $C_{min} = C_c$ and $C_{max} = C_h$

In the simulation $C_{min,r}$ /$C_{h,r}$ is taken as 0.478, assuming $c_{pc,r}$ is equal to 2.0 kJ/kg-K, and $c_{ph,r}$ is equal to 4.18 kJ/kg.K, while the effectiveness of the same heat exchanger, ε_r, is taken as 0,7.

T_{12} is calculated from:

$$T_{12} := T_{11} + \frac{C_{hr}}{C_{cr}} \cdot (T_8 - T_9) \tag{3.65}$$

h_9 is computed via equation (3.49), while due to the throttling process in the expansion valve:

$$h_{10} = h_9 \tag{3.66}$$

s_{10} is calculated with the equation 3.52

Energy balance across the refrigerant heat exchanger gives h_{12} as

$$h_{12} = h_8 + h_{11} - h_9 \tag{3.67}$$

s_{12} is calculated with the equation 3.52

Overall heat transfer coefficient area- product (UA value) (kW/K) of the refrigerant heat exchanger is given by:

$$(AU_r) := \frac{\left(\ln\left(1 - \varepsilon_r \cdot C_{rr}\right) - \ln\left(1 - \varepsilon_r\right)\right)}{1 - C_{rr}} \tag{3.68}$$

where

$$C_r = C_{min} / C_{max} \tag{3.69}$$

This value of $(UA)_r$ is a fixed design parameter, and is used to find ε_r in the latter simulations, in case $C_{min,r}$ (or $C_{pc.r}$) and $C_{max,r}$ (or $C_{ph,r}$) are varied.

For the mixture (solution) heat exchanger, low-temperature end approach is taken as 10°C, for the design simulation, therefore:

$$T_5 = T_2 + 10 \tag{3.70}$$

h_5 is also calculated from equation (3.49).

Due to the throttling process occurring through the pressure reducing valve:

$$h_6 = h_5 \tag{3.71}$$

s_6 is calculated with the equation 3.52

Then, mass flow rates are found. Energy balance across the absorber yields:

$$Q_a = m_6 h_6 + m_{12} h_{12} - m_1 h_1 \tag{3.72}$$

where m_1, and m_6 can be written in terms of m_{12} :

$$m_1 = CR.m_{12} \tag{3.73}$$

$$m_6 = m_1 - m_{12} \tag{3.74}$$

Thus, absorber capacity becomes:

$$Q_a = m_{12}.[(CR-1).h_6 + h_{12} - CR.h_1] \tag{3.75}$$

Condenser capacity is:

$$Q_c = m_7 (h_7 - h_8) \tag{3.76}$$

Sum of the capacities of the absorber and condenser, which should be delivered to the floor heating system, is given as a design condition and is represented by Q_{ahp} .

$$Q_{ahp} = Q_a + Q_c \tag{3.77}$$

Therefore, m_{12} is obtained by using the following relation:

$$m_{12} := \frac{Q_{ahp}}{\left[(CR-1)\cdot h_6 + h_{12} - CR\cdot h_1 + h_7 - h_8\right]} \qquad (3.78)$$

m_1, and m_6 are calculated via equation (3.73) and equation (3.74), respectively.

The obtained m_1, value, which is the mass flow rate of the solution pump, is one of the constants of the AHP.

Mass flow rates of the remaining state points are then:

$$m_3 = m_2 = m_1 \qquad (3.79)$$

$$m_4 = m_5 = m_6 \qquad (3.80)$$

$$m_7 = m_8 = m_9 = m_{10} = m_{11} = m_{12} \qquad (3.81)$$

Next, C_{min} and C_{max} for the mixture heat exchanger are determined, after calculating c_{ph} and c_{pc} for the micture heat exchanger via equation (3.50).

T_3 is found by the following relation, treating the mixture heat exchanger as counterflow type:

$$T_3 := T_2 + \frac{C_{hm}}{C_{cm}}\cdot\left(T_4 - T_5\right) \qquad (3.82)$$

Energy balance across the mixture heat exchanger yields h_3 as:

$$h_3 := \frac{m_2 \cdot h_2 + m_4 \cdot h_4 - m_5 \cdot h_5}{m_3} \tag{3.83}$$

Effectiveness of the same device is computed from:

$$\varepsilon_m := \frac{C_{hm} \cdot (T_4 - T_5)}{C_{minm} \cdot (T_4 - T_2)} \tag{3.84}$$

$(UA)_m$ is then calculated from ε_m , as done in the refrigerant heat exchanger.

$$UA_m := \frac{\ln(1 - \varepsilon_m \cdot C_{rm}) - \ln(1 - \varepsilon_m)}{1 - C_{rm}} \cdot C_{minm} \tag{3.85}$$

Finally, capacities of the AHP components are calculated.

Generator heat input rate:

$$Q_g = m_4 h_4 + m_2 h_7 - m_3 h_3 \tag{3.86}$$

Cooling effect rate of the evaporator:

$$Q_e = m_{11}(h_{11} - h_{10}) \tag{3.87}$$

Heating effect rate of the absorber:

$$Q_a = m_6 h_6 + m_{12} h_{12} - m_1 h_1 \tag{3.88}$$

Heating effect rate of the condenser:

$$Q_c = m_7(h_7 - h_8) \tag{3.89}$$

Solution pump power (if not neglected):

$$W_p = m_1(h_2 - h_1) \tag{3.90}$$

Mathcad program output written for the design simulation algorithm is given in Appendix D.1.

3.5.2. Simulation Algorithm for Storage Tank Temperature Greater Than $T_{s,ref}$

Following assumptions are made:

- Temperature difference, of the water circulating between the storage tank and the evaporator, ΔT_L is constant (4.5 K) for the evaporator. Hence, evaporator capacity is constant.
- Performance of the AHP does not change as long as T_s does not exceed $T_{s,ref}$(34.5°C).
- Temperatures of the absorber and condenser are constant and equal to the values in the design simulation.

First, heat capacity rate on the waterside of the evaporator, C_h is determined. Since phase change occurs on the refrigerant (vapour) side of the evaporator,

$$C_{c,e} = C_{max,e} = \infty \quad \text{and} \quad C_{h,e} = C_{min,e}$$

$$C_{mine} := \frac{Q_c}{\Delta T_L} \tag{3.91}$$

Treating the evaporator as a counterflow heat exchanger, its effectiveness is computed

$$\varepsilon_e := \frac{Q_e}{C_{mine} \cdot (T_{hi} - T_{ci})} \tag{3.92}$$

where $T_{h,i}$ and $T_{c,i}$ are the design values of $T_{s,ref}$ and T_{10}. ε_e is constant, since $C_{r.e}$= $C_{min.e}$ / $C_{max.e}$= 0

$(UA)_e$ can be calculated from the following relation.

$$(UA)_e = -\ln(1-\varepsilon_e).C_{min.e} \tag{3.93}$$

When T_s exceeds $T_{s\cdot ref}$, T_{10} is found from

$$T_{10} := T_5 - \frac{Q_e}{\varepsilon_e \cdot C_{mine}} \tag{3.94}$$

While P_L and h_{11} are again obtained from equation (3.44) and equation (3.51), respectively. P_H does not change since T_8 is assumed constant.

Then, m_{11} is computed from the following equation after finding T_9, h_9 and h_{10} as in the design simulation.

$$m_{11} := \frac{Q_e}{h_{11} - h_{10}} \tag{3.95}$$

Since m_1 is constant, m_6 is again found by using equation (3.74).

X_1, is computed using equation (3.46), at T_1 and P_L, while x_6 is calculated from

$$x_6 := x_1 \cdot \frac{CR}{CR - 1} \tag{3.96}$$

Knowing X_4 and P_H, T_{r4} is computed via equation (3.46), then T_4 is obtained by using equation 3.45.

h_1, h_2, h_4, h_8, h_7 and T_{12} are found in the same way as in the design simulation.

ε_m is calculated as follows. $(UA)_m$ being a fixed parameter, after computing C_{min} and C_{max}

$$\varepsilon_m := \frac{1 - \exp\left[-NTU_m \cdot \left(1 - C_{rm}\right)\right]}{1 - C_{rm} \cdot \exp\left[-NTU_m \cdot \left(1 - C_{rm}\right)\right]} \tag{3.97}$$

where NTU is the number of transfer units for a heat exchanger and is defined as:

$$NTU := \frac{UA}{C_{mir}} \tag{3.98}$$

Then, T_5 is calculated from:

$$T_5 := T_4 - \varepsilon_m \cdot \frac{C_{minm}}{C_{hm}} \cdot \left(T_4 - T_2\right) \tag{3.99}$$

Finally, h_5, h_6, T_3 and h_3 are found in the same way as in the design simulation, before calculating the capacities of the AHP components.

All the entropy values at states are calculated with respect to the equation 3.52 at the given temperature and concentration values at that states.

3.5.3. Exergy Analysis of the Absorption Heat Pump.

3.5.3.1. Exergy

Second law analysis calculates the system performance based on exergy, which always decreases owing to thermodynamics irreversibilities. Exergy is defined as maximum possible reversible work that can be obtained from a system.

In the absence of electricity, magnetism, surface tension and nuclear reaction the specific exergy of a flow is

$$\psi := \psi_{ph} + \psi_{ch} + \psi_{k} + \psi_{p} \tag{3.100}$$

where ph , ch , k and p stand for physical,chemical , kinetic and potential exergies ,respectively. Neglecting changes in kinetic and potential exergy the specific exergy content of a single component can be expressed as

$$\psi := h - T_o \cdot s - g_o \tag{3.101}$$

Since there is no chemical reaction in the components of the AHP, including the last term of equation 3.99 is the same for incoming and outcoming substances. Steady-flow exergy loss for a component can be expressed as

$$\Delta\psi := \sum_i m_i \cdot \psi_i - \sum_e m_e \cdot \psi_e - \left(1 - \frac{T_o}{T}\right) \cdot Q - W \tag{3.102}$$

The first two terms on the right hand side are the sums of exergy carried by incoming mass and exergy leaving with the outgoing mass. The third term is the exergy of heat Q, which is transfered at constant temperature T.

The last term is the mechanical work transfer to or from the system.

3.5.3.2 Exergy loss in system components

For each component of the absorption cycle, exergy loss can be calculated as follows:

Condenser : Condenser rejects heat during condensing process to the cooling water by an amount Q_c .It is assumed that the heat Q_c is rejected to the cooling water at constant temperatute T_c . Exergy loss in the condenser is given by

$$\Delta\psi_c := m_7 \cdot (\psi_7 - \psi_8) - \psi_c \tag{3.103}$$

which can be written as

$$\Delta\psi_c := m_7 \cdot \left[(h_7 - h_8) - T_0 \cdot (S_7 - s_8) \right] - Q_c \cdot \left(1 - \frac{T_0}{T_c} \right) \tag{3.104}$$

Evaporator : Evaporator gains heat during evaporation process by an amount Q_e at constant temperature T_e . Exergy loss in the evaporator is given by

$$\Delta\psi_{eva} := m_{10} \cdot (\psi_{11} - \psi_{10}) - \psi_e \tag{3.105}$$

which can be written as

$$\Delta\psi_{eva.} := m_{10} \cdot \left[(h_{11} - h_{10}) - T_0 \cdot (S_{11} - s_{10}) \right] - Q_e \cdot \left(1 - \frac{T_0}{T_e} \right) \tag{3.106}$$

Absorber: Absorber rejects heat during condensing process to the cooling water by an amount Q_a .

It is assumed that the heat Q_a is rejected to the cooling water at constant temperature T_a Exergy loss in the absorber is given by

$$\Delta\psi_{abs} := m_{11}\cdot\psi_{12} + m_6\cdot\psi_6 - m_1\cdot\psi_1 - \psi_a \tag{3.107}$$

$$\begin{aligned}\Delta\psi_{abs.} := {} & m_{11}\cdot\left[\left(h_{12} - h_0\right) - T_0\cdot\left(s_{12} - s_0\right)\right] + m_6\cdot\left[\left(h_6 - h_0\right) - T_0\cdot\left(s_6 - s_0\right)\right] \dots \\ & + -\left[m_1\cdot\left[\left(h_1 - h_0\right) - T_0\cdot\left(s_1 - s_0\right)\right]\right] - Q_a\cdot\left(1 - \frac{T_0}{T_a}\right)\end{aligned} \tag{3.108}$$

Generator: Generator gains heat during evaporation process by an amount Q_g at constant temperature T_g. Exergy change in the generator is given by

$$\Delta\psi_{gen} := m_4\cdot\psi_4 + m_7\cdot\psi_7 - m_3\cdot\psi_3 - \psi_g \tag{3.109}$$

which can be written as

$$\begin{aligned}\Delta\psi_{gen.} := {} & m_4\cdot\left[\left(h_4 - h_0\right) - T_0\cdot\left(s_4 - s_0\right)\right] + m_7\cdot\left[\left(h_7 - h_0\right) - T_0\cdot\left(s_7 - s_0\right)\right] \dots \\ & + -\left[m_3\cdot\left[\left(h_3 - h_0\right) - T_0\cdot\left(s_3 - s_0\right)\right]\right] - Q_g\cdot\left(1 - \frac{T_0}{T_g}\right)\end{aligned} \tag{3.110}$$

Pump: The exergy loss in the pump is given by the following equation

$$\Delta\psi_{pp} := m_1\cdot\left(\psi_1 - \psi_2\right) \tag{3.111}$$

which can be written as

$$\Delta\psi_{pp} := m_1\cdot\left[\left(h_1 - h_2\right) - T_0\cdot\left(s_1 - s_2\right)\right] + W_p \tag{3.112}$$

Refrigerant Heat Exchanger : The exergy loss in RHE is given by

$$\Delta\psi_{rhe} := m_{11}\cdot(\psi_{11} - \psi_{12}) + m_{8}\cdot(\psi_{8} - \psi_{9}) \tag{3.113}$$

which can be written as;

$$\Delta\psi_{r.h.e.} := m_{11}\cdot\left[\left(h_{11} - h_{12}\right) - T_{0}\cdot\left(s_{11} - s_{12}\right)\right] + m_{8}\cdot\left[\left(h_{8} - h_{9}\right) - T_{0}\cdot\left(s_{8} - s_{9}\right)\right] \tag{3.114}$$

Solution Heat Exchanger : The exergy loss in SHE is given by

$$\Delta\psi_{she} := m_{2}\cdot(\psi_{2} - \psi_{3}) + m_{4}\cdot(\psi_{4} - \psi_{5}) \tag{3.115}$$

which can be written as ;

$$\Delta\psi_{s.h.e.} := m_{2}\cdot\left[\left(h_{2} - h_{3}\right) - T_{0}\cdot\left(s_{2} - s_{3}\right)\right] + m_{4}\cdot\left[\left(h_{4} - h_{5}\right) - T_{0}\cdot\left(s_{4} - s_{5}\right)\right] \tag{3.116}$$

Total exergy loss in AHP can be calculated as follow:

$$\Delta\psi_{tot} := \Delta\psi_{c} + \Delta\psi_{eva.} + \Delta\psi_{abs.} + \Delta\psi_{gen.} + \Delta\psi_{pp} + \Delta\psi_{s.h.e.} + \Delta\psi_{r.h.e.} \tag{3.117}$$

Percent exergy losses in each component of AHP can be calculated by dividing the exergy loss of the component to the total exergy loss.

Mathcad program output for the above simulation is presented in Appendix D.2 .

3.6. Performance Parameters

Circulation ratio (CR) is the ratio of the strong solution mass flow rate to the refrigerant mass flow rate.

$$CR = \frac{m_1}{m_{12}} \tag{3.118}$$

CR determines the feasibility of the operation besides the physical dimensions and pumping power requirement, of the system. Low values of CR (around 10) are desired in AHP systems.

The performance of an absorption cycle is usually expressed in terms of its coefficient of performance (COP), defined as the ratio of the heating or cooling supplied by the unit, to the sum of the high temperature heat input and the pump power, if the latter term is not neglected. Two different parameters may be defined, one for the absorption cycle and the other for the whole system, such that the COP of the cycle is given as:

$$COP := \frac{Q_a + Q_c}{Q_g + W_p} \tag{3.119}$$

and the COP of the system is expressed by [48]:

$$COP_{sys} := \frac{E}{I \cdot A_c + \left(Q_{auxt} + Q_{st}\right) + W_p} \tag{3.120}$$

In an absorption heat pump the products are the absorber and condenser heat.Thermal energy is supplied to the generator and low grade energy is supplied to the evaporator i.e. outdoor ambient winter.Exergetic efficiencies are calculated by the following formula.

$$ECOP := \frac{\left[Q_c \cdot \left(1 - \frac{T_o}{T_c}\right) + Q_a \cdot \left(1 - \frac{T_o}{T_a}\right)\right]}{Q_g \cdot \left(1 - \frac{T_o}{T_g}\right) + W_p} \tag{3.121}$$

However, for a solar-aided unit the contribution of solar input to the total energy needed to operate the unit is more convenient and informative. This ratio, named the “fraction of non-purchased energy” is expressed as:

$$FNP := \frac{Q_i}{Q_i + \left(Q_{auxt} + Q_{st}\right) + W_p} \tag{3.122}$$

and it indicates "how much" solar energy is used to operate the system.

Solar-aided systems may be judged with another parameter when FNP is used to see what fraction of the input is solar energy. This parameter, called the “solar performance coefficient”, which compares the auxiliary input with the load and indicates “how effectively” solar energy is used, was first proposed by Ileri [22] and was defined as:

$$SPC := 1 - \frac{\left(Q_{auxt} + Q_{st}\right) + W_p}{E} \tag{3.123}$$

3.7.Simulation Method and Comments about the Simulation

- Total heating load (Q_H) of the sample building is calculated, followed by the computation of the probable hourly heating load (E) using the degree-hour method [18]. In this method, it is assumed that on a long term average, solar and internal gains would offset heat loss when the hourly ambient air temperature is 18°C, and that the fuel consumption would be proportional to the difference between the hourly ambient air temperature and 18°C. The equation for calculating E using the degree-hour method is [19]:

$$E := Q_H \cdot \frac{(18 - T_o)}{\Delta T_d} \tag{3.124}$$

- Where ΔT_d (°C) is the difference between the inside air design temperature (taken as avarage 17°C) and the ambient air design temperature (-12°C).

- Using the calculated heat loss of each room, tube spacing, panel temperature, heating loads etc.for floor heating system is determined for each space. The system is designed for $T_{f,i}$ and $T_{f,o}$ values of in 50°C and 40°C, respectively.

- After finding the rate of total energy, which should be supplied by the AHP to the floor heating system, Simulation calculation for AHP are performed. It was found that the optimum generator temperature, T_g was 82°C for absorber and condenser temperatures of 50°C. This is parallel to the conclusion of Lei and Bunn [28], stating that the optimum generator temperature is 85°C for a solar-driven absorption unit with flat plate solar collectors.

- Then, simulation of the whole system was performed for the characteristic day (21st day) of January, February, March, April, October, November and December, using the hourly ambient air temperature and the hourly solar insolation values of these days.

- Daily average values of heating load, working time, rate of heat addition by the collectors and theauxiliary energy consumed was calculated, and the performance parameters of the whole system (AHP, Solar collector , storage tank etc.) were obtained.

- Fraction of working time length (FWT) of the AHP unit in each hour, was taken as the ratio of the probable hourly heating load, E to the designvalue of the rate of energy which should be supplied by the AHP unit to the floor heating system, Q_{ahp}. It was assumed that this was achieved by an automatic control unit operating according to the inside air temperature.

- It was observed by Ergül [19] that such a solar-assisted system AHP requires large collector areas for effective operation and to get high performance values. Furthermore. Antonopoulos has shown that , the optimum value of the ratio of the floor heating area to the collector area, f is in the range of 1 to 3. Therefore, almost all of the roof area of the sample building (except the terrace) was utilised for placing collectors of 100 m^2 total area, which gives values of f 1.5, since total heated floor area is 150 m^2. Ms was taken as 7500 kg (75 kg per m^2 of collector area) and (UA)s as 0.015kW/K

- Lower values for $T_{s.ref}$ is preferable and FNP improves at lower values. Because auxiliary heating of the tank outlet decreases. So , $T_{s.ref}$ is taken as 34.5°C (6.5°C above T_e). Below this value, the system could not supply all of the heating demands.

- As mentioned in section 3.3. some of the equations that are used to obtain the thermodynamic properties of the H_2O-LiBr solution are applicable in the range of $45 \leq X \leq 70\%$.The AHP is switched-off when T_s exceeds 41.5 °C,Therefore, for cases of $41°C \leq T_s \leq 50^{Q}C$, water leaving the storage tank is delivered to the floor heating system after its temperature is boosted to 50°C via AUX_2.

When T_s exceeds 50°C, again water leaving the storage tank is circulated through the floor heating system, but this time no auxiliary energy is used

- Starting temperatures of the storage tank (at 00:00) in each month was optimised, and was determined as 40°C for all months.

- When Q_u is a negative quantity, that is when the collector inlet temperature is greater than its outlet temperature, collector is not operated (its pump is switched off so that water is not circulated through it).

- Heating load of the sample building if it was heated by a conventional system of storey boiler was calculated by the equation below. It was also assumed that this system would use natural gas with a thermal efficiency of 0.8.

$$E_{conv} := \left(E_{floorheating}\right) \cdot \frac{Q_{H_{conv}}}{Q_{H_{floorheating}}} \tag{3.125}$$

- All the above operation of the system is assumed to be happening via an automatic control system, receiving information from temperature sensors.

3.8 Exergy Analysis of the Overall System

Exergy analysis can be performed by three different cases. As mentioned in section 3.7. Reference storage tank temperature is taken as 34.5 ^{o}C and AHP is schwitched off when T_s exceeds 41.5 ^{o}C.There are three cases that should be considered.

i) The storage tank temperature is less than 41.5 ^{o}C

When the storage tank temperature is less than 34.5 ^{o}C .The water coming from storage tank is heated by AUX $_2$ up to the temperature of 34.5 ^oC.Then the water flows through evaporator and AHP is switched-off.

The overall efficiency is calculated by the following equation

$$\eta_{overall} := \frac{\psi_{rec}}{\psi_i} \tag{3.126}$$

When the storage tank temperature is greater than 34.5 ^oC and less than 41.5 ^oC , water coming from the storage tank is directly circulated through evaporator and AHP is switched on.Then the overall efficiency becomes

$$\eta_{overall} := \frac{\psi_{rec}}{\psi_i + \psi_{aux2}} \tag{3.127}$$

In the above equations , ψ_{rec} , ψ_i and ψ_{aux} are calculated with the equations 3.37 , 3.4 and 3.17, respectively.

The exergetic efficiency of the collector evaporator subsystem without preheating with AUX_2 is calculated with the following equation as

$$\eta_{IIsub} := \frac{\psi_e}{\psi_i} \tag{3.128}$$

In this equation , ψ_e is the exergy supplied to the evaporator and given as

$$\psi_e := m_L \cdot C_{pwL} \cdot \left(T_{fe} - T_L - T_0 \cdot \ln\left(\frac{T_{fe}}{T_L} \right) \right) \tag{3.129}$$

where T_{fe} and T_L are the temperature of the water leaving and entering to the evaporator ,respectively.

The exergetic efficiency of the collector evaporator subsystem with preheating by AUX_2 is calculated with the following equation as

$$\eta_{IIsub} := \frac{\Psi_e}{\Psi_i + \Psi_{aux2}} \tag{3.130}$$

ii) The storage tank temperature is greater than 41.5 °C and less than 50 °C

When the storage tank temperature is greater than 41.5 °C ,the water temperature is increased upto 50 °C and the AHP is switched off.

The overall efficiency is again calculated by the equation 3.127.

iii) The storage tank temperature is greater than 50 °C

When the storage tank temperature is greater than 50 °C ,the water coming from storage tank is directly circulated through the heating panel ,the second law efficiency for the overall system is given by equation 3.126

The Mathcad Program written for the second law efficiencies are given in Appendix E

CHAPTER 4

DISCUSSION AND RESULTS

4.1. Discussion

A thermodynamic modelling of a solar assisted absorption heat pump was presented, which is to be coupled to the floor heating system of an existing duplex house in Ankara. Floor heating panel was designed for each heated space in the building after calculating their heating loads. An AHP was designed to supply energy to the floor heating system, where its generator is heated with an auxiliary heater and its evaporator receives solar energy. Water-lithium bromide was used as the refrigerant-absorbent pair in the AHP cycle. Only heating was performed in this study with the AHP.

Performance parameters of the system was calculated after determining the useful energy supplied to the system and the auxiliary energy consumed by the system, besides the storage tank temperature, working time length for each hour. Results are tabulated in Tables 4.1 to 4.13 and are presented in Figures 4.1 to 4.18. Optimum storage tank temperature at the beginning of a 24-hour period was determined after a number of successive simulations. This leads to a more uniform auxiliary energy consumption and storage tank temperature throughout the day as can be observed from Tables 4.1 and 4.7. Increasing the storage tank temperature to 40°C at the beginning of the day in all months however decreases useful energy gain and collector efficiency. The reason for this is that losses from the collectors increase, as presented in Table 4.2, besides the increase in the auxiliary energy required to boost the temperature to 40°C, but it was observed that the reduction in total auxiliary energy was greater than the increase in useful energy.

Results obtained for the whole year are tabulated in Table 4.8. It is deduced that heating load and total auxiliary energy are in phase with each other throughout the year. Also, utilisation of solar energy in winter months are quite low, but this is compensated in spring and fall leading to a yearly FNP of 0.463 and SPC of 0.305, which means that 30.5% of the heating demand is met by solar energy while 46.3% of the energy consumed in supplying this demand is solar energy. Moreover, value of Q_{st} at the at the end of the year is about 7% of the total auxiliary energy consumed, and it does not have a big effect on the yearly performance of the system.

Figure 4.4 shows the comparison between the energy which would be consumed by a conventional heating system with radiators and the floor heating system receiving energy from solar-driven AHP. It is assumed that the conventional system (which has about 20% higher design heating load) uses natural gas with a thermal efficiency of 0.8 as the auxiliary heaters of the system with AHP. Furthermore, it is found that the savings in energy in one year would amount to 49,545 kWh, by instead of using a conventional system that would spend 91,995 kWh, which yields about 54% saving in energy consumption. However, higher initial cost of the system with AHP should also be considered.

For the generator temperature increase from 60 to 120 ^{o} C and the evaporator temperature of 20, 25, 28 ,30 and 35 ^{o} C percent exergy losses are calculated for all components , first and second law efficiencies of the AHP system are given in the Figure 4.12 to 4.18 for the condenser , evaporator , absorber , generator , pump of the AHP respectively.

When the generator temperature increases from 60 to 120 ^{o}C. As it is seen Figure 4.12 percent exergy loss in the condenser chances. As the temperature of the generator increases up to 82 ^{o} C the percent exergy loss decreases to 0,2416. At the temperatures above this value results in the increase in the percent exergy loss in the condenser .This means that there is a generator temperature at which the percent exergy loss is minimum for an evoparotor temperature. That is, when the evapotator temperature is increased from 20 to 35 ^{o} C the same behaviour is obtained with different exergy loss. Increasing evaporator temperature or reducing the condenser temperature not only increases heat transfer irreversibilitiy but also increases solution circulation ratio and thus irreversibilities in the generator and absorber. Irreversibility in condenser results from the water vapour coming from the generator entering with a certain degree of superheat.

The exergy loss in the evaporator results mainly from the temperature difference between the environment and the evaporating refrigerant. Similar plots are obtained for the percent exergy losses in the evaporator with the increase in the generator temperature from 60 to 120 ^{o}C. The values are comparatively higher.It is seen from figure 4.13 that when choosing evaporator, generator values must be chosen around 80-85 ^{o}C and evaporator temperature can be choosen as small as possible. Although evaporator heat transfer irreveribility is lower than the exergy loss occuring in the absorber and generator ,it has a high impact on the system performance ,in order to improve cycle performance first priority should be given to the evaporator.

As can be seen in Figure 4.14. Increase in the generator temperature result in the increase in the dimensionless exergy losses in the absorber .To determine the working conditions for the absorber or to make the absorber more efficient ,generator temperature must be as low as possible. Increase in the evaporator temperature couldn't change the dimensionless exergy loss in the absorber considerabely. As can be seen from the percent loss ,increasing the generator temperature from 60 to 90 ^{o} C increases the losses from 24 to 26. Increasing the evaporator temperature from 20 to 35 ^{o} C increases the percent losses from 24 to 27 .This means that evaporator temperature must be taken as low as possible.Increasing the generator temperature results in a lower solution circulation rate.However,this may not reduce the irreversibilities at the absorber,because the irreversibility increases along with the temperature of the solution entering the absorber

Increase in the generator temperature decreases the percent exergy loss in the generator (Figure 4.15). Therefore, the generator temperature must be chosen as high as possible.The changes in the evaporator temperatures has an important effect in the exergy loss. The internal irreversibility at the generator also increases,even with the reduced solution circulation rate, because of the higher solution temperature.However the generator heat transfer irreversibility is reduced significantly,causing the overall generator irreversibility to fall.When considering the overall effect of increasing generator temperature,the sum of the absorber and the generator irreversibilities decreased,and so the cycle is more efficient.It must be remembered , however, that the generator temperature cannot be increased without limit.,because at some point the solution will be saturated (Crystalization will take place).Also when the system is operated with the solution in the generator close to saturation condition,the effectiveness of the solution heat exchanger must be reduced to avoid crystallization before it enters the throttling valve.

To obtain low pump loss ,evaporator temperature must be chosen moderate (28 C). As in figure 4.16 ,the percent exergy losses decrease with the increase in the generator temperatue up to 90 ^{o}C. At temperature above this value does not chance the dimensionless exergy loss in the pump.

Percent exergy loses in each component are calculated and results of the calculations are shown in figure 4.12 to 4.18. Decreasing the evaporator temperature and increasing the generator temperature cause a decrease in exergy losses for all components.Results are support the chance of 28 ^{o}C for evaporator and 82 ^{o}C for generator temperature .

In figure 4.18 ,second law efficiencies are shown. Second law efficiencies increases with an increase in the generator temperature up to 82 ^{o}C. Above this value ,the efficiency starts to decrease.

Consequently , the results of the second law analysis can be used to identify the most efficient components of the system and also modify them.Morover, suitibility of the selected components can be jugded by this anaysis.The second law analsis may be a good tool for determination of the optimum working conditions of such systems.It can be deduced that there is a generator temperature at which the second law efficiency is maximum while the percent exergy loss in each components are minimum.According to second law analysis presented here,it is more thermodynamically efficient if absorption systems are operated using low-grade waste heat rather than high temperature heat sources.

While calculating the overall efficiency ,the system can be considered three subsytems.These are Collector–Evaporator ,Absorption Heat Pump and Floor Heating Panel,respectively.

In table 4.10 , The irreversibilities in the Collector-Storage tank system are shown. It can be seen when the system is more irreversible in this table.In January,it can be deduced that early in the morning is much more irreversible than noon.The irreversibilities decreases as the hourly solar insolation values and water leaving the storage tank start to increase.In that table ,it is also shown how efficiently solar radiation can be used.

In table 4.11, Second law efficiencies are shown for the Collector-Evaporator subsystem. When the water leaving the storage tank is less than 34.5 ^{o}C ,the water is heated and then fed into evaporator,If the storage tank temperature is between 34.5 ^{o}C and 41.5 ^{o}C ,the water is fed to evaporator without preheating.When the water leaving the evaporator is between 41.5 ^{o}C and 50 ^{o}C , water is heated to 50 ^{o}C and then fed into floor heating system. Finally,if the water leaving the storage tank is larger than 50 ^{o}C ,it is directly circulated through the floor heating panel.

In Table 4.12 ,the efficiency values of the AHP subsystem are shown .When the water leaving the storage tank is less than 34.5 ^{o}C ,the efficiency value is constant.When temperature is larger than this value the efficiency values of the AHP starts to increase with the increase in the storage tank temperature.

In table 4.13, The exergies at floor heating panel are shown.The exergetic efficiency is constant when the water enters the panel at a temperature of 50 ^{o}C.

When the floor panel is modelled, mean plate temperature is taken into account.When the temperature exceeds 50 ^{o}C, the exergetic efficiencies also increases.

When the Storage tank temperature is less than 34.5 ^{o}C , The fluid leaving the storage tank is heated up to 34.5 ^{o}C via auxiliary system and then fed into evaporator. In january (Figure 4.5), Since the evaporator temperature is constant ,the exergetic efficiency (Second law efficiency) decreases by 3 % as the inlet exergy increase (ψ_i) with an increase in solar radiation. The Storage tank temperature reaches its maximum at 14:00. Then the efficiency starts to decrease with the decrease of the storage tank and solar insolation value.

As can be seen from Figure 4.6, In February, the starting temperature is greater than 34.5 ^{o}C. Therefore,There is an increase in the efficiency with an increase of storage tank temperature and solar insolation value.Solar insolation value reaches its maximum at 13:00 p.m. So the maximum value for the efficiency is obtained at that hour.After this hour the efficiency start to decrease.It can be seen in the efficiencies, the efficiency in January is less than the efficiency in February.The reason for this behaviour is that the hourly insolation values on a surface in February are greater than that of January.

As for March, Figure 4.7, The behaviour of the effciency is similar to that of January,There is a decrease in efficiency till 34.5 ^{o}C .Followed by this value,the efficiency increases sharply. It can be also seen that when the maximum value for efficiency is reached (27 %).The storage tank temperature is 43.7 ^{o}C.After this maximum value ,there is still an increase in the storage tank temperature but the efficiency is not greater than that of maximum because of the decrease in the hourly solar insolation .

In April , October and November the figures are similar to that of February.The efficiency increase with the increase at storage tank temperature and insolation value.The maximum value is reached at 13:00 p.m.The values are shown in the figures 4.8 to 4.10,respectively.

In December, There is a decrease in the second law efficiency till 35.4 ^{o}C.Since the evaporator capacity is constant and solar insolation value increase steadily.The increase in the efficiency occur at a temperature of 37.2 ^{o}C .As can be seen in Figure 4.11, the maximum value is reached at 13:00 p.m. After this value efficiency starts to decrease.

It can also be deduced that, such a system of solar assisted AHP supplying energy to a floor heating system would give higher performance values and would perform more effectively from energy economy point of view, in warmer climates with higher solar insolation values and higher ambient temperatures (eg. southern Turkey).

4.2. Results

Table 4.1 Probable hourly heating load ,E[kWh]

Month / Hour	1 Jan	2 Feb	3 Mar.	4 Apr	10 Oct	11 Nov	12 Dec
1	15,82	13,19	17,67	9,67	10,11	17,67	17,76
2	15,82	13,19	18,02	9,58	10,11	17,14	17,76
3	15,82	13,27	18,46	9,58	10,55	16,96	17,93
4	15,82	13,27	19,25	9,67	11,51	17,05	18,28
5	15,91	13,36	19,51	9,93	11,69	16,53	18,37
6	16,09	13,45	19,78	10,72	11,60	16,00	18,37
7	16,70	13,45	20,04	10,72	11,60	15,38	18,20
8	16,79	14,24	18,46	9,84	10,81	14,94	17,76
9	16,00	14,94	16,00	9,23	9,93	13,36	16,61
10	15,12	15,03	15,12	8,09	9,23	11,43	14,86
11	13,36	14,77	14,24	6,50	8,61	11,25	13,19
12	12,13	13,71	12,66	6,33	7,30	10,46	10,90
13	12,04	13,36	11,34	5,36	7,03	9,67	10,64
14	11,87	12,83	10,37	5,45	6,68	10,20	10,28
15	11,95	12,13	9,67	7,03	6,94	10,28	10,20
16	12,22	12,22	9,23	6,68	6,59	10,37	10,46
17	13,62	12,31	8,61	7,03	6,59	10,55	11,25
18	14,94	13,45	9,23	7,91	7,47	10,55	11,95
19	16,00	13,71	10,55	8,00	8,26	11,08	13,01
20	17,32	13,89	11,78	8,79	9,05	11,08	13,01
21	17,58	14,24	13,54	8,97	9,14	11,08	13,01
22	18,11	15,12	13,54	9,58	10,28	11,51	13,10
23	18,37	15,47	13,62	10,02	10,72	12,22	13,98
24	18,81	15,82	14,59	10,55	11,08	12,13	14,59

Daily avarage heating load[kWh]

Jan	Feb	Mar.	Apr	Oct	Nov	Dec
368,2	330,42	345,27	205,25	222,91	308,88	345,45

Table 4.2 Rate of heat addition by the Collectors,Q_u[kW]

Month / Hour	1 Jan	2 Feb	3 Mar.	4 Apr	10 Oct	11 Nov	12 Dec
1,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
2,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
3,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
4,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
5,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
6,00	0,00	0,00	0,00	0,30	0,00	0,00	0,00
7,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
8,00	0,00	0,00	2,30	5,40	5,70	2,00	0,00
9,00	4,90	8,90	13,70	15,90	17,00	12,90	2,80
10,00	13,40	17,20	22,50	24,60	26,20	24,00	11,10
11,00	20,30	23,40	29,30	31,50	33,30	31,90	17,50
12,00	23,50	27,10	33,60	35,10	37,40	35,90	20,80
13,00	23,00	26,60	33,20	34,50	36,60	35,20	20,40
14,00	19,30	22,30	28,20	29,50	36,40	29,80	171,00
15,00	12,40	15,80	20,90	21,40	23,20	20,90	10,70
16,00	3,90	7,30	11,90	12,30	13,50	9,80	2,90
17,00	0,00	0,00	1,50	1,70	2,20	0,00	0,00
18,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
19,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
20,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
21,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
22,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
23,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
24,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00

Daily avarage heat addition by the collectors[kWh]

Jan	Feb	Mar.	Apr	Oct	Nov	Dec
120,7	148,6	197,1	211,9	226,5	202,4	103,3

Table 4.3 Working time length ,WT[min] of the AHP

Month / Hour	1	2	3	4	10	11	12
	Jan	Feb	Mar.	Apr	Oct	Nov	Dec
1,00	38,13	31,806	42,687	0	0	42,594	42,78
2,00	37,944	31,713	43,245	0	0	41,106	42,594
3,00	37,851	31,806	44,082	23,25	0	40,548	42,873
4,00	37,665	31,713	45,663	23,436	0	40,548	43,524
5,00	37,758	31,806	46,221	23,901	28,179	39,153	43,524
6,00	38,13	31,899	46,686	25,854	27,9	37,758	43,338
7,00	39,246	31,806	47,151	25,761	27,807	36,177	42,78
8,00	39,432	33,573	43,431	23,436	25,854	35,154	41,757
9,00	37,665	35,154	37,665	22,134	23,715	31,434	39,06
10,00	35,433	35,34	35,6	19,53	22,227	26,877	34,968
11,00	31,527	34,875	33,48	0	0	26,598	30,969
12,00	28,644	32,643	30,132	0	0	25,11	25,668
13,00	28,644	32,178	27,528	0	0	0	25,11
14,00	28,644	0	25,9	0	0	0	24,459
15,00	28,83	0	0	0	0	0	24,366
16,00	0	0	0	0	0	0	25,11
17,00	32,922	0	0	0	0	0	26,97
18,00	33,294	0	0	0	0	0	28,551
19,00	38,502	33,108	0	0	0	0	31,062
20,00	41,385	33,387	0	0	0	0	30,969
21,00	41,943	34,131	0	0	0	0	30,783
22,00	42,966	36,084	0	0	0	0	30,969
23,00	43,338	36,828	32,736	0	0	0	32,922
24,00	44,175	37,479	33,201	0	0	29,202	34,317

Daily avarage working time length [min :]

Jan	Feb	Mar.	Apr	Oct	Nov	Dec
844,068	637,329	615,408	187,302	155,682	452,259	819,423

Table 4.4. Storage tank temperature,T_{snew}[^oC] at the end of the hour

Month / Hour	1 Jan	2 Feb	3 Mar.	4 Apr	10 Oct	11 Nov	12 Dec
1	39,1	39,3	39	41,9	43,8	39	39
2	38,3	38,6	38,1	40,8	42,7	38,1	38,1
3	37,4	37,8	37,1	40,2	41,5	37,2	37,1
4	36,6	37,1	36,1	39,7	40,1	36,3	36,1
5	35,7	36,4	35	39,1	39,5	35,4	35,2
6	34,9	35,7	34	38,5	38,9	34,6	34,2
7	34	35	32,9	38	38,2	33,8	33,2
8	33,10	34,20	32,20	38,00	38,30	33,20	32,30
9	32,80	34,40	33,00	39,30	39,70	34,00	31,80
10	33,60	35,60	34,80	41,70	42,20	36,10	32,30
11	35,20	37,50	37,40	44,60	45,00	39,20	33,60
12	37,20	39,90	40,50	47,80	48,40	42,70	35,40
13	39,20	42,20	43,70	51,20	51,80	45,60	37,20
14	40,80	43,30	45,80	53,90	54,70	47,90	38,60
15	41,50	43,70	47,00	55,60	56,60	49,10	39,20
16	40,60	43,20	47,30	56,40	57,50	49,00	39,00
17	39,80	41,80	46,50	55,80	57,10	47,80	38,30
18	39	40,2	45,4	55	56,4	46,6	37,7
19	38,2	39,5	44,2	54,2	55,6	45,3	37
20	37,2	38,7	42,9	53,3	54,7	44	36,3
21	36,3	37,9	41,3	52,4	53,7	42,7	35,6
22	35,3	37,1	39,8	51,3	52,7	41,4	34,9
23	34,4	36,3	39	50,2	51,5	40	34,1
24	33,4	35,4	38,2	49	50,3	39,4	33,4

	Jan	Feb	Mar.	Apr	Oct	Nov	Dec
T_s initial at 0:00	40	40	40	40	40	40	40
$T_{s,f}$-$T_{s,i}$[^oC]	-6,6	-4,6	-1,8	9	10,3	0,4	-6,6
$Q_{st,o}$[kWh]	-57,5	-40,1	-15,7	61,8	56,6	3,5	-57,5

Table 4.5 Rate of auxiliary energy, $Q_{aux,so}$[kW] delivered to the storage tank to raise its temperature to T_{sref} or T_{fi}

Month / Hour	1	2	3	4	10	11	12
	Jan	Feb	Mar.	Apr	Oct	Nov	Dec
1	0	0	0	6,4	5,9	0	0
2	0	0	0	7,4	7,3	0	0
3	0	0	0	0	9,3	0	0
4	0	0	0	0	0	0	0
5	0	0	0	0	0	0	0
6	0	0	0	0	0	0	0
7	0	0	1	0	0	0	0,5
8	0,80	0,00	2,80	0,00	0,00	1,10	2,10
9	2,10	0,40	3,50	0,00	0,00	1,70	3,40
10	2,40	0,10	2,10	0,00	0,00	0,60	3,90
11	1,20	0,00	0,00	5,10	6,40	0,00	2,80
12	0,00	0,00	0,00	3,30	3,50	0,00	1,00
13	0,00	0,00	0,00	1,10	1,10	6,70	0,00
14	0,00	7,60	6,52	0,00	0,00	4,20	0,00
15	0,00	7,40	3,90	0,00	0,00	2,10	0,00
16	9,80	7,30	2,60	0,00	0,00	0,90	0,00
17	0,00	8,00	2,20	0,00	0,00	1,00	0,00
18	0	10,5	3,1	0	0	2,2	0
19	0	0	4,6	0	0	3,6	0
20	0	0	6,5	0	0	5	0
21	0	0	9,2	0	0	6,3	0
22	0	0	11,1	0	0	7,9	0
23	0	0	0	0	0	9,9	0
24	0,3	0	0	0	0	0	0,5

Daily avarage auxiliary energy required by the storage tank [kWh]

Jan	Feb	Mar.	Apr	Oct	Nov	Dec
16,6	43,5	59,116	23,3	38,3	53,2	14,2

Table 4.6 Rate of auxiliary energy required by the generator, $Q_{aux,go}$[kW]

Month / Hour	1 Jan	2 Feb	3 Mar.	4 Apr	10 Oct	11 Nov	12 Dec
1	8,10	6,70	9,00	0,00	0,00	9,00	9,10
2	8,10	6,80	9,20	0,00	0,00	8,80	9,10
3	8,10	6,80	9,50	4,90	0,00	8,70	9,20
4	8,20	6,80	9,90	5,00	0,00	8,80	9,40
5	8,20	6,90	10,10	5,10	6,00	8,60	9,50
6	8,40	7,00	10,30	5,50	5,90	8,30	9,60
7	8,70	7,00	10,40	5,50	6,00	8,00	9,50
8	8,70	7,40	9,60	5,00	5,60	7,80	9,30
9	8,40	7,80	8,40	4,80	5,10	7,00	8,70
10	7,90	7,80	7,80	4,20	4,70	6,00	7,70
11	7,00	7,70	7,40	0,00	0,00	5,80	6,90
12	6,30	7,10	6,50	0,00	0,00	5,40	5,70
13	6,20	6,80	5,80	0,00	0,00	0,00	5,50
14	6,10	0,00	0,00	0,00	0,00	0,00	5,30
15	6,10	0,00	0,00	0,00	0,00	0,00	5,20
16	0,00	0,00	0,00	0,00	0,00	0,00	5,40
17	6,90	0,00	0,00	0,00	0,00	0,00	5,80
18	7,70	0,00	0,00	0,00	0,00	0,00	6,10
19	8,20	7,00	0,00	0,00	0,00	0,00	6,70
20	8,90	7,10	0,00	0,00	0,00	0,00	6,70
21	9,10	7,30	0,00	0,00	0,00	0,00	6,70
22	9,40	7,80	0,00	0,00	0,00	0,00	6,80
23	9,50	8,00	7,00	0,00	0,00	0,00	7,30
24	9,80	8,20	7,40	0,00	0,00	6,20	7,60

Daily avarage auxiliary energy required by the generator [kWh]

Jan	Feb	Mar.	Apr	Oct	Nov	Dec
184	138	128,4	40	33,3	98,4	178,8

Table 4.7 Rate of auxiliary energy required by the system, $Q_{aux,to}$[kW]

Month / Hour	1	2	3	4	10	11	12
	Jan	Feb	Mar.	Apr	Oct	Nov	Dec
1	8,1	6,7	9	6,4	4,8	9	9,1
2	8,1	6,8	9,2	7,4	5,9	8,8	9,1
3	8,1	6,8	9,5	4,9	7,3	8,7	9,2
4	8,2	6,8	9,9	5	9,3	8,8	9,4
5	8,2	6,9	10,1	5,1	6	8,6	9,5
6	8,4	7	10,3	5,5	5,9	8,3	9,6
7	8,7	7	11,4	5,5	6	8	10
8	9,60	7,40	12,40	5,00	5,60	8,90	11,40
9	10,50	8,20	11,80	4,80	5,10	8,60	12,10
10	10,30	8,00	9,90	4,20	4,70	6,60	11,60
11	8,20	7,70	7,40	5,10	6,40	5,80	9,70
12	6,30	7,10	6,50	3,30	3,50	5,40	6,60
13	6,20	6,80	5,80	1,10	1,10	6,70	5,50
14	6,10	9,50	6,20	0,00	0,00	4,20	5,30
15	6,10	7,70	3,90	0,00	0,00	2,10	5,20
16	9,80	7,30	2,60	0,00	0,00	0,90	5,40
17	6,90	8,00	2,20	0,00	0,00	1,00	5,80
18	7,7	10,5	3,1	0	0	2,2	6,1
19	8,2	7	4,6	0	0	3,6	6,7
20	8,9	7,1	6,5	0	0	5	6,7
21	9,1	7,3	9,2	0	0	6,3	6,7
22	9,4	7,8	11,1	0	0	7,9	6,8
23	9,5	8	7	0	0	9,9	7,3
24	10,1	8,2	7,4	0	0	6,2	8,1

Daily avarage auxiliary energy required by the system [kWh]

Jan	Feb	Mar.	Apr	Oct	Nov	Dec
200,7	181,6	187,2	63,3	71,6	151,5	192,9

Table 4.8 Seasonal Results

Month	1	2	3	4	10	11	12	Total
	Jan	Feb	Mar.	Apr	Oct	Nov	Dec	
IA_c[kWh]	275,8	331,6	412,2	437,2	462,2	407,4	235,6	77.583
E[kWh]	368,2	330,42	345,27	205,25	222,91	308,88	345,45	61.104
WT[min]	844,068	637,329	615,408	187,302	155,682	452,259	819,42	120.258
Q_u[kWh]	120,7	148,6	197,1	211,9	226,5	202,4	103,3	36.665
$Q_{aux,so}$[kWh]	16,6	43,5	58,9	23,3	38,3	53,2	14,2	7.481
$Q_{aux,go}$[kWh]	184,4	138	128,4	40	33,3	98,4	178,8	24.276
$Q_{aux,to}$[kWh]	200,6	181,5	187,3	63,3	71,6	151,6	193	31.757
$Q_{aux,s}$[kWh]	20,8	54,4	73,6	29,1	66,5	66,5	17,8	9.351
$Q_{aux,g}$[kWh]	230	172,5	160,5	50	123	123	223,5	30.344
$Q_{aux,t}$[kWh]	250,8	226,9	234,1	79,1	89,5	189,5	241,3	39.696
$Q_{st,o}$[kWh]	-57,5	-40,1	-15,7	61,8	56,6	3,5	-57,5	-1.461
Q_{st}[kWh]	-71,9	-50,1	-19,6	61,8	56,6	3,5	-71,9	-2.755
$Q_{aux,t}$-Q_{st}	322,6	277	253,8	17,3	32,9	186	313,1	42.450
FNP	0,272	0,349	0,437	0,924	0,873	521	0,248	0.463
SPC	0,076	0,117	0,225	0,911	0,844	0,366	0,043	0.305
$E_{conv,o}$[kWh]	420,6	377,8	394,5	234,7	254,6	353,1	394,2	73.596
E_{conv}[kWh]	527,7	472,3	493,1	293,4	318,3	441,4	492,8	91.995
E_{conv}-(Q_{aux},t-Q_{st})	203,1	195,3	239,3	276,1	285,4	255,4	179,6	49.545

Table 4.9 Percent Exery Loss(%)

T_{eva}=20 ^oC

	Generator Temperature and Exergy loss													
	Tg=60	%	Tg=70	%	Tg=82	%	Tg=90	%	Tg=100	%	Tg=110	%	Tg=120	%
Abs.	0,2365	24	0,2286	23	0,2146	22	0,2539	26	0,2629	27	0,2682	27	0,2728	27
Eva.	0,2881	29	0,2877	29	0,2773	29	0,2655	27	0,2723	28	0,2743	27	0,2757	28
Con.	0,2592	26	0,2483	25	0,2385	25	0,2295	24	0,2341	24	0,2346	24	0,2345	23
Gen.	0,2118	21	0,214	22	0,22	23	0,2159	22	0,2165	22	0,2132	21	0,2093	21
Pump	0,00434	0,4	0,0099	1	0,0067	1	0,003	0	0,0022	0,2	0,0076	0,8	0,00762	0,8
Total	1,000		0,989		0,957		0,968		0,988		0,998		1,000	

T_{eva}=25 ^oC

	Generator Temperature and Exergy loss													
	Tg=60	%	Tg=70	%	Tg=82	%	Tg=90	%	Tg=100	%	Tg=110	%	Tg=120	%
Abs.	0,2366	23	0,231	23	0,234	23	0,2602	26	0,267	27	0,275	27	0,28	28
Eva.	0,28889	29	0,286	29	0,27	27	0,2695	27	0,271	27	0,2735	27	0,275	27
Con.	0,262	26	0,251	25	0,24	24	0,235	24	0,239	24	0,24	24	0,241	24
Gen.	0,22	22	0,217	22	0,227	23	0,2199	22	0,2185	22	0,2147	21	0,21	21
Pump	0,0009	0,1	0,003	0	0,005	0	0,0022	0	0,0012	0,1	0,00103	0,1	0,0008	0,1
Total	1,008		0,988		1,000		0,987		0,997		1,004		1,007	

T_{eva}=28 ^oC

	Generator Temperature and Exergy loss													
	Tg=60	%	Tg=70	%	Tg=82	%	Tg=90	%	Tg=100	%	Tg=110	%	Tg=120	%
Abs.	0,2366	23	0,2326	23	0,2528	25	0,2617	26	0,27	27	0,28	28	0,283	28
Eva.	0,2902	28	0,2847	29	0,2643	26	0,2707	27	0,2719	27	0,273	27	0,274	27
Con.	0,2676	26	0,257	26	0,2421	24	0,2418	24	0,2445	24	0,2438	24	0,245	24
Gen.	0,2247	22	0,2185	22	0,2298	23	0,2238	22	0,2194	22	0,2148	21	0,2103	21
Pump	0,0009	0,1	0,0042	0	0,00348	0	0,00187	0	0,0012	0,1	0,00103	0,1	0,0008	0,1
Total	1,020		0,997		1,000		1,000		1,007		1,013		1,013	

T_{eva}=30 ^oC

	Generator Temperature and Exergy loss													
	Tg=60	%	Tg=70	%	Tg=82	%	Tg=90	%	Tg=100	%	Tg=110	%	Tg=120	%
Abs.	0,239	23	0,241	24	0,255	25	0,2623	26	0,269	27	0,276	27	0,28	28
Eva.	0,287	28	0,281	28	0,266	26	0,2702	27	0,2705	27	0,2725	27	0,2737	28
Con.	0,266	26	0,258	25	0,244	24	0,243	24	0,246	24	0,245	24	0,225	23
Gen.	0,2252	22	0,225	22	0,264	26	0,2242	22	0,22	22	0,2151	21	0,2107	21
Pump	0,01214	1,2	0,01	1	0,005	0	0,004	0	0,0025	0,2	0,0025	0,2	0,0042	0,4
Total	1,029		1,015		1,034		1,004		1,008		1,011		0,994	

T_{eva}=35 ^oC

	Generator Temperature and Exergy loss													
	Tg=60	%	Tg=70	%	Tg=82	%	Tg=90	%	Tg=100	%	Tg=110	%	Tg=120	%
Abs.	0,2414	24	0,2604	25	0,2614	24	0,2638	26	0,2674	26	0,271	27	0,2748	27
Eva.	0,2796	27	0,2707	26	0,269	24	0,2695	27	0,2705	27	0,2715	27	0,2725	27
Con.	0,2617	26	0,2585	25	0,2483	22	0,2476	24	0,247	24	0,2466	24	0,2463	24
Gen.	0,2294	22	0,2343	23	0,3202	29	0,2258	22	0,2206	22	0,2156	21	0,2107	21
Pump	0,01214	1,2	0,01	1	0,0089	1	0,00672	1	0,00543	0,5	0,00471	0,5	0,0042	0,4
Total	1,024		1,034		1,108		1,013		1,011		1,009		1,009	

Table-4.10. Exergy Losses in the Collector-Storage Tank Subsystem

January

Hour	Q_i	Q_u	ψ_i	ψ_u	$\psi_i - \psi_u$	loss(%)
9	18,40	4,90	17,13	1,18	15,95	93,12
10	29,90	13,40	27,84	5,17	22,67	81,42
11	39,10	20,30	36,40	9,42	26,98	74,12
12	43,70	23,50	40,68	11,48	29,20	71,78
13	43,70	23,00	41,43	5,46	35,97	86,82
14	39,10	19,30	36,40	8,75	27,66	75,97
15	29,90	12,40	27,84	4,77	23,07	82,87
16	18,40	3,90	17,13	0,88	16,25	94,86

February

Hour	Q_i	Q_u	ψ_i	ψ_u	$\psi_i - \psi_u$	loss(%)
9	23,90	8,90	22,25	2,80	19,45	87,41
10	35,70	17,20	33,23	7,35	25,88	77,88
11	44,80	23,40	41,71	11,42	30,29	72,62
12	50,20	27,10	46,74	13,99	32,74	70,06
13	50,20	26,60	46,74	13,76	32,98	70,57
14	44,80	22,30	41,43	5,45	35,97	86,83
15	35,70	15,80	33,23	5,20	28,03	84,35
16	23,90	7,30	22,25	2,19	20,06	90,16

March

Hour	Q_i	Q_u	ψ_i	ψ_u	$\psi_i - \psi_u$	loss(%)
8	15,16	2,30	14,52	0,37	14,15	97,42
9	30,40	13,70	28,30	5,24	23,06	81,49
10	42,80	22,50	39,85	10,75	29,10	73,03
11	52,60	29,30	48,97	15,57	33,40	68,20
12	58,90	33,60	54,84	18,82	36,02	65,68
13	58,90	33,20	54,84	18,61	36,22	66,06
14	52,60	28,20	48,97	14,99	33,98	69,39
15	42,80	15,80	39,85	6,74	33,11	83,10
16	30,40	7,30	28,30	2,24	26,06	92,08
17	15,60	1,50	14,52	0,26	14,26	98,19

Table 4.10. Cont'd

April

Hour	Q_i	Q_u	ψ_i	ψ_u	$\psi_i - \psi_u$	loss(%)
8	18,30	5,40	17,04	1,36	15,68	92,01
9	32,90	15,90	30,63	6,64	23,99	78,32
10	45,20	24,60	42,08	12,35	29,74	70,66
11	55,10	31,50	51,30	17,37	33,93	66,14
12	61,00	35,10	56,79	20,15	36,64	64,51
13	61,00	34,50	56,79	19,73	37,06	65,26
14	55,10	29,50	51,30	16,01	35,29	68,79
15	45,20	21,40	42,08	10,48	31,60	75,09
16	32,90	12,30	30,63	4,88	25,75	84,06
17	18,30	1,70	17,04	0,35	16,69	97,96

October

Hour	Q_i	Q_u	ψ_i	ψ_u	$\psi_i - \psi_u$	loss(%)
8	19,20	5,70	17,88	1,48	16,40	91,73
9	34,80	17,00	32,40	7,31	25,09	77,44
10	47,90	26,20	44,60	13,48	31,12	69,78
11	58,50	33,30	54,46	18,72	35,74	65,63
12	64,80	37,40	60,33	21,93	38,40	63,65
13	64,80	36,60	60,33	21,41	38,92	64,52
14	58,50	31,40	54,46	17,55	36,91	67,77
15	47,90	23,20	44,60	11,73	32,87	73,71
16	34,80	13,50	32,40	5,58	26,82	82,79
17	19,20	2,20	17,88	0,49	17,39	97,26

November

Hour	Q_i	Q_u	ψ_i	ψ_u	$\psi_i - \psi_u$	loss(%)
8	14,00	2,00	13,03	0,34	12,69	97,37
9	28,60	12,90	26,60	4,82	21,78	81,88
10	43,70	24,00	40,69	11,80	28,89	71,01
11	55,50	31,90	51,67	17,44	34,23	66,24
12	61,90	35,90	57,63	20,63	37,00	64,21
13	61,90	35,20	57,63	20,17	37,46	65,00
14	55,50	29,80	51,67	16,20	35,47	68,64
15	43,70	20,90	40,69	10,01	30,68	75,40
16	28,60	9,80	26,63	3,34	23,29	87,45

Table 4.10. Cont'd

December

Hour	Q_i	Q_u	ψ_i	ψ_u	$\psi_i - \psi_u$	loss(%)
8	15,16	2,30	14,52	0,37	14,15	97,42
9	30,40	13,70	28,30	5,24	23,06	81,49
10	42,80	22,50	39,85	10,75	29,10	73,03
11	52,60	29,30	48,97	15,57	33,40	68,20
12	58,90	33,60	54,84	18,82	36,02	65,68
13	58,90	33,20	54,84	18,61	36,22	66,06
14	52,60	28,20	48,97	14,99	33,98	69,39
15	42,80	15,80	39,85	6,74	33,11	83,10
16	30,40	7,30	28,30	2,24	26,06	92,08
17	15,60	1,50	14,52	0,26	14,26	98,19

Tablo 4.11 Exergetic Efficiencies of Collector-Evaporator Subsystem[kWh]

Month / Hour	1 Jan	2 Feb	3 Mar	4 Apr	10 Oct	11 Nov	12 Dec
8			0,152	0,071	0,070	0,063	
9	0,036	0,020	0,094	0,091	0,163	0,046	0,132
10	0,018	0,040	0,041	0,135	0,231	0,040	0,100
11	0,033	0,083	0,080	0,248	0,271	0,125	0,063
12	0,065	0,144	0,150	0,310	0,302	0,257	0,050
13	0,081	0,215	0,270	0,342	0,319	0,345	0,035
14	0,126	0,132	0,246	0,308	0,209	0,231	0,115
15	0,086	0,130	0,184	0,243	0,237	0,175	0,159
16	0,006	0,051	0,074	0,154	0,167	0,095	
17			0,027	0,029	0,027		

Tablo 4.12 Exergetic Efficiencies of Absorption Heat Pump[kWh]

Month / Hour	1 Jan	2 Feb	3 Mar	4 Apr	10 Oct	11 Nov	12 Dec
8			0,521	0,732	0,722	0,521	
9	0,521	0,521	0,521	0,741	0,734	0,521	0,521
10	0,521	0,657	0,525	0,748	-	0,692	0,521
11	0,654	0,692	0,690	-	-	0,696	0,521
12	0,701	0,758	0,767	-	-	-	0,647
13	0,764	-	-	-	-	-	0,652
14	0,772	-	-	-	-	-	0,658
15	0,781	-	-	-	-	-	0,610
16	0,770	-	-	-	-	-	-
17	-	-	-	-	-	-	-

Tablo 4.13 Exergetic Efficiencies of Floor Heating Panel[kWh]

Month / Hour	1 Jan	2 Feb	3 Mar	4 Apr	10 Oct	11 Nov	12 Dec
8			0,808	0,808	0,808	0,808	
9	0,808	0,808	0,808	0,808	0,808	0,808	0,808
10	0,808	0,808	0,808	0,808	0,808	0,808	0,808
11	0,808	0,808	0,808	0,808	0,808	0,808	0,808
12	0,808	0,808	0,808	0,808	0,808	0,808	0,808
13	0,808	0,808	0,808	0,812	0,814	0,808	0,808
14	0,808	0,808	0,808	0,818	0,819	0,808	0,808
15	0,808	0,808	0,808	0,823	0,827	0,808	0,808
16	0,808	0,808	0,808	0,826	0,831	0,808	
17			0,808	0,824	0,829		

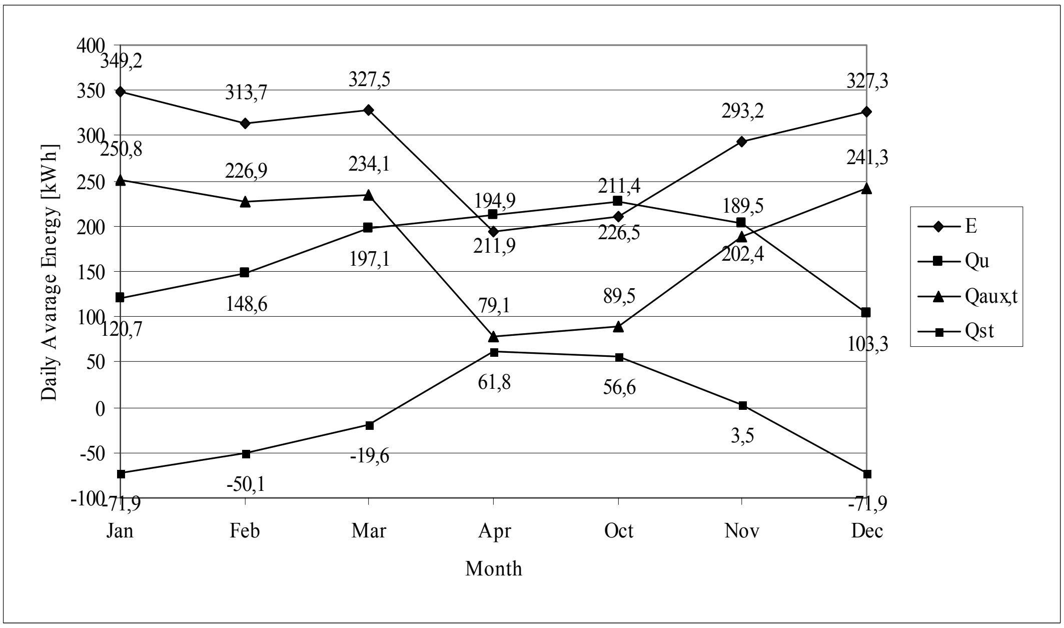

Figure 4.1. Energy Utilisation during the Year

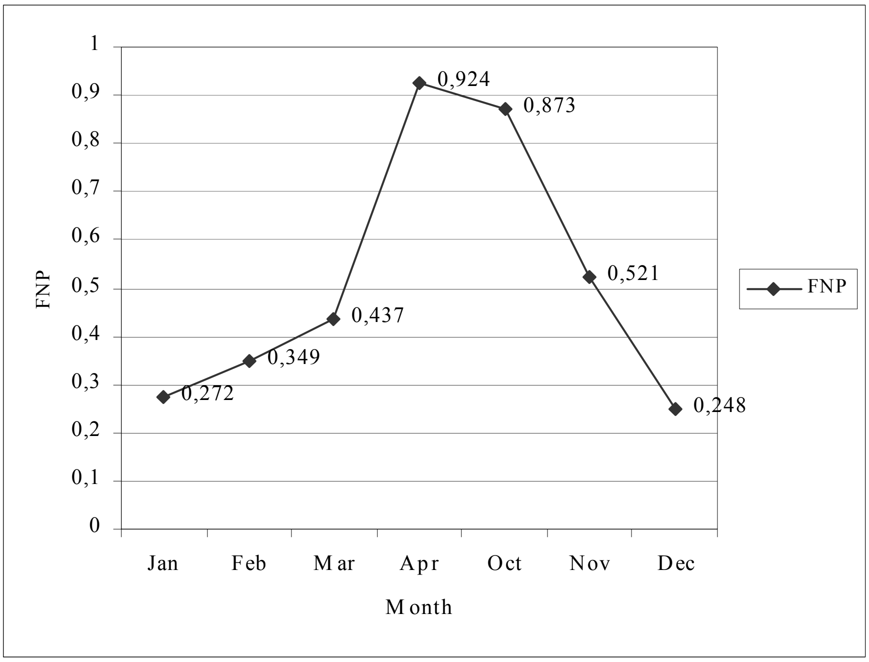

Figure 4.2 Variation of FNP during the Year

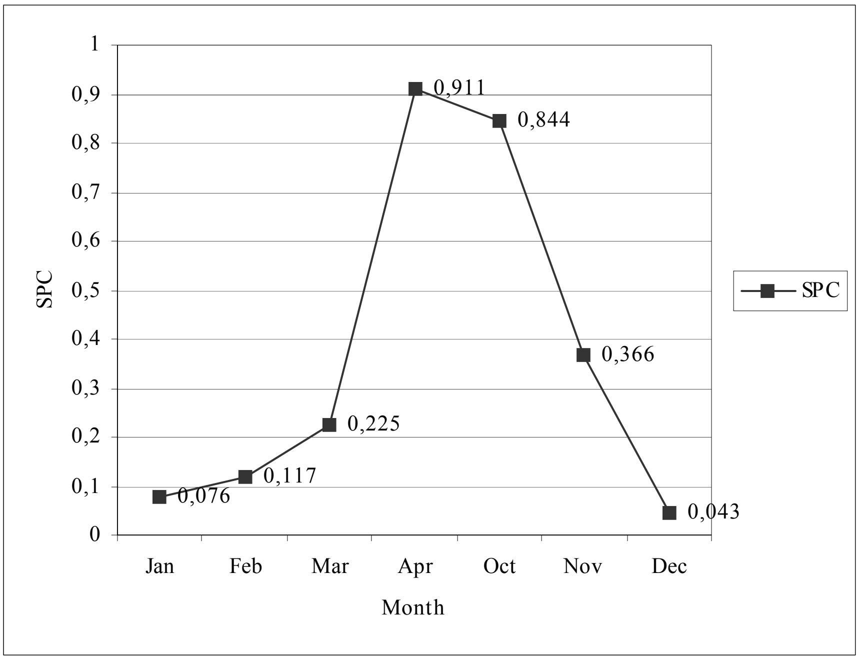

Figure 4.3 Variation of SPC during Year

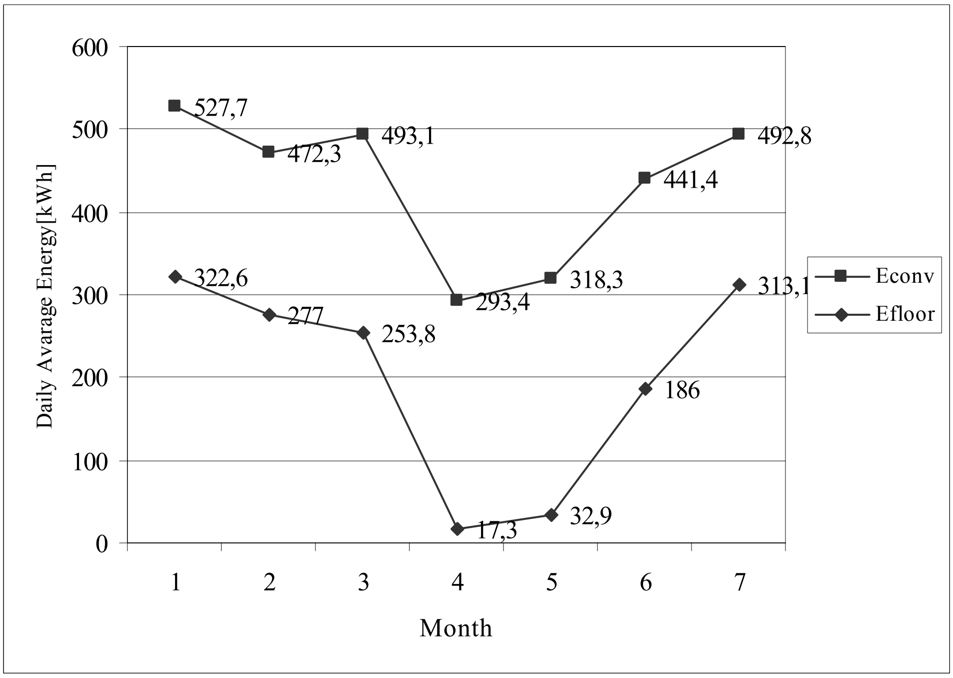

Figure 4.4 Comparison of the energy consumed by a conventional system and a Floor heating system coupled to an AHP during the year

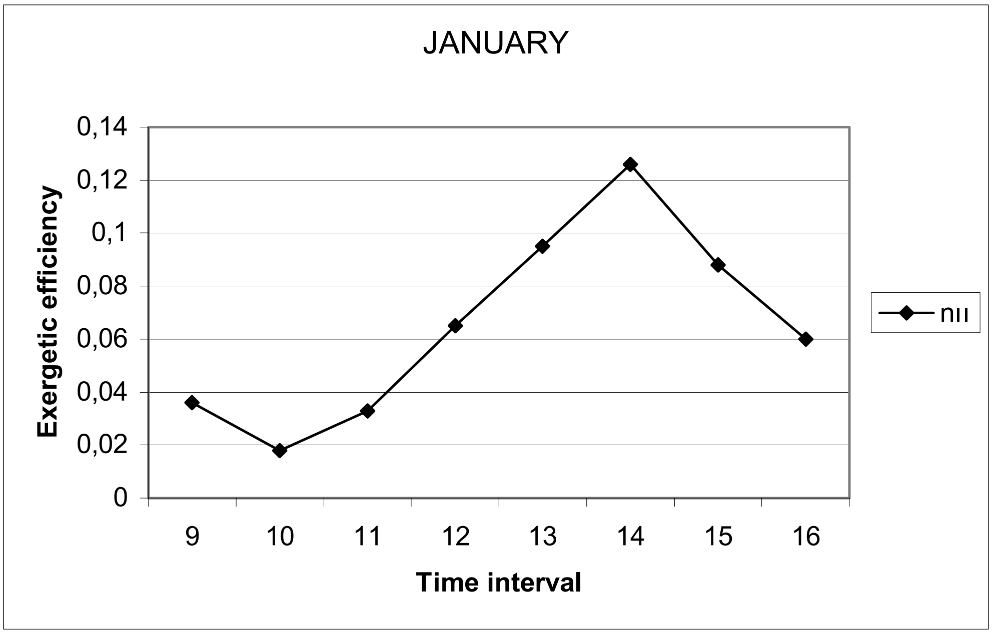

Figure 4.5 Exergetic Efficiency of Collector-Evaporator Subsystem in January

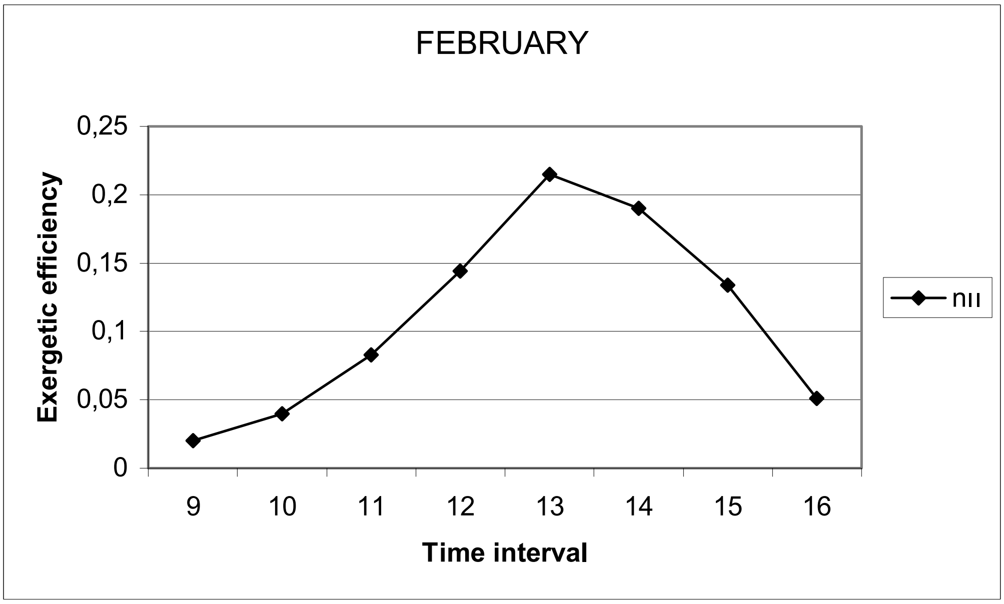

Figure 4.6 Exergetic Efficiency of Collector-Evaporator Subsystem in February

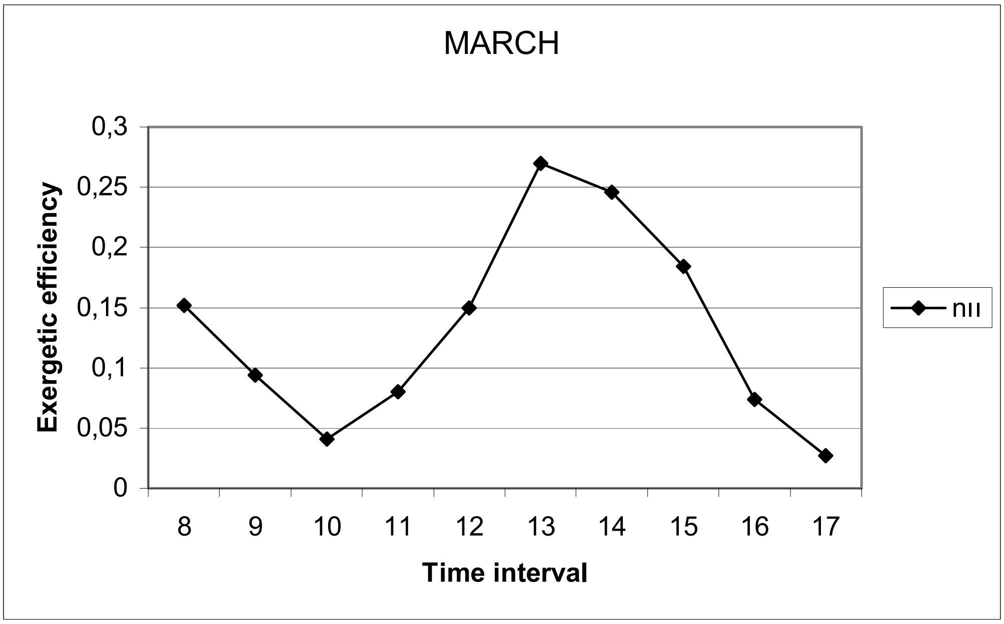

Figure 4.7 Exergetic Efficiency of Collector-Evaporator Subsystem in March

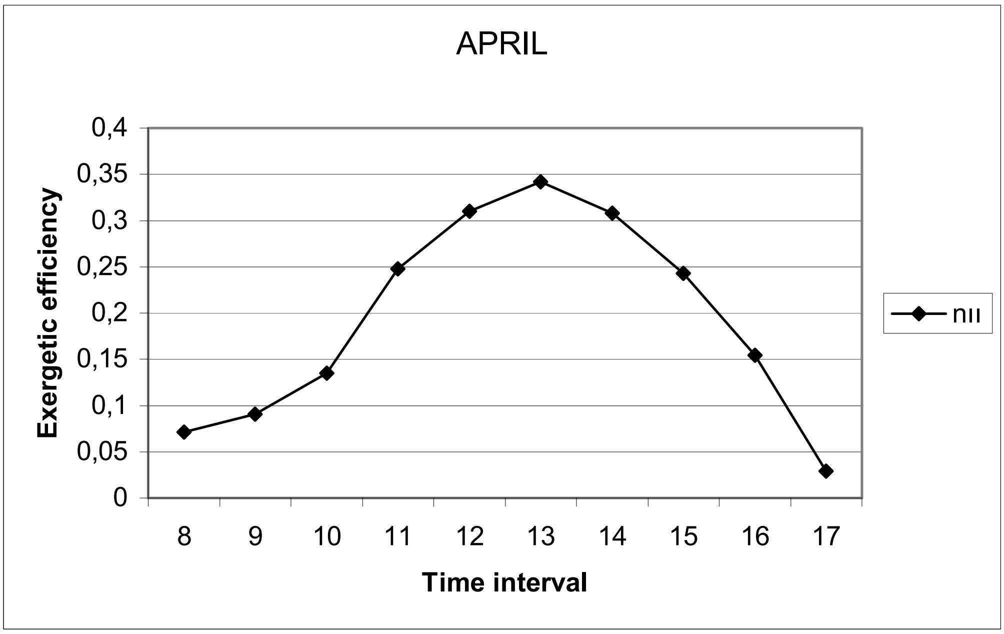

Figure 4.8 Exergetic Efficiency of Collector-Evaporator Subsystem in April

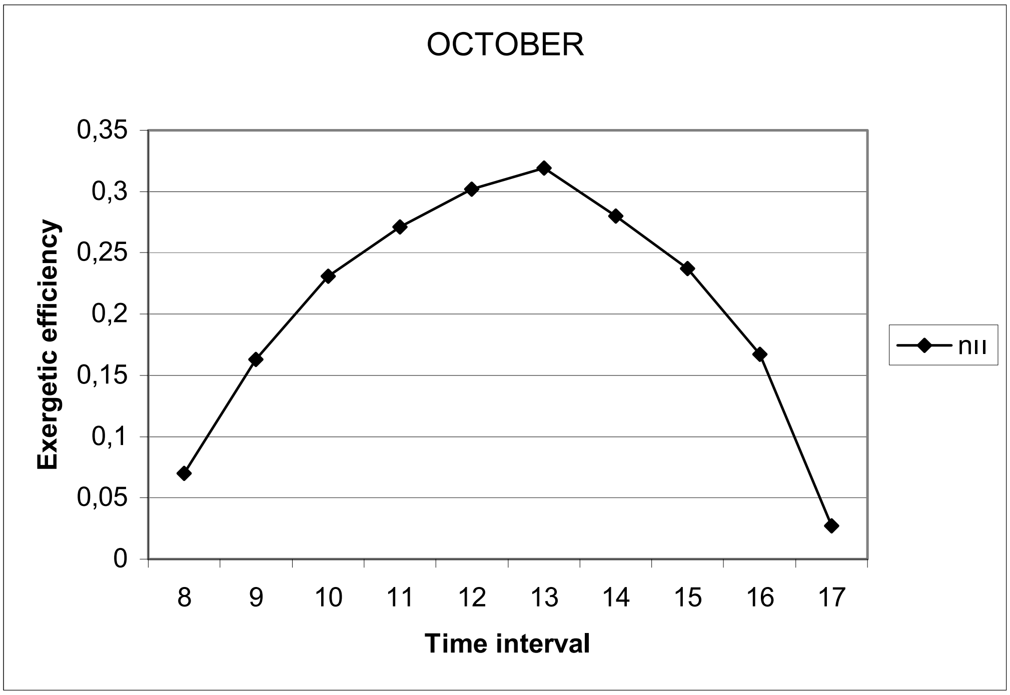

Figure 4.9 Exergetic Efficiency of Collector-Evaporator Subsystem in October

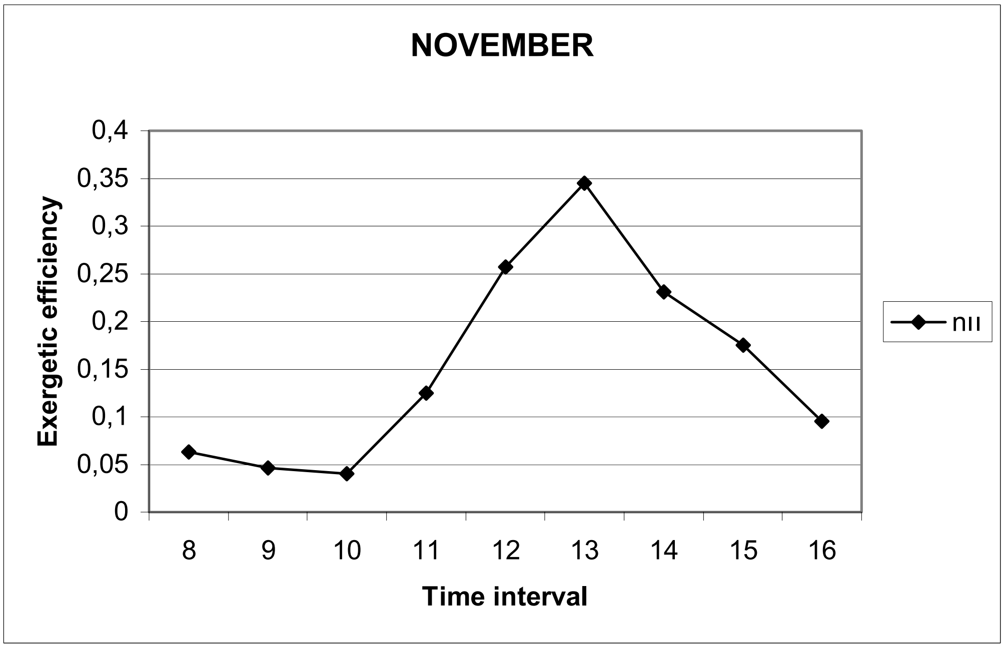

Figure 4.10 Exergetic Efficiency of Collector-Evaporator Subsystem in November

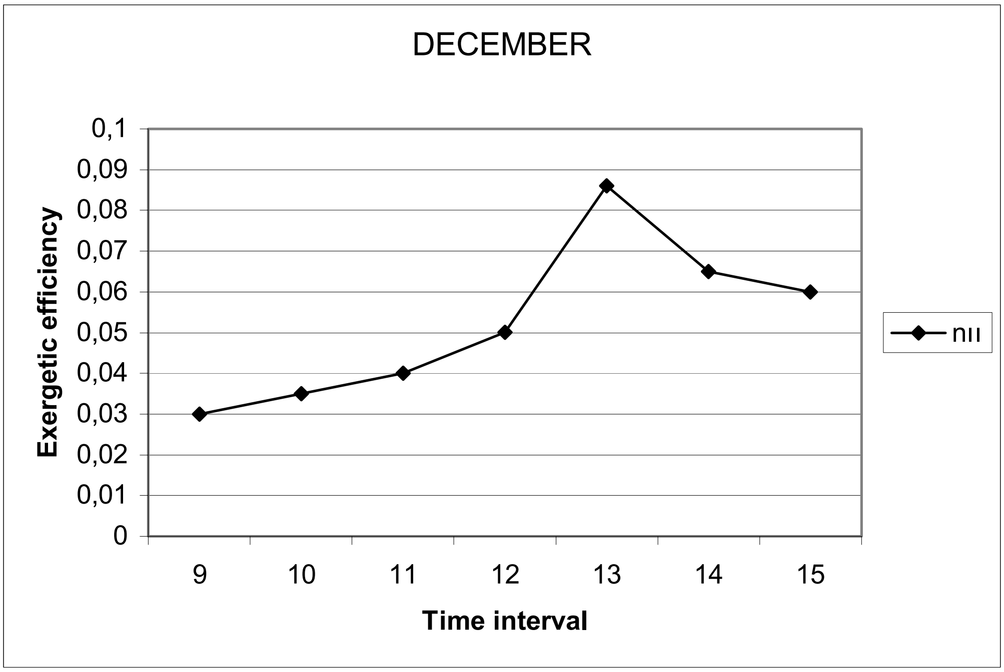

Figure 4.11 Exergetic Efficiency of Collector-Evaporator Subsystem in December

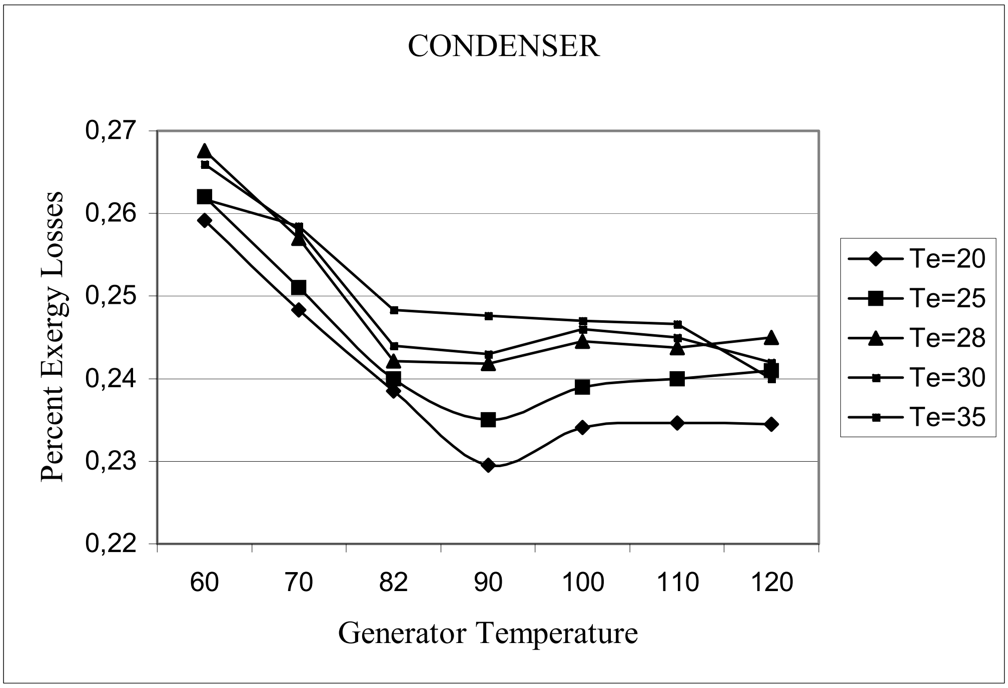

Figure 4.12 Percent Exergy Loss in Condenser

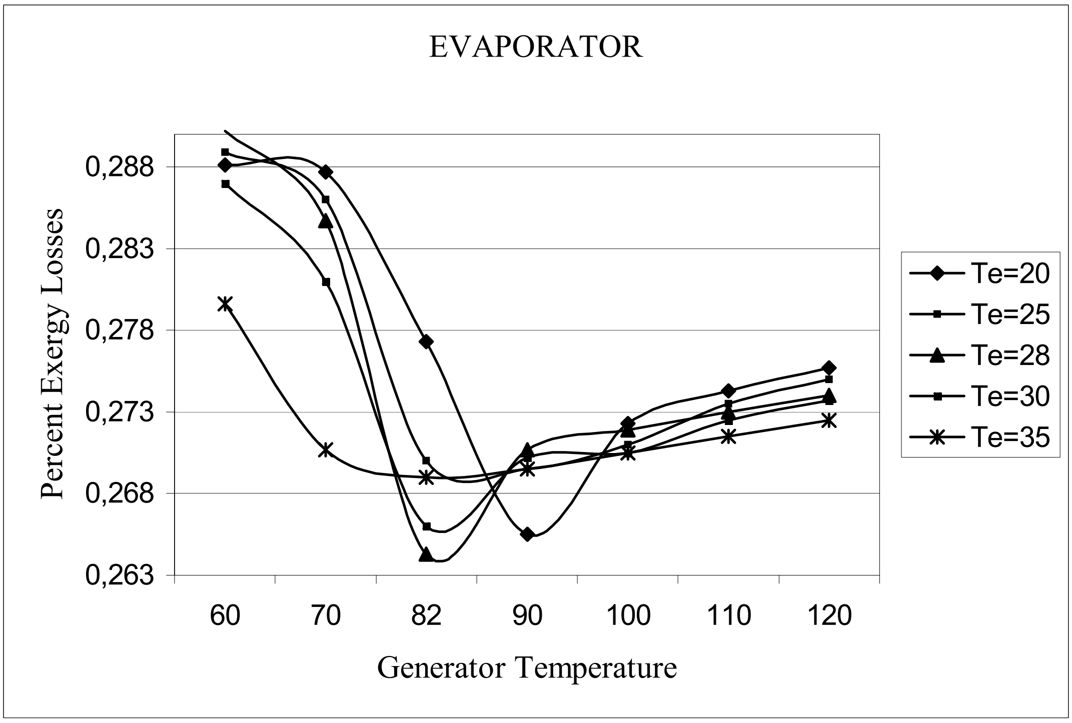

Figure 4.13 Percent Exergy Loss in Evaporator

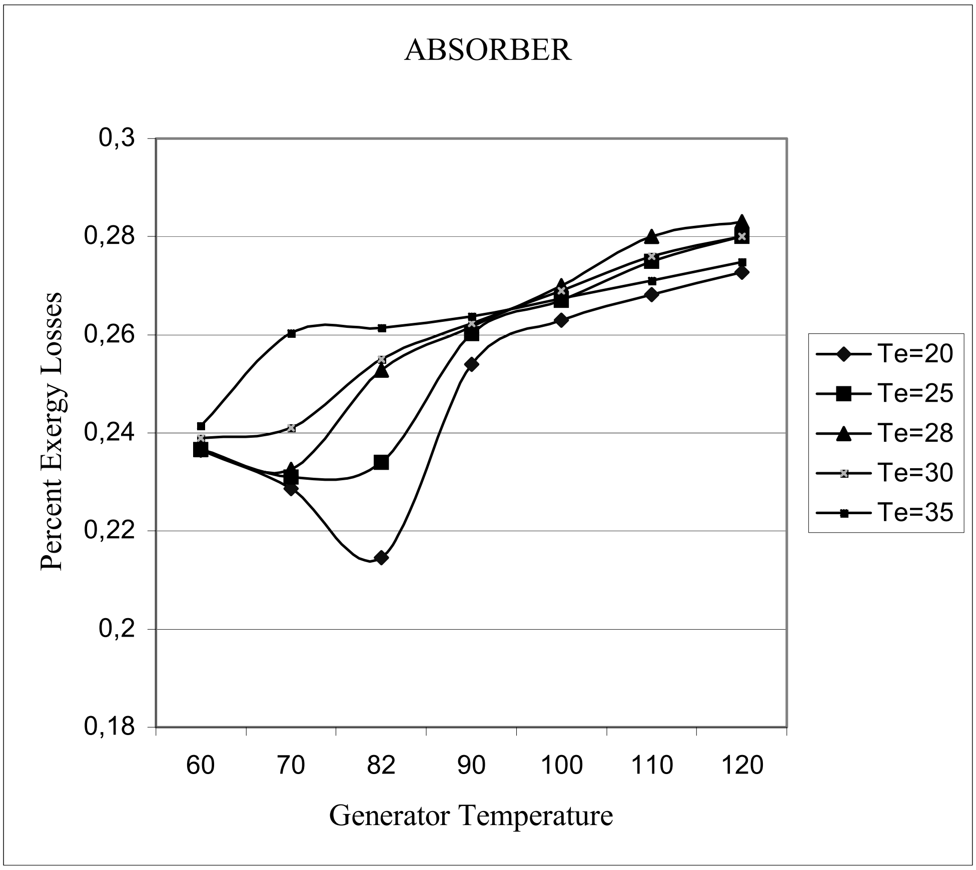

Figure 4.14 Percent Exergy Loss in Absorber

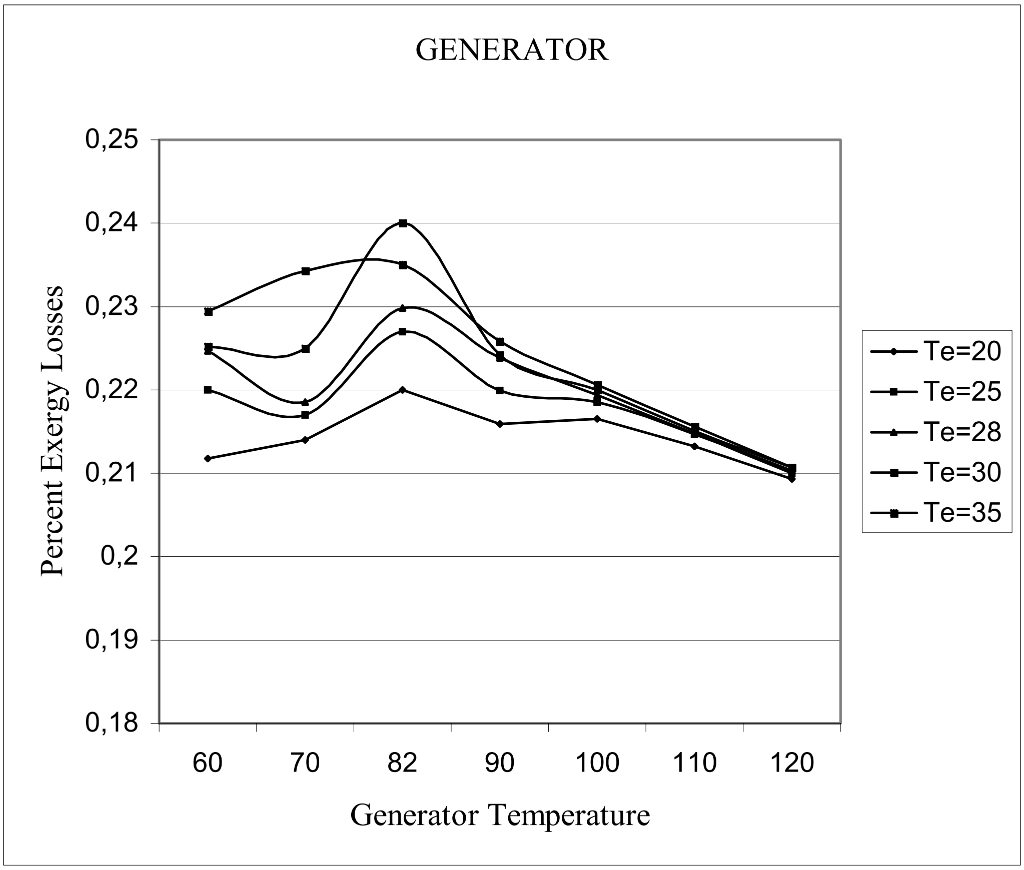

Figure 4.15 Percent Exergy Loss in Generator

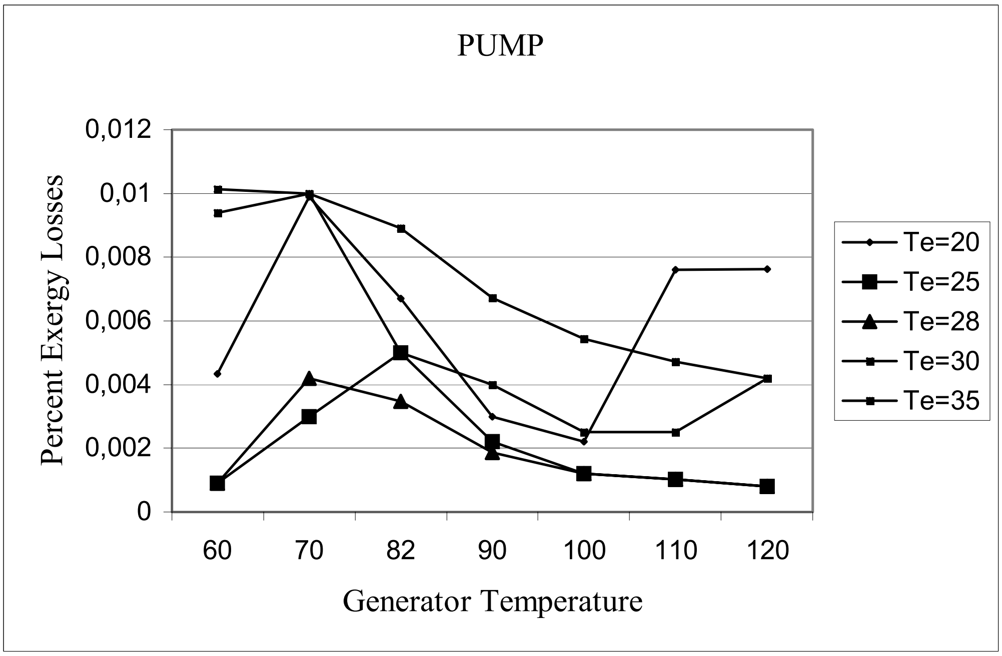

Figure 4.16 Percent Exergy Loss in Pump

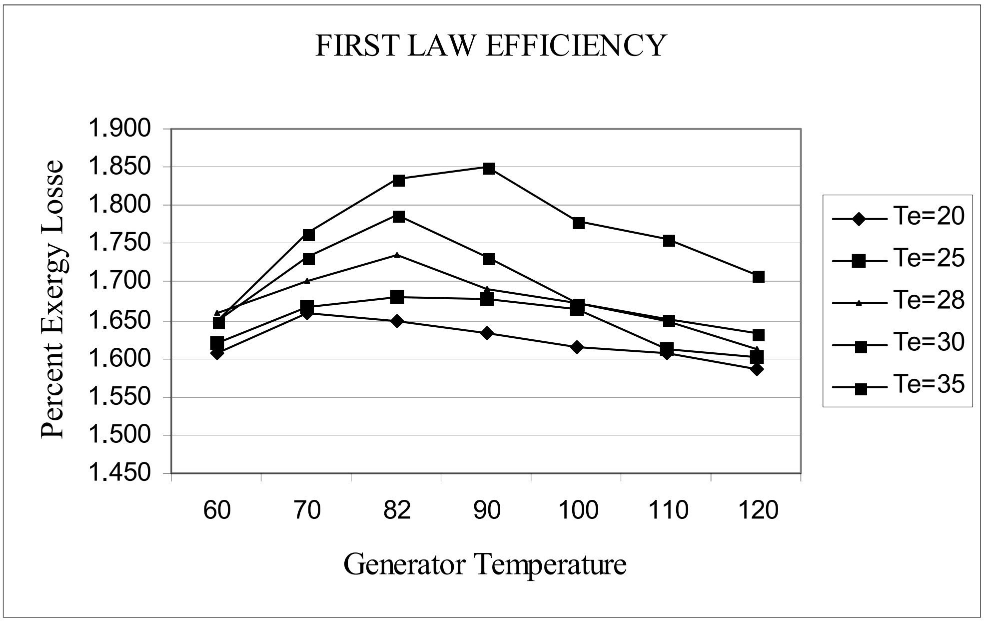

Figure 4.17 Coefficient of Performance of AHP

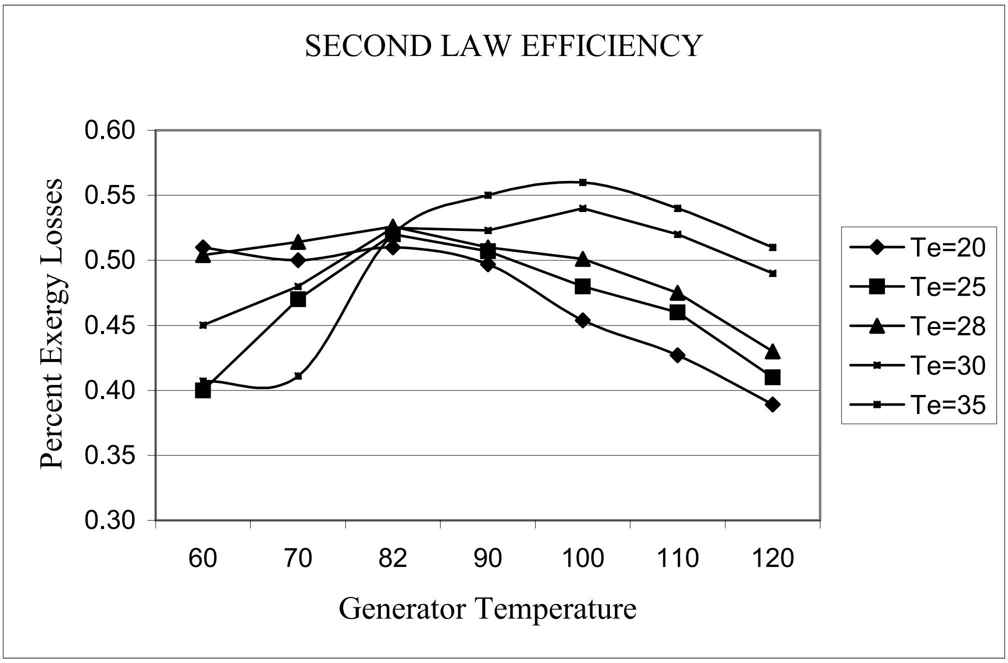

Figure 4.18 Exergetic Coefficient of Performance of AHP

4.3. Conclusions

In the analysis of absorption systems high values for useful exergy supplied to the system is important. Therefore, parabolic solar collector may be an alternative for AHP systems to obtain high input exergy to the system.

Another possibility to enhance system performance would be to use rather new and promising refrigerant-absorbent pair such as TFE-TEGDME.

Instead of using single stage heat pump, double or triple stage absorption heat pumps may be used to increase system efficiency.

APPENDIX A

HEATING LOAD CALCULATIONS

A.1. Design Criteria and Data for Heating Load Calculation

A.1.1. Design Conditions

- Location :Ankara
- Ambient air design temperature :- 12 ^oC
- Wind condition :Windy

A.1.2. Inside Air Design Temperature

Living room	: 18 ^oC
Bedrooms	: 17 ^oC
Ground floor –hall and entrance	: 17 ^oC
First floor hall	: 17 ^oC
Kitchen	: 17 ^oC
Bathroom	: 20 ^oC
WC	: 7 ^oC
Roof	: 6 ^oC

- Unheated spaces in 2nd floor : 1 °C
- Ground : 6 °C

A.2. Structural Elements of the Sample Building

Floor	Structural Material	L_p	k_p	h_{pi}	h_{po}	U_p	t_y	R_y	R_a	k_{eq}
(GSF)	Floor finish	0,02	0,15	5	~	0,66	0,08	0,2	0,1	0,4
	Cover layer	0,07	1,2	5	~	0,66	0,08	0,2	0,1	0,4
	Insulation	0,03	0,034	5	~	0,66	0,08	0,2	0,1	0,4
	Concrete	0,15	1,3	5	~	0,66	0,08	0,2	0,1	0,4
	Caging	0,2	1,5	5	~	0,66	0,08	0,2	0,1	0,4
(FSF)	Floor finish	0,02	0,15	7	7	0,68	0,08	0,2	0,1	0,4
	Cover layer	0,07	1,2	7	7	0,68	0,08	0,2	0,1	0,4
	Insulation	0,03	0,034	7	7	0,68	0,08	0,2	0,1	0,4
	Supporting slab	0,15	1,8	7	7	0,68	0,08	0,2	0,1	0,4
	Interior Plaster	0,02	0,75	7	7	0,68	0,08	0,2	0,1	0,4
(SSF)	Floor finish	0,02	0,15	7	7	0,72				
	Insulation	0,03	0,034	7	7	0,72				
	Supporting slab	0,13	1,8	7	7	0,72				
	Interior Plaster	0,2	0,75	7	7	0,72				
(TR)	Floor finish	0,02	2,5	7	20	0,42				
	Concrete	0,04	1,2	7	20	0,42				
	Insulation	0,07	0,034	7	20	0,42				
	Supporting slab	0,15	1,8	7	20	0,42				
	Interior Plaster	0,02	0,75	7	20	0,42				
(EW)	Exterior Plaster	0,03	1,2	7	20	1,7				
	Perforated brick	0,19	0,43	7	20	1,7				
	Interior Plaster	0,02	0,75	7	20	1,7				
Bath. (EW)	Exterior Plaster	0,03	1,2	7	20	0,74				
	Perforated brick	0,19	0,43	7	20	0,74				
	Insulation	0,03	0,034	7	20	0,74				
	Interior Plaster	0,02	0,75	7	20	0,74				
(IW1)	Interior Plaster	0,02	0,75	7	7	1,45				
	Perforated brick	0,15	0,43	7	7	1,45				
	Interior Plaster	0,02	0,75	7	7	1,45				
(IW2)	Interior Plaster	0,02	0,75	7	7	2,2				
	Perforated brick	0,05	0,43	7	7	2,2				
	Interior Plaster	0,02	0,75	7	7	2,2				

A.3. Heating Load Calculations for Floor Heating System

Table A.1. Ground Floor (Living Room) Heating Load,kW

Ground Floor :Living Room (18 ^oC)													
Structural element													
Type	Direction	Height	Width	A_o	Quantity	A_{ex}	A_{net}	U_p	T_p	Q_o	Z_H	C	Q_T
		m	m	m^2	pcs	m^2	m^2	kcal/hm^2 C	^oC	kcal/h			kcal/h
EW	W	3,1	4,57	14,17	1	1,28	12,89	1,46	30	564,58			
EW	N	3,1	7,67	23,78	1	9,84	13,94	1,46	30	610,57			
EW	E	3,1	5,31	16,46	1	0	16,46	1,46	30	720,95			
EW	S	3,1	1,31	4,06	1	1,28	2,78	1,46	30	121,76			
EW1	W	1,6	0,8	1,28	1	0	1,28	4,5	30	172,8			
EW1	S	1,6	0,8	1,28	1	0	1,28	4,5	30	172,8			
ED	N	2,4	1,4	9,84	1	0	9,84	4,3	30	1269,4			
ID	S	2,4	1,5	3,6	1	0	3,6	2	3	21,6			
IW1	S	3,1	1,52	4,71	1	0	4,71	1,45	13	88,784			
IW1	S	3,1	3	9,3	1	0	9,3	1,45	3	40,455			
FSF		7,67	5,31	40,73	1	1,05	39,68	0,68	3	80,947			
GSF		7,67	5,31	40,73	1	1,05	39,68	0,66	14	366,64			
										4231,3	0,05	0,8	3554,254

a	l	Quantity	T_p	R	H	Z_E	E_Y	Q_L
		pcs	^oC					kcal/h
2,5	11,8	1	30	0,8	0,7	1,2	1	594,72

Q_H(kcal/h)= **4148,974**

Q_H(W)= **4824,178**

qf,des(W/m2)= **121,5771**

Conventional System **Q_H(W)=** **7285,9**

Table A.2. Ground Floor (Kitchen)Heating Load,kW

Ground Floor :Kitchen (17 ^oC)													
Structural element													
Type	Direction	Height	Width	A_o	Quantity	A_{ex}	A_{net}	U_p	T_p	Q_o	Z_H	C	Q_T
		m	m	m^2	pcs	m^2	m^2	$kcal/hm^2 C$	^oC	kcal/h			kcal/h
EW	N	2,6	1,56	4,06	1	1,92	2,14	1,46	29	90,608			
ED	N	2,4	0,8	1,92	1	0	1,92	3	29	167,04			
EW	W	2,6	6,63	17,24	1	1,44	15,8	1,46	29	668,97			
EW1	W	0,6	0,6	0,36	4	0	1,44	4,5	29	187,92			
EW	S	2,6	4,22	10,97	1	1,95	9,02	1,46	29	381,91			
EW1	S	1,3	1,5	1,95	1	0	1,95	2,2	29	124,41			
EW	E	2,6	2,01	5,23	1	2,16	3,07	1,46	29	129,98			
ED	E	2,4	0,9	2,16	1	0	2,16	5	29	313,2			
IW1	N	2,6	1,52	3,95	1	0	3,95	1,45	12	68,73			
IW1	E	2,6	2,37	6,16	1	0	6,16	1,45	12	107,18			
GSF		4,22	4,26	17,98	1	0	17,98	0,66	13	154,27			
GSF		2,37	1,56	3,7	1	0	3,7	0,66	13	31,746			
										2426	0,05	0,8	2037,814

a	l	Quantity	T_p	R	H	Z_E	E_Y	Q_L
		pcs	^oC					kcal/h
2,5	7,13	1	30	0,8	0,7	1,2	1	359,352

Q_H(kcal/h)= **2397,166**

Q_H(W)= **2787,28**

qf,des(W/m2)= **155,0212**

Conventional System **Q_H(W)=** **3890,9**

Table A.3. Ground Floor (Hall and Entrance) Heating Load,kW

Ground Floor :Hall & Entrance (17 ^{o}C)													
Structural element													
Type	Direction	Height	Width	A_o	Quantity	A_{ex}	A_{net}	U_p	T_p	Q_o	Z_H	C	Q_T
		m	m	m^2	pcs	m^2	m^2	kcal/hm^2 C	^{o}C	kcal/h			kcal/h
EW	E	2,6	2,32	6,03	1	1,56	4,47	1,46	29	189,26			
EG	E	2,6	0,6	1,56	1	0	1,56	0,7	29	31,668			
EW	S	2,6	1,01	2,63	1	0	5,36	1,46	29	226,94			
EW	E	2,6	2,06	5,36	1	0	5,36	1,46	29	226,94			
EW	S	2,6	2,06	5,36	1	3,36	2	1,46	29	84,68			
ED	S	2,4	1,4	3,36	1	0	3,36	3	29	292,32			
IW1	W	2,6	3,06	7,96	1	2,16	5,8	1,45	12	100,92			
ID	W	2,4	0,9	2,16	1	0	2,16	2	12	51,84			
GSF		2,32	4,56	10,58	1	0	10,58	0,66	13	90,776			
GSF		2,06	2,06	4,24	1	0	4,24	0,66	13	36,379			
										1331,7	-0,05	0,8	1012,113

a	l	Quantity	T_p	R	H	Z_E	E_Y	Q_L
		pcs	^{o}C					kcal/h
2,5	10,08	1	30	0,8	0,7	1	1	423,36

Q_H(kcal/h)= **1435,473**

Q_H(W)= **1669,082**

qf,des(W/m2)= **149,1584**

Conventional System **Q_H(W)=** **2118**

Table A.4. First Floor (Bedroom) Heating Load,kW

First Floor :Bedroom-NW (17 °C)													
Structural element													
Type	Direction	Height	Width	A_o	Quantity	A_{ex}	A_{net}	U_p	T_p	Q_o	Z_H	C	Q_T
		m	m	m^2	pcs	m^2	m^2	kcal/hm^2 C	°C	kcal/h			kcal/h
EW	W	2,7	5,41	14,61	1	1,28	13,33	1,46	29	564,39			
EW1	W	1,6	0,8	1,28	1	0	1,28	4,5	29	167,04			
EW	N	2,7	3,07	8,29	1	1,5	6,79	1,46	29	287,49			
EW1	N	1,5	1	1,5	1	0	1,5	4,5	29	195,75			
CCR		5,41	3,07	16,61	1	0	16,61	0,72	25	298,98			
										1513,7	0,05	0,8	1271,467

a	l	Quantity	T_p	R	H	Z_E	E_Y	Q_L
		pcs	°C					kcal/h
2,5	5,55	1	30	0,8	0,7	1,2	1	279,72

Q_H(kcal/h)= **1551,187**

Q_H(W)= **1803,627**

qf,des(W/m2)= **108,5868**

Conventional System **Q_H(W)=** **2808,8**

Table A.5 First Floor (Bedroom) Heating Load,kW

First Floor :Bedroom-NE(17 ^oC)													
Structural element													
Type	Direction	Height	Width	A_o	Quantity	A_{ex}	A_{net}	U_p	T_p	Q_o	Z_H	C	Q_T
		m	m	m^2	pcs	m^2	m^2	kcal/hm^2 C	^oC	kcal/h			kcal/h
EW	E	2,7	5,41	14,61	1	1,28	13,33	1,46	29	564,39			
EW1	E	1,6	0,8	1,28	1	0	1,28	4,5	29	167,04			
EW	N	2,7	3,1	8,37	1	1,5	6,87	1,46	29	290,88			
EW1	N	1,5	1	1,5	1	0	1,5	4,5	29	195,75			
EW	S	2,65	2,15	5,7	1	0	5,7	1,46	29	241,34			
CCR		5,41	3,1	16,77	1	0	16,77	0,72	25	301,86			
										1761,3	0,05	0,8	1479,455
					a	l	Quantity	T_p	R	H	Z_E	E_Y	Q_L
							pcs	^oC					kcal/h
					2,5	5,55	1	30	0,8	0,7	1,2	1	279,72
									Q_H(kcal/h)=				**1759,175**

Q_H(W)= **2045,463**

qf,des(W/m2)= **121,9716**

Conventional System **Q_H(W)=** **3150,4**

Table A.6. First Floor (Hall) Heating Load,kW

First Floor :Hall(17 °C)													
Structural element													
Type	Direction	Height	Width	A_o	Quantity	A_{ex}	A_{net}	U_p	T_p	Q_o	Z_H	C	Q_T
		m	m	m^2	pcs	m^2	m^2	kcal/hm^2 C	°C	kcal/h			kcal/h
EW	N	2,7	2,52	6,8	1	2,55	4,25	1,46	29	179,95			
EW1	N	1,5	1,7	2,55	1	0	2,55	4,5	29	332,78			
EW	E	2,95	2,32	6,84	1	1,74	5,1	1,46	29	215,93			
EG	E	2,95	0,6	1,77	1	0	1,77	0,7	29	35,931			
EW	S	2,95	1,01	2,98	1	0	2,98	1,46	29	126,17			
CCR		2,52	5,51	13,89	1	0	13,89	0,72	25	250,02			
SSF		2,32	4,56	10,58	1	0	10,58	0,72	18	137,12			
										1277,9	0,05	0,8	1073,432

a	l	Quantity	T_p	R	H	Z_E	E_Y	Q_L
		pcs	°C					kcal/h
2,5	9,44	1	30	0,8	0,7	1,2	1	475,776

Q_H(kcal/h)= **1549,208**

Q_H(W)= **1801,326**

qf,des(W/m2)= **129,6851**

Table A.7. First Floor (Parent's Bedroom) Heating Load,kW

First Floor :Parent's Bedroom(17 °C)													
Structural element													
Type	Direction	Height	Width	A_o	Quantity	A_{ex}	A_{net}	U_p	T_p	Q_o	Z_H	C	Q_T
		m	m	m^2	pcs	m^2	m^2	kcal/hm^2 C	°C	kcal/h			kcal/h
EW	W	2,6	4,31	11,21	1	1,28	9,93	1,46	29	420,44			
EW1	W	1,6	0,8	1,28	1	0	1,28	4,5	29	167,04			
EW	S	2,6	4,22	10,97	1	2,25	8,72	1,46	29	369,2			
EW1	S	1,5	1,5	2,25	1	0	2,25	2,2	29	143,55			
EW	E	2,6	2,06	5,36	1	1,92	3,44	1,46	29	145,65			
ED	E	2,4	0,8	1,92	1	0	1,92	3	29	167,04			
TR		4,22	4,31	18,19	1	0	18,19	0,42	29	221,55			
										1634,5	0	0,8	1307,58

a	l	Quantity	T_p	R	H	Z_E	E_Y	Q_L
		pcs	°C					kcal/h
2,5	8,33	1	30	0,8	0,7	1,2	1	419,832

Q_H(kcal/h)= **1727,412**

Q_H(W)= **2008,531**

qf,des(W/m2)= **110,4195**

Conventional System **Q_H(W)=** **3097,9**

Table A.8. First Floor (Bathroom) Heating Load,kW

First Floor :Bathroom(20 ^{o}C)													
Structural element													
Type	Direction	Height	Width	A_o	Quantity	A_{ex}	A_{net}	U_p	T_p	Q_o	Z_H	C	Q_T
		m	m	m^2	pcs	m^2	m^2	kcal/hm^2 C	^{o}C	kcal/h			kcal/h
EWB	W	2,95	2,32	6,84	1	0,54	6,3	0,64	32	129,02			
EWI	W	0,6	0,9	0,54	1	0	0,54	4,5	32	77,76			
EWB	N	2,95	1,2	3,54	1	0	3,54	0,64	32	72,499			
IW1	S	2,95	3,17	9,35	1	0	9,35	1,45	5	67,788			
IW1	N	2,95	1,97	5,81	1	0	5,81	1,45	5	42,123			
IW1	E	2,95	2,32	6,84	1	2,16	4,68	1,45	5	33,93			
ID	E	2,4	0,9	2,16	1	0	2,16	2	5	21,6			
SSF		2,32	3,17	7,35	1	0	7,35	0,72	21	111,13			
FSF		2,32	3,17	7,35	1	4,65	2,7	0,68	5	9,18			
FSF		3,06	4,65	4,65	1	0	4,65	0,68	15	47,43			
										612,47	0,05	0,8	514,4708
					a	l	Quantity	T_p	R	H	Z_E	E_Y	Q_L
							pcs	^{o}C					kcal/h
					2,5	2,65	1	32	0,8	0,7	1	1	118,72
									Q_H(kcal/h)=				**633,1908**

Q_H(W)= **736,2362**

qf,des(W/m2)= **100,1682**

Conventional System **Q_H(W)=** **1255**

Table A.9. First Floor (Bathroom) Heating Load,kW

First Floor :Parent's Bathroom(20 °C)													
Structural element													
Type	Direction	Height	Width	A_o	Quantity	A_{ex}	A_{net}	U_p	T_p	Q_o	Z_H	C	Q_T
		m	m	m^2	pcs	m^2	m^2	kcal/hm^2 C	°C	kcal/h			kcal/h
EWB	S	2,6	2,11	5,49	1	0,45	5,04	0,64	32	103,22			
EWI	S	0,5	0,9	0,45	1	0	0,45	4,5	32	64,8			
EWB	E	2,6	2,06	5,36	1	0	5,36	0,64	32	109,77			
IW1	N	2,6	2,11	5,49	1	0	5,49	1,45	5	39,803			
IW1	W	2,6	2,06	5,36	1	1,92	3,44	1,45	5	24,94			
ID	W	2,4	0,8	1,92	1	0	1,92	2	5	19,2			
TR		2,06	2,11	4,35	1	0	4,35	0,42	32	58,464			
FSF		2,06	2,1	4,35	1	0	4,35	0,68	5	14,79			
										434,99	-0,05	0,8	330,5913
					a	l	Quantity	T_p	R	H	Z_E	E_Y	Q_L
							pcs	°C					kcal/h
					2,5	2,21	1	32	0,8	0,7	1	1	99,008
									Q_H(kcal/h)=				429,5993

Q_H(W)= **499,5122**

qf,des(W/m2)= **114,8304**

$(Q_H)_{Total}$(kcal/h)= 15632 kcal/h

$(Q_H)_{Total}$(kcal/h)= 23259 kcal/h (Conventional system)

Ground floor area = 80,91 m^2

First floor area = 87,74 m^2

Roof floor area = 66,36 m^2

Total area = 235 m^2

Total area to be heated = 149,7 m^2

APPENDIX B

FLOOR HEATING SYSTEM MODELLING

B.1. Floor Heating System Modelling Results

Table B.1. Floor heating system modelling results

Floor	T_a	Q_H	A_p	$q_{f,des}$	$q_{f,max}$	T_f	S
	^oC	W	m^2	W/m^2	W/m^2	^oC	m
GF-Living room	18	4824,18	39,68	121,577	124,7	45	0,2
GF-Kitchen	17	2787,28	21,68	155,021	137,2	45	0,1
GF-Hall&Ent.	17	1669,08	11,19	149,158	137,2	45	0,1
FF-Bedroom-NW	17	1803,63	16,61	108,587	137,2	45	0,3
FF-Bedroom-NE	17	2045,46	16,77	121,972	137,2	45	0,25
FF-Hall	17	1801,33	13,89	129,685	137,2	45	0,25
FF-Parent's Bed.	17	2008,53	18,19	110,42	137,2	45	0,3
FF-Bathroom	20	736,236	7,35	100,168	100	45	0,3
FF-Parent's Bath.	20	499,512	4,35	114,83	100	45	0,25

Floor	q_f	T_p	X	q_s	Q_s	m_w	V_w
	W/m^2	^oC		W/m^2	W	kg/s	m^3/h
GF-Living room	118,8	28,5	0,737	161,2	6395,7	0,15	0,56
GF-Kitchen	152,8	30,2	0,755	202,5	4389,6	0,09	0,31
GF-Hall&Ent.	152,8	30,2	0,755	202,5	2265,7	0,04	0,16
FF-Bedroom-NW	117,7	27,4	0,803	146,5	2434,1	0,05	0,19
FF-Bedroom-NE	136,6	28,9	0,803	170,1	2852,2	0,06	0,21
FF-Hall	136,6	28,9	0,803	170,1	2362,3	0,05	0,18
FF-Parent's Bed.	117,7	27,4	0,798	147,6	2684,2	0,06	0,2
FF-Bathroom	105,1	29,4	0,724	145,1	1066,7	0,02	0,08
FF-Parent's Bath.	121,9	30,8	0,782	155,9	678	0,01	0,05

B.2. Floor Heating System Modelling Outputs of the Sample Building

Ground Floor Living Room:

Design heating load

$\boxed{q_{fdes} := 121.58}$ W/m^2

Panel area

$\boxed{A_p := 39.68}$ m^2

Inside air design temperature

$\boxed{T_a := 18}$ °C

Downside space temperature

$\boxed{T_n := 6}$ °C

Maximum temperature

$\boxed{T_{pmax} := 29}$ °C

Convection heat trans.of panel surf.

$\boxed{h_{ps} := 5.8}$ W/m^{2}K

Equivalent thermal cond.

$\boxed{k_{eq} := 0.51}$ W/mK

Total thickness above pipe

$\boxed{t_y := 0.08}$ m

Resistance between pipe center and panel surface

$\boxed{R_y := 0.16}$ m^{2}K/W

Resistance between pipe center and downide space

$\boxed{R_a := 0.98}$ m^{2}K/W

Outside pipe diameter

$\boxed{D_o := 0.02}$ m

Inside pipe diameter

$\boxed{D_i := 0.016}$ m

Conv. heat transfer on the pipe surface.

$\boxed{h_f := 3000}$ W/m^{2}K

Mean fluid temperature

$\boxed{T_f := 45}$ °C

Fluid inlet temperature

$\boxed{T_{fi} := 50}$ °C

Fluid exit temperature

$\boxed{T_{fo} := 40}$ °C

Specific heat of water

$\boxed{c_{pw} := 4180}$ kJ/kgK

density

$\boxed{\rho_w := 990}$ kg/m^3

$$q_{fmax} := 8.92 \cdot (T_{pmax} - T_a)^{1.1}$$

$$q_{fmax} = 124.7 \quad W/m^2$$

$$m := \sqrt{\frac{h_{ps}}{k_{eq} \cdot t_y}}$$

$$W(S) := \frac{S - D_o}{2}$$

$$F(S) := \frac{\tanh(m \cdot W(S))}{m \cdot W(S)}$$

$$q_{fdes} := 121.6 \quad W/m^2$$

$$F_1(S) := \left[\frac{S}{D_o + (S - D_o) \cdot F(S)} + \frac{h_{ps} \cdot S}{\pi \cdot D_i \cdot h_f} \right]^{-1}$$

$$S := 0.1 \quad m$$

Given

$$q_{fdes} = F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$$

$$\text{Find}(S) = 0.191 \quad m$$

$$S := 0.2 \quad m$$

$$q_f(S) := F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$$

$q_f(S) = 118.8$ W/m^2

$q_f := 118.8$ W/m^2

$$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}}$$

$T_p = 28.5$ °C

$$T_b := T_p + q_f \cdot R_y$$

$T_b = 47.5$ °C

$$X := \frac{1}{1 + \frac{R_y \cdot (T_b - T_n)}{R_a \cdot (T_b - T_p)}}$$

$X = 0.737$

$$q_s := \frac{q_f}{X}$$

$q_s = 161.2$ W/m^2

$$Q_s := q_s \cdot A_p$$

$Q_s = 6395.7$ W

$$m_w := \frac{Q_s}{c_{pw} \cdot (T_{fi} - T_{fo})}$$

$m_w = 0.153$ kg/s

$$V_w := \frac{m_w}{\rho_w} \cdot 3600$$

$V_w = 0.556$ m^3/h

Ground Floor - Kitchen:

Design heating load

$\boxed{q_{fdes} := 155.2}$ W/m^2

Panel area

$\boxed{A_p := 21.68}$ m^2

Inside air design temperature

$\boxed{T_a := 17}$ °C

Downside space temperature

$\boxed{T_n := 6}$ °C

Maximum temperature

$\boxed{T_{pmax} := 29}$ °C

Convection heat trans.of panel surf.

$\boxed{h_{ps} := 5.8}$ W/m^{2}K

Equivalent thermal cond.

$\boxed{k_{eq} := 0.51}$ W/mK

Total thickness above pipe

$\boxed{t_y := 0.08}$ m

Resistance between pipe center and panel surface

$\boxed{R_y := 0.16}$ m^{2}K/W

Resistance between pipe center and downide space

$\boxed{R_a := 0.98}$ m^{2}K/W

Outside pipe diameter

$\boxed{D_o := 0.02}$ m

Inside pipe diameter

$\boxed{D_i := 0.016}$ m

Conv. heat transfer o the pipe surface.

$\boxed{h_f := 3000}$ W/m^{2}K

Mean fluid temperature

$\boxed{T_f := 45}$ °C

Fluid inlet temperature

$\boxed{T_{fi} := 50}$ °C

Fluid exit temperature

$\boxed{T_{fo} := 40}$ °C

Specific heat of water

$\boxed{c_{pw} := 4180}$ kJ/kgK

density

$\boxed{\rho_w := 990}$ kg/m^3

$q_{fmax} := 8.92 \cdot (T_{pmax} - T_a)^{1.1}$

$q_{fmax} = 137.2$ W/m^2

$$m := \sqrt{\frac{h_{ps}}{k_{eq} \cdot t_y}}$$

$$W(S) := \frac{S - D_o}{2}$$

$$F(S) := \frac{\tanh(m \cdot W(S))}{m \cdot W(S)}$$

$q_{fdes} := 155.02$ W/m^2

$$F_1(S) := \left[\frac{S}{D_o + (S - D_o) \cdot F(S)} + \frac{h_{ps} \cdot S}{\pi \cdot D_i \cdot h_f}\right]^{-1}$$

$S := 0.1$ m

Given

$$q_{fdes} = F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$$

$\text{Find}(S) = 0.09$ m

$S := 0.10$m

$$q_f(S) := F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$$ $$q_f(S) = 152.8 \quad \text{W/m}^2$$

$$q_f := 152.8 \quad \text{W/m}^2$$

$$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}}$$ $$T_p = 30.2 \quad {}^\circ\text{C}$$

$$T_b := T_p + q_f \cdot R_y$$ $$T_b = 54.7 \quad {}^\circ\text{C}$$

$$X := \frac{1}{1 + \frac{R_y \cdot (T_b - T_n)}{R_a \cdot (T_b - T_p)}}$$ $$X = 0.755$$

$$q_s := \frac{q_f}{X}$$ $$q_s = 202.5 \quad \text{W/m}^2$$

$$Q_s := q_s \cdot A_p$$ $$Q_s = 4389.6 \quad \text{W}$$

$$m_w := \frac{Q_s}{c_{pw} \cdot (T_{fi} - T_{fo})}$$ $$m_w = 0.105 \quad \text{kg/s}$$

$$V_w := \frac{m_w}{\rho_w} \cdot 3600$$ $$V_w = 0.382 \quad \text{m}^3\text{/h}$$

Ground Floor - Hall and Entrance:

Design heating load

$\boxed{q_{fdes} := 149.15}$ W/m^2

Panel area

$\boxed{A_p := 11.19}$ m^2

Inside air design temperature

$\boxed{T_a := 17}$ °C

Downside space temperature

$\boxed{T_n := 6}$ °C

Maximum temperature

$\boxed{T_{pmax} := 29}$ °C

Convection heat trans.of panel surf.

$\boxed{h_{ps} := 5.8}$ W/m^{2}K

Equivalent thermal cond.

$\boxed{k_{eq} := 0.51}$ W/mK

Total thickness above pipe

$\boxed{t_y := 0.08}$ m

Resistance between pipe center and panel surface

$\boxed{R_y := 0.16}$ m^{2}K/W

Resistance between pipe center and downide space

$\boxed{R_a := 0.98}$ m^{2}K/W

Outside pipe diameter

$\boxed{D_o := 0.02}$ m

Inside pipe diameter

$\boxed{D_i := 0.016}$ m

Conv. heat transfer o the pipe surface.

$\boxed{h_f := 3000}$ W/m^{2}K

Mean fluid temperature

$\boxed{T_f := 45}$ °C

Fluid inlet temperature

$\boxed{T_{fi} := 50}$ °C

Fluid exit temperature

$\boxed{T_{fo} := 40}$ °C

Specific heat of water

$\boxed{c_{pw} := 4180}$ kJ/kgK

density

$\boxed{\rho_w := 990}$ kg/m^3

$$q_{fmax} := 8.92 \cdot \left(T_{pmax} - T_a\right)^{1.1}$$

$$q_{fmax} = 137.2 \quad W/m^2$$

$$m := \sqrt{\frac{h_{ps}}{k_{eq} \cdot t_y}}$$

$$W(S) := \frac{S - D_o}{2}$$

$$F(S) := \frac{\tanh(m \cdot W(S))}{m \cdot W(S)}$$

$$q_{fdes} := 149.15 \quad W/m^2$$

$$F_1(S) := \left[\frac{S}{D_o + \left(S - D_o\right) \cdot F(S)} + \frac{h_{ps} \cdot S}{\pi \cdot D_i \cdot h_f}\right]^{-1}$$

$$S := 0.1 \quad m$$

Given

$$q_{fdes} = F_1(S) \cdot h_{ps} \cdot \left(T_f - T_a\right)$$

$$\mathrm{Find}(S) = 0.115 \quad m$$

$S := 0.10\text{m}$

$q_f(S) := F_1(S)\cdot h_{ps}\cdot(T_f - T_a)$ $\qquad q_f(S) = 152.8 \quad \text{W/m}^2$

$q_f := 152.8 \quad \text{W/m}^2$

$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}}$ $\qquad T_p = 30.2 \quad °\text{C}$

$T_b := T_p + q_f\cdot R_y$ $\qquad T_b = 54.7 \quad °\text{C}$

$X := \frac{1}{1 + \frac{R_y\cdot(T_b - T_n)}{R_a\cdot(T_b - T_p)}}$ $\qquad X = 0.755$

$q_s := \frac{q_f}{X}$ $\qquad q_s = 202.5 \quad \text{W/m}^2$

$Q_s := q_s\cdot A_p$ $\qquad Q_s = 2265.7 \quad \text{W}$

$m_w := \frac{Q_s}{c_{pw}\cdot(T_{fi} - T_{fo})}$ $\qquad m_w = 0.054 \quad \text{kg/s}$

$V_w := \frac{m_w}{\rho_w}\cdot 3600$ $\qquad V_w = 0.197 \quad \text{m}^3\text{/h}$

First Floor - Bedroom-NW:

Design heating load

$q_{fdes} := 108.58$ W/m^2

Panel area

$A_p := 16.61$ m^2

Inside air design temperature

$T_a := 17$ °C

Downside space temperature

$T_n := 18$ °C

Maximum temperature

$T_{pmax} := 29$ °C

Convection heat trans.of panel surf.

$h_{ps} := 5.8$ W/m^{2}K

Equivalent thermal cond.

$k_{eq} := 0.51$ W/mK

Total thickness above pipe

$t_y := 0.08$ m

Resistance between pipe center and panel surface

$R_y := 0.16$ m^{2}K/W

Resistance between pipe center and downide space

$R_a := 0.98$ m^{2}K/W

Outside pipe diameter

$D_o := 0.02$ m

Inside pipe diameter

$D_i := 0.016$ m

Conv. heat transfer o the pipe surface.

$h_f := 3000$ W/m^{2}K

Mean fluid temperature

$T_f := 45$ °C

Fluid inlet temperature

$T_{fi} := 50$ °C

Fluid exit temperature

$T_{fo} := 40$ °C

Specific heat of water

$c_{pw} := 4180$ kJ/kgK

density

$\rho_w := 990$ kg/m^3

$$q_{fmax} := 8.92\cdot\left(T_{pmax} - T_a\right)^{1.1}$$

$$q_{fmax} = 137.2 \quad \text{W/m}^2$$

$$m := \sqrt{\frac{h_{ps}}{k_{eq}\cdot t_y}}$$

$$W(S) := \frac{S - D_o}{2}$$

$$F(S) := \frac{\tanh(m\cdot W(S))}{m\cdot W(S)}$$

$$q_{fdes} := 108.58 \quad \text{W/m}^2$$

$$F_1(S) := \left[\frac{S}{D_o + \left(S - D_o\right)\cdot F(S)} + \frac{h_{ps}\cdot S}{\pi\cdot D_i\cdot h_f}\right]^{-1}$$

$$S := 0.1 \quad \text{m}$$

Given

$$q_{fdes} = F_1(S)\cdot h_{ps}\cdot\left(T_f - T_a\right)$$

$$\text{Find}(S) = 0.248 \quad \text{m}$$

$S := 0.30$ m

$q_f(S) := F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$ $\quad q_f(S) = 94.8$ W/m^2

$q_f := 117.7$ W/m^2

$$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}}$$ $\quad T_p = 27.4$ °C

$T_b := T_p + q_f \cdot R_y$ $\quad T_b = 46.3$ °C

$$X := \frac{1}{1 + \frac{R_y \cdot (T_b - T_n)}{R_a \cdot (T_b - T_p)}}$$ $\quad X = 0.803$

$$q_s := \frac{q_f}{X}$$ $\quad q_s = 146.5$ W/m^2

$Q_s := q_s \cdot A_p$ $\quad Q_s = 2434.1$ W

$$m_w := \frac{Q_s}{c_{pw} \cdot (T_{fi} - T_{fo})}$$ $\quad m_w = 0.058$ kg/s

$$V_w := \frac{m_w}{\rho_w} \cdot 3600$$ $\quad V_w = 0.212$ m^3/h

First Floor - Bedroom-NE:

Design heating load

$q_{fdes} := 1121.97$ W/m²

Panel area

$A_p := 16.77$ m²

Inside air design temperature

$T_a := 17$ °C

Downside space temperature

$T_n := 18$ °C

Maximum temperature

$T_{pmax} := 29$ °C

Convection heat trans.of panel surf.

$h_{ps} := 5.8$ W/m²K

Equivalent thermal cond.

$k_{eq} := 0.51$ W/mK

Total thickness above pipe

$t_y := 0.08$ m

Resistance between pipe center and panel surface

$R_y := 0.16$ m²K/W

Resistance between pipe center and downide space

$R_a := 0.98$ m²K/W

Outside pipe diameter

$D_o := 0.02$ m

Inside pipe diameter

$D_i := 0.016$ m

Conv. heat transfer o the pipe surface.

$h_f := 3000$ W/m²K

Mean fluid temperature

$T_f := 45$ °C

Fluid inlet temperature

$T_{fi} := 50$ °C

Fluid exit temperature

$T_{fo} := 40$ °C

Specific heat of water

$c_{pw} := 4180$ kJ/kgK

density

$\rho_w := 990$ kg/m³

$$q_{fmax} := 8.92 \cdot (T_{pmax} - T_a)^{1.1}$$

$$q_{fmax} = 137.2 \quad W/m^2$$

$$m := \sqrt{\frac{h_{ps}}{k_{eq} \cdot t_y}}$$

$$W(S) := \frac{S - D_o}{2}$$

$$F(S) := \frac{\tanh(m \cdot W(S))}{m \cdot W(S)}$$

$$q_{fdes} := 121.97 \quad W/m^2$$

$$F_1(S) := \left[\frac{S}{D_o + (S - D_o) \cdot F(S)} + \frac{h_{ps} \cdot S}{\pi \cdot D_i \cdot h_f}\right]^{-1}$$

$$S := 0.1 \quad m$$

Given

$$q_{fdes} = F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$$

$$\mathrm{Find}(S) = 0.204 \quad m$$

$S := 0.25m$

$q_f(S) := F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$ $\qquad q_f(S) = 108.1$ W/m^2

$q_f := 136.6$ W/m^2

$$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}} \qquad T_p = 28.9 \quad °C$$

$$T_b := T_p + q_f \cdot R_y \qquad T_b = 50.8 \quad °C$$

$$X := \frac{1}{1 + \frac{R_y \cdot (T_b - T_n)}{R_a \cdot (T_b - T_p)}} \qquad X = 0.803$$

$$q_s := \frac{q_f}{X} \qquad q_s = 170.1 \quad W/m^2$$

$$Q_s := q_s \cdot A_p \qquad Q_s = 2852.2 \quad W$$

$$m_w := \frac{Q_s}{c_{pw} \cdot (T_{fi} - T_{fo})} \qquad m_w = 0.068 \quad kg/s$$

$$V_w := \frac{m_w}{\rho_w} \cdot 3600 \qquad V_w = 0.248 \quad m^3/h$$

First Floor - Hall:

Design heating load

$q_{fdes} := 129.6$ W/m^2

Panel area

$A_p := 13.89$ m^2

Inside air design temperature

$T_a := 17$ °C

Downside space temperature

$T_n := 18$ °C

Maximum temperature

$T_{pmax} := 29$ °C

Convection heat trans.of panel surf.

$h_{ps} := 5.8$ W/m^{2}K

Equivalent thermal cond.

$k_{eq} := 0.51$ W/mK

Total thickness above pipe

$t_y := 0.08$ m

Resistance between pipe center and panel surface

$R_y := 0.16$ m^{2}K/W

Resistance between pipe center and downide space

$R_a := 0.98$ m^{2}K/W

Outside pipe diameter

$D_o := 0.02$ m

Inside pipe diameter

$D_i := 0.016$ m

Conv. heat transfer o the pipe surface.

$h_f := 3000$ W/m^{2}K

Mean fluid temperature

$T_f := 45$ °C

Fluid inlet temperature

$T_{fi} := 50$ °C

Fluid exit temperature

$T_{fo} := 40$ °C

Specific heat of water

$c_{pw} := 4180$ kJ/kgK

density

$\rho_w := 990$ kg/m^3

$$q_{fmax} := 8.92 \cdot (T_{pmax} - T_a)^{1.1}$$

$$q_{fmax} = 137.2 \quad W/m^2$$

$$m := \sqrt{\frac{h_{ps}}{k_{eq} \cdot t_y}}$$

$$W(S) := \frac{S - D_o}{2}$$

$$F(S) := \frac{\tanh(m \cdot W(S))}{m \cdot W(S)}$$

$$q_{fdes} := 129.6 \quad W/m^2$$

$$F_1(S) := \left[\frac{S}{D_o + (S - D_o) \cdot F(S)} + \frac{h_{ps} \cdot S}{\pi \cdot D_i \cdot h_f} \right]^{-1}$$

$$S := 0.1 \quad m$$

Given

$$q_{fdes} = F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$$

$$Find(S) = 0.18 \quad m$$

$S := 0.25$ m

$q_f(S) := F_1(S)\cdot h_{ps}\cdot(T_f - T_a)$ $\quad q_f(S) = 108.1$ W/m^2

$q_f := 136.6$ W/m^2

$$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}}$$ $T_p = 28.9$ °C

$T_b := T_p + q_f\cdot R_y$ $\quad T_b = 50.8$ °C

$$X := \frac{1}{1 + \frac{R_y\cdot(T_b - T_n)}{R_a\cdot(T_b - T_p)}}$$ $X = 0.803$

$$q_s := \frac{q_f}{X}$$ $q_s = 170.1$ W/m^2

$Q_s := q_s\cdot A_p$ $\quad Q_s = 2362.3$ W

$$m_w := \frac{Q_s}{c_{pw}\cdot(T_{fi} - T_{fo})}$$ $m_w = 0.057$ kg/s

$$V_w := \frac{m_w}{\rho_w}\cdot 3600$$ $V_w = 0.206$ m^3/h

First Floor - Parent's Bedroom:

Design heating load	Panel area	
$\boxed{q_{fdes} := 110.42}$ W/m^2	$\boxed{A_p := 18.19}$ m^2	

Inside air design temperature	Downside space temperature	Maximum temperature
$\boxed{T_a := 17}$ °C	$\boxed{T_n := 17}$ °C	$\boxed{T_{pmax} := 29}$ °C

Convection heat trans.of panel surf.	Equivalent thermal cond.	Total thickness above pipe
$\boxed{h_{ps} := 5.8}$ W/m^{2}K	$\boxed{k_{eq} := 0.51}$ W/mK	$\boxed{t_y := 0.08}$ m

Resistance between pipe center and panel surface	Resistance between pipe center and downide space
$\boxed{R_y := 0.16}$ m^{2}K/W	$\boxed{R_a := 0.98}$ m^{2}K/W

Outside pipe diameter	Inside pipe diameter	Conv. heat transfer o the pipe surface.
$\boxed{D_o := 0.02}$ m	$\boxed{D_i := 0.016}$ m	$\boxed{h_f := 3000}$ W/m^{2}K

Mean fluid temperature	Fluid inlet temperature	Fluid exit temperature
$\boxed{T_f := 45}$ °C	$\boxed{T_{fi} := 50}$ °C	$\boxed{T_{fo} := 40}$ °C

Specific heat of water	density
$\boxed{c_{pw} := 4180}$ kJ/kgK	$\rho_w := 990$ kg/m^3

$$q_{fmax} := 8.92 \cdot \left(T_{pmax} - T_a\right)^{1.1}$$

$$q_{fmax} = 137.2 \quad \text{W/m}^2$$

$$m := \sqrt{\frac{h_{ps}}{k_{eq} \cdot t_y}}$$

$$W(S) := \frac{S - D_o}{2}$$

$$F(S) := \frac{\tanh(m \cdot W(S))}{m \cdot W(S)}$$

$$q_{fdes} := 110.42 \quad \text{W/m}^2$$

$$F_1(S) := \left[\frac{S}{D_o + \left(S - D_o\right) \cdot F(S)} + \frac{h_{ps} \cdot S}{\pi \cdot D_i \cdot h_f}\right]^{-1}$$

$$S := 0.1 \quad \text{m}$$

Given

$$q_{fdes} = F_1(S) \cdot h_{ps} \cdot \left(T_f - T_a\right)$$

$$\text{Find}(S) = 0.242 \quad \text{m}$$

$S := 0.30$ m

$q_f(S) := F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$ $q_f(S) = 94.8$ W/m²

$q_f := 117.7$ W/m²

$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}}$ $T_p = 27.4$ °C

$T_b := T_p + q_f \cdot R_y$ $T_b = 46.3$ °C

$X := \frac{1}{1 + \frac{R_y \cdot (T_b - T_n)}{R_a \cdot (T_b - T_p)}}$ $X = 0.798$

$q_s := \frac{q_f}{X}$ $q_s = 147.6$ W/m²

$Q_s := q_s \cdot A_p$ $Q_s = 2684.2$ W

$m_w := \frac{Q_s}{c_{pw} \cdot (T_{fi} - T_{fo})}$ $m_w = 0.064$ kg/s

$V_w := \frac{m_w}{\rho_w} \cdot 3600$ $V_w = 0.234$ m³/h

First Floor - Bathroom:

Design heating load

$q_{fdes} := 100.17$ W/m²

Panel area

$A_p := 7.35$ m²

Inside air design temperature

$T_a := 20$ °C

Downside space temperature

$T_n := 7$ °C

Maximum temperature

$T_{pmax} := 29$ °C

Convection heat trans.of panel surf.

$h_{ps} := 5.8$ W/m²K

Equivalent thermal cond.

$k_{eq} := 0.51$ W/mK

Total thickness above pipe

$t_y := 0.08$ m

Resistance between pipe center and panel surface

$R_y := 0.16$ m²K/W

Resistance between pipe center and downide space

$R_a := 0.98$ m²K/W

Outside pipe diameter

$D_o := 0.02$ m

Inside pipe diameter

$D_i := 0.016$ m

Conv. heat transfer on the pipe surface.

$h_f := 3000$ W/m²K

Mean fluid temperature

$T_f := 45$ °C

Fluid inlet temperature

$T_{fi} := 50$ °C

Fluid exit temperature

$T_{fo} := 40$ °C

Specific heat of water

$c_{pw} := 4180$ kJ/kgK

density

$\rho_w := 990$ kg/m³

$$q_{fmax} := 8.92 \cdot (T_{pmax} - T_a)^{1.1}$$

$$q_{fmax} = 100 \quad W/m^2$$

$$m := \sqrt{\frac{h_{ps}}{k_{eq} \cdot t_y}}$$

$$W(S) := \frac{S - D_o}{2}$$

$$F(S) := \frac{\tanh(m \cdot W(S))}{m \cdot W(S)}$$

$$q_{fdes} := 100.17 \quad W/m^2$$

$$F_1(S) := \left[\frac{S}{D_o + (S - D_o) \cdot F(S)} + \frac{h_{ps} \cdot S}{\pi \cdot D_i \cdot h_f} \right]^{-1}$$

$$S := 0.1 \quad m$$

Given

$$q_{fdes} = F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$$

$$\text{Find}(S) = 0.236 \quad m$$

$S := 0.30$ m

$q_f(S) := F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$ $\qquad q_f(S) = 84.6$ W/m^2

$q_f := 105.1$ W/m^2

$$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}} \qquad T_p = 29.4 \quad °C$$

$$T_b := T_p + q_f \cdot R_y \qquad T_b = 46.2 \quad °C$$

$$X := \frac{1}{1 + \frac{R_y \cdot (T_b - T_n)}{R_a \cdot (T_b - T_p)}} \qquad X = 0.724$$

$$q_s := \frac{q_f}{X} \qquad q_s = 145.1 \quad W/m^2$$

$$Q_s := q_s \cdot A_p \qquad Q_s = 1066.7 \, W$$

$$m_w := \frac{Q_s}{c_{pw} \cdot (T_{fi} - T_{fo})} \qquad m_w = 0.026 \quad kg/s$$

$$V_w := \frac{m_w}{\rho_w} \cdot 3600 \qquad V_w = 0.093 \quad m^3/h$$

First Floor - Parent's Bathroom:

Design heating load

$\boxed{q_{fdes} := 114.83}$ W/m^2

Panel area

$\boxed{A_p := 4.35}$ m^2

Inside air design temperature

$\boxed{T_a := 20}$ °C

Downside space temperature

$\boxed{T_n := 17}$ °C

Maximum temperature

$\boxed{T_{pmax} := 29}$ °C

Convection heat trans.of panel surf.

$\boxed{h_{ps} := 5.8}$ W/m^2K

Equivalent thermal cond.

$\boxed{k_{eq} := 0.51}$ W/mK

Total thickness above pipe

$\boxed{t_y := 0.08}$ m

Resistance between pipe center and panel surface

$\boxed{R_y := 0.16}$ m^2K/W

Resistance between pipe center and downide space

$\boxed{R_a := 0.98}$ m^2K/W

Outside pipe diameter

$\boxed{D_o := 0.02}$ m

Inside pipe diameter

$\boxed{D_i := 0.016}$ m

Conv. heat transfer o the pipe surface.

$\boxed{h_f := 3000}$ W/m^2K

Mean fluid temperature

$\boxed{T_f := 45}$ °C

Fluid inlet temperature

$\boxed{T_{fi} := 50}$ °C

Fluid exit temperature

$\boxed{T_{fo} := 40}$ °C

Specific heat of water

$\boxed{c_{pw} := 4180}$ kJ/kgK

density

$\rho_w := 990$ kg/m^3

$$q_{fmax} := 8.92 \cdot \left(T_{pmax} - T_a\right)^{1.1}$$

$$q_{fmax} = 100 \quad \text{W/m}^2$$

$$m := \sqrt{\frac{h_{ps}}{k_{eq} \cdot t_y}}$$

$$W(S) := \frac{S - D_o}{2}$$

$$F(S) := \frac{\tanh(m \cdot W(S))}{m \cdot W(S)}$$

$$q_{fdes} := 114.83 \quad \text{W/m}^2$$

$$F_1(S) := \left[\frac{S}{D_o + \left(S - D_o\right) \cdot F(S)} + \frac{h_{ps} \cdot S}{\pi \cdot D_i \cdot h_f}\right]^{-1}$$

$$S := 0.1 \quad \text{m}$$

Given

$$q_{fdes} = F_1(S) \cdot h_{ps} \cdot \left(T_f - T_a\right)$$

$$\text{Find}(S) = 0.183 \quad \text{m}$$

$S := 0.25\,m$

$q_f(S) := F_1(S) \cdot h_{ps} \cdot (T_f - T_a)$ $\qquad q_f(S) = 96.5 \quad W/m^2$

$q_f := 121.9 \quad W/m^2$

$$T_p := T_a + \left(\frac{q_f}{8.92}\right)^{\frac{1}{1.1}} \qquad T_p = 30.8 \quad °C$$

$T_b := T_p + q_f \cdot R_y$ $\qquad T_b = 50.3 \quad °C$

$$X := \frac{1}{1 + \frac{R_y \cdot (T_b - T_n)}{R_a \cdot (T_b - T_p)}} \qquad X = 0.782$$

$$q_s := \frac{q_f}{X} \qquad q_s = 155.9 \quad W/m^2$$

$Q_s := q_s \cdot A_p$ $\qquad Q_s = 678 \quad W$

$$m_w := \frac{Q_s}{c_{pw} \cdot (T_{fi} - T_{fo})} \qquad m_w = 0.016 \quad kg/s$$

$$V_w := \frac{m_w}{\rho_w} \cdot 3600 \qquad V_w = 0.059 \quad m^3/h$$

APPENDIX C

SIMULATION OUTPUTS FOR MODELLING OF THE SYSTEM COMPONENTS

Constants of the simulation:

Evaporator capacity	Design heating load	Generator design capacity
$\boxed{Q_e := 10.809}$ kW	$\boxed{Q_{ahpdes} := 25.5}$ kW	$\boxed{Q_{gdes} := 13.17}$ kW
Collector area	Overall heat loss coeff area product for storage tank	Storage tank mass
$\boxed{A_c := 100}$ m^2	$\boxed{UA_s := 0.015}$ kW/K	$\boxed{M_s := 7500}$ kg
Storage tank reference temperature	Specific heat of load	Spesicif heat of water
$\boxed{T_{sref} := 34.5}$ °C	$\boxed{c_{pwL} := 4.18}$ kj/kgK	$\boxed{c_{pws} := 4.18}$ kj/kgK

Input data:

$T_{sold} := 33.0$ °C $\quad T_o := 0.8$ °C $\quad I := 0.428$ kW/m^2

$$E := \frac{25.5 \cdot (18 - T_o)}{29} \qquad E = 15.1 \text{ kW}$$

$Q_{ahp} := Q_{ahpdes}$ $\qquad$ $Q_g := Q_{gdes}$

$$Q_u := A_c \cdot I \cdot \left(0.71 - 0.00244 \cdot \frac{T_{sold} - T_o}{I}\right)$$

$Q_u = 22.5$ kW

$$FWT := \frac{E}{Q_{ahp}}$$

$FWT = 0.593$

$WT := 60 \cdot FWT$

$WT = 35.6$ min

$$T_{snew} := T_{sold} + \frac{3600}{M_s \cdot c_{pws}} \cdot \left[Q_u - FWT \cdot Q_e - UA_s \cdot \left(T_{sold} - T_o\right) \right]$$

$T_{snew} = 34.8$ °C

$$m_L := \frac{Q_e}{4.5 \cdot c_{pwL}}$$

$m_L = 0.5746$ kg/s

$Q_{auxso} := m_L \cdot c_{pwL} \cdot \left(T_{sref} - T_{sold}\right) \cdot FWT$

$$Q_{auxso} := m_L \cdot c_{pwL} \cdot (T_{sref} - T_{sold}) \cdot FWT$$

$Q_{auxso} = 2.1$ kW

$$Q_{auxgo} := Q_g \cdot FWT$$

$Q_{auxgo} = 7.8$ kW

$$Q_{auxto} := Q_{auxso} + Q_{auxgo}$$

$Q_{auxto} = 9.9$ kW

APPENDIX D

SIMULATION OUTPUTS FOR ABSORPTION HEAT PUMP

D.1. Design Simulation

$T_1 := 50$ °C $\quad T_2 := T_1$ $\quad$ $T_{ab}=T_c=50°C$, equal to floor heatin panel design inlet temperature

$T_8 := T_1$

$T_{11} := 28$ °C $\quad T_{10} := T_{11}$ $\quad$ Te= 6.5°C below storage tank reference temperature, $T_{s,ref}=34.5°C$

$T_4 := 82$ °C $\quad T_7 := T_4$ $\quad$ $T_g=82$ C by auxiliary heating

$T_5 := T_2 + 10$ $\quad$ Low-temperature end approach assumed as 10°C

$k_0 := 6.21147 \qquad k_1 := -2886.373 \qquad k_2 := -337269.46$

$$P_L := 10^{\left[k_0+\frac{k_1}{T_{11}\cdot 1.8+491.69}+\frac{k_2}{\left(T_{11}\cdot 1.8+491.69\right)^2}\right]}\cdot(6.89474)$$

$P_L = 3.78 \times 10^0$ kPa $\qquad T_{11} = 28 \times 10^0$

$P_L=P_e=3.782$ kPa

$$P_H := 10^{\left[k_0+\frac{k_1}{T_8\cdot 1.8+491.69}+\frac{k_2}{\left(T_8\cdot 1.8+491.69\right)^2}\right]}\cdot(6.89474)$$

T_8=50°C=581.69°R

$P_H = 12.34\times 10^0$ kPa

P_H=P_c=12.336 kPa

$$a(X) := -2.008 + 0.17\cdot X - 3.133\cdot 10^{-3}\cdot X^2 + 1.977\cdot 10^{-5}\cdot X^3$$

$$b(X) := 178.404 - 10.734\cdot X + 0.208\cdot X^2 - 1.147\cdot 10^{-3}\cdot X^3$$

$T_1 = 50\times 10^0$ °C $\qquad$ $P_L = 3.78\times 10^0$ kPa

$$T_{r1}(X) := \frac{T_1 - b(X)}{a(X)}$$

$X := 70$

Given

$$P_L = 10^{\left[7.050-\left(\frac{1603.5406}{T_{r1}(X)+273.15}\right)-\frac{104095.51}{\left(T_{r1}(X)+273.15\right)^2}\right]}$$

$X_1 := \mathrm{Find}(X) = \left(50\times 10^0\right)$ $\qquad$ $X_2 := X_1$ $\qquad$ $X_3 := X_1$

$$x_1 := \frac{X_1}{100}$$

$x_1 = 499.99\times 10^{-3}$ $\qquad$ $x_2 := x_1$ $\qquad$ $x_3 := x_1$

$T_4 = 82\times 10^0$ °C $\qquad$ $P_H = 12.34\times 10^0$ kPa

$$T_{r4}(X) := \frac{T_4 - b(X)}{a(X)}$$

$X := 40$

Given

$$P_H = 10^{\left[7.050 - \left(\frac{1603.5406}{T_{r4}(X)+273.15}\right) - \frac{104095.51}{\left(T_{r4}(X)+273.15\right)^2}\right]}$$

$X_4 := \mathrm{Find}(X) = \left(53.92 \times 10^0\right)$ $\quad X_5 := X_4 \quad X_6 := X_4$

$$x_4 := \frac{X_4}{100}$$

$x_4 = 0.5392 \quad x_5 := x_4 \quad x_6 := x_4$

$$CR := \frac{x_6}{\left|x_6 - x_1\right|}$$

$CR = 13.759$

$X_7 := 0 \quad X_8 := 0 \quad X_9 := 0 \quad X_{10} := 0 \quad X_{11} := 0 \quad X_{12} := 0$

$A_{0,0} := 1.134125 \quad A_{1,0} := -4.800450 \cdot 10^{-1} \quad A_{2,0} := -2.161438 \cdot 10^{-3}$

$A_{3,0} := 2.336235 \cdot 10^{-4} \quad A_{4,0} := -1.188679 \cdot 10^{-5} \quad A_{5,0} := 2.291532 \cdot 10^{-7}$

$A_{0,1} := 4.124891 \quad A_{1,1} := -7.643903 \cdot 10^{-2} \quad A_{2,1} := 2.589577 \cdot 10^{-3}$

$A_{3,1} := -9.500522 \cdot 10^{-5} \quad A_{4,1} := 1.708026 \cdot 10^{-6} \quad A_{5,1} := -1.102363 \cdot 10^{-8}$

$A_{0,2} := 5.743693 \cdot 10^{-4} \quad A_{1,2} := 5.870921 \cdot 10^{-5} \quad A_{2,2} := -7.375319 \cdot 10^{-6}$

$A_{3,2} := 3.277592 \cdot 10^{-7} \quad A_{4,2} := -6.062304 \cdot 10^{-9} \quad A_{5,2} := 3.901897 \cdot 10^{-11}$

$B_{0,0} := 5.127558 \cdot 10^{-1}$ $B_{0,1} := -1.3939 \cdot 10^{-2}$ $B_{0,2} := 2.92414 \cdot 10^{-5}$

$B_{0,3} := 9.03 \cdot 10^{-7}$ $B_{1,0} := 1.227 \cdot 10^{-2}$ $B_{1,1} := -9.156 \cdot 10^{-5}$

$B_{1,2} := 1.821 \cdot 10^{-8}$ $B_{1,3} := -7.99 \cdot 10^{-10}$ $B_{2,0} := -1.36489 \cdot 10^{-5}$

$B_{2,1} := 1.0689 \cdot 10^{-7}$ $B_{2,2} := -1.381 \cdot 10^{-9}$ $B_{2,3} := 1.529 \cdot 10^{-11}$

$B_{3,0} := 1.021 \cdot 10^{-8}$ $B_{3,1} := 0$ $B_{3,2} := 0$

$B_{3,3} := 0$

$$h_1(X_1, T_1) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_1^{\,i} \cdot T_1^{\,j}$$

$$h_1(X_1, T_1) = 105.69 \times 10^0$$

$h_1 := h_1(X_1, T_1)$ $h_1 = 105.69 \times 10^0$ kJ/kg

$$X_2 = 50 \times 10^0$$

Given

$$P_H = 10^{\left[7.3 - \left(\frac{1603.5406}{\frac{T_2 - b(X_2)}{a(X_2)} + 273.15}\right) - \frac{104095.51}{\left(\frac{T_2 - b(X_2)}{a(X_2)} + 273.15\right)^2}\right]}$$

$T_2 := \text{Find}(T_2) = (62.26 \times 10^0)$ $T_2 = 62.26 \times 10^0$

$$s_1(X_1,T_1) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_1^{\,j} \cdot T_1^{\,i}$$

$s_1 := s_1(X_1,T_1)$ $\qquad$ $s_1 = 360.34 \times 10^{-3}$ kJ/kg K

$$s_2(X_2,T_2) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_2^{\,j} \cdot T_2^{\,i}$$

$s_2 := s_2(X_2,T_2)$ $\qquad$ $s_2 = 441.62 \times 10^{-3}$

$$h_2(X_2,T_2) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_2^{\,i} \cdot T_2^{\,j}$$

$h_2 := h_2(X_2,T_2)$ $\qquad$ $h_2 = 132.3 \times 10^{0}$

$$h_4(X_4,T_4) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_4^{\,i} \cdot T_4^{\,j}$$

$h_4(X_4,T_4) = 179.28 \times 10^{0}$

$h_4 := h_4(X_4,T_4)$ $\qquad$ $h_4 = 179.28 \times 10^{0}$ kJ/kg

$$s_4(X_4,T_4) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_4^{\,j} \cdot T_4^{\,i}$$

$s_4 := s_4(X_4,T_4)$ $\qquad$ $s_4 = 524.88 \times 10^{-3}$ kJ/kg K

$$h_8(X_8,T_8) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_8^{\,i} \cdot T_8^{\,j} \qquad h_8(X_8,T_8) = 208.81 \times 10^{0}$$

$h_8 := h_8(X_8, T_8) \qquad h_8 = 208.81 \times 10^0$ kJ/kg (hf at T8)

$T_8 := 50$

$$s_8(X_8, T_8) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_8^{\,j} \cdot T_8^{\,i}$$

$s_8 := s_8(X_8, T_8) \qquad s_8 = 1.09 \times 10^0$ kJ/kg K

$$h_{11} := \left[\begin{array}{l} \left[0.00274 \cdot (1.8 \cdot T_{11} + 32) - 0.989805\right] \cdot \dfrac{P_L}{6.89474} \;... \\ + 0.44942 \cdot (1.8 \cdot T_{11} + 32) + 1060.8 \end{array} \right] \cdot 2.326$$

$h_{11} = 2.55 \times 10^3$ kJ/kg (h_g at T_{11})

$$S_{11}(X_{11}, T_{11}) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_{11}^{\,j} \cdot T_{11}^{\,i}$$

$S_{11} := S_{11}(X_{11}, T_{11}) = (845.84 \times 10^{-3})$

$S_{11} = 845.84 \times 10^{-3}$

$$h_7 := \left[\begin{array}{l} \left[0.00274 \cdot (1.8 \cdot T_7 + 32) - 0.989805\right] \cdot \dfrac{P_H}{6.89474} \;... \\ + 0.44942 \cdot (1.8 \cdot T_7 + 32) + 1060.8 \end{array} \right] \cdot 2.326$$

$h_7 = 2653.095$ kJ/kg (at T_7 and P_H)

$$S_7(X_7, T_7) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_7^{\,j} \cdot T_7^{\,i}$$

$S_7 := S_7(X_7, T_7) = (1.43 \times 10^0)$

$S_7 = 1.43 \times 10^0$

Refrigerant heat exchanger:

$C_{cr} := 2.0$ kW/K $\quad C_{hr} := 4.18$ kW/K $\quad \varepsilon_r := 0.7$

$C_{minr} := C_{cr} \quad C_{maxr} := C_{hr} \quad C_{rr} := \dfrac{C_{minr}}{C_{maxr}}$

$\dfrac{C_{minr}}{C_{hr}} = 478.47 \times 10^{-3} \quad \dfrac{C_{hr}}{C_{cr}} = 2.09 \times 10^0 \quad C_{rr} = 478.47 \times 10^{-3}$ kW/K

$$T_9 := T_8 - \frac{C_{minr}}{C_{hr}} \cdot \varepsilon_r \cdot \left(T_8 - T_{11}\right)$$

$T_9 = 42.63 \times 10^0$ °C

$$h_9\left(X_9, T_9\right) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_9{}^i \cdot T_9{}^j$$

$h_9\left(X_9, T_9\right) = 178.03 \times 10^0$

$h_9 := h_9\left(X_9, T_9\right) \quad h_9 = 178.03 \times 10^0$ kJ/kg $\quad$ (h_f at T_9)

$h_{10} := h_9 \quad h_{10} = 178.03 \times 10^0$ kJ/kg

$$s_9\left(X_9, T_9\right) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_9{}^j \cdot T_9{}^i$$

$s_9 := s_9\left(X_9, T_9\right)$

$s_9 = 1.01 \times 10^0$ kJ/kg K

$$s_{10}(X_{10}, T_{10}) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_{10}^{\ j} \cdot T_{10}^{\ i}$$

$s_{10}(X_{10}, T_{10}) = 845.84 \times 10^{-3}$ kJ/kg K

$s_{10} := s_{10}(X_{10}, T_{10})$

$$UA_r := \frac{\ln(1 - \varepsilon_r \cdot C_{rr}) - \ln(1 - \varepsilon_r)}{1 - C_{rr}} \cdot C_{minr}$$

$UA_r = 3.05 \times 10^0$ kW/K

$$T_{12} := T_{11} + \frac{C_{hr}}{C_{cr}} \cdot (T_8 - T_9)$$

$T_{12} = 43.4$ °C

$h_{12} := h_8 + h_{11} - h_9$

$h_{12} = 2583.369$ kJ/kg

$$s_{12}(X_{12}, T_{12}) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_{12}^{\ j} \cdot T_{12}^{\ i}$$

$s_{12}(X_{12}, T_{12}) = 1.02 \times 10^0$

$s_{12} := s_{12}(X_{12}, T_{12})$ $s_{12} = 1.02 \times 10^0$

Mixture heat exchanger:

$T_5 := T_2 + 10$ (Low-temperature end approach assumed as 10°C

$T_5 = 72.26 \times 10^0$ °C

$T_6 := 75$ °C

$$h_5(X_5, T_5) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_5{}^{i} \cdot T_5{}^{j}$$

$h_5(X_5, T_5) = 158.97 \times 10^0$

$h_5 := h_5(X_5, T_5)$ $h_5 = 158.97 \times 10^0$ kJ/kg

$h_6 := h_5$ $h_6 = 158.97 \times 10^0$ kJ/kg

$$s_6(X_6, T_6) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_6{}^{j} \cdot T_6{}^{i}$$

$s_6 := s_6(X_6, T_6)$ $s_6 = 483.17 \times 10^{-3}$ kJ/kg K

$$s_5(X_5, T_5) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_5{}^{j} \cdot T_5{}^{i}$$

$s_5 := s_5(X_5, T_5)$ $s_5 = 466.66 \times 10^{-3}$

$Q_{ahp} := 25.5$ kW (design condition) kJ/kg K

$$m_{12} := \frac{Q_{ahp}}{\left[(CR-1)\cdot h_6 + h_{12} - CR\cdot h_1 + h_7 - h_8\right]}$$

$m_{12} = 4.55 \times 10^{-3}$ kg/s

$m_7 := m_{12}$ $m_8 := m_{12}$ $m_9 := m_{12}$ $m_{10} := m_{12}$ $m_{11} := m_{12}$

$m_7 = m_8 = m_9 = m_{10} = m_{11} = m_{12} = 0.00402$ kg/s

$m_1 := CR\cdot m_{12}$

$m_1 = 6.2632 \times 10^{-2}$ kg/s

$m_2 := m_1$ $m_3 := m_1$

$m_3 = m_2 = m_1 = 0.05868$ kg/s

$m_6 := m_1 - m_{12}$

$m_6 = 0.05808$ kg/s

$m_4 := m_6$ $m_5 := m_6$

$m_4=m_5=m_6=0.05441$ kg/s

$x_w := x_4$ $x_w = 0.5392$

$c_{phm} := -3.09\cdot x_w + 4.18$

$c_{phm} = 2.5139$ kJ/kgK

$x_r := x_1$ $x_r = 499.99 \times 10^{-3}$

$c_{pcm} := -3.09 \cdot x_r + 4.18$

$c_{pcm} = 2.635$ kJ/kgK

$C_{hm} := m_6 \cdot c_{phm}$

$C_{hm} = 0.146$ kW/K

$C_{minm} := C_{hm}$

$C_{cm} := m_1 \cdot c_{pcm}$

$C_{cm} = 0.165$ kW/K

$C_{maxm} := C_{cm}$

$$T_3 := T_2 + \frac{C_{hm}}{C_{cm}} \cdot (T_4 - T_5)$$

$T_3 = 70.88 \times 10^0$ °C

$$h_3 := \frac{m_2 \cdot h_2 + m_4 \cdot h_4 - m_5 \cdot h_5}{m_3}$$

$h_3 := 148.848$ kJ/kg

$$s_3(X_3, T_3) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_3{}^j \cdot T_3{}^i$$

$s_3 := s_3(X_3, T_3)$

$s_3 = 497.31 \times 10^{-3}$ kJ/kg K

$$C_{rm} := \frac{C_{minm}}{C_{maxm}}$$

$C_{rm} = 884.71 \times 10^{-3}$ kW/K

$$\varepsilon_m := \frac{C_{hm} \cdot (T_4 - T_5)}{C_{minm} \cdot (T_4 - T_2)}$$

$\varepsilon_m = 0.4934$

$$UA_m := \frac{\ln(1 - \varepsilon_m \cdot C_{rm}) - \ln(1 - \varepsilon_m)}{1 - C_{rm}} \cdot C_{minn}$$

$UA_m = 134.77 \times 10^{-3}$ kW/K

$$Q_a := m_6 \cdot h_6 + m_{12} \cdot h_{12} - m_1 \cdot h_1$$

$Q_a = 14.37 \times 10^{0}$ kW

$$Q_c := m_7 \cdot (h_7 - h_8)$$

$Q_c = 11.127$ kW

$$Q_g := m_4 \cdot h_4 + m_7 \cdot h_7 - m_3 \cdot h_3$$

$Q_g = 13.17 \times 10^{0}$ kW

$$Q_e := m_{11} \cdot (h_{11} - h_{10})$$

$Q_e = 10.809$ kW

$Q_a + Q_c = 25.5 \times 10^0$ kW $\qquad Q_g + Q_e = 23.98 \times 10^0$ kW

$T_0 := 298.15$

$h_0 := 104.7$

$s_0 := 0.3679$

$W_p := m_1 \cdot (h_2 - h_1)$

$W_p = 1.67 \times 10^0$

$\Delta\psi_{pp} := m_1 \cdot \left[(h_1 - h_2) - T_0 \cdot (s_1 - s_0) \right] + W_p$

$\Delta\psi_{pp} = 141.17 \times 10^{-3}$

$T_c := 323.15$ K

$$\Delta\psi_c := m_7 \cdot \left[(h_7 - h_8) - T_0 \cdot (S_7 - s_8) \right] - Q_c \cdot \left(1 - \frac{T_0}{T_c} \right)$$

$\Delta\psi_c = 9.81 \times 10^0$ kW

$T_e := 301.15$ K

$$\Delta\psi_{eva.} := m_{10} \cdot \left[(h_{11} - h_{10}) - T_0 \cdot (S_{11} - s_{10}) \right] - Q_e \cdot \left(1 - \frac{T_0}{T_e} \right)$$

$\Delta\psi_{eva.} = 10.7 \times 10^0$ kW

$T_a := 323.15 \quad K$

$$\Delta\psi_{abs.} := m_{11}\cdot\left[\left(h_{12} - h_0\right) - T_0\cdot\left(s_{12} - s_0\right)\right] + m_6\cdot\left[\left(h_6 - h_0\right) - T_0\cdot\left(s_6 - s_0\right)\right] \ldots$$
$$+ -\left[m_1\cdot\left[\left(h_1 - h_0\right) - T_0\cdot\left(s_1 - s_0\right)\right]\right] - Q_a\cdot\left(1 - \frac{T_0}{T_a}\right)$$

$\Delta\psi_{abs.} = 10.24 \times 10^0 \quad kW$

$T_g := 355.15 \quad K$

$$\Delta\psi_{gen.} := m_4\cdot\left[\left(h_4 - h_0\right) - T_0\cdot\left(s_4 - s_0\right)\right] + m_7\cdot\left[\left(h_7 - h_0\right) - T_0\cdot\left(S_7 - s_0\right)\right] \ldots$$
$$+ -\left[m_3\cdot\left[\left(h_3 - h_0\right) - T_0\cdot\left(s_3 - s_0\right)\right]\right] - Q_g\cdot\left(1 - \frac{T_0}{T_g}\right)$$

$\Delta\psi_{gen.} = 9.31 \times 10^0$

$$\Delta\psi_{s.h.e.} := m_2\cdot\left[\left(h_2 - h_3\right) - T_0\cdot\left(s_2 - s_3\right)\right] + m_4\cdot\left[\left(h_4 - h_5\right) - T_0\cdot\left(s_4 - s_5\right)\right]$$

$\Delta\psi_{s.h.e.} = 174.94 \times 10^{-3}$

$$\Delta\psi_{r.h.e.} := m_{11}\cdot\left[\left(h_{11} - h_{12}\right) - T_0\cdot\left(S_{11} - s_{12}\right)\right] + m_8\cdot\left[\left(h_8 - h_9\right) - T_0\cdot\left(s_8 - s_9\right)\right]$$

$\Delta\psi_{r.h.e.} = 126.2 \times 10^{-3}$

$$\Delta\psi_{tot} := \Delta\psi_c + \Delta\psi_{eva.} + \Delta\psi_{abs.} + \Delta\psi_{gen.} + \Delta\psi_{pp} + \Delta\psi_{s.h.e.} + \Delta\psi_{r.h.e.}$$

$\Delta\psi_{tot} = 40.49 \times 10^0$

$$\frac{\Delta\psi_{c}}{\Delta\psi_{tot}} = 0.2421 \qquad \frac{\Delta\psi_{eva.}}{\Delta\psi_{tot}} = 0.2643$$

$$\frac{\Delta\psi_{pp}}{\Delta\psi_{tot}} = 3.49 \times 10^{-3} \qquad \frac{\Delta\psi_{gen.}}{\Delta\psi_{tot}} = 0.2298$$

$$\frac{\Delta\psi_{abs.}}{\Delta\psi_{tot}} = 0.2528$$

$T_c := 323 \qquad T_a := 323$

$T_g := 355.15 \qquad T_o := 298$

$$COP_h := \frac{Q_a + Q_c}{Q_g + W_p} \qquad COP_h = 1.719$$

$$ECOP := \frac{\left[Q_c \cdot \left(1 - \frac{T_o}{T_c}\right) + Q_a \cdot \left(1 - \frac{T_o}{T_a}\right)\right]}{Q_g \cdot \left(1 - \frac{T_o}{T_g}\right) + W_p} \qquad ECOP = 521.39 \times 10^{-3}$$

D.2. Design Simulation for Cases of T_s Greater Than $T_{s,ref}$

Evaporator heating capacity

$\boxed{Q_{edes} := 10.809}$ kW

Design temerature of storage tai

$\boxed{T_{sdes} := 34.5}$ °C

Design temerature of evaporator

$\boxed{T_{10des} := 28}$ °C

Specific heat of water

$\boxed{c_{pw} := 4.18}$ kj/kgK

$$m_L := \frac{Q_{edes}}{4.5 \cdot c_{pw}} \qquad m_L = 0.5746 \quad \text{kg/s}$$

$$C_{mine} := \frac{Q_{edes}}{4.5} \qquad C_{mine} = 2.402 \quad \text{kW/K}$$

$$\varepsilon_e := \frac{Q_{edes}}{C_{mine} \cdot (T_{sdes} - T_{10des})} \qquad \varepsilon_e = 0.692$$

$$NTU_e := -\ln(1 - \varepsilon_e) \qquad NTU_e = 1.179$$

$$UA_e := NTU_e \cdot C_{mine} \qquad UA_e = 2.831 \quad \text{kW/K}$$

$Q_e := Q_{edes}$ $\qquad T_{sold} := 39.2$ °C

$$T_{10} := T_{sold} - \frac{Q_e}{\varepsilon_e \cdot C_{mine}} \qquad T_{10} = 32.7 \quad {}^\circ\text{C}$$

$T_{11} := T_{10}$ $\qquad\qquad$ $T_{11} = 32.7$ °C

$k_0 := 6.21147 \qquad k_1 := -2886.373 \qquad k_2 := -337269.46$

$$P_L := 10^{\left[k_0 + \frac{k_1}{T_{11}\cdot 1.8 + 491.69} + \frac{k_2}{\left(T_{11}\cdot 1.8 + 491.69\right)^2}\right]}\cdot(6.89474)$$

$P_L = 4.95$ kPa

$$h_{11} := \left[\left[0.00274\cdot\left(1.8\cdot T_{11} + 32\right) - 0.989805\right]\cdot\frac{P_L}{6.89474} \;...\; + 0.44942\cdot\left(1.8\cdot T_{11} + 32\right) + 1060.8\right]\cdot 2.326$$

$h_{11} = 2561.164$ kJ/kg $\qquad$ (h_g at T11)

$s_{11} := 8.40$ kJ/kg $\qquad$ (s_g at T11)

$T_8 := 50$ °C

$$P_H := 10^{\left[k_0 + \frac{k_1}{T_8\cdot 1.8 + 491.69} + \frac{k_2}{\left(T_8\cdot 1.8 + 491.69\right)^2}\right]}\cdot(6.89474)$$

$P_H = 12.336$ kPa

$X_7 := 0 \quad X_8 := 0 \quad X_9 := 0 \quad X_{10} := 0 \quad X_{11} := 0 \quad X_{12} := 0$

$A_{0,0} := 1.134125 \quad A_{1,0} := -4.800450\cdot10^{-1} \quad A_{2,0} := -2.161438\cdot10^{-3}$

$A_{3,0} := 2.336235\cdot10^{-4} \quad A_{4,0} := -1.188679\cdot10^{-5} \quad A_{5,0} := 2.291532\cdot10^{-7}$

$A_{0,1} := 4.124891 \quad A_{1,1} := -7.643903\cdot10^{-2} \quad A_{2,1} := 2.589577\cdot10^{-3}$

$A_{3,1} := -9.500522\cdot10^{-5} \quad A_{4,1} := 1.708026\cdot10^{-6} \quad A_{5,1} := -1.102363\cdot10^{-8}$

$A_{0,2} := 5.743693\cdot10^{-4} \quad A_{1,2} := 5.870921\cdot10^{-5} \quad A_{2,2} := -7.375319\cdot10^{-6}$

$A_{3,2} := 3.277592\cdot10^{-7} \quad A_{4,2} := -6.062304\cdot10^{-9} \quad A_{5,2} := 3.901897\cdot10^{-11}$

$B_{0,0} := 5.127558\cdot10^{-1} \quad B_{0,1} := -1.3939\cdot10^{-2} \quad B_{0,2} := 2.92414\cdot10^{-5}$

$B_{0,3} := 9.03\cdot10^{-7} \quad B_{1,0} := 1.227\cdot10^{-2} \quad B_{1,1} := -9.156\cdot10^{-5}$

$B_{1,2} := 1.821\cdot10^{-8} \quad B_{1,3} := -7.99\cdot10^{-10} \quad B_{2,0} := -1.36489\cdot10^{-5}$

$B_{2,1} := 1.0689\cdot10^{-7} \quad B_{2,2} := -1.381\cdot10^{-9} \quad B_{2,3} := 1.529\cdot10^{-11}$

$B_{3,0} := 1.021\cdot10^{-8} \quad B_{3,1} := 0 \quad B_{3,2} := 0$

$B_{3,3} := 0$

$C_{cr} := 2 \quad C_{hr} := 4.18$

$C_{minr} := C_{cr} \quad C_{maxr} := C_{hr} \quad \frac{C_{minr}}{C_{hr}} = 0.478$

$UA_r := 3.053$ kW/K $\qquad C_{rr} := \dfrac{C_{minr}}{C_{maxr}} \qquad C_{rr} = 0.478$

$$NTU_r := \frac{UA_r}{C_{minr}} \qquad NTU_r = 1.526$$

$$\varepsilon_r := \frac{1 - \exp\left[-NTU_r \cdot \left(1 - C_{rr}\right)\right]}{1 - C_{rr} \cdot \exp\left[-NTU_r \cdot \left(1 - C_{rr}\right)\right]} \qquad \varepsilon_r = 0.7$$

$$T_9 := T_8 - \frac{C_{minr}}{C_{hr}} \cdot \varepsilon_r \cdot \left(T_8 - T_{11}\right)$$

$T_9 = 44.206$ °C

$$h_9\left(X_9, T_9\right) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_9^{\,i} \cdot T_9^{\,j}$$

$h_9\left(X_9, T_9\right) = 184.6$

$h_9 := h_9\left(X_9, T_9\right) \qquad h_9 = 184.6$ kJ/kg $\qquad$ (h_f at T9)

$h_{10} := h_9 \qquad h_{10} = 184.6$ kJ/kg

$$s_9\left(X_9, T_9\right) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_9^{\,j} \cdot T_9^{\,i}$$

$s_9 := s_9\left(X_9, T_9\right) \qquad s_9 = 1.029$ kJ/kg K

$$s_{10}\left(X_{10}, T_{10}\right) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_{10}^{\,j} \cdot T_{10}^{\,i}$$

$s_{10} := s_{10}(X_{10}, T_{10}) \qquad s_{10} = 0.9$

$m_{11} := \dfrac{Q_e}{h_{11} - h_{10}} \qquad m_{11} = 4.55 \times 10^{-3}$ kg/s

$m_7 := m_{11} \qquad m_8 := m_{11} \qquad m_9 := m_{11}$

$m_{10} := m_{11} \qquad m_{12} := m_{11}$

$m_1 := 0.05868$ kg/s

$m_2 := m_1 \qquad m_3 := m_1$

$m_6 := m_1 - m_{12}$

$m_6 = 0.05413$ kg/s

$m_4 := m_6 \qquad m_5 := m_6$

$CR := \dfrac{m_1}{m_{12}} \qquad CR = 12.902$

$T_1 := 50$ °C

$a(X) := -2.008 + 0.17 \cdot X - 3.133 \cdot 10^{-3} \cdot X^2 + 1.977 \cdot 10^{-5} \cdot X^3$

$b(X) := 178.404 - 10.734 \cdot X + 0.208 \cdot X^2 - 1.147 \cdot 10^{-3} \cdot X^3$

$P_L = 4.95$ kPa

$T_{r1}(X) := \dfrac{T_1 - b(X)}{a(X)}$

$X := 60$

Given

$$P_L = 10^{\left[7.050-\left(\frac{1603.5406}{T_{r1}(X)+273.15}\right)-\frac{104095.51}{\left(T_{r1}(X)+273.15\right)^2}\right]}$$

$\mathrm{Find}(X) = 46.49$

$X_1 := 45.92$ $\quad X_2 := X_1$ $\quad X_3 := X_1$

$$x_1 := \frac{X_1}{100}$$

$x_1 = 0.4592$ $\quad x_2 := x_1$ $\quad x_3 := x_1$

$x_6 := x_1 \cdot \frac{CR}{CR - 1}$ $\quad x_6 = 0.4978$

$x_4 := x_6$ $\quad x_5 := x_6$

$X_6 := x_6 \cdot 100$ $\quad X_4 := X_6$ $\quad X_5 := X_6$

$X_6 = 49.78$ $\quad X_4 = 49.78$

$P_H = 12.336$ kPa

$T_{r4} := 30$ °C

Given

$$P_H = 10^{\left[7.050-\left(\frac{1603.5406}{T_{r4}+273.15}\right)-\frac{104095.51}{\left(T_{r4}+273.15\right)^2}\right]}$$

$\mathrm{Find}(T_{r4}) = 50.011$ $\quad T_{r4} := 50$

$$T_4 := a(X_4)\cdot T_{r4} + b(X_4)$$

$T_4 = 74.488$ °C

$$T_7 := T_4$$

$T_7 = 74.488$ °C

$$h_1(X_1, T_1) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j}\cdot X_1^{\,i}\cdot T_1^{\,j}$$

$$h_1(X_1, T_1) = 103.246$$

$h_1 := h_1(X_1, T_1)$ $\quad\quad$ $h_1 = 103.246$ kJ/kg

$$T_2 := 55$$

$$X_2 := 45.92$$

Given

$$P_H = 10^{\left[7.350 - \left(\frac{1603.5406}{\frac{T_2 - b(X_2)}{a(X_2)} + 273.15}\right) - \frac{104095.51}{\left(\frac{T_2 - b(X_2)}{a(X_2)} + 273.15\right)^2}\right]}$$

$$\mathrm{Find}(T_2) = 53.666$$

$$h_2(X_2, T_2) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j}\cdot X_2^{\,i}\cdot T_2^{\,j}$$

$$h_2(X_2, T_2) = 114.59$$

$$h_2 := h_2(X_2, T_2) \qquad h_2 = 114.59 \qquad \text{kJ/kg}$$

$$s_1(X_1, T_1) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_1^{\,j} \cdot T_1^{\,i}$$

$$s_1(X_1, T_1) = 0.399$$

$$s_1 := s_1(X_1, T_1) \qquad s_1 = 0.399 \qquad \text{kJ/kg K}$$

$$s_2(X_2, T_2) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_2^{\,j} \cdot T_2^{\,i}$$

$$s_2(X_2, T_2) = 0.434$$

$$s_2 := s_2(X_2, T_2) \qquad s_2 = 0.434 \qquad \text{kJ/kg K}$$

$$h_4(X_4, T_4) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_4^{\,i} \cdot T_4^{\,j}$$

$$h_4(X_4, T_4) = 158.82$$

$$h_4 := h_4(X_4, T_4) \qquad h_4 = 158.82 \qquad \text{kJ/kg}$$

$$s_4(X_4, T_4) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_4^{\,j} \cdot T_4^{\,i}$$

$$s_4 := s_4(X_4, T_4) \qquad s_4 = 0.523 \qquad \text{kJ/kg K}$$

$$h_8(X_8, T_8) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_8^{\,i} \cdot T_8^{\,j} \qquad \text{kJ/kg}$$

$$h_8(X_8, T_8) = 208.815$$

$h_8 := h_8(X_8, T_8)$ $\qquad$ $h_8 = 208.815$ kJ/kg $\qquad$ (h_f at T8)

$$s_8(X_8, T_8) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_8^{\,j} \cdot T_8^{\,i}$$

$s_8 := s_8(X_8, T_8)$ $\qquad$ $s_8 = 1.093$ $\qquad$ kJ/kg K

$$h_7 := \left[\left[0.00274 \cdot (1.8 \cdot T_7 + 32) - 0.989805\right] \cdot \frac{P_H}{6.89474} \dots + 0.44942 \cdot (1.8 \cdot T_7 + 32) + 1060.8 \right] \cdot 2.326$$

$h_7 = 2638.806$ kJ/kg $\qquad$ (at T_7 and P_H)

$$s_7(X_7, T_7) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_7^{\,j} \cdot T_7^{\,i}$$

$s_7 := s_7(X_7, T_7)$ $\qquad$ $s_7 = 1.355$ $\qquad$ kJ/kg K

$T_{12} := T_{11} + \frac{C_{hr}}{C_{cr}} \cdot (T_8 - T_9)$ $\qquad$ $T_{12} = 44.81$ °C

$h_{12} := h_8 + h_{11} - h_9$ $\qquad$ $h_{12} = 2585.379$ kJ/kg

$$s_{12}(X_{12}, T_{12}) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_{12}^{\,j} \cdot T_{12}^{\,i}$$

$s_{12}(X_{12}, T_{12}) = 1.036$

$s_{12} := s_{12}(X_{12}, T_{12})$ $\quad s_{12} = 1.036$ kJ/kg K

$x_w := x_4$ $\quad x_w = 0.4978$

$c_{phm} := -3.09 \cdot x_w + 4.18$ $\quad c_{phm} = 2.6419$ kJ/kgK

$x_r := x_1$ $\quad x_r = 0.4592$

$c_{pcm} := -3.09 \cdot x_r + 4.18$ $\quad c_{pcm} = 2.7611$ kJ/kgK

$C_{hm} := m_6 \cdot c_{phm}$ $\quad C_{hm} = 0.143$ kW/K

$C_{minm} := C_{hm}$ $\quad C_{minm} = 0.143$ kW/K

$C_{cm} := m_1 \cdot c_{pcm}$ $\quad C_{cm} = 0.162$ kW/K

$C_{maxm} := C_{cm}$ $\quad C_{maxm} = 0.162$ kW/K

$UA_m := 0.268$ kW/K $\quad C_{rm} := \dfrac{C_{minm}}{C_{maxm}}$ $\quad C_{rm} = 0.883$

$NTU_m := \dfrac{UA_m}{C_{minm}}$ $\quad NTU_m = 1.874$

$$\varepsilon_m := \frac{1 - \exp\left[-NTU_m \cdot \left(1 - C_{rm}\right)\right]}{1 - C_{rm} \cdot \exp\left[-NTU_m \cdot \left(1 - C_{rm}\right)\right]} \qquad \varepsilon_m = 0.677$$

$T_2 = 55$ °C

$$T_5 := T_4 - \varepsilon_m \cdot \frac{C_{minm}}{C_{hm}} \cdot \left(T_4 - T_2\right) \qquad T_5 = 61.295 \text{ °C}$$

$T_6 := 65$ °C

$$h_5\left(X_5, T_5\right) := \sum_{i=0}^{5} \sum_{j=0}^{2} A_{i,j} \cdot X_5{}^{i} \cdot T_5{}^{j}$$

$h_5\left(X_5, T_5\right) = 130.006$

$h_5 := h_5\left(X_5, T_5\right)$ $\qquad h_5 = 130.006$ kJ/kg

$h_6 := h_5$ $\qquad h_6 = 130.006$ kJ/kg

$$s_5\left(X_5, T_5\right) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_5{}^{j} \cdot T_5{}^{i}$$

$s_5 := s_5\left(X_5, T_5\right)$ $\qquad s_5 = 0.437$ kJ/kg K

$$s_6\left(X_6, T_6\right) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_6{}^{j} \cdot T_6{}^{i} \qquad \text{kJ/kgK}$$

$s_6 := s_6\left(X_6, T_6\right)$ $\qquad s_6 = 0.462$

$$T_3 := T_2 + \frac{C_{hm}}{C_{cm}} \cdot (T_4 - T_5)$$

$T_3 = 66.645$ °C

$$h_3 := \frac{m_2 \cdot h_2 + m_4 \cdot h_4 - m_5 \cdot h_5}{m_3}$$

$h_3 = 141.17$ kJ/kg

$$s_3(X_3, T_3) := \sum_{i=0}^{3} \sum_{j=0}^{3} B_{i,j} \cdot X_3{}^j \cdot T_3{}^i$$

$s_3(X_3, T_3) = 0.515$ $\qquad s_3 := s_3(X_3, T_3)$ $\qquad s_3 = 0.515$ kJ/kg K

$Q_{ades} := 14.37$ kW $\qquad Q_{cdes} := 11.127$ kW

$Q_{gdes} := 13.17$ kW $\qquad Q_{edes} := 10.809$ kW

$$Q_a := m_6 \cdot h_6 + m_{12} \cdot h_{12} - m_1 \cdot h_1$$

$Q_a = 12.738$ kW

$$Q_c := m_7 \cdot (h_7 - h_8)$$

$Q_c = 11.052$ kW

$$Q_g := m_4 \cdot h_4 + m_7 \cdot h_7 - m_3 \cdot h_3$$

$Q_g = 12.315$ kW

$$Q_e := m_{11} \cdot (h_{11} - h_{10})$$

$Q_e = 10.809$ kW

$Q_{ahp} := Q_a + Q_c$ $\qquad Q_{ahp} = 23.79$ $\qquad \frac{Q_{ahp}}{25.5} = 0.933$

$$m_{fc} := \frac{Q_{cdes}}{c_{pw} \cdot 10}$$ $m_{fc} = 0.266$ kg/s

$$m_{fa} := \frac{Q_{ades}}{c_{pw} \cdot 10}$$ $m_{fa} = 0.344$ kg/s

$T_{17} := 40$ °C

$$T_{18} := T_{17} + \frac{Q_c}{m_{fc} \cdot c_{pw}}$$ $T_{18} = 49.933$ °C

$T_{15} := 40$ °C

$$T_{16} := T_{15} + \frac{Q_a}{m_{fa} \cdot c_{pw}}$$ $T_{16} = 48.864$ °C

$T_0 := 298.15$

$h_0 := 104.7$

$s_0 := 0.3674$

$$W_p := m_1 \cdot (h_2 - h_1)$$

$W_p = 0.666$ $\quad$ $s_1 := 0.399$

$$\Delta\psi_{pp} := m_1 \cdot \left[(h_1 - h_2) - T_0 \cdot (s_1 - s_2) \right] + W_p$$

$\Delta\psi_{pp} = 0.615$

$T_c := 323.15$ K

$$\Delta\psi_c := m_7 \cdot \left[(h_7 - h_8) - T_0 \cdot (s_7 - s_8) \right] - Q_c \cdot \left(1 - \frac{T_0}{T_c} \right)$$

$\Delta\psi_c = 9.842$ kW

$T_e := 301.15$ K

$$\Delta\psi_{eva.} := m_{10}\cdot\left[\left(h_{11} - h_{10}\right) - T_0\cdot\left(s_{11} - s_{10}\right)\right] - Q_e\cdot\left(1 - \frac{T_0}{T_e}\right)$$

$\Delta\psi_{eva.} = 0.531$ kW

$T_a := 323.15$ K

$$\Delta\psi_{abs.} := m_{11}\cdot\left[\left(h_{12} - h_0\right) - T_0\cdot\left(s_{12} - s_0\right)\right] + m_6\cdot\left[\left(h_6 - h_0\right) - T_0\cdot\left(s_6 - s_0\right)\right] \cdots$$
$$+ -\left[m_1\cdot\left[\left(h_1 - h_0\right) - T_0\cdot\left(s_1 - s_0\right)\right]\right] - Q_a\cdot\left(1 - \frac{T_0}{T_a}\right)$$

$\Delta\psi_{abs.} = 9.877$ kW

$T_g := 378.15$ K

$$\Delta\psi_{gen.} := m_4\cdot\left[\left(h_4 - h_0\right) - T_0\cdot\left(s_4 - s_0\right)\right] + m_7\cdot\left[\left(h_7 - h_0\right) - T_0\cdot\left(s_7 - s_0\right)\right] \cdots$$
$$+ -\left[m_3\cdot\left[\left(h_3 - h_0\right) - T_0\cdot\left(s_3 - s_0\right)\right]\right] - Q_g\cdot\left(1 - \frac{T_0}{T_g}\right)$$

$\Delta\psi_{gen.} = 8.438$

$$\Delta\psi_{tot} := \Delta\psi_c + \Delta\psi_{eva.} + \Delta\psi_{abs.} + \Delta\psi_{gen.} + \Delta\psi_{pp}$$

$\Delta\psi_{tot} = 29.302$

$\frac{\Delta\psi_c}{\Delta\psi_{tot}} = 0.3359$ $\frac{\Delta\psi_{eva.}}{\Delta\psi_{tot}} = 0.0181$

$$\frac{\Delta\psi_{pp}}{\Delta\psi_{tot}} = 0.021$$

$$\frac{\Delta\psi_{abs.}}{\Delta\psi_{tot}} = 0.3371$$

$$\frac{\Delta\psi_{gen.}}{\Delta\psi_{tot}} = 0.288$$

$T_c := 323$ $\qquad$ $T_a := 323$

$T_g := 347.15$ $\qquad$ $T_o := 298$

$$COP_h := \frac{Q_a + Q_c}{Q_g + W_p}$$

$$COP_h = 1.8327$$

$$ECOP := \frac{\left[Q_c \cdot \left(1 - \frac{T_o}{T_c}\right) + Q_a \cdot \left(1 - \frac{T_o}{T_a}\right)\right]}{\left[Q_g \cdot \left(1 - \frac{T_o}{T_g}\right)\right] + W_p}$$

$$ECOP = 0.764$$

APPENDIX E

SIMULATION OUTPUTS FOR THE EXERGETIC EFFICIENCY

E.1. Floor Heating Panel

Constants of Simulation

Total mass flow rate: $\boxed{m_f := 0.535}\ \frac{kg}{s}$

Specific heat of water: $\boxed{C_{pf} := 4.18}\ \frac{kj}{kg \cdot K}$

Water inlet temp.: $\boxed{T_{fi} := 323}\ K$

Water outlet temp.: $\boxed{T_{fo} := 313}\ K$

$\boxed{Q_s := 25.33}\ kW$

Ambient temperature: $\boxed{T_o := 298.15}\ K$

Plate temperature: $\boxed{T_p := 314}\ K$

$$\psi_f := m_f \cdot C_{pf} \cdot \left[\left(T_{fi} - T_{fo} \right) - T_o \cdot \ln\left(\frac{T_{fi}}{T_{fo}} \right) \right]$$

$$\psi_f = 1.394 \quad kW$$

$$\psi_{rec} := Q_s \cdot \left(1 - \frac{T_o}{T_p} \right) \qquad \psi_{rec} = 1.279 \quad kW$$

$$n_{floor} := \left(\frac{\psi_{rec}}{\psi_f} \right) \qquad n_{floor} = 0.917$$

E.2. Storage tank temperature is less than 41.5 ^oC

Constants of Simulation

Mass flow rate of water in collector	mass flow rate of load	Collector area
$\boxed{m_{col} := 0.05}\ \frac{kg}{s}$	$\boxed{m_L := 0.5746}\ \frac{kg}{s}$	$\boxed{A_c := 100}\ m^2$

Specific heat of water in collector	Specific heat of load stream	Boiler temperature
$\boxed{C_{pwcol} := 4.18}\ \frac{kj}{kg \cdot K}$	$\boxed{C_{pwL} := 4.18}\ \frac{kj}{kg \cdot K}$	$\boxed{T_F := 350}\ K$

Input data :

$Q_u := 19.3$ kW $\quad I := 437\ \frac{W}{m^2} \quad Q_{aux2} := 0$ kW $\quad T_0 := 273.15$ K

$T_s := 39.2 + 273.15 \qquad T_s = 312.35$ K

$$Q_i := \frac{(A_c \cdot I)}{1000} \qquad Q_i = 43.7 \text{ kW}$$

$\psi_i := Q_i \cdot 0.985 \qquad \psi_i = 43.044$ kW

$$T_{fo} := \left[\left[\frac{Q_u}{(m_{col} \cdot C_{pwcol})}\right]\right] + (T_s)$$

$T_{fo} = 404.694$ K

$$\psi_u := m_{col} \cdot C_{pwL} \cdot \left(T_{fo} - T_s - T_0 \cdot \ln\left(\frac{T_{fo}}{T_s}\right)\right)$$

$\psi_u = 4.514$ kW

$T_{fe} := 39.2$ $\quad T_{01} := 25$

$T_L := T_{fe} - 4.5$

$$\psi_e := m_L \cdot C_{pwL} \cdot \left(T_{fe} - T_L - T_{01} \cdot \ln\left(\frac{T_{fe}}{T_L} \right) \right)$$

$\psi_e = 3.486$ kW

$T_{fe} := 39.2 + 273.15$ $\quad T_{fe} = 312.35$ $\quad T_{02} := 25 + 273.15$

$$\psi_{aux2} := m_L \cdot C_{pwL} \cdot \left[\left(T_{fe} - T_s \right) - T_{02} \cdot \ln\left(\frac{T_{fe}}{T_s} \right) \right]$$

$\psi_{aux2} = 0$

$$\eta_{IIsub} := \left(\frac{\psi_u}{\psi_i + \psi_{aux2}} \right) \cdot \left(\frac{\psi_e}{\psi_u} \right)$$

$\eta_{IIsub} = 0.081$

$$\eta_{overall} := \frac{\psi_{rec}}{\psi_i + \psi_{aux2}}$$

$\eta_{overall} = 0.03$

E.3. Storage tank temperature is between 41.5 and 50C

Constants of Simulation

Mass flow rate of water in collector

$\boxed{m_{col} := 0.05} \ \frac{kg}{s}$

Mass flow rate of load

$\boxed{m_L := 0.5746} \frac{kg}{s}$

Collector area

$\boxed{A_c := 100} \ m^2$

Specific heat of water in collector

$\boxed{C_{pwcol} := 4.18} \ \frac{kj}{kg \cdot K}$

Specific heat of load stream

$\boxed{C_{pwL} := 4.18} \frac{kj}{kg \cdot K}$

Boiler temperature

$\boxed{T_F := 390} \ K$

Input data :

$Q_u := 22.3 \quad kW \qquad I := 448 \ \frac{W}{m^2} \qquad Q_{aux2} := 7.6 \ \ kW \quad T_0 := 273.15 \ \ K$

$T_s := 43.3 + 273.15 \qquad T_s = 316.45 \quad K$

$Q_i := \frac{(A_c \cdot I)}{1000} \qquad Q_i = 44.8 \qquad kW$

$\psi_i := Q_i \cdot 0.985 \qquad \psi_i = 44.128 \qquad kW$.

$$T_{fo} := \left[\left[\frac{Q_u}{(m_{col} \cdot C_{pwcol})}\right]\right] + (T_s)$$

$T_{fo} = 423.149 \quad K$

$$\psi_u := m_{col} \cdot C_{pwL} \cdot \left(T_{fo} - T_s - T_0 \cdot \ln\left(\frac{T_{fo}}{T_s}\right)\right)$$

$\psi_u = 5.713 \qquad kW$

$T_{in} := 43.3 + 273.15 \qquad T_{in} = 316.45$

$T_{out} := 50 + 273.15 \qquad T_{out} = 323.15$

$m_L := 0.538 \qquad C_{pwL} := 4.18 \quad \frac{kj}{kg \cdot K}$

$$\psi_f := m_L \cdot C_{pwL} \cdot \left[\left(T_{out} - T_{in} \right) - T_0 \cdot \ln\left(\frac{T_{out}}{T_{in}} \right) \right]$$

$\psi_f = 2.197 \quad$ kW

$$\eta_{overall} := \left(\frac{\psi_{rec}}{\psi_i + \psi_f} \right)$$

$\eta_{overall} = 0.028$

E.4. Storage tank temperature is greater than 50°C

Constants of Simulation

Mass flow rate of water in collector	mass flow rate of load	Collector area
$m_{col} := 0.05 \ \frac{kg}{s}$	$m_L := 0.5746 \ \frac{kg}{s}$	$A_c := 100 \ m^2$

Specific heat of water in collector	Specific heat of load stream
$C_{pwcol} := 4.18 \ \frac{kj}{kg \cdot K}$	$C_{pwL} := 4.18 \ \frac{kj}{kg \cdot K}$

Input data :

$Q_u := 36.4$ kW $\quad I := 585 \ \frac{W}{m^2} \quad Q_{aux2} := 0$ kW $\quad T_0 := 273.15$ K

$T_s := 43.3 + 273.15 \qquad T_s = 316.45$ K

$$Q_i := \frac{(A_c \cdot I)}{1000} \qquad Q_i = 58.5 \quad kW$$

$\psi_i := Q_i \cdot 0.985 \qquad \psi_i = 57.623$ kW

$$T_{fo} := \left[\left[\frac{Q_u}{(m_{col} \cdot C_{pwcol})}\right]\right] + (T_s)$$

$T_{fo} = 490.613$ K

$$\psi_u := m_{col} \cdot C_{pwL} \cdot \left(T_{fo} - T_s - T_0 \cdot \ln\left(\frac{T_{fo}}{T_s}\right)\right)$$

$\psi_u = 11.367$ kW

$$\eta_{overall} := \frac{\psi_u}{\psi_i} \cdot \frac{\psi_{rec}}{\psi_u}$$

$$\eta_{overall} = 0.022$$

APPENDIX F

AMBIENT AIR TEMPERATURE AND INSOLATION VALUES

F.1. Hourly Ambient Air Temperature Values

Table F.1. Hourly Ambient Air Temperature, T_o(^oC) in Ankara for the 21 st Day of the Months of the Heating Season

Month / Hour	1 Jan	2 Feb	3 Mar.	4 Apr	10 Oct	11 Nov	12 Dec
1	0,00	3,00	-2,10	7,00	6,50	-2,10	-2,20
2	0,00	3,00	-2,50	7,10	6,50	-1,50	-2,20
3	0,00	2,90	-3,00	7,10	6,00	-1,30	-2,40
4	0,00	2,90	-3,90	7,00	4,90	-1,40	-2,80
5	-0,10	2,80	-4,20	6,70	4,70	-0,80	-2,90
6	-0,30	2,70	-4,50	5,80	4,80	-0,20	-2,90
7	-1,00	2,70	-4,80	5,80	4,80	0,50	-2,70
8	-1,10	1,80	-3,00	6,80	5,70	1,00	-2,20
9	-0,20	1,00	-0,20	7,50	6,70	2,80	-0,90
10	0,80	0,90	0,80	8,80	7,50	5,00	1,10
11	2,80	1,20	1,80	10,60	8,20	5,20	3,00
12	4,20	2,40	3,60	10,80	9,70	6,10	5,60
13	4,30	2,80	5,10	11,90	10,00	7,00	5,90
14	4,50	3,40	6,20	11,80	10,40	6,40	6,30
15	4,40	4,20	7,00	10,00	10,10	6,30	6,40
16	4,10	4,10	7,50	10,40	10,50	6,20	6,10
17	2,50	4,00	8,20	10,00	10,50	6,00	5,20
18	1,00	2,70	7,50	9,00	9,50	6,00	4,40
19	-0,20	2,40	6,00	8,90	8,60	5,40	3,20
20	-1,70	2,20	4,60	8,00	7,70	5,40	3,20
21	-2,00	1,80	2,60	7,80	7,60	5,40	3,20
22	-2,60	0,80	2,60	7,10	6,30	4,90	3,10
23	-2,90	0,40	2,50	6,60	5,80	4,10	2,10
24	-3,40	0,00	1,40	6,00	5,40	4,20	1,40

F.2. Hourly Solar Insolation Values

Table F.2. Hourly Solar Insolation ,[W/m^2] on a Surface in Ankara,with 60^o inclination and 0^o azimuth angle,for the 21st Day of the Months of the heating Seasons

Month	1	2	3	4	10	11	12
Hour	Jan	Feb	Mar.	Apr	Oct	Nov	Dec
1	0,00	0,00	0,00	0,00	0,00	0,00	0,00
2	0,00	0,00	0,00	0,00	0,00	0,00	0,00
3	0,00	0,00	0,00	0,00	0,00	0,00	0,00
4	0,00	0,00	0,00	0,00	0,00	0,00	0,00
5	0,00	0,00	0,00	0,00	0,00	0,00	0,00
6	0,00	0,00	0,00	3,00	0,00	0,00	0,00
7	0,00	0,00	58,00	58,00	59,00	0,00	0,00
8	68,00	112,00	156,00	183,00	192,00	140,00	27,00
9	184,00	239,00	304,00	329,00	358,00	286,00	154,00
10	299,00	357,00	428,00	452,00	479,00	437,00	261,00
11	391,00	448,00	526,00	551,00	585,00	555,00	347,00
12	437,00	502,00	589,00	610,00	648,00	619,00	389,00
13	437,00	502,00	589,00	610,00	648,00	619,00	389,00
14	391,00	448,00	526,00	551,00	585,00	555,00	347,00
15	299,00	357,00	428,00	452,00	479,00	437,00	261,00
16	184,00	239,00	304,00	329,00	348,00	286,00	154,00
17	68,00	112,00	156,00	183,00	192,00	140,00	27,00
18	0,00	0,00	58,00	58,00	59,00	0,00	0,00
19	0,00	0,00	0,00	3,00	0,00	0,00	0,00
20	0,00	0,00	0,00	0,00	0,00	0,00	0,00
21	0,00	0,00	0,00	0,00	0,00	0,00	0,00
22	0,00	0,00	0,00	0,00	0,00	0,00	0,00
23	0,00	0,00	0,00	0,00	0,00	0,00	0,00
24	0,00	0,00	0,00	0,00	0,00	0,00	0,00

Daily avarage insolation [kWh/m^2] :

2758	3316	4122	4372	4622	4074	2356

Annual Insolation [kWh/m^2] : 775826

F3. Sample Building

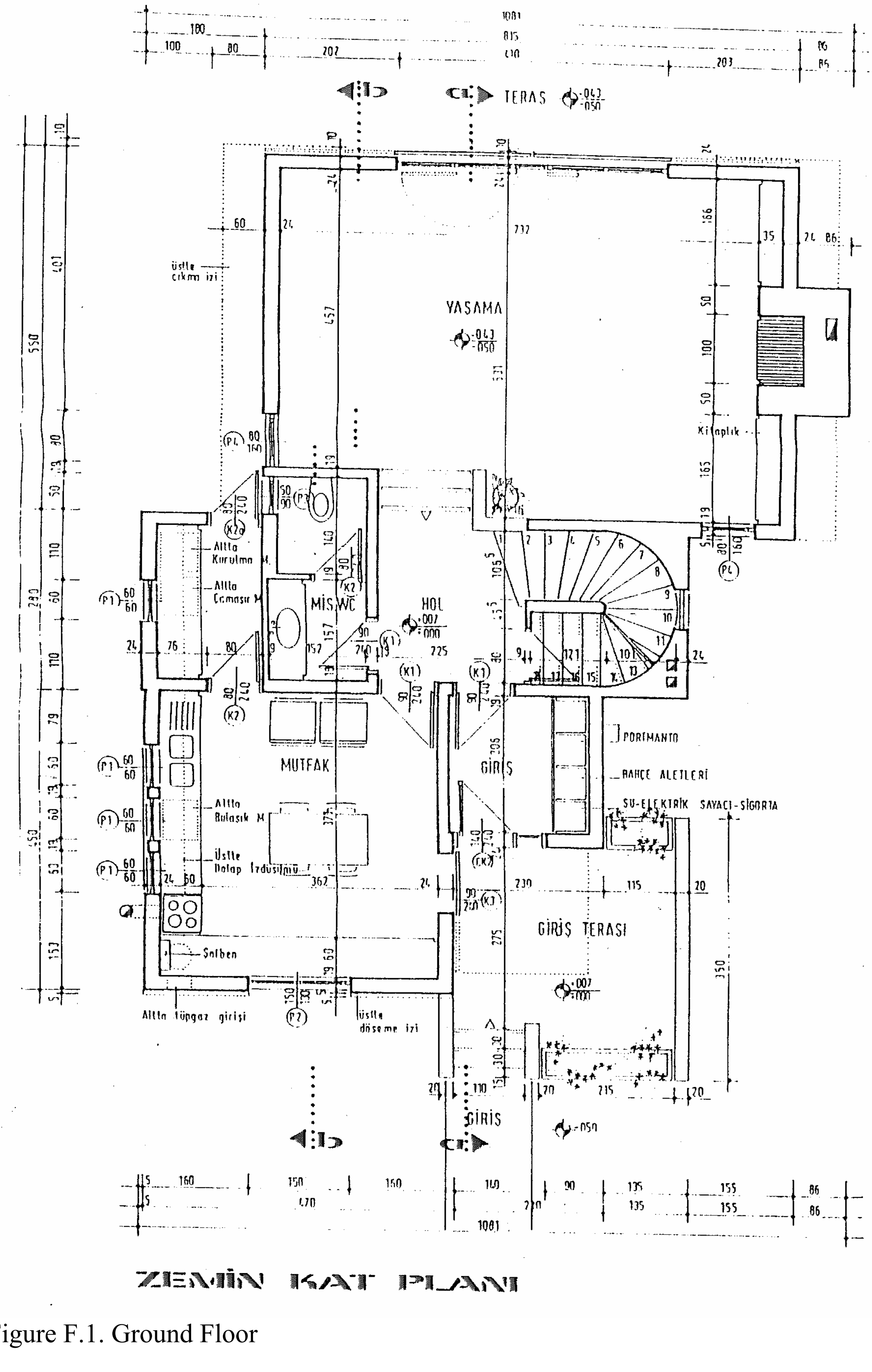

Figure F.1. Ground Floor

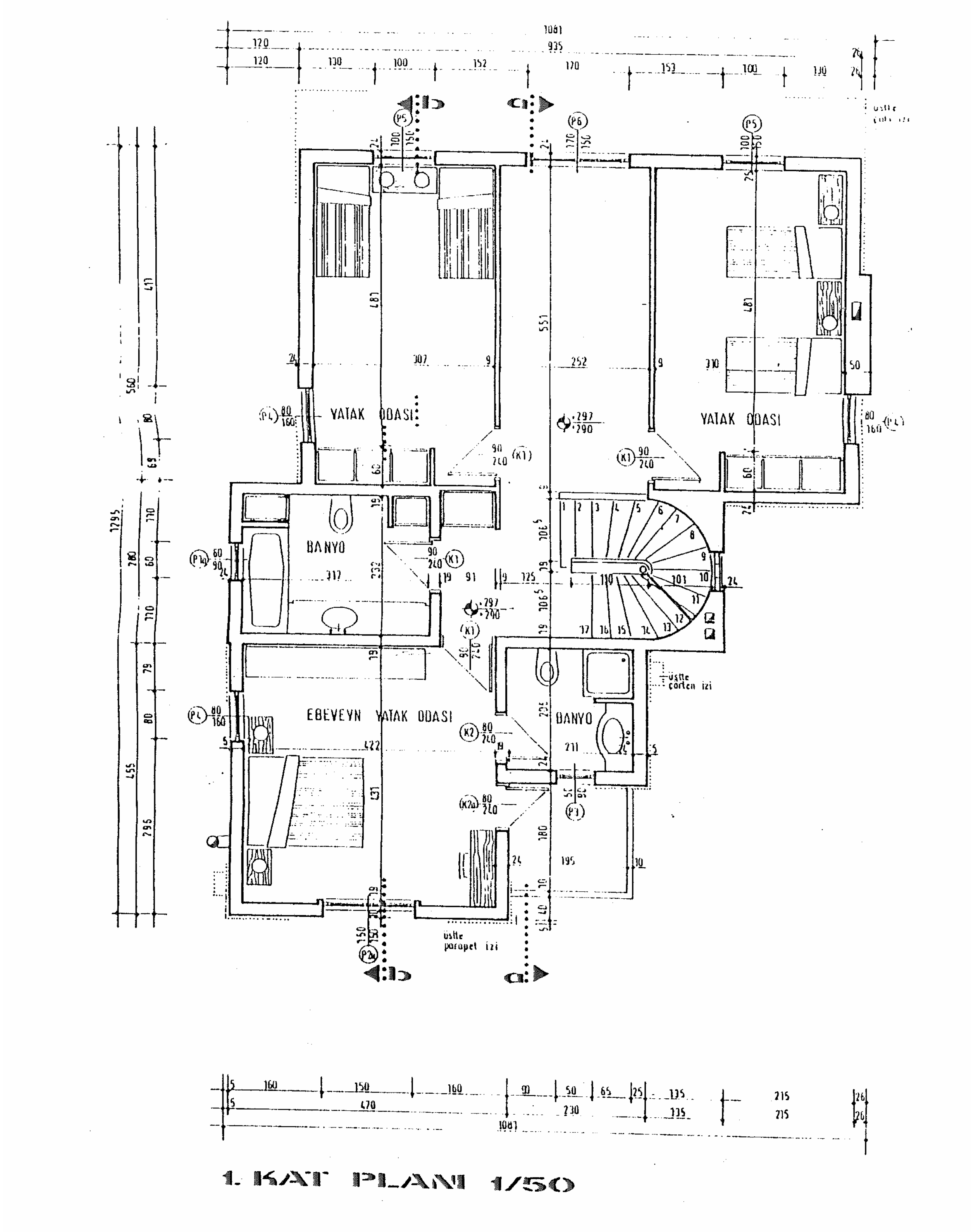

Figure F.2. First Floor

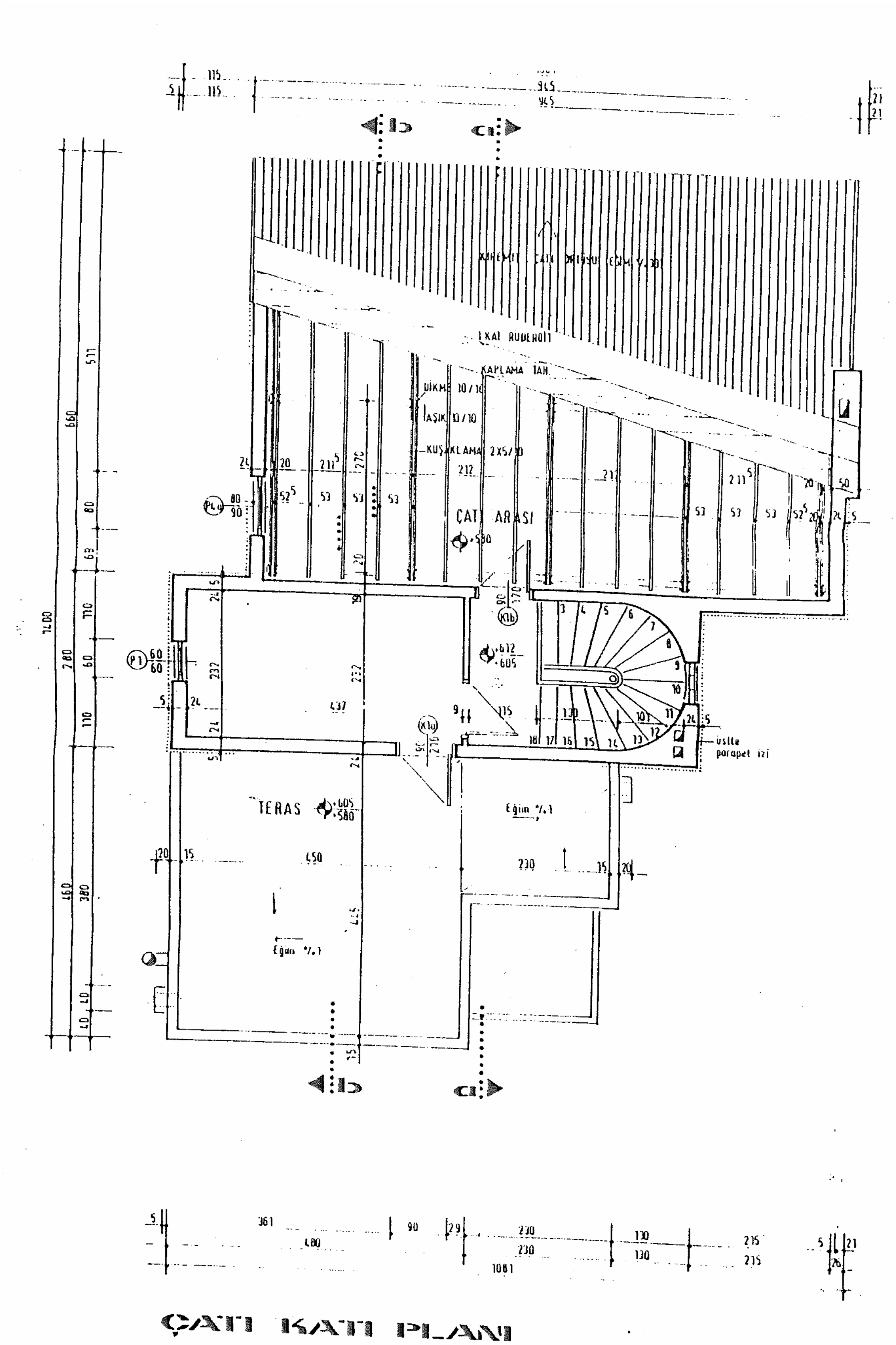

Figure F.3. Second and Roof floors

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