DEVELOPMENT OF PROPERTY EQUATIONS FOR BUTANE AND
ISOBUTANE – PRELIMINARY DESIGN OF HOUSEHOLD REFRIGERATION
CYCLE

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DEVELOPMENT OF PROPERTY EQUATIONS FOR BUTANE AND ISOBUTANE – PRELIMINARY DESIGN OF HOUSEHOLD REFRIGERATION CYCLE

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ABSTRACT

DEVELOPMENT OF PROPERTY EQUATIONS FOR BUTANE AND ISOBUTANE – PRELIMINARY DESIGN OF HOUSEHOLD REFRIGERATION CYCLE

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This study aims to simulate a vapor compression refrigeration cycle, working with either butane (R-600) or isobutane (R-600a). For this purpose a computer program is written to design a household refrigerator, by modeling a steady-state, vapor compression cycle, with user defined input data. Each refrigerator component can be designed separately, as well as parts of a single refrigeration system in the program.

In order to determine the refrigerant thermophysical properties at different states, least squares polynomial equations for different properties of R-600 and R-600a have been developed.

Computer program is used for refrigeration cycle analysis, variable speed compressor design and calculating coefficient of performance (COP) and irreversibility of the cycle.

Sample-preliminary designs have been carried out for different refrigeration loads, room and cold space temperatures with the program, to compare the

performance characteristics of the refrigerants. Designs have been performed at different refrigeration loads, room and cold space temperatures. It is observed that for the same conditions R-600 has slightly better performance characteristics than those of R-600a.

ÖZ

BÜTAN VE İZOBÜTAN İÇİN ÖZELLİK DENKLEMLERİ

GELİŞTİRİLMESİ – EV TİPİ BUZDOLABI SOĞUTMA ÇEVRİMİ ÖN

TASARIMI

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Bu çalışma, bütan (R-600) yada izobütan (R-600a) ile çalışan, buhar sıkıştırmalı bir soğutma çevriminin benzeşimini gerçekleştirmeyi hedeflemektedir. Bu amaçla kullanıcı girdisini kullanarak; sabit durum, buhar sıkıştırmalı soğutma çevrimi modelleyerek, ev tipi bir buzdolabı tasarlayan bir bilgisayar programı yazılmıştır. Programda, her bir buzdolabı elemanı ayrı ayrı tasarlanabileceği gibi, tek bir soğutma sistemine ait parçalar olarak da tasarlanabilmektedir.

Soğutucu akışkanların değişik durumlardaki termofiziksel özelliklerinin hesaplanabilmesi amacıyla, R-600 ve R-600a' nın farklı özelliklerine ait en küçük kareler polinomu denklemleri geliştirilmiştir.

Bilgisayar programı soğutma çevrimi analizleri, değişken hızlı kompresör tasarımı, performans katsayısı hesaplamaları ve çevrim tersinmezliği hesaplarında kullanılmıştır.

Soğutucu akışkanların performans özelliklerinin karşılaştırabilmesi amacıyla, bilgisayar programı kullanılarak, farklı soğutma yükler, oda ve soğuk hacim sıcaklıkları için, örnek ön tasarımlar gerçekleştirilmiştir. Tasarımlar farklı soğutma yükleri, oda ve soğuk hacim sıcaklıkları için gerçekleştirilmiştir. Aynı şartlarda R-600'ün R-600a'ya göre daha iyi performans özelliklerine sahip olduğu gözlemlenmiştir.

Anahtar Kelimeler: Bütan, izobütan, termofiziksel özellikler, en küçük kareler polinomu, ev tipi buzdolabı.

To My Family…

ACKNOWLEDGEMENTS

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LIST OF SYMBOLS AND ABBREVIATIONS

A   :Area (m^2)

a,b,c,

d,e,f   :Constant coefficient

C   :Clearance factor

COP   :Coefficient of performance

c_p :Constant pressure specific heat (kJ/kg.K)

D   :Diameter (mm)

f   :fraction, friction factor

F   :View factor

G   :Mass flow rate per unit area (kg/m^2.s)

g   :Gravitational acceleration (m/s^2)

h   :Heat transfer coefficient (J/m^2.K)

i   :Enthalpy (kj/kg)

k   :Thermal conductivity (mW/m.K), isentropic compression

           index

L   :Length, stroke, fin length (m)

L_t : Tube length (m)

m   :Mass, re-expansion index

n   :Compression index, number of cylinders

N   : Rotational speed (rpm)

Nu   :Nusselt number (ratio of convective to conductive heat

           transfer)

P   :pressure (Pa, MPa), perimeter (m)

Pr   :Prandtl number (ratio of kinematic viscosity and thermal

           diffusivity)

q   :Specific heat transfer (J/m^3)

Q   :Heat transfer rate (J/m^2)

Ra   :Rayleigh number

Re   :Reynolds number (ratio of inertial forces to viscous forces)

S   :Entropy (kJ/kg.K)

t_p : pitch (mm)

T   :Temperature (K)

u   :Velocity (m/s)

U   :Overall heat transfer coefficient ($J/m^2.K$)

V   :Volume (m^3), velocity (m/s)

w   :Specific work (J/kg)

W   :Work (J)

x   :Two-phase mixture quality

z   :Length corresponding to unit quality change (mm)

Greek Letters:

α : Thermal diffusivity (m^2/s)

β : Reciprocal of air temperature (1/K)

X : Lockart-Martinelli parameter (liquid fraction of a flowing

          fluid)

ε : Emissivity

η : Efficiency, for compressor

ηo : Temperature effectiveness, for fins

ηf : Efficiency, for fins

μ : Dynamic viscosity (μPa.s)

ν : Specific volume (m^3/kg)

ρ : Density (kg/m^3)

σ : Stefan-Boltzman constant ($W/m^2.K^4$)

Δ : Difference

Γ : Liquid film flow rate per unit perimeter

$''$: Flux

$'$: Per length

Subscripts:

1,2…..	:States, sections, coefficients
2p	:Two-phase
b	:Base, boiling
c	:Clearance, compression, cross-sectional, convective
comp	:Compressor
cond	:Condenser
cv	:Clearance volumetric
d	:Discharge
D	:Diameter based
e	:Exit
evap	:Evaporator
expd.	:Expansion device
f	:Fluid, friction, fin
fc	:Forced convection
fi	:Inner scale
g	:Vapor, gravity
h	:Higher
i	:Inner
l	:Liquid, lower
lm	:Log-mean
m	:Mean, momentum
o	:Outer
p	:Piston
r	:Refrigerant
pd	:Discharge pressure of the compressor
ps	:Suction pressure of the compressor
rad	:Radiative
s	:Suction, surface
sat	:Saturation

sat,ib	:Saturation incipient
sb	:subcooled
sh	:superheated
t	:Tube, isothermal
v	:Vapor, volumetric
w	:wall
z	:Length corresponding to unit quality change
∞	:air

CHAPTER 1

INTRODUCTION

Refrigeration is the process of removing heat from an enclosed space, or from a substance, and moving it to a place where it is unobjectionable. The primary purpose of refrigeration is lowering the temperature of the enclosed space or substance and then maintaining that lower temperature. Today refrigeration is used for various purposes from air conditioning of buildings to petrochemical plants; however refrigeration is most widely used in the process of keeping food at the desired temperature in households.

The first public demonstration of a refrigeration process was in Edinburgh, in 1755, by a Scottish chemist and doctor; William Cullen. By using a pump to produce a partial vacuum over a container of diethyl ether, which then boiled to absorb heat from the surroundings, he managed to produce a small amount of ice [1]. However, the idea of using a vapor compression cycle for the purpose of refrigeration was proposed by Oliver Evans in 1805 in "The Young Steam Engineers Guide" [1]. The first patented vapor compression cycle was invented by an American scientist named Jacob Perkins in 1834 in Great Britain [1]. The system was using a hand operated compressor and ether as the working fluid. Although the system was working properly it did not have commercial success. The first commercially successful vapor compression refrigerators was designed and built by Alexander Twinning in 1855 [1].

In the 19th century the vapor compression refrigeration cycles were widely being used for producing large amounts of ice and were using working fluids like methyl formate, ammonia, methyl chloride, and sulfur dioxide which are hazardous and not suitable for use at homes. In 1930 Chlorofluorocarbon (CFC)

refrigerants synthesized by General Motors Research Lab began to be used by Frigidaire in household refrigerators [1]. However CFC's are found to be dangerous for the atmospheric ozone and in 1990 in Montreal Protocol it is stated that use of CFC's should be terminated by year 2000. As an alternative to CFC's, hydrochlorofluorocarbon (HCFC) and hydrofluorocarbon (HFC) which have less or no ozone depletion effect are began to be used in household refrigerators. First in 1993 R-134a (tetrafluoroethane) is began to be used as an alternative for R-12 and then in recent years use of R-600a (isobutane) in household refrigerators is increased. This study is focused on the refrigerants R-600 (butane) and R-600a (isobutane).

1.1 Vapor Compression Refrigeration Cycles

The most widely used system for household refrigerators is vapor-compression cycle. In a vapor-compression refrigeration cycle a fluid cycles through the system absorbing the heat from the refrigerated space and giving it away to the environment. This fluid is called a refrigerant. While cycling through the system, the thermophysical state of the refrigerant changes between different stages of the cycle.

There are four main processes in a vapor-compression refrigeration cycle; compression, condensation, expansion and evaporation. First the refrigerant at the lower working pressure of the cycle is compressed to the higher working pressure in the compressor. This is the most important part of the cycle. Also almost all of the energy consumption of the cycle comes from this process.

After compression the refrigerant vapor at high pressure and temperature is cooled to until the liquid state is achieved, which is called condensation. The heat transfer occurs in a component called condenser which has high surface area and thermal conductivity for better heat removal.

For the third stage the cooled refrigerant liquid is again taken to the lower working pressure of the cycle by an expansion device. During the expansion process flow rate of the refrigerant is also adjusted.

The last process of the cycle is evaporation, which means the removal of heat from the cold room by transferring it to the refrigerant. Similar to a condenser a component with high surface area and thermal conductivity is used which is called an evaporator. After the evaporator the low pressure high temperature refrigerant is compressed in the compressor, which restarts the whole vapor-compression refrigeration cycle. The schematic diagram of a refrigeration cycle is given in Figure 1.1.
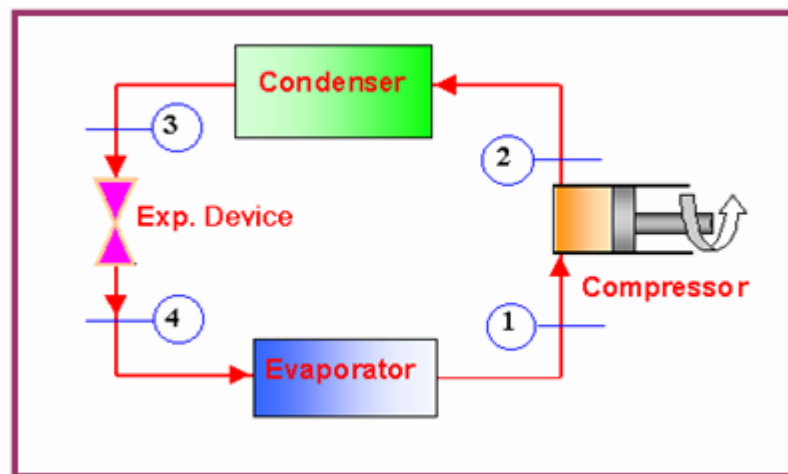
Figure 1.1 Schematic equipment diagram of a refrigeration cycle [2].

1.1.1 Vapor Compression Refrigeration Cycle Principles

As it is the case with all thermodynamic cycles there is an ideal and an actual cycle for vapor-compression refrigeration. In the ideal case; the compression process is isentropic (constant entropy), the evaporation and condensation processes are isobaric (constant pressure) and the expansion is isenthalpic

(constant enthalpy). A T-s diagram depicting an ideal refrigeration cycle is given in Figure 1.2.
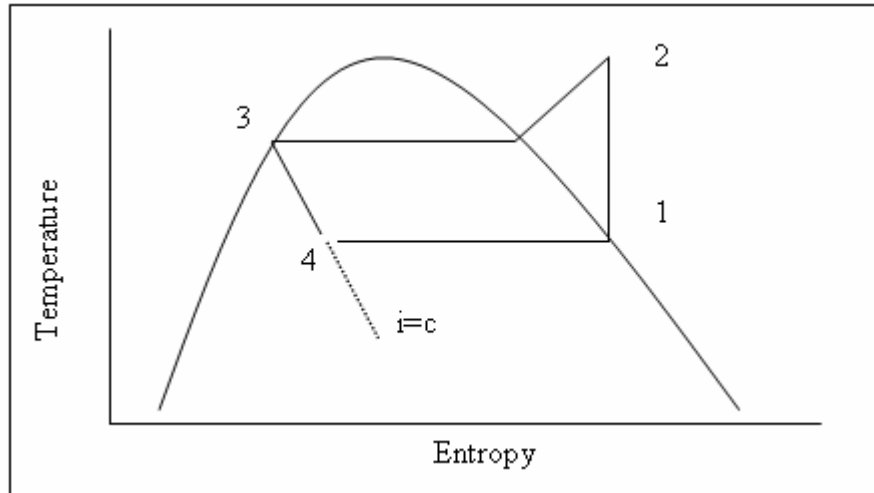
Figure 1.2 T-s diagram of an ideal vapor compression cycle [2].

Unfortunately because of inefficiencies and other variables in an actual cycle all of the processes show deviations from the ideal case. A comparison of ideal and actual cycles can be observed in a P-i diagram given in Figure 1.3.
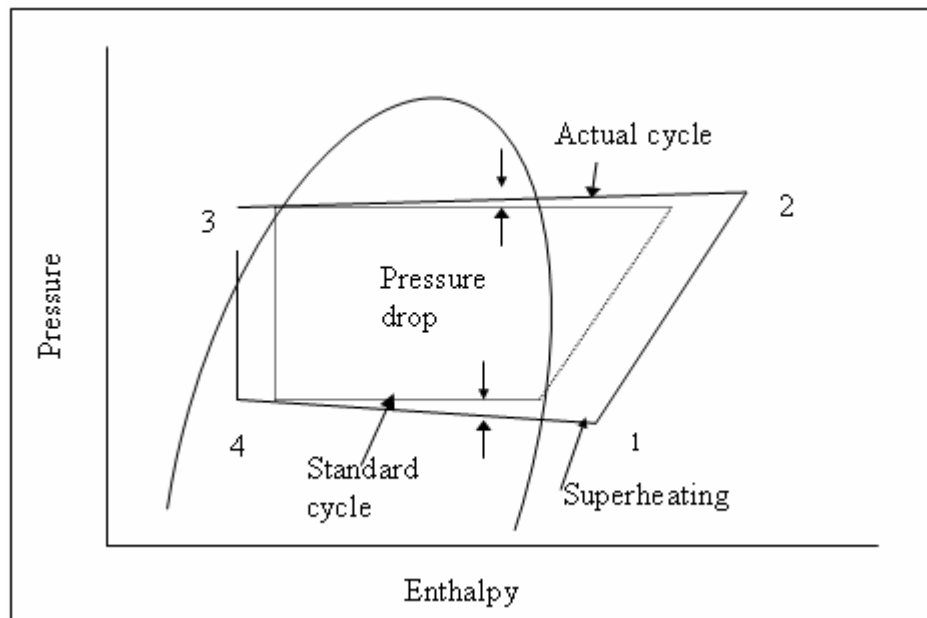
Figure 1.3 P-i diagrams of actual and ideal vapor compression cycles [2].

As it can be seen in Figure 1.3 the condensation and evaporation processes are not isobaric due to unavoidable friction in the evaporator and condenser. Also the expansion process is not isenthalpic and the exit state is saturated liquid (with a small amount of vapor) which is the desired state in the ideal cycle.

Another deviation from the ideal cycle is the superheating of the refrigerant in the evaporator in order to avoid any liquid droplets from going into the compressor, which may be hazardous. Also as it can be seen in the figure, in order to compensate the pressure loss in the condenser, the higher working pressure of the actual cycle is slightly higher than those of the ideal cycle.

1.2 Present Study

In this study, the aim is to develop a computer program that designs a household refrigerator working with R-600 or R-600a. Program simulates a steady-state, vapor compression refrigeration cycle. It uses user-defined cold space and ambient temperatures. The aim is to decrease the number of experiments required to design an actual refrigerator, by pre-determining the dimensions of the components with the program, before beginning experiments

Computer program is capable of designing individual refrigerator components; compressor, evaporator, condenser and capillary tube. Combining all individual design modules all four components of a given cycle can be designed.

The program is capable of determining coefficient of performance (COP) and irreversibility of the cycle. It can also be used for designing a variable speed compressor by altering the input parameters.

In order to design refrigerator components, fluid thermophysical properties at different states are required. There is an independent module in the program that

can calculate refrigerant property values, according to user-defined temperature and/or pressure values.

In Chapter 2 the basic characteristics of hydrocarbon refrigerants and the reason behind selection of R-600 and R-600a as the main subjects of this study will be explained. In Chapter 3 method used for thermophysical property calculations - least squares polynomial approximation- is introduced. In Chapters 4 through 7 governing equations used for compressor, condenser, expansion device and evaporator design is explained respectively. In Chapter 8 design procedure of the computer algorithm for both individual component and cycle designs are explained. In Chapter 9 a sample cycle run is conducted and results of different cycles with both R-600 and R-600a are compared. In Chapter 10 the results of the cycle comparison and other output data of the computer program is evaluated, and suggestions for future studies in this area have been made.

CHAPTER 2

REFRIGERANTS FOR HOUSEHOLD REFRIGERATORS

Main purpose of refrigeration is to remove heat from a space and give it away to another. In order to achieve the required amount of heat removal a substance with high capacity of heat absorption is required. The substance that is used in refrigerators for heat removal is called the refrigerant.

There are numerous applications of refrigeration in a wide range of cooling capacities from keeping food at households, to storing tons of dairy consumer products before shipment. Although the capacities change, the basic principle is the same for all refrigeration processes. Since refrigerants are utilized in equipment that are in close contact with humans, there are some safety and environmental parameters together with the performance parameters for a refrigerant.

Main parameter that is used for comparison of refrigerant performance is coefficient of performance (COP). COP is a dimensionless ratio and defined by [3]

$$COP = \frac{\text{useful refrigeration effect}}{\text{net energy supplied from external sources}} \qquad (2.1)$$

In a vapor-compression refrigerator the useful refrigeration effect is the amount of cooling load supplied by the evaporator whereas the net energy supplied from external sources is the compressor power.

For the comparison of refrigerants from an environmental point of view the global warming potential (GWP) and ozone depletion potential (ODP) is used. ODP of a refrigerant is the amount of degradation it can cause to the ozone layer compared with trichlorofluoromethane (R-11 or CFC-11) which is taken as the base substance and has an ODP of 1.0. GWP is a measure of how much is the effect of a refrigerant vapor (or any greenhouse gas) to global warming. Similar to ODP, GWP is also a relative value; GWP of a gas is the relative effect of a gas compared with the effect of same amount of mass of carbon dioxide (which has a GWP of 1.0 by definite) GWP is calculated for a specific period of time and it should be denoted when a calculation result will be presented. A refrigerant with small GWP is desired for an environment-friendly system.

As mentioned before, safety is an important parameter for refrigerant selection. There are two main parameters for safety evaluation; flammability and toxicity. As the name indicates flammability states how easily a substance ignites. Since refrigerators are in close contact with humans refrigerants are required to be either non flammable or have very small flammability. Similar to flammability because of human safety refrigerants should be non-toxic.

2.1 Refrigerant Designation Numbering System

Because the chemical names of typical refrigerants are long and complex a method of referring to refrigerants by number was developed by DuPont Company. The numbering system was released for general use in 1956 and has become a standard then. The system uses the chemical composition of the refrigerant in order to number it. Since various refrigerants will be mentioned in this study it will be useful to know how they are numbered. The basic principle for refrigerant numbering is given in Figure 2.1. When there is bromine instead of all or part of the chlorine, the same system is used except that the capital letter "B" after the designation for the parent compound shows the presence of the bromine (Br). The number following the letter "B" shows the number of

Bromine atoms present. If there is lower-case letter that follows the R number, it refers to the form of the molecule when different forms (isomers) are possible, with the most symmetrical form indicated by the number alone. As the form becomes increasingly unsymmetrical, the letters a, b, and c (lower case) are appended.

```
R – X  X  X  X
          └─────────────→ Number of fluorine atoms

       └────────────────→ Number of hydrogen atoms plus one

                          Number of carbon atoms minus one
    └───────────────────→ (not used when equal to zero)

                          Number of unsaturated carbon-carbon
 └──────────────────────→ bonds in the compound
                          (not used when equal to zero)
```
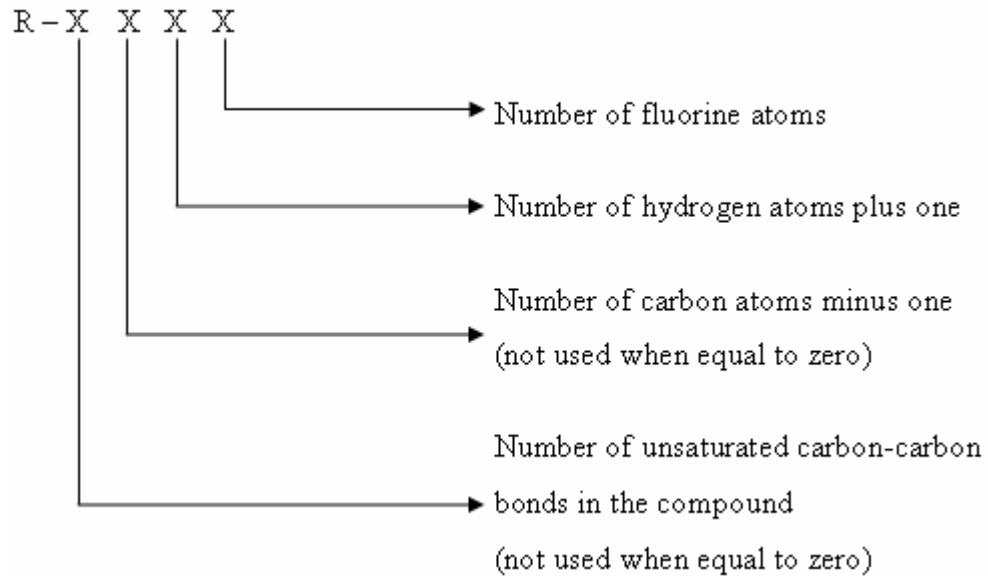
Figure 2.1 R numbering system

The chemical formulae of tetrafluoroethane (R-134a) is CH_2FCF_3 which means it has 2 carbon, 4 fluorine and 1 hydrogen atom as a result the R number will be:

```
R –   1  3  4
            └─────────→ 4 = 4 Fluorine atoms

         └────────────→ 3 = 2 Hydrogen atoms + 1

      └───────────────→ 1 = 2 Carbon atoms - 1

                         Not used since there is no carbon-carbon
 └─────────────────────→ bonds
```
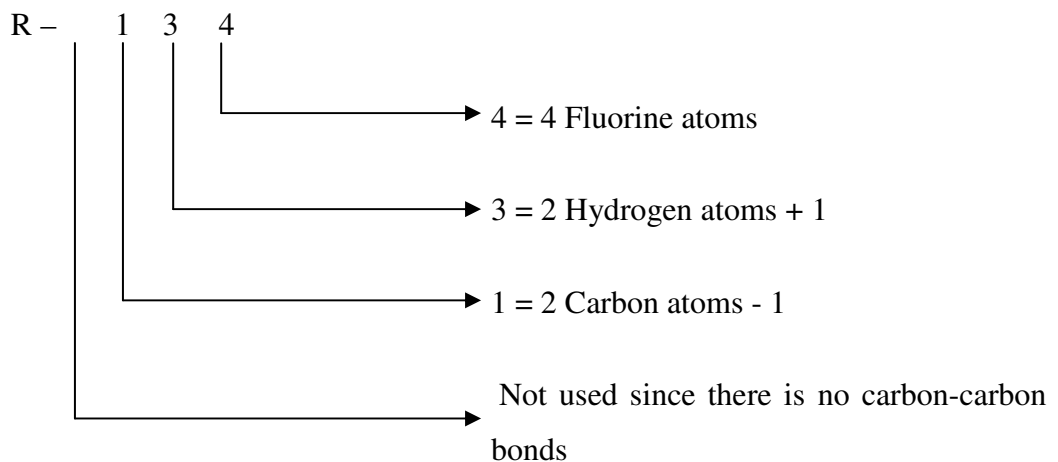
Since it is an isomer the final form of the number will be R-134a.

2.2 Hydrocarbon Refrigerants

In the first vapor-compression refrigerator developed by Perkins, main refrigerant was ethyl ether [4]. Ethyl ether is not a good refrigerant because it is both anaesthetic and flammable. Also since it is odorless and denser than air it was very dangerous to be used with the ignition technology which was lamp or candle.

In the 20th century, beginning with the development of household refrigerators search for a refrigerant that was suitable for domestic use was required. It should be stable, non-toxic, non-flammable, miscible with lubricating oil, capable of operating in a domestic refrigerator without going below atmospheric pressure, a good electrical insulator and a substance with a low index of compression so that the compressor would run cool [4]. However environmental properties were not a concern, since there was no threat at the time. Hydrocarbons (HCs), hydrofluorocarbons (HFCs) and chlorofluorocarbons (CFCs) were investigated and R-12, which was a CFC was selected as the main refrigerant for domestic refrigerator use. It was non-toxic, non-flammable and had a very high COP. The only problem was that R-12 was requiring a large compressor, which is solved by another CFC, R-22. However in 1974 Rowland and Molina stated in their hypothesis that the CFCs are dangerous to ozone layer and their effects will be lethal [5]. Although at the beginning the response to the hypothesis was negative, with further research it was revealed that it was correct and search for ozone-friendly refrigerants began.

After the Montreal Protocol, which forbid the use of CFCs, tetrafluoroethane (R-134a), another refrigerant which was known at the time when R-12 was developed, was chosen to be the best candidate as the alternative. R-134a was a HFC, which was non-flammable with 0 ODP and had similar performance parameters to R-12. It was not selected at the time of R-12's development, because it was not miscible with the conventional lubricants at that time. In

Table 2.1 alternative refrigerants at the time of R-12's development and their properties are given.

Table 2.1 Properties of CFCs and HFCs at the time of R-12's development [6].

Refrigerant	COP	ODP	GWP
R-12	4.70	0.82	10600
R-40	4.90	0.02	16
R-764	4.87	0	0
R-717	4.76	0	0
R-134a	4.50	0	1300

After its integration into domestic refrigerators, R-134a became the main refrigerant in the market. However it had a very high GWP and it cannot satisfy the new and very strict refrigerant criteria. There was no refrigerant in the family of CFCs and HFCs that meet the newly introduced environmental criteria as a result another search for the ideal refrigerant began.

In the early days of refrigerant development HCs were not considered as candidates because they are flammable. However they are very environment-friendly with 0 ODP and very small GWP. As a result they are considered as good candidates for use in household refrigerators. Some of hydrocarbons are generally accepted and tested as an alternative for R-134a. These are; isobutane (R-600a), butane (R-600), propane (R-290) and propylene (R-1270). In Table 2.2 properties of these refrigerants together with R-134a are given.

Table 2.2 Properties of refrigerants [7].

Refrigerant	Boiling Point (°C)	Lower Flame Limit (LFL)	ODP	GWP
R-134a	-26.1	Non-flammable	0	1300
R-290	-42.2	2.1	0	20
R-1270	-47.7	2.0	0	20
R-600	-2	1.5	0	20
R-600a	-11.7	1.7	0	20

As seen in Table 2.2, HC refrigerants have 0 ODP and very low GWP. However, they are flammable. Flammability is a parameter in most of the

refrigeration standards including ASHRAE Standard 34 and prEN 378 and most of the European standards. There are three flammability classes which are; Class 1 denoting non-flammable fluids, Class 2 denoting low flammability and Class 3 denoting high flammability fluids [8]. The use of HCs which are Class 3 refrigerants in household refrigerators is forbidden in USA and Japan, however they can be used in Europe. The safety is satisfied by strict regulations about the maximum charge of refrigerant that can be used in the refrigerator. By keeping the maximum allowable amount very far from the required concentration of flammability a dangerous situation is avoided. It is stated in several standards that if the charge is less than 0.15 kg in hermetically sealed (all joints are welded or brazed) systems HCs can be used [8]. Such small amounts indicate that HCs can be used in small applications like domestic refrigeration.

After safety and environmental concerns are satisfied the performance of the HCs comes into consideration. There are several studies in the literature focusing on selection of a HC as an alternative for R-134a. In some of these researches the possibility of using a pure hydrocarbon is investigated whereas another group focuses on the mixture of HCs as a refrigerant. The main parameter for performance comparison was COP. Other parameters including evaporator and condenser temperatures, total irreversibility, volumetric cooling capacity (VCC) and compressor power are also used [7-12].

Previous studies done for use of hydrocarbons are generally focused on using HCs as a drop in substitute (without changing any component in the refrigerator) for R-134a with some focusing on refrigerators specially designed for hydrocarbon usage. It is observed in several studies that even with a refrigerator designed for R-134a usage hydrocarbons can give better COP values. In Figure 2.2 results of an experimental comparison of COPs of a refrigerator designed for R-134a with some HCs and their blends are given.
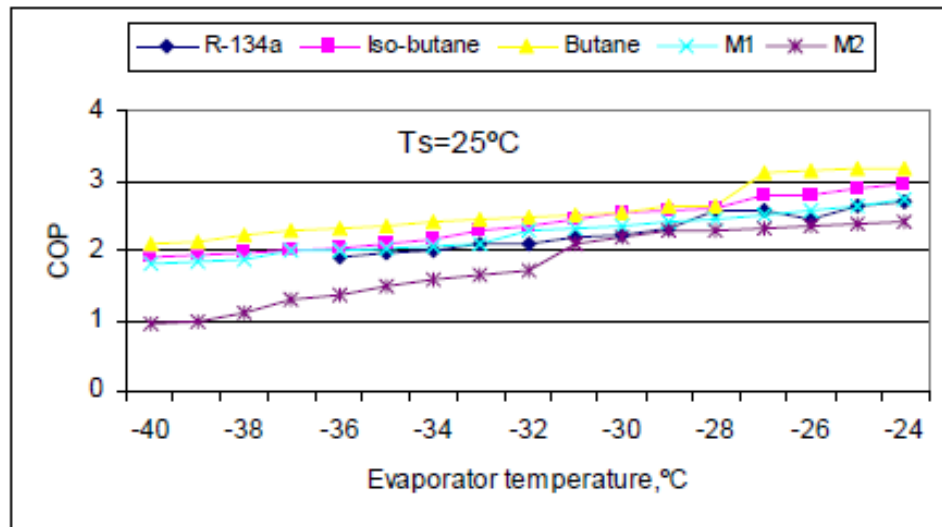
Figure 2.2 COP vs evaporator temperature for R-134a, R-600a, R-600 and their blends [11].

It is observed in Figure 2.2 that even in a refrigerator designed for R-134a, hydrocarbons can give better COP values. In another study, a theoretical analysis is done for performance comparison of various hydrocarbons and R-134a. In Figures 2.3 and 2.4 the results of the comparison in terms of COP and VCC are given. It is observed that R-600 and R-600a give COP values higher than that of R-134a by about 2.4 % and 6.9% respectively [7]. As it comes to the VCC, R-600 and R-600a have significantly lower values than those of R-134a. VCC is the cooling capacity per unit of vapor volume at the exit of the evaporator [8]. As a result, a decrease in VCC requires an increase in the size of compressor. Figure 2.4 indicates that to be able to use R-600 and R-600a, a modification in the compressor is required for a given system.
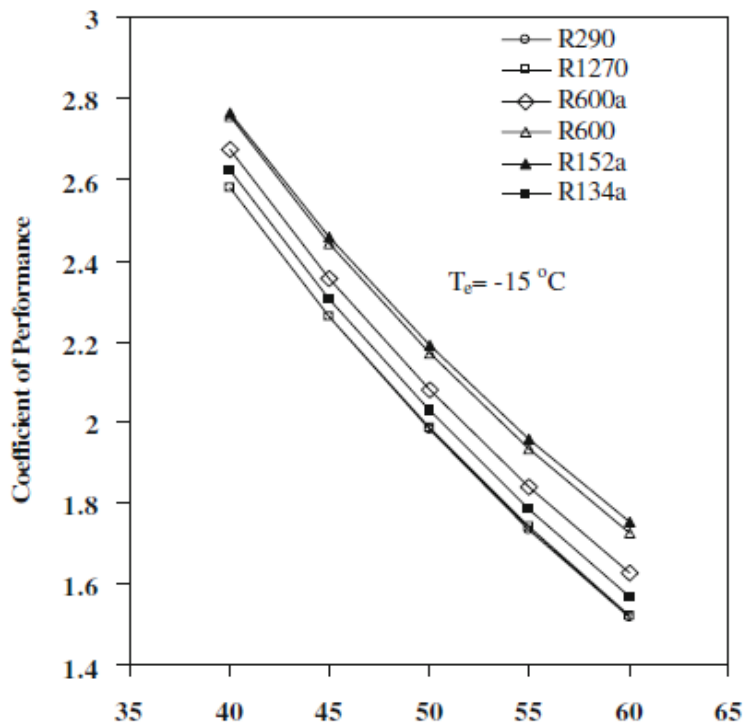
Figure 2.3 COP vs evaporator temperature for R-134a, R-152a and various
hydrocarbons [7].
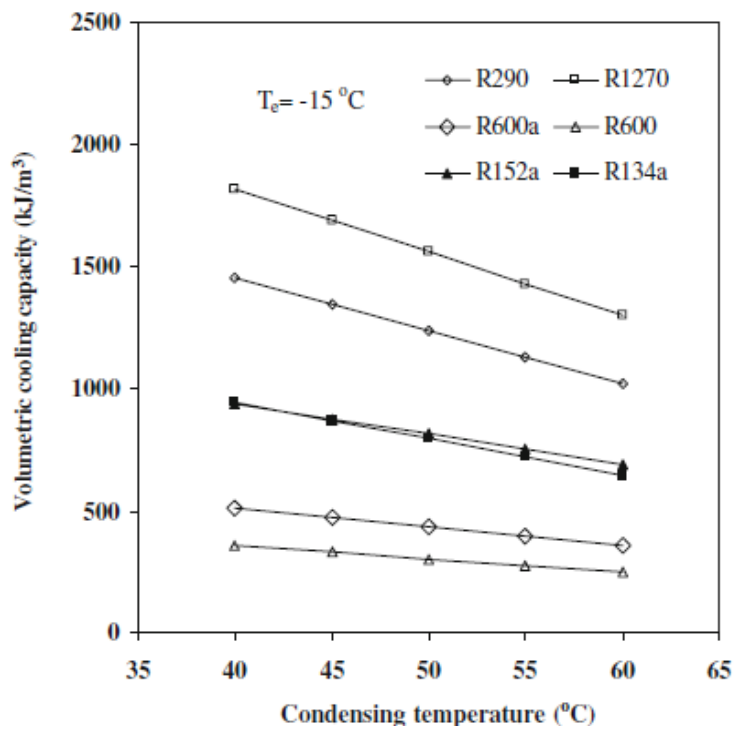
Figure 2.4 VCC vs evaporator temperature for R-134a, R-152a and various
hydrocarbons [7].

Further studies have been done for comparing household refrigerators in the market that are using R-12 and R-600a. In Table 2.3 results of this study are give. It is seen that although the refrigerators working with isobutane have higher cooling capacities they require significantly less power. Further studies about replacing R-134a with hydrocarbons and their blends can be found in the literature [13-17].

Table 2.3 Comparison of refrigerators working with R-12 and R-600a [9].

Make	Model	Refrigerant	Capacity (L)	Power Consumption (kWhr / 24hr)
UK	A	R-12	129	0.75
UK	B	R-12	160	0.71
Liebherr	KT1580	R-600a	155	0.38
Siemens	KT15RSO	R-600a	144	0.52

In this study development of a database for thermophysical properties and vapor-compression refrigeration cycle design of refrigerants R-600 and R-600a will be investigated.

CHAPTER 3

REFRIGERANT THERMOPHYSICAL PROPERTY EQUATIONS

Due to the facts discussed in Chapter 2, there is a need for developing refrigerators working with environmentally-friendly refrigerants. Since thermophysical properties of each refrigerant are unique, each refrigeration system must be designed for its particular refrigerant. To decrease the number of experiments required for system design, computer algorithms are used for predetermination of experimental parameters. In order to have accurate and fast results from the computer algorithms, the thermophysical properties of the refrigerants at the desired thermophysical state should be calculated and collected in a database that can be accessed by the software.

Equations of state (EOS) methods are commonly utilized to calculate the thermophysical properties of a refrigerant under a wide range of temperatures and pressures [18-21]. However the use of EOS requires stability and a high number of iterations, and therefore EOS usage for a computer algorithm may result in several hours of calculations [22]. For an accurate and fast computer algorithm to design a household refrigerator, a simpler approach for refrigerant property calculation is required. Different studies have been carried out for thermophysical property calculations in computer algorithms [23-26].

In this study, a fast and accurate method for calculating thermophysical properties of refrigerants suitable for household refrigerators is developed. The method is based on least squares polynomial approximation for the calculation procedure and is designed to be used for a computer algorithm to design a household refrigeration system charged with the selected refrigerants. It is

required that a majority of computer systems in the market be able to use the algorithm with very short computation times.

In this study, least squares polynomial approximation is used for the calculation of thermophysical properties of refrigerants R-600 and R-600a.

3.1 Basic Principles of Least Squares Polynomial Approximation

Least squares polynomial approximation or least squares polynomial regression analysis is a mathematical method utilized for curve fitting of a given set of data. For n pairs of data in the form of (x_1, y_1), (x_2, y_2)..., (x_n, y_n), a kth order polynomial can be fit to the given set of data:

$$y = a_0 + a_1 \cdot x + a_2 \cdot x^2 + ... + a_k \cdot x^k \tag{3.1}$$

provided that $n \geq k+1$. In Eq. (3.1) a_0, a_1, ..., a_k are independent coefficients. Since the polynomial does not give an exact fit to the data there will be a residual (i.e. error) for each calculated data:

$$R^2 = \sum_{i=1}^{n} \left[y_i - \left(\begin{array}{l} a_0 + a_1 \cdot x_i + a_2 \cdot x_i^2 \\ + ... + a_k \cdot x_i^k \end{array} \right) \right]^2 \tag{3.2}$$

The partial derivatives of the residuals are:

$$\frac{\partial (R^2)}{\partial a_0} = -2 \cdot \sum_{i=1}^{n} \left[y_i - \left(\begin{array}{l} a_0 + a_1 \cdot x_i + \\ a_2 \cdot x_i^2 + ... + \\ a_k \cdot x_i^k \end{array} \right) \right] \tag{3.3}$$

$$\frac{\partial(R^2)}{\partial a_1} = -2 \cdot \sum_{i=1}^{n} \left[y_i - \left(\begin{array}{c} a_0 + a_1 \cdot x_i \\ + a_2 \cdot x_i^2 \\ + ... + \\ a_k \cdot x_i^k \end{array} \right) \right] \cdot x_i \qquad (3.4)$$

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$$\frac{\partial(R^2)}{\partial a_k} = -2 \cdot \sum_{i=1}^{n} \left[y_i - \left(\begin{array}{c} a_0 + a_1 \cdot x_i \\ + a_2 \cdot x_i^2 \\ + ... + \\ a_k \cdot x_i^k \end{array} \right) \right] \cdot x_i^k \qquad (3.5)$$

By equating the partial derivates to zero corresponding to minimum R^2's, the following equations can be obtained:

$$\left(\begin{array}{c} a_0 \cdot n + a_1 \cdot \sum_{i=1}^{n} x_i + a_2 \cdot \sum_{i=1}^{n} x_i^2 \\ + ... + a_k \cdot \sum_{i=1}^{n} x_i^k \end{array} \right) = \sum_{i=1}^{n} y_i \qquad (3.6)$$

$$\left(\begin{array}{c} a_0 \cdot \sum_{i=1}^{n} x_i + a_1 \cdot \sum_{i=1}^{n} x_i^2 \\ + a_2 \cdot \sum_{i=1}^{n} x_i^3 + ... + \\ a_k \cdot \sum_{i=1}^{n} x_i^{k+1} \end{array} \right) = \sum_{i=1}^{n} y_i \cdot x_i \qquad (3.7)$$

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$$\begin{pmatrix} a_0 \cdot \sum_{i=1}^{n} x_i^{\ k} + a_1 \cdot \sum_{i=1}^{n} x_i^{\ 1+k} \\ + a_2 \cdot \sum_{i=1}^{n} x_i^{\ 2+k} + ... + \\ a_k \cdot \sum_{i=1}^{n} x_i^{\ k+k} \end{pmatrix} = \sum_{i=1}^{n} y_i \cdot x_i^{\ k} \qquad (3.8)$$

Representing the above equations in the matrix form:

$$\begin{pmatrix} n & \sum_{i=1}^{n} x_i & ... & \sum_{i=1}^{n} x_i^{\ k} \\ \sum_{i=1}^{n} x_i & \sum_{i=1}^{n} x_i^{\ 2} & ... & \sum_{i=1}^{n} x_i^{\ k+1} \\ ... & ... & ... & ... \\ \sum_{i=1}^{n} x_i^{\ k} & \sum_{i=1}^{n} x_i^{\ 1+k} & ... & \sum_{i=1}^{n} x_i^{\ k+k} \end{pmatrix} \cdot \begin{pmatrix} a_0 \\ a_1 \\ ... \\ a_k \end{pmatrix} = \begin{pmatrix} \sum_{i=1}^{n} y_i \\ \sum_{i=1}^{n} y_i \cdot x_i \\ ... \\ \sum_{i=1}^{n} y_i \cdot x_i^{\ k} \end{pmatrix} \qquad (3.9)$$

Premultiplying both sides by the transpose of the first matrix in Eq. (3.9) gives:

$$\begin{pmatrix} 1 & x_1 & ... & x_1^{\ k} \\ 1 & x_2 & ... & x_2^{\ k} \\ ... & ... & ... & ... \\ 1 & x_n & ... & x_n^{\ k} \end{pmatrix} \cdot \begin{pmatrix} a_0 \\ a_1 \\ ... \\ a_k \end{pmatrix} = \begin{pmatrix} y_1 \\ y_2 \\ ... \\ y_n \end{pmatrix} \qquad (3.10)$$

Finally the coefficients a_0, a_1, …, a_k can be found by inverting Eq. (3.10). The simplification of the given procedure will be:

$$a = (X^T X)^{-1} X^T Y \qquad (3.11)$$

Least squares polynomial approximation can also be applied to data sets with two independent variables. For example, consider n pairs of data in the form of (x_1, y_1, z_1), (x_2, y_2, z_2)…, (x_n, y_n, z_n) where x and y are two independent variables and z is the dependent variable. The same procedure applied for the single independent variable case is valid for the two independent variable case. In this case the coefficient equations will be:

$$\begin{pmatrix} a_0 \cdot n + a_1 \cdot \sum_{i=1}^{n} x_i + a_2 \cdot \sum_{i=1}^{n} y_i \\ + a_3 \cdot \sum_{i=1}^{n} x_i^2 + a_4 \cdot \sum_{i=1}^{n} y_i^2 + \\ a_5 \cdot \sum_{i=1}^{n} x_i \cdot y_i \end{pmatrix} = \sum_{i=1}^{n} z_i \qquad (3.12)$$

$$\begin{pmatrix} a_0 \cdot \sum_{i=1}^{n} x_i + a_1 \cdot \sum_{i=1}^{n} x_i^2 \\ + a_2 \cdot \sum_{i=1}^{n} x_i \cdot y_i + a_3 \cdot \sum_{i=1}^{n} x_i^3 \\ + a_4 \cdot \sum_{i=1}^{n} x_i \cdot y_i^2 + \\ a_5 \cdot \sum_{i=1}^{n} x_i^2 \cdot y_i \end{pmatrix} = \sum_{i=1}^{n} x_i \cdot z_i \qquad (3.13)$$

$$\begin{pmatrix} a_0 \cdot \sum_{i=1}^{n} x_i \cdot y_i + a_1 \cdot \sum_{i=1}^{n} x_i^2 \cdot y_i \\ + a_2 \cdot \sum_{i=1}^{n} x_i \cdot y_i^2 \\ + a_3 \cdot \sum_{i=1}^{n} x_i^3 \cdot y_i \\ + a_4 \cdot \sum_{i=1}^{n} x_i \cdot y_i^3 \\ + a_5 \cdot \sum_{i=1}^{n} x_i^2 \cdot y_i^2 \end{pmatrix} = \sum_{i=1}^{n} x_i \cdot y_i \cdot z_i \qquad (3.14)$$

The rest of the calculation procedure is the same as the single variable case.

3.2 Curve Fitting with Least Squares Polynomial Approximation to Thermophysical Properties of Refrigerants

For a vapor-compression household refrigerator system design, the thermophysical properties of the refrigerants are required in two main states: two-phase (i.e. saturated) state and superheated state (For subcooled liquid state,

saturated liquid properties have been used). The curve fitting polynomials are developed for these two subgroups.

3.2.1 Two-Phase Region Thermophysical Property Calculations

For the two-phase region property calculations of a refrigerant either the temperature or the pressure of the refrigerant is required. If the known parameter is the pressure, the computer algorithm will calculate the corresponding saturation temperature, and the rest of the thermophysical properties will be calculated by using the saturation temperature. The required properties will be calculated by a fifth degree polynomial derived by least squares polynomial approximation:

$$y = a_0 + a_1 \cdot T + a_2 \cdot T^2 + a_3 \cdot T^3 + a_4 \cdot T^4 + a_5 \cdot T^5 \qquad (3.15)$$

The required properties are the saturation pressure, surface tension and each of the following at the saturated liquid and vapor states:

- Density
- Enthalpy
- Entropy
- Dynamic Viscosity
- Constant-Pressure Specific Heat
- Thermal Conductivity

For a more accurate and fast algorithm each property is divided into subgroups with different temperature ranges. For the thermophysical property datasets of R-600 and R-600a the ASHRAE Handbook: Fundamentals-SI (Metric) Edition is used [27].

For the saturation temperature calculation the least squares polynomial approximation is rearranged with pressure as the known parameter:

$$T = a_0 + a_1 \cdot \ln(P) + a_2 \cdot \ln(P)^2 + a_3 \cdot \ln(P)^3 + a_4 \cdot \ln(P)^4 + a_5 \cdot \ln(P)^5 \qquad (3.16)$$

Similarly for the temperature calculations the polynomial coefficients are derived for different pressure ranges for better accuracy and fast calculation. Temperature calculations are done with logarithms of the pressure, which makes the dataset distribute more linearly, resulting in better accuracy.

3.2.2 Superheated Phase Region Thermophysical Property Calculations

For the superheated region property calculations, a least squares polynomial approximation with two independent variables is required. In this case a second degree polynomial with the independent variables x and y and the dependent variable z is obtained:

$$z = a_0 + a_1 \cdot x + a_2 \cdot y + a_3 \cdot x^2 + a_4 \cdot y^2 + a_5 \cdot x \cdot y \qquad (3.17)$$

The required superheated properties of a refrigerant for a household refrigeration system design are:

- Superheated Vapor Enthalpy
- Superheated Vapor Entropy
- Superheated Vapor Density

For the calculation of enthalpy the required independent variables are either temperature and pressure or entropy and pressure, for density calculations the independent variables are pressure and enthalpy. Similar to the two-phase region property calculations, for better accuracy and fast calculation the least squares approximation polynomials are obtained for incremental pressure ranges.

3.2.2.1 Solution for the Unknown Input Parameter

For superheated state thermophysical property calculations, both pressure and temperature of the refrigerant should be known. However due to the operating characteristics of vapor-compression refrigeration cycles, at some points only one input parameter and one thermophysical property is known. As a result the other input parameter is required to be calculated in order to obtain the other thermophysical properties.

In order to obtain the unknown input parameter (temperature or pressure), together with one thermophysical property (entropy, enthalpy or density) Eqn. (3.17) is required to be rearranged. Let the unknown input parameter be x and parameters y and z be known, then Eqn. (3.17) can be rearranged as:

$$b_1 \cdot x^2 + b_2 \cdot x + b_3 = 0 \tag{3.18}$$

where,

$$b_1 = a_3$$

$$b_2 = a_1 + a_5 \cdot y$$

$$b_3 = a_0 + a_2 \cdot y + a_4 \cdot y^2 - z$$

This is a simple 2$^{\text{nd}}$ order equation with two roots, one positive and the other negative or both positive. These roots can be found from the equation below:

$$x_{1,2} = \frac{-b_2 \pm \sqrt{b_2^2 - 4 \cdot b_1 \cdot b_3}}{2 \cdot b_1} \tag{3.19}$$

The positive root or if both roots are positive, the smaller one, is the required input parameter (x).

3.3 Refrigerant Thermophysical Property Calculation Equations and Coefficients

Thermophysical properties of the refrigerants R-600 and R-600a are required at saturated and superheated states for the design software. The required thermophysical properties are mentioned in Section 3.2. Properties are calculated at designated ranges for each refrigerant and the absolute percentage deviations from the tabulated data are given. For the absolute percentage deviation calculations the following equation is used:

$$\text{deviation} = \left| \frac{\text{Tabulated Value} - \text{Calculated Value}}{\text{Tabulated Value}} \right| \cdot 100 \qquad (3.20)$$

The absolute percentage deviation is calculated for each tabulated data and the mean of the calculated deviation is given in the tables.

3.3.1 Two-phase Region Equations

Temperatures between 220 K and 425 K and pressure values of 28 kPa to 3800 kPa are the range for R-600 property calculations whereas temperatures between 220 K and 407 K and pressure values of 5 kPa to 3650 kPa are the range for R-600a property calculations. For each property, calculation equation and the required coefficients are given. For simplicity, coefficients are rounded to two digits after decimal point, however 14 digits are used in the algorithm for better percentage deviations.

3.3.1.1 Coefficients of Thermophysical Property Equations

As mentioned before saturation temperature equation is the form:

$$T(P) = a_0 + a_1 \cdot \ln(P) + a_2 \cdot \ln(P)^2 + a_3 \cdot \ln(P)^3 + a_4 \cdot \ln(P)^4 + a_5 \cdot \ln(P)^5 \qquad (3.16)$$

where T is the saturation temperature of the refrigerant in K, and P is the saturation pressure in kPa. Polynomial coefficients and the maximum absolute percentage deviations of each refrigerant are tabulated in Tables 3.1 and 3.2.

Table 3.1 Coefficients of saturation temperature equation of R-600

P (kPa)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
5,02 - 90	1.90E+02	1.44E+01	-9.94E-01	6.59E-01	-9.25E-02	6.76E-03	4.80E-04
90 - 200	-3.20E+03	3.47E+03	-1.41E+03	2.87E+02	-2.92E+01	1.19E+00	3.50E-04
200 - 500	7.71E+02	-5.02E+02	1.83E+02	-3.21E+01	2.84E+00	-9.87E-02	4.00E-05
500 - 1000	-7.56E+03	5.93E+03	-1.81E+03	2.76E+02	-2.10E+01	6.42E-01	3.00E-05
1000 - 2000	-8.10E+03	5.74E+03	-1.58E+03	2.18E+02	-1.50E+01	4.12E-01	1.70E-04
2000 - 3000	-1.70E+05	1.08E+05	-2.72E+04	3.43E+03	-2.16E+02	5.44E+00	1.00E-08
3000 - 3796	-1.09E+04	4.08E+03	-4.96E+02	2.04E+01			2.61E-11

Table 3.2 Coefficients of saturation temperature equation of R-600a

P (kPa)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
28,31 - 90	2.63E+02	-8.75E+01	4.95E+01	-1.19E+01	1.46E+00	-7.00E-02	6.68E-05
90 - 200	1.61E+03	-1.45E+03	5.97E+02	-1.22E+02	1.24E+01	-5.03E-01	2.64E-05
200 - 500	-6.54E+02	7.35E+02	-2.49E+02	4.32E+01	-3.73E+00	1.30E-01	4.49E-05
500 - 1000	9.92E+02	-6.15E+02	1.95E+02	-3.02E+01	2.35E+00	-7.19E-02	3.29E-05
1000 - 2000	7.36E+03	-5.06E+03	1.43E+03	-2.03E+02	1.43E+01	-4.05E-01	1.16E-04
2000 - 3000	-3.39E+04	2.11E+04	-5.20E+03	6.40E+02	-3.91E+01	9.54E-01	1.61E-08
3000 - 3640	2.49E+02	-2.17E+01	5.00E+00				1.75E-13

Other thermophysical properties are calculated by using saturation temperature as the known independent variable. Equations are of the form:

$$X(T) = a_0 + a_1 \cdot T + a_2 \cdot T^2 + a_3 \cdot T^3 + a_4 \cdot T^4 + a_5 \cdot T^5 \qquad (3.21)$$

where T is the saturation temperature of the refrigerant in K, and X is the unknown thermophysical property. The calculated thermophysical parameters are:

- Saturation Pressure (MPa)
- Saturated Liquid Density (kg/m^3)
- Saturated Vapor Specific Volume (m^3/kg)
- Saturated Liquid and Vapor Enthalpy (kJ/kg)

- Saturated Liquid and Vapor Entropy (kJ/kg.K)
- Saturated Liquid and Vapor Constant Pressure Specific Heat (kJ/kg.K)
- Saturated Liquid and Vapor Dynamic Viscosity (µPa.s)
- Saturated Liquid and Vapor Thermal Conductivity (mW/m.K)
- Surface Tension (mN/m)

Polynomial coefficients and the maximum absolute percentage deviations of each refrigerant and thermophysical property for 50K intervals are given in Tables 3.3 through 3.30.

Table 3.3 Coefficients of saturation pressure equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	3.60E+00	-7.54E-02	6.20E-04	-2.46E-06	4.53E-09	-2.84E-12	7.30E-03
270 - 320	-6.92E+00	1.03E-01	-6.00E-04	1.72E-06	-2.65E-09	2.14E-12	1.84E-03
320 - 370	-2.19E+01	3.19E-01	-1.83E-03	5.21E-06	-7.55E-09	4.86E-12	1.18E-03
370 - 425	-1.68E+03	2.19E+01	-1.14E-01	2.97E-04	-3.87E-07	2.02E-10	3.28E-03

Table 3.4 Coefficients of saturation pressure equation of R-600a

T (K)	a0	a1	a2	a3	a4	a5	% deviation
220 - 270	-2.34E-12	3.91E-09	-2.08E-06	5.04E-04	-5.84E-02	2.64E+00	4.03E-03
270 - 320	-3.87E-12	6.29E-09	-3.55E-06	9.59E-04	-1.29E-01	6.95E+00	4.35E-04
320 - 370	1.99E-11	-3.26E-08	2.19E-05	-7.42E-03	1.25E+00	-8.44E+01	9.02E-04
370 - 407	-2.87E-10	5.64E-07	-4.42E-04	1.73E-01	-3.38E+01	2.63E+03	1.56E-03

Table 3.5 Coefficients of saturation liquid density equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	2.04E+04	-4.00E+02	3.26E+00	-1.33E-02	2.71E-05	-2.20E-08	2.40E-03
270 - 320	5.55E+02	9.93E+00	-1.12E-01	5.04E-04	-1.06E-06	8.44E-10	3.91E-03
320 - 370	-5.56E+04	8.17E+02	-4.73E+00	1.37E-02	-1.97E-05	1.13E-08	3.52E-03
370 - 425	5.09E+06	-6.55E+04	3.37E+02	-8.67E-01	1.12E-03	-5.74E-07	1.75E-02

Table 3.6 Coefficients of saturation liquid density equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	1.27E+04	-2.44E+02	2.00E+00	-8.21E-03	1.68E-05	-1.38E-08	2.84E-03
270 - 320	5.22E+03	-7.52E+01	5.04E-01	-1.71E-03	2.92E-06	-2.00E-09	4.30E-03
320 - 370	4.15E+04	-5.98E+02	3.52E+00	-1.04E-02	1.55E-05	-9.23E-09	2.77E-03
370 - 407	2.25E+08	-2.93E+06	1.52E+04	-3.97E+01	5.16E-02	-2.68E-05	5.00E-03

Table 3.7 Coefficients of saturated vapor specific volume equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	1.18E+04	-2.26E+02	1.74E+00	-6.69E-03	1.29E-05	-1.00E-08	1.37E-02
270 - 320	6.41E+02	-9.99E+00	6.26E-02	-1.97E-04	3.11E-07	-1.97E-10	1.61E-02
320 - 370	7.33E+01	-9.30E-01	4.75E-03	-1.22E-05	1.57E-08	-8.12E-12	1.83E-02
370 - 425	1.21E+03	-1.54E+01	7.84E-02	-2.00E-04	2.54E-07	-1.29E-10	1.90E-02

Table 3.8 Coefficients of saturated vapor specific volume equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	5.70E+03	-1.09E+02	8.36E-01	-3.22E-03	6.21E-06	-4.80E-09	4.36E-03
270 - 320	4.06E+02	-6.37E+00	4.02E-02	-1.28E-04	2.03E-07	-1.30E-10	2.10E-02
320 - 370	9.86E+01	-1.35E+00	7.43E-03	-2.06E-05	2.86E-08	-1.60E-11	1.59E-02
370 - 407	4.42E+03	-5.74E+01	2.99E-01	-7.77E-04	1.01E-06	-5.26E-10	1.79E-02

Table 3.9 Coefficients of saturated liquid enthalpy equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-2.32E+03	4.23E+01	-3.29E-01	1.34E-03	-2.70E-06	2.19E-09	1.71E-03
270 - 320	-3.19E+03	5.10E+01	-3.37E-01	1.16E-03	-1.98E-06	1.36E-09	9.79E-04
320 - 370	-7.34E+03	1.06E+02	-6.23E-01	1.86E-03	-2.79E-06	1.68E-09	5.19E-04
370 - 425	-2.73E+07	3.49E+05	-1.78E+03	4.53E+00	-5.77E-03	2.94E-06	2.55E-03

Table 3.10 Coefficients of saturated liquid enthalpy equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-4.72E+03	9.08E+01	-7.19E-01	2.90E-03	-5.81E-06	4.67E-09	1.21E-02
270 - 320	-1.55E+04	2.62E+02	-1.78E+00	6.09E-03	-1.04E-05	7.07E-09	1.03E-02
320 - 370	-1.17E+03	2.73E+01	-2.29E-01	9.14E-04	-1.71E-06	1.23E-09	3.66E-03
370 - 407	-1.02E+08	1.33E+06	-6.93E+03	1.80E+01	-2.35E-02	1.22E-05	1.55E-02

Table 3.11 Coefficients of saturated vapor enthalpy equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	1.04E+03	-1.53E+01	1.37E-01	-5.72E-04	1.21E-06	-1.03E-09	2.95E-04
270 - 320	2.86E+03	-4.26E+01	2.92E-01	-9.73E-04	1.63E-06	-1.10E-09	4.27E-04
320 - 370	1.40E+04	-2.05E+02	1.24E+00	-3.71E-03	5.58E-06	-3.37E-09	1.73E-04
370 - 425	6.11E+05	-8.05E+03	4.24E+01	-1.12E-01	1.48E-04	-7.79E-08	1.87E-04

Table 3.12 Coefficients of saturated vapor enthalpy equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	1.22E+03	-1.83E+01	1.52E-01	-5.96E-04	1.18E-06	-9.46E-10	2.81E-04
270 - 320	1.20E+03	-1.57E+01	1.18E-01	-4.16E-04	7.47E-07	-5.47E-10	3.07E-04
320 - 370	2.74E+04	-4.08E+02	2.47E+00	-7.45E-03	1.13E-05	-6.86E-09	3.70E-04
370 - 407	1.07E+08	-1.39E+06	7.23E+03	-1.88E+01	2.45E-02	-1.27E-05	1.24E-02

Table 3.13 Coefficients of saturated liquid entropy equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	8.93E+00	-2.16E-01	1.90E-03	-7.91E-06	1.62E-08	-1.32E-11	1.74E-03
270 - 320	4.07E+00	-9.09E-02	7.11E-04	-2.48E-06	4.25E-09	-2.88E-12	2.10E-03
320 - 370	3.00E+01	-4.48E-01	2.65E-03	-7.63E-06	1.09E-08	-6.22E-12	1.65E-03
370 - 425	-6.43E+04	8.20E+02	-4.18E+00	1.07E-02	-1.36E-05	6.91E-09	1.62E-02

Table 3.14 Coefficients of saturated liquid entropy equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	7.46E-01	-4.45E-02	4.76E-04	-1.99E-06	4.01E-09	-3.17E-12	2.07E-03
270 - 320	-2.34E+01	3.83E-01	-2.55E-03	8.71E-06	-1.50E-08	1.03E-11	1.61E-03
320 - 370	-6.00E+01	8.71E-01	-5.07E-03	1.50E-05	-2.21E-08	1.32E-11	1.25E-03
370 - 407	-2.50E+05	3.25E+03	-1.69E+01	4.40E-02	-5.72E-05	2.98E-08	1.02E-02

Table 3.15 Coefficients of saturated vapor entropy equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-1.74E+00	1.38E-01	-1.41E-03	6.52E-06	-1.43E-08	1.23E-11	5.67E-04
270 - 320	5.59E-01	5.13E-02	-4.40E-04	1.67E-06	-2.95E-09	1.99E-12	1.02E-03
320 - 370	5.99E+01	-8.41E-01	4.92E-03	-1.45E-05	2.13E-08	-1.26E-11	6.85E-04
370 - 425	4.32E+03	-5.66E+01	2.96E-01	-7.77E-04	1.02E-06	-5.33E-10	2.99E-03

Table 3.16 Coefficients of saturated vapor entropy equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	9.65E-01	7.57E-02	-8.74E-04	4.23E-06	-9.48E-09	8.18E-12	8.58E-04
270 - 320	4.21E+00	-1.07E-02	-3.25E-05	3.34E-07	-7.53E-10	5.47E-13	1.10E-03
320 - 370	4.42E+01	-6.35E-01	3.87E-03	-1.18E-05	1.83E-08	-1.13E-11	3.95E-04
370 - 407	2.61E+05	-3.39E+03	1.77E+01	-4.60E-02	5.98E-05	-3.11E-08	8.75E-03

Table 3.17 Coefficients of saturated liquid constant-pressure specific heat

equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-9.46E+01	2.00E+00	-1.65E-02	6.80E-05	-1.40E-07	1.15E-10	7.11E-03
270 - 320	5.49E+01	-9.66E-01	6.98E-03	-2.50E-05	4.45E-08	-3.15E-11	8.71E-03
320 - 370	-4.21E+02	6.37E+00	-3.85E-02	1.17E-04	-1.77E-07	1.08E-10	8.69E-03
370 - 425	-2.39E+05	3.11E+03	-1.62E+01	4.22E-02	-5.49E-05	2.86E-08	5.93E-03

Table 3.18 Coefficients of saturated liquid constant-pressure specific heat

equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-2.13E+01	5.04E-01	-4.40E-03	1.92E-05	-4.16E-08	3.59E-11	7.82E-03
270 - 320	-1.09E+02	1.92E+00	-1.34E-02	4.67E-05	-8.15E-08	5.71E-11	1.10E-02
320 - 370	-3.82E+03	5.69E+01	-3.38E-01	1.01E-03	-1.50E-06	8.91E-10	9.20E-03
370 - 407	-9.63E+06	1.26E+05	-6.61E+02	1.73E+00	-2.27E-03	1.19E-06	5.74E-03

Table 3.19 Coefficients of saturated vapor constant-pressure specific heat

equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-4.95E+01	1.01E+00	-8.12E-03	3.25E-05	-6.49E-08	5.18E-11	9.51E-03
270 - 320	2.09E+02	-3.53E+00	2.39E-02	-8.07E-05	1.36E-07	-9.14E-11	1.18E-02
320 - 370	-1.02E+03	1.53E+01	-9.14E-02	2.74E-04	-4.10E-07	2.46E-10	5.87E-03
370 - 425	-2.37E+05	3.11E+03	-1.63E+01	4.27E-02	-5.59E-05	2.93E-08	1.19E-02

Table 3.20 Coefficients of saturated vapor constant-pressure specific heat

equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-6.82E+01	1.44E+00	-1.19E-02	4.95E-05	-1.03E-07	8.50E-11	1.46E-02
270 - 320	-1.61E+02	2.82E+00	-1.96E-02	6.81E-05	-1.18E-07	8.26E-11	1.45E-02
320 - 370	-4.35E+03	6.53E+01	-3.92E-01	1.18E-03	-1.77E-06	1.06E-09	7.89E-03
370 - 407	-1.28E+07	1.67E+05	-8.76E+02	2.30E+00	-3.01E-03	1.58E-06	1.77E-02

Table 3.21 Coefficients of saturated liquid dynamic viscosity equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	2.89E+04	-5.04E+02	3.66E+00	-1.36E-02	2.55E-05	-1.94E-08	5.95E-03
270 - 320	2.50E+04	-3.94E+02	2.55E+00	-8.40E-03	1.39E-05	-9.29E-09	1.27E-02
320 - 370	-1.00E+05	1.46E+03	-8.47E+00	2.45E-02	-3.54E-05	2.04E-08	2.10E-02
370 - 425	1.53E+06	-1.97E+04	1.01E+02	-2.59E-01	3.32E-04	-1.70E-07	1.27E-02

Table 3.22 Coefficients of saturated liquid dynamic viscosity equation of

R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	2.82E+04	-4.69E+02	3.26E+00	-1.17E-02	2.12E-05	-1.57E-08	7.84E-03
270 - 320	2.50E+04	-3.88E+02	2.49E+00	-8.17E-03	1.35E-05	-9.04E-09	1.55E-02
320 - 370	-1.40E+04	2.18E+02	-1.31E+00	3.84E-03	-5.56E-06	3.19E-09	1.50E-02
370 - 407	3.42E+06	-4.48E+04	2.34E+02	-6.14E-01	8.03E-04	-4.21E-07	8.99E-03

Table 3.23 Coefficients of saturated vapor dynamic viscosity equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-1.35E+01	1.02E-01	1.06E-03	-1.13E-05	3.69E-08	-4.10E-11	1.62E-02
270 - 320	4.66E+02	-7.59E+00	4.97E-02	-1.62E-04	2.62E-07	-1.69E-10	3.11E-02
320 - 370	3.97E+03	-5.71E+01	3.28E-01	-9.40E-04	1.34E-06	-7.68E-10	2.63E-02
370 - 425	-8.08E+05	1.04E+04	-5.32E+01	1.37E-01	-1.75E-04	8.97E-08	1.27E-02

Table 3.24 Coefficients of saturated vapor dynamic viscosity equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	5.47E+02	-1.14E+01	9.58E-02	-4.00E-04	8.32E-07	-6.90E-10	3.84E-02
270 - 320	6.95E-01	-1.01E-01	1.69E-03	-8.60E-06	1.93E-08	-1.58E-11	3.34E-02
320 - 370	7.23E+04	-1.05E+03	6.09E+00	-1.77E-02	2.56E-05	-1.48E-08	1.12E-02
370 - 407	-6.61E+04	1.13E+03	-7.27E+00	2.27E-02	-3.44E-05	2.05E-08	1.47E-02

Table 3.25 Coefficients of saturated liquid thermal conductivity equation of

R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-4.82E+03	1.06E+02	-8.85E-01	3.66E-03	-7.50E-06	6.12E-09	1.42E-02
270 - 320	2.52E+01	7.87E+00	-8.76E-02	3.99E-04	-8.43E-07	6.81E-10	2.01E-02
320 - 370	6.01E+04	-8.74E+02	5.10E+00	-1.49E-02	2.16E-05	-1.26E-08	2.39E-02
370 - 425	-1.10E+05	1.60E+03	-9.20E+00	2.61E-02	-3.67E-05	2.05E-08	1.79E-02

Table 3.26 Coefficients of saturated liquid thermal conductivity equation of

R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	2.55E+03	-4.72E+01	3.69E-01	-1.43E-03	2.70E-06	-2.01E-09	1.65E-02
270 - 320	-2.41E+04	4.10E+02	-2.77E+00	9.31E-03	-1.57E-05	1.05E-08	2.82E-02
320 - 370	-1.40E+04	2.06E+02	-1.19E+00	3.45E-03	-4.97E-06	2.85E-09	2.80E-02
370 - 407	-4.18E+06	5.51E+04	-2.91E+02	7.67E-01	-1.01E-03	5.33E-07	3.33E-02

Table 3.27 Coefficients of saturated vapor thermal conductivity equation of

R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-2.55E+03	5.17E+01	-4.19E-01	1.69E-03	-3.42E-06	2.76E-09	9.17E-03
270 - 320	-1.15E+03	1.92E+01	-1.27E-01	4.23E-04	-7.01E-07	4.65E-10	1.28E-02
320 - 370	-3.51E+03	5.20E+01	-3.08E-01	9.12E-04	-1.35E-06	8.03E-10	8.18E-03
370 - 425	-1.24E+06	1.61E+04	-8.40E+01	2.19E-01	-2.85E-04	1.49E-07	3.49E-02

Table 3.28 Coefficients of saturated vapor thermal conductivity equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	1.82E+03	-3.72E+01	3.04E-01	-1.24E-03	2.53E-06	-2.06E-09	1.23E-02
270 - 320	-7.21E+02	1.26E+01	-8.74E-02	3.04E-04	-5.26E-07	3.64E-10	1.30E-02
320 - 370	-1.01E+04	1.51E+02	-8.99E-01	2.68E-03	-4.01E-06	2.40E-09	7.41E-03
370 - 407	-9.73E+06	1.28E+05	-6.72E+02	1.77E+00	-2.32E-03	1.22E-06	6.28E-02

Table 3.29 Coefficients of surface tension equation of R-600

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-1.87E+03	3.90E+01	-3.19E-01	1.29E-03	-2.62E-06	2.12E-09	1.29E-02
270 - 320	-3.17E+02	5.55E+00	-3.47E-02	1.04E-04	-1.53E-07	8.84E-11	1.91E-02
320 - 370	-3.61E+03	5.22E+01	-2.99E-01	8.53E-04	-1.22E-06	6.94E-10	1.53E-02
370 - 425	-5.71E+04	7.29E+02	-3.72E+00	9.47E-03	-1.21E-05	6.15E-09	1.18E-01

Table 3.30 Coefficients of surface tension equation of R-600a

T (K)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
220 - 270	-5.34E+02	1.20E+01	-1.01E-01	4.16E-04	-8.59E-07	7.08E-10	8.12E-03
270 - 320	6.37E+02	-1.04E+01	7.16E-02	-2.49E-04	4.32E-07	-2.99E-10	2.01E-02
320 - 370	-7.96E+03	1.15E+02	-6.66E-01	1.92E-03	-2.77E-06	1.59E-09	3.66E-02
370 - 407	-9.68E+04	1.27E+03	-6.63E+00	1.73E-02	-2.27E-05	1.19E-08	3.45E-01

3.3.2 Superheated Region Equations

For the superheated region, the analysis is made in the range of 101.325 kPa to 2 MPa considering 200 kPa intervals with the data of an increment of 20 kPa.

3.3.2.1 Pressure-Temperature-Enthalpy Equation

Superheated vapor pressure-temperature-enthalpy equation is the form:

$$i(P,T) = a_0 + a_1 \cdot P + a_2 \cdot T + a_3 \cdot P^2 + a_4 \cdot T^2 + a_5 \cdot P \cdot T \qquad (3.22)$$

where T is the temperature of the refrigerant in K, P is the pressure of the refrigerant in MPa and i is the superheated vapor enthalpy in kJ/kg. Polynomial coefficients and the maximum absolute percentage deviations of each refrigerant for 200 kPa intervals are given in Tables 3.31 and 3.32

Table 3.31 Coefficients of superheated vapor pressure-temperature-enthalpy

equation of R-600

P(kPa)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
101,325 - 200	3.03E+02	-9.33E+01	4.89E-01	-9.80E-01	2.06E-03	1.74E-01	1.21E-02
200 - 400	2.86E+02	-9.16E+01	5.78E-01	2.07E-01	1.95E-03	1.67E-01	1.70E-02
400 - 600	2.60E+02	-9.23E+01	7.12E-01	-2.11E+00	1.78E-03	1.72E-01	1.90E-02
600 - 800	2.32E+02	-9.49E+01	8.57E-01	-3.84E+00	1.59E-03	1.83E-01	2.31E-02
800 - 1000	2.00E+02	-9.88E+01	1.02E+00	-4.66E+00	1.38E-03	1.97E-01	2.74E-02
1000 - 1200	1.65E+02	-1.05E+02	1.21E+00	-4.48E+00	1.13E-03	2.13E-01	3.14E-02
1200 - 1400	1.20E+02	-1.09E+02	1.43E+00	-4.90E+00	8.45E-04	2.25E-01	3.62E-02
1400 - 1600	7.85E+01	-1.32E+02	1.71E+00	1.09E-01	4.89E-04	2.46E-01	4.26E-02
1600 - 1800	1.75E+01	-1.34E+02	2.00E+00	-5.56E+00	5.88E-05	2.90E-01	5.12E-02
1800 - 2000	-7.38E+01	-1.34E+02	2.42E+00	-6.65E+00	-4.52E-04	3.01E-01	5.89E-02

Table 3.32 Coefficients of superheated vapor pressure-temperature-enthalpy

equation of R-600a

P(kPa)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
101,325 - 200	2.95E+02	-8.38E+01	3.75E-01	8.90E-01	2.19E-03	1.56E-01	1.29E-02
200 - 400	2.80E+02	-8.20E+01	4.55E-01	-1.30E+00	2.09E-03	1.53E-01	1.85E-02
400 - 600	2.57E+02	-8.33E+01	5.78E-01	-9.22E-01	1.93E-03	1.54E-01	2.07E-02
600 - 800	2.32E+02	-8.56E+01	7.09E-01	-2.16E+00	1.76E-03	1.63E-01	2.59E-02
800 - 1000	2.04E+02	-8.85E+01	8.57E-01	-2.95E+00	1.57E-03	1.73E-01	3.15E-02
1000 - 1200	1.72E+02	-9.50E+01	1.03E+00	-2.15E+00	1.35E-03	1.84E-01	3.80E-02
1200 - 1400	1.42E+02	-1.12E+02	1.23E+00	1.16E+00	1.08E-03	2.03E-01	4.66E-02
1400 - 1600	9.57E+01	-1.10E+02	1.45E+00	-4.60E+00	7.60E-04	2.35E-01	5.71E-02
1600 - 1800	2.92E+01	-1.07E+02	1.75E+00	-6.00E+00	3.89E-04	2.41E-01	6.69E-02
1800 - 2000	-7.72E-01	-1.51E+02	2.08E+00	1.53E-01	-1.07E-04	2.90E-01	8.44E-02

3.3.2.2 Pressure-Enthalpy-Density Equation

Superheated vapor pressure-enthalpy-density equation is the form:

$$\rho(i, P) = a_0 + a_1 \cdot \ln(i) + a_2 \cdot P + a_3 \cdot (\ln(i))^2 + a_4 \cdot P^2 + a_5 \cdot \ln(i) \cdot P \qquad (3.23)$$

where i is the superheated vapor enthalpy in kJ/kg, P is the pressure of the refrigerant in MPa and ρ is the superheated vapor density in kg/m^3. Polynomial coefficients and the maximum absolute percentage deviations of each refrigerant for 200 kPa intervals are given in Tables 3.31 and 3.32

Table 3.33 Coefficients of superheated vapor pressure-enthalpy-density equation

of R-600

P(kPa)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
101,325 - 200	1.44E+02	-4.34E+01	1.70E+02	3.27E+00	1.71E+00	-2.26E+01	6.89E-01
200 - 400	3.02E+02	-9.11E+01	1.78E+02	6.87E+00	1.50E+00	-2.38E+01	5.99E-01
400 - 600	5.74E+02	-1.74E+02	1.96E+02	1.31E+01	2.07E+00	-2.66E+01	4.39E-01
600 - 800	9.16E+02	-2.78E+02	2.19E+02	2.10E+01	2.37E+00	-3.00E+01	4.01E-01
800 - 1000	1.34E+03	-4.07E+02	2.43E+02	3.09E+01	2.56E+00	-3.37E+01	3.90E-01
1000 - 1200	1.86E+03	-5.67E+02	2.70E+02	4.30E+01	2.64E+00	-3.77E+01	3.94E-01
1200 - 1400	2.52E+03	-7.65E+02	2.96E+02	5.82E+01	2.70E+00	-4.16E+01	4.00E-01
1400 - 1600	3.34E+03	-1.02E+03	3.34E+02	7.74E+01	1.18E+00	-4.66E+01	4.26E-01
1600 - 1800	4.35E+03	-1.33E+03	3.81E+02	1.01E+02	3.38E+00	-5.46E+01	4.68E-01
1800 - 2000	5.60E+03	-1.71E+03	4.10E+02	1.30E+02	3.35E+00	-5.89E+01	4.91E-01

Table 3.34 Coefficients of superheated vapor pressure-enthalpy-density equation

of R-600a

P(kPa)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
101,325 - 200	1.39E+02	-4.23E+01	1.67E+02	3.21E+00	1.61E+00	-2.23E+01	8.05E-01
200 - 400	2.86E+02	-8.72E+01	1.76E+02	6.64E+00	1.87E+00	-2.37E+01	7.24E-01
400 - 600	5.41E+02	-1.65E+02	1.91E+02	1.26E+01	1.65E+00	-2.59E+01	5.28E-01
600 - 800	8.54E+02	-2.61E+02	2.10E+02	1.99E+01	2.04E+00	-2.89E+01	4.85E-01
800 - 1000	1.24E+03	-3.79E+02	2.32E+02	2.90E+01	2.30E+00	-3.21E+01	4.80E-01
1000 - 1200	1.72E+03	-5.26E+02	2.55E+02	4.02E+01	2.03E+00	-3.56E+01	4.92E-01
1200 - 1400	2.31E+03	-7.10E+02	2.88E+02	5.44E+01	7.80E-01	-3.99E+01	5.26E-01
1400 - 1600	3.03E+03	-9.33E+02	3.24E+02	7.18E+01	3.15E+00	-4.63E+01	5.76E-01
1600 - 1800	3.92E+03	-1.20E+03	3.44E+02	9.25E+01	3.37E+00	-4.95E+01	6.08E-01
1800 - 2000	5.04E+03	-1.56E+03	4.13E+02	1.20E+02	1.28E+00	-5.85E+01	6.94E-01

3.3.2.3 Pressure-Entropy-Enthalpy Equation

Superheated vapor pressure-entropy-enthalpy equation is the form:

$$i(s,P) = a_0 + a_1 \cdot s + a_2 \cdot P + a_3 \cdot s^2 + a_4 \cdot P^2 + a_5 \cdot s \cdot P \qquad (3.24)$$

where, s is the superheated vapor entropy in kJ/kg.K, P is the pressure of the refrigerant in MPa and i is the superheated vapor enthalpy in kJ/kg. Polynomial coefficients and the maximum absolute percentage deviations of each refrigerant for 200 kPa intervals are given in Tables 3.31 and 3.32

Table 3.35 Coefficients of superheated vapor pressure-entropy-enthalpy

equation of R-600

P(kPa)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
101,325 - 200	4.57E+02	-1.88E+02	1.65E+02	9.11E+01	-1.18E+03	1.87E+02	2.90E-02
200 - 400	4.36E+02	-1.67E+02	8.18E+01	9.07E+01	-3.04E+02	9.78E+01	2.33E-02
400 - 600	4.17E+02	-1.50E+02	4.32E+01	9.02E+01	-1.07E+02	5.99E+01	9.60E-03
600 - 800	4.02E+02	-1.36E+02	2.69E+01	8.94E+01	-5.52E+01	4.44E+01	8.33E-03
800 - 1000	3.89E+02	-1.24E+02	1.73E+01	8.85E+01	-3.37E+01	3.60E+01	7.68E-03
1000 - 1200	3.78E+02	-1.14E+02	9.14E+00	8.76E+01	-2.22E+01	3.08E+01	6.84E-03
1200 - 1400	-1.12E-03	-4.93E+01	4.52E+02	8.17E+01	-1.65E+02	5.99E+00	6.18E-02
1400 - 1600	-6.42E-04	-4.94E+01	3.92E+02	8.18E+01	-1.27E+02	8.29E+00	4.64E-02
1600 - 1800	3.43E+02	-8.43E+01	3.18E-01	8.43E+01	-9.87E+00	2.24E+01	6.70E-03
1800 - 2000	3.29E+02	-7.43E+01	-2.92E-01	8.31E+01	-8.18E+00	2.05E+01	6.54E-03

Table 3.36 Coefficients of superheated vapor pressure-entropy-enthalpy

equation of R-600a

P(kPa)	a_0	a_1	a_2	a_3	a_4	a_5	% deviation
101,325 - 200	4.20E+02	-1.80E+02	1.69E+02	9.15E+01	-1.16E+03	1.88E+02	3.12E-02
200 - 400	3.99E+02	-1.59E+02	8.67E+01	9.11E+01	-2.99E+02	9.71E+01	2.48E-02
400 - 600	3.80E+02	-1.41E+02	4.58E+01	9.04E+01	-1.05E+02	5.95E+01	1.09E-02
600 - 800	3.66E+02	-1.28E+02	2.96E+01	8.97E+01	-5.38E+01	4.38E+01	9.56E-03
800 - 1000	3.54E+02	-1.16E+02	2.04E+01	8.88E+01	-3.30E+01	3.53E+01	8.65E-03
1000 - 1200	3.44E+02	-1.06E+02	1.28E+01	8.79E+01	-2.17E+01	2.99E+01	8.55E-03
1200 - 1400	3.34E+02	-9.61E+01	7.42E+00	8.70E+01	-1.52E+01	2.62E+01	8.87E-03
1400 - 1600	3.23E+02	-8.73E+01	5.08E+00	8.60E+01	-1.21E+01	2.38E+01	8.80E-03
1600 - 1800	3.09E+02	-7.77E+01	5.97E+00	8.49E+01	-1.02E+01	2.14E+01	8.67E-03
1800 - 2000	3.04E+02	-6.87E+01	-1.45E+00	8.36E+01	-7.28E+00	2.03E+01	9.15E-03

CHAPTER 4

COMPRESSOR

Almost all of the household refrigeration systems are mechanical vapor compression type refrigeration systems. For this type of refrigerators compressor is the most important element of the system. Not only most of the energy consumption of the systems comes from the compressor, but also it is the main component that determines the efficiency and power consumption of the system.

There are five types of compressors:

- Reciprocating compressors
- Rotary compressors
- Scroll compressors
- Screw compressors
- Centrifugal compressors

Reciprocating compressors are positive displacement type compressors and they were the first ones that were used for refrigeration systems [3]. Rotary compressors have been also used for domestic refrigeration systems because of their low noise and vibration levels compared to reciprocating compressors [28]. Scroll compressors are also positive displacement compressors with few moving parts and low noise and vibration characteristics. However the possibility of leakage of the compressed gas adversely affects the volumetric efficiency and makes it difficult to control the system volumetric efficiency. Screw type compressors are divided into two groups: single screw and twin screw. Pressure ration of the compressor may not match the rest of the compression systems thus it may require extra work which will result in a drop in the overall system

efficiency. Centrifugal compressors are the only non-positive displacement compressors among the five. They are generally used for large capacity refrigerators used in air conditioning like chillers or rooftop units. More detailed information about the compressor types that are used in refrigeration systems can be found in the literature [3, 28].

Among the five compressor types reciprocating compressors are the most widely used ones because of their ease of use, controllability and efficiency. Basically it is a piston moving back and forth in a cylinder. In this study, a reciprocating compressor is chosen as the compression device of the refrigeration system

4.1 Principle Dimensions of a Reciprocating Compressor

Working mechanism of a reciprocating compressor consists of mainly four parts: suction, compression, discharge and re-expansion. First, the piston moves outwards until the cylinder pressure becomes equal to the pressure at the suction valve, then the piston movement continues to begin the suction of the vapor coming from the evaporator. Secondly, the suction valve closes and the compression process begins the high pressure vapor at the end of the compression process is discharged to the condenser by opening the discharge valve. Then the whole process repeats itself.

Principle dimensions of a reciprocating compressor are, bore diameter (D), stroke (L), clearance volume (V_c), swept volume (V_p) and suction volume (V_s) as shown on Figures 4.1 and 4.2.
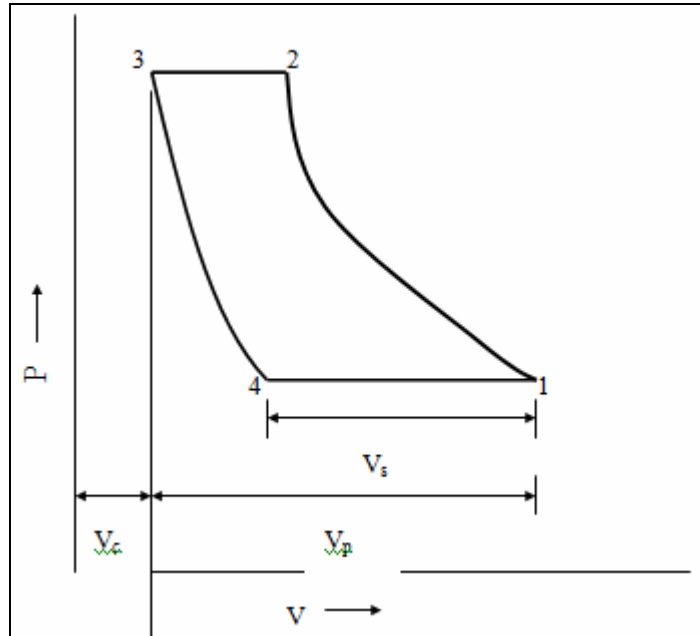
Figure 4.1 P-V Diagram of a reciprocating compressor [2].
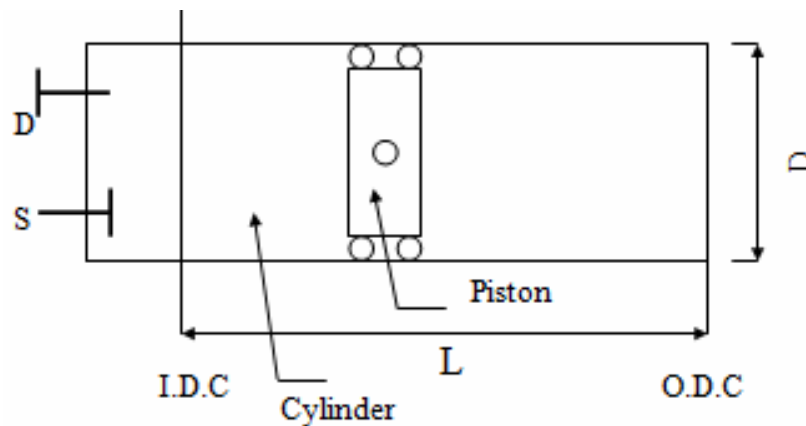
Figure 4.2. Cylinder and piston arrangement of a reciprocating compressor [2].

As it is shown on Figure 4.2 bore diameter (D) is the diameter of the cylinder, whereas the stroke is the distance that the piston covers during compression and re-expansion (i.e. each stroke). Volume covered for each stroke is called the swept volume (V_p).

$$V_p = V_1 - V_3 = \frac{\pi \cdot D^2}{4} \cdot L \tag{4.1}$$

Clearance volume is the result of the distance left between the top dead center (TDC) of the compressor and the valve plate in order to avoid a clash between the piston and the valves.

$$V_c = C \cdot V_p \qquad (4.2)$$

where C is the clearance factor.

The suction volume is the volume between states 4 and 1.

$$V_s = V_1 - V_4 \qquad (4.3)$$

Figure 4.1 represents the P-V diagram of an ideal reciprocating compressor. However in an actual compression cycle there will be pressure losses at the suction and discharge valves due to friction. As a result higher (P_h) and lower (P_l) working pressures of the compressor are slightly different than suction (P_s) and discharge (P_d) pressures. P-V diagram of an actual cycle is shown on Figure 4.3.
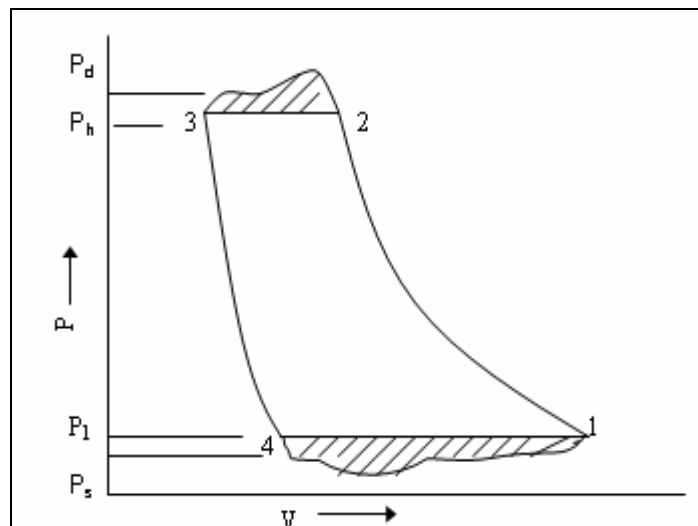
Figure 4.3 Pressure differences at discharge and suction valve [2].

Suction pressure is also the vapor pressure at the evaporator exit.

$$P_s = f_{ps} \cdot P_l \tag{4.4}$$

where f_{ps} is the pressure fraction through suction valve and generally about 0.95.

Similarly the discharge pressure is equal to the condenser inlet pressure. The discharge pressure fraction f_{pd} is also about 0.95.

$$P_d = \frac{P_h}{f_{pd}} \tag{4.5}$$

4.2 Work Done in a Reciprocating Compressor

As shown in Figure 4.1 working mechanism of a reciprocating compressor consists of four main states either with different pressures, or different volumes or both. As a result the work done in a reciprocating compressor can be calculated by simply taking the integral of P.dV between each state:

$$
\begin{aligned}
W = \oint P \, dV &= \int_1^2 P \cdot dV + \int_2^3 P \cdot dV + \int_3^4 P \cdot dV + \int_4^1 P \cdot dV \\
&= \int_1^2 P \cdot dV + \int_3^4 P \cdot dV + P_h \cdot (V_3 - V_2) + P_l \cdot (V_1 - V_4)
\end{aligned}
\tag{4.6}
$$

If the specific volume v and the mass m of the refrigerant vapor are known the power consumption can also be represented as:

$$W = \oint P \, dV = -\oint V \, dP = -m \cdot \oint v \, dP \tag{4.7}$$

and the specific work can be calculated as:

$$w = -\oint \nu \cdot dP \qquad\qquad (4.8)$$

4.3 Reciprocating Compressor Processes

As mentioned before the basic processes in a reciprocating compressor are compression, discharge, re-expansion and suction.

4.3.1 Compression Processes

There are three types of compression: isothermal, isentropic and polytropic. In isothermal compression it is assumed that the temperature remains constant during compression. For isentropic compression it is assumed that the entropy will remain constant during the process. Both isothermal and isentropic processes are ideal cases and not valid in actual refrigeration. However they are used for efficiency analysis thus will be examined in this study. For the actual cycle both temperature and entropy are variable as a result the actual compression is a polytropic process. A P-V diagram showing different compression processes is given in Figure 4.4.

4.3.1.1 Isothermal Compression Process

As mentioned before the temperature of the isothermal compression process is constant and the lower and higher working pressures are known. By knowing pressure and temperature of a state all of its properties can be calculated, thus the work done during compression can be calculated from:

$$q = T \cdot (s_2 - s_1) = (i_2 - i_1) - w \qquad\qquad (4.9)$$
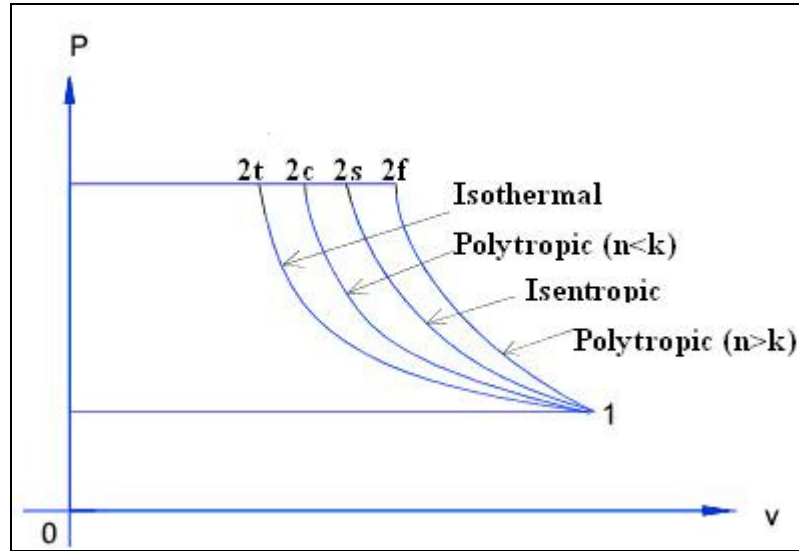
Figure 4.4 P-V diagram for different compression processes

where s_1 and s_2 are entropies and i_1 and i_2 are enthalpies at the given states. Please note that enthalpy is represented as i in the equations instead of h representation.

4.3.1.2 Isentropic Compression Process

For the isentropic compression the entropy remains constant during the whole process. For this type of compression the isentropic index of compression k can be used:

$$P \cdot v^k = P_1 \cdot v_1^{\ k} = P_2 \cdot v_2^{\ k}$$ (4.10)

where

$$k = \frac{\ln\left(\dfrac{P_2}{P_1}\right)}{\ln\left(\dfrac{v_1}{v_2}\right)} \cong \frac{c_p}{c_v}$$ (4.11)

After calculating the isentropic index of compression the specific work can be calculated as:

$$w = -\int v \cdot dP = -\int v_1 \cdot \left(\frac{P_1}{P_2}\right)^{\frac{1}{k}} \cdot dP = -\frac{k}{k-1} \cdot P_1 \cdot v_1 \cdot \left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right] \quad (4.12)$$

4.3.1.3 Polytropic Compression Process

Calculation of the polytropic compression is very similar to the calculation of isentropic compression. The only difference is instead of a isentropic index of compression there is polytropic index of compression n:

$$P \cdot v^n = P_1 \cdot v_1^{\ n} = P_2 \cdot v_2^{\ n} \quad (4.13)$$

where

$$n = \frac{\ln\left(\dfrac{P_2}{P_1}\right)}{\ln\left(\dfrac{v_1}{v_2}\right)} \quad (4.14)$$

Polytropic index of compression can vary with external cooling. As shown in figure 4.4 if enough cooling is supplied it can be smaller than isentropic index. However if frictional heat gain is higher than the cooling (i.e. there is not sufficient cooling or no cooling) it can be higher than isentropic index. Specific work of polytropic compression can be calculated as:

$$w = -\int v \cdot dP = -\int v_1 \cdot \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} \cdot dP = -\frac{n}{n-1} \cdot P_1 \cdot v_1 \cdot \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \quad (4.15)$$

4.3.2 Discharge Process

During the discharge process the refrigerant vapor is pushed out of the cylinder to the condenser via the discharge valve. The process occurs very quickly and the exposed cylinder area is very small, which results in a very small heat transfer from the vapor that it can be neglected. As a result the discharge process can be assumed as isenthalpic, i.e.:

$$i_3 = i_2 \qquad\qquad (4.16)$$

Please note that the state of the refrigerant at the discharge should be superheated vapor.

4.3.3 Re-Expansion Processes

Similar to the compression process there are three types of re-expansion: isothermal, isentropic and polytropic. Again isothermal and isentropic re-expansion processes are ideal whereas the actual process can be taken as a polytropic re-expansion.

4.3.3.1 Isothermal Re-Expansion Process

Similar to the isothermal compression work done during re-expansion can be calculated from:

$$q = T \cdot (s_4 - s_3) = (i_4 - i_3) - w \qquad\qquad (4.17)$$

where s_1 and s_2 are entropies and i_1 and i_2 are enthalpies at the given states.

4.3.3.2 Isentropic Re-Expansion Process

For the isentropic re-expansion the entropy remains constant during the whole process. For this type of re-expansion the isentropic index of re-expansion k can be used:

$$P \cdot v^k = P_3 \cdot v_3{}^k = P_4 \cdot v_4{}^k \qquad (4.18)$$

where

$$k = \frac{\ln\left(\dfrac{P_3}{P_4}\right)}{\ln\left(\dfrac{v_4}{v_3}\right)} \cong \frac{c_p}{c_v} \qquad (4.19)$$

After calculating the isentropic index of compression the specific work can be calculated as:

$$w = -\int v \cdot dP = \int v_4 \cdot \left(\frac{P_4}{P_3}\right)^{\frac{1}{k}} \cdot dP = -\frac{k}{k-1} \cdot P_4 \cdot v_4 \cdot \left[\left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}} - 1\right] \qquad (4.20)$$

4.3.3.3 Polytropic Re-Expansion Process

Calculation of the polytropic re-expansion is very similar to the calculation of polytropic compression. The only difference is instead of a polytropic index of compression n there is polytropic index of re-expansion m:

$$P \cdot v^m = P_1 \cdot v_1{}^m = P_2 \cdot v_2{}^m \qquad (4.21)$$

where

$$m = \frac{\ln\left(\dfrac{P_3}{P_4}\right)}{\ln\left(\dfrac{V_4}{V_3}\right)} \qquad\qquad (4.22)$$

Specific work of polytropic re-expansion can be calculated as:

$$w = -\int V \cdot dP = \int V_4 \cdot \left(\frac{P_4}{P_3}\right)^{\frac{1}{m}} \cdot dP = -\frac{m}{m-1} \cdot P_4 \cdot V_4 \cdot \left[\left(\frac{P_3}{P_4}\right)^{\frac{m-1}{m}} - 1\right] \qquad (4.23)$$

4.3.4 Suction Process

Similar to the discharge process the heat transfer during the suction process can be neglected.

$$i_1 = i_4 \qquad\qquad (4.24)$$

Please note that the state of the refrigerant at the suction should be superheated vapor.

4.4 Compression Performance Evaluation Parameters

In order to evaluate the performance of a compressor certain efficiencies and characteristics have been defined. These are isothermal efficiency, isentropic efficiency, clearance volumetric efficiency, overall volumetric efficiency and specific work of compression. There are also mechanical characteristics of the compressor like compressor speed, bore diameter and stroke, which are used for compressor sizing and / or evaluation.

4.4.1 Isothermal Efficiency

$$\text{Isothermal Efficiency} = \frac{\text{Isothermal Work}}{ActualWork} \tag{4.25}$$

4.4.2 Isentropic Efficiency

$$\text{Isentropic Efficiecny} = \frac{\text{Isentropic Work}}{\text{ActualWork}} \tag{4.26}$$

4.4.3 Clearance Volumetric Efficiency

As mentioned before there is a clearance volume in reciprocating compressors which adversely affect the volume of the sucked vapor i.e. swept volume for a given cylinder size. The clearance volume can be represented as:

$$V_4 = V_3 \cdot \left(\frac{P_h}{P_s}\right)^{\frac{1}{m}} = V_c \cdot \left(\frac{P_h}{P_s}\right)^{\frac{1}{m}} = C \cdot V_p \cdot \left(\frac{P_h}{P_s}\right)^{\frac{1}{m}} \tag{4.27}$$

and the clearance volumetric efficiency is calculated as:

$$\eta_{cv} = \frac{V_1 - V_4}{V_p} = \frac{V_s}{V_p} = \frac{(V_p + V_c) - C \cdot V_p \left(\frac{P_h}{P_s}\right)^{\frac{1}{m}}}{V_p} \tag{4.28}$$

Simplifying:

$$\eta_{cv} = 1 + C - C \cdot \left(\frac{P_h}{P_s}\right)^{\frac{1}{m}} \tag{4.29}$$

4.4.4 Overall Volumetric Efficiency

Overall volumetric efficiency is similar to clearance volumetric efficiency, however in this case the pressure drops and leakage losses in compression is taken into account. The overall volumetric efficiency is calculated as:

$$\eta_v = (1+C) \cdot \left(\frac{P_s}{P_1}\right)^{\frac{1}{m}} - C \cdot \left(\frac{P_d}{P_1}\right)^{\frac{1}{m}} - f_{leakage} \cdot \left(\frac{P_d}{P_s}\right) \tag{4.30}$$

where $f_{leakage}$ is leakage fraction and generally is about 0.01.

4.4.5 Specific Work of Compression

Specific work of compression based on overall volumetric efficiency can be calculated as:

$$w_c = \frac{P_1 \cdot v_1}{\eta_v} \cdot \left[\frac{-n}{n-1} \cdot (1+C) \cdot \left(\left(\frac{P_d}{P_1}\right)^{\frac{n-1}{n}} - 1\right) + \frac{m}{m-1} \cdot (1+C-\eta_v) \cdot \left(\left(\frac{P_h}{P_s}\right)^{\frac{m-1}{m}} - 1\right)\right]$$

(4.31)

If the indexes of polytropic compression and re-expansion are the same i.e. n = m then the specific work can be simplified as:

$$w_c = \frac{-n}{n-1} \cdot P_1 \cdot v_1 \cdot \left(\left(\frac{P_d}{P_1}\right)^{\frac{n-1}{n}} - 1\right) \tag{4.32}$$

4.4.6 Speed of Compressor

Important characteristics for the compressor speed are rotational speed N (rpm), and the mean piston speed C_m (m/s).

$$N = \frac{\dot{m} \cdot \upsilon_4}{\eta_v \cdot V_p} \cdot 60 \tag{4.33}$$

$$C_m = \frac{2 \cdot L \cdot N}{60} = \frac{2 \cdot L \cdot \dot{m} \cdot \upsilon_4}{\eta_v \cdot V_p} \tag{4.34}$$

where υ_4 is the specific volume in m^3/kg and $\dot{m}$ is the mass flow rate of the refrigerant in kg/s.

To simulate the varying internal loads, change in room temperature or cold space temperature a variable speed compressor will be designed in this study.

CHAPTER 5

CONDENSER

In a household refrigerator after the superheated refrigerant vapor is discharged from the compressor it is received by the condenser. Condenser is basically a heat exchanger. Its duty is to cool the superheated refrigerant to a certain level in order to be reprocessed in the evaporator. The superheated vapor is first desuperheated to saturation temperature, then condensed to saturated liquid and at the end it will come out of the condenser as subcooled liquid.

As it is the general case for heat exchangers condensers use two types of fluids with different temperatures in order to keep one of them at the desired temperature.

There are three types of condensers:
- Air-cooled condensers
- Water-cooled condensers
- Evaporative condensers

As their names suggest air-cooled condensers use air and water-cooled condensers use water as second fluids. Evaporative condensers on the other hand use evaporation of water as the main source. Due to compact size and cost requirements the use of water in condensers is not an applicable option. As a result, air-cooled condensers are the most widely used type of condensers in household refrigerators. In this study the designed condenser will be air-cooled type.

5.1 Air-cooled Condensers

As mentioned before air cooled condensers use air for the purpose of cooling the refrigerant. The heat removal can be either by natural or forced convection. Since forced convection will give better results and it is possible to produce compact and silent fans for air flow forced convection type air-cooled condensers are being widely used in household refrigerators. For the condenser configuration the general preference is wire-on-tube type condensers with vertex configuration. A vertex configuration, wire-on-tube type condenser is show in Figure 5.1. In this study force cooled, wire-on-tube, vertex configuration condensers will be designed.

Figure 5.1 Wire-on-tube type, vertex configuration condenser

5.2 Heat Transfer in Air-cooled Condensers

Basic heat transfer equation for heat exchangers is:

$$Q = U \cdot A \cdot \Delta T_{lm} \qquad (5.1)$$

Where U is the overall heat transfer coefficient, A is the surface area and ΔT_{lm} is the log-mean temperature difference. The overall heat transfer coefficient based on the outer surface area can be calculated as:

$$U_o = \left[\frac{A_o}{A_i} \cdot \frac{1}{h_i} + \frac{A_o}{A_i} \cdot \frac{1}{h_{fi}} + \frac{A_o}{A_m} \cdot \left(\frac{x}{k} \right)_t + \frac{1}{h_o \cdot \eta_o} \right]^{-1} \qquad (5.2)$$

where;

A_o: Outside surface area,

A_i: Inside surface area,

h_i: Inside convective heat transfer coefficient,

h_{fi}:   Coefficient of heat transfer through the fouling (accumulated material on the inner pipe surface),

A_m: Mean surface area,

x: Thickness of the tube wall,

k: Thermal conductivity of the tube material,

h_o: Outside convective heat transfer coefficient,

η_o: Overall surface efficiency,

$$\eta_o = 1 - \frac{A_f}{A_o} \cdot \left(1 - \eta_f \right) \qquad (5.3)$$

where;

A$_f$:   Finned surface area,

η_f: Fin efficiency,

With the advances in technology and chemistry scale forming is a very controllable and preventable situation, thus the second term can be neglected. The thickness of the tube wall is very small and the tube materials are selected from highly conductive materials, which makes the third term very small compared to the others, as a result it can be neglected also. Then, the overall outer surface heat transfer coefficient will be simplified as:

$$U_o = \left[\frac{A_o}{A_i} \cdot \frac{1}{h_i} + \frac{1}{h_o \cdot \eta_o} \right]^{-1} \tag{5.4}$$

5.3 Heat Transfer Processes in Condensers

As mentioned before there are three steps of condensation in refrigeration systems; desuperheating, condensing and subcooling. The process occurs in the same order and each stage will have a different heat transfer characteristic.

5.3.1 Desuperheating Process in Condensers

In vapor compression refrigeration systems, the refrigerant is discharged from the compressor at superheated vapor state and received by the condenser. It is required that at the condenser exit the state of the refrigerant should be subcooled liquid. For this purpose the superheated vapor should be desuperheated to saturated vapor first. This is a sensible cooling (the water vapor ratio is constant during heat exchange) process which takes place in the first few coils of the condenser.

Since until the desuperheating process ends the refrigerant will be in vapor phase only, it can be taken as a single-phase flow, which can be either laminar or turbulent. The flow can be either in entrance region or fully developed. Since the tube length is very long compared to tube diameter in the condenser, the flow will be taken as fully developed in this study.

To decide whether the flow is laminar or turbulent the common approach is to use the Reynolds number. If it is higher than 2300 the flow is turbulent, otherwise it is laminar. Reynolds number of the flow can be calculated as:

$$Re_D = \frac{\rho \cdot u_m \cdot D}{\mu} = \frac{4 \cdot \dot{m}}{\pi \cdot D \cdot \mu} \qquad (5.5)$$

where D is the inner tube diameter and μ is the viscosity of the refrigerant.

5.3.1.1 Laminar Flow Desuperheating

Since desuperheating process is a phase change process it can be assumed as a constant surface temperature process in a circular tube. For this arrangement the Nusselt number will be [29]:

$$Nu_D = \frac{h \cdot D}{k} = 3.66 \qquad (5.6)$$

where k is the conductivity of the refrigerant, and h is the convective heat transfer coefficient.

For the pressure drop inside the tube the following equation can be used:

$$\Delta p = f \cdot \frac{\rho \cdot u_m^2}{2 \cdot D} \cdot (x_2 - x_1) \qquad (5.7)$$

where u_m is the mean flow velocity, ρ is the refrigerant density, x_1 and x_2 are the locations of the tube where the pressure drop is calculated between and f is the friction factor and can be calculated as:

$$f = \frac{64}{Re_D}$$ (5.8)

for fully developed laminar flow [29].

5.3.1.2 Turbulent Flow Desuperheating

For fully developed flow in a circular tube with constant surface temperature the Nusselt number can be calculated in the range of $0.5 < Pr < 2000$ and $3000 < Re_D < 5 \times 10^6$ by the correlations found by Petukhov and later modified by Gnielinski [29]:

$$Nu_D = \frac{\left(\frac{f}{8}\right) \cdot (Re_D - 1000) \cdot Pr}{1 + 12.7 \cdot \left(\frac{f}{8}\right)^{0.5} \cdot \left(Pr^{\frac{2}{3}} - 1\right)}$$ (5.9)

where Pr is the Prandtl number. The friction factor for fully developed turbulent flow in smooth circular tubes can be calculated from:

$$f = \left(0.790 \cdot \ln(Re_D) - 1.64\right)^2$$ (5.10)

Which is also derived by Petukhov for the range $3000 \le Re_D \le 5 \times 10^6$ [29]. The pressure difference can be calculated from Eqn. (5.7).

5.3.2 Condensation Process in Condensers

After the superheated vapor is desuperheated to saturated vapor, the condensation of the refrigerant to the saturated liquid state begins. Condensation may occur in two ways; dropwise condensation or film condensation. In dropwise condensation, condensate forms in drops whereas in film condensation, a condensate film is formed. Dropwise condensation is not a desired situation in refrigerators and it can be prevented by simply applying a coating inside the tube surface. In this study film condensation will be considered as the condensation mechanism. Condensation may occur as laminar film or forced-convection film condensation.

5.3.2.1 Laminar Flow Condensation

For film condensation in horizontal tubes the conditions inside the tube is very complicated and affected by the velocity of the fluid. If the velocity is low, the condensate flow is from the upper surface of the tube to the bottom and the condensate flow is at the lower portion of the tube. This type of flow is also called stratified flow. For higher velocities the two-phase flow becomes annular. The vapor is at the center of the annulus, and as the condensation overcomes the thickness of the layer decreases. Stratified and annular flows are shown in Figure 5.2.
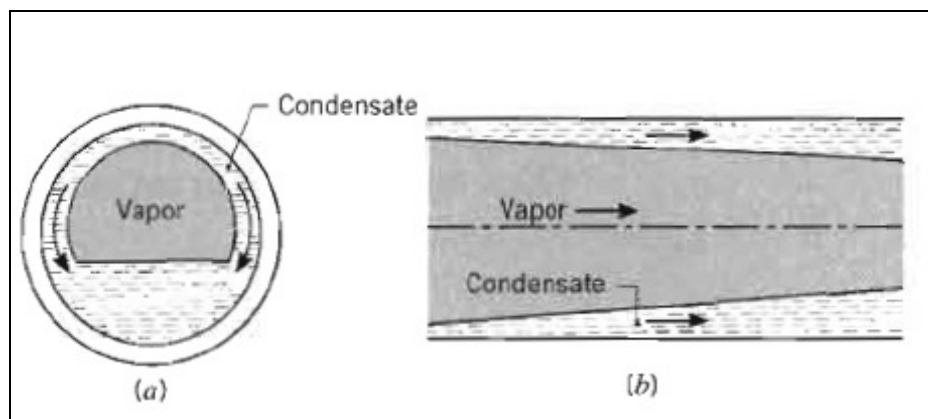

Figure 5.2 Film condensation in horizontal tubes (a) Low vapor velocity (stratified) flow (b) High vapor velocity (annular) flow [29].

For low vapor velocities the flow will be laminar and the heat transfer coefficient will be calculated in the range of $Re_D < 35000$ from:

$$\overline{h}_D = 0.555 \cdot \left[\frac{g \cdot \rho_1 \cdot (\rho_1 - \rho_v) \cdot k_1^{\ 3} \cdot i'_{fg}}{\mu_1 \cdot (T_{sat} - T_s) \cdot D} \right]^{1/4} \tag{5.11}$$

where ρ_l and ρ_v are saturated liquid and vapor densities respectively, k_l is the saturated liquid conductivity, T_{sat} is the saturation temperature at the given pressure and $h`_{fg}$ is the modified latent heat and can calculated as:

$$i'_{fg} = i_{fg} + \frac{3}{8} \cdot c_{p,l} \cdot (T_{sat} - T_s) \tag{5.12}$$

5.3.2.2 Forced-Convection Condensation

High velocity flow results in annular film condensation, which is called forced-convection condensation. There are different approaches to this type of condensation in the literature. The most accepted correlation is derived by Rohsenow for local heat transfer [30]:

$$i_z = \frac{k_1}{D} \cdot \frac{Pr_1 \cdot Re_1^{\ 0.9}}{F_2} \cdot \left[\frac{1}{X} + \frac{2.85}{X^{0.476}} \right] \tag{5.13}$$

where

$$Re_1 = \frac{G \cdot (1 - x) \cdot D}{\mu_1} \tag{5.14}$$

$$X = \left(\frac{\mu_v}{\mu_1} \right)^{0.1} \cdot \left(\frac{1 - x}{x} \right)^{0.9} \cdot \left(\frac{\rho_v}{\rho_1} \right)^{0.5} \tag{5.15}$$

$$F_2 = \begin{array}{ll} Pr_1 + 5 \cdot \ln(1 + 5 \cdot Pr_1) + 2.5 \cdot \ln(0.0031 \cdot Re_1{}^{0.812}) & Re_1 > 1125 \\ 5 \cdot Pr_1 + 5 \cdot \ln(1 + Pr_1 \cdot (0.0964 \cdot Re_1{}^{0.585} - 1)) & 60 < Re_1 < 1125 \\ 0.707 \cdot Pr_1 \cdot Re_1{}^{0.5} & Re_1 < 50 \end{array}$$

(5.16)

where G is mass velocity, x is the local quality of the fluid.

The mean heat transfer coefficient can be found as dividing the condensation into small quality increments (Δx) and finding the h_z at the mid-point of each increment (taking increments as small as possible, thus assuming uniform heat transfer coefficient for each increment). After finding h_z the required tube length (Δz) for the given quality increment can be calculated:

$$\Delta z = \frac{D \cdot G \cdot i_{fg} \cdot \Delta x}{4 \cdot h_z \cdot \Delta T}$$

(5.17)

The total required tube length to condense the refrigerant from an initial quantity to the desired will be sum of all incremental tube lengths:

$$L = \Sigma \Delta z$$

(5.18)

After calculating the total tube length and knowing local heat transfer coefficients for each increment the overall heat transfer coefficient can be found by:

$$h_m = \frac{1}{L} \cdot \int_0^L h_z \cdot dz$$

(5.19)

Eqn. (5.19) can be represented in terms of quality by taking the inlet quality as 1 (saturated vapor) and the exit quality x_e:

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$$h_m = \frac{1}{(1 - x_e)} \cdot \int_{x_e}^{1} \frac{dx}{h_z} \tag{5.20}$$

The pressure drop inside the tube should also be calculated. The important parameters are the changes in friction, momentum and gravity:

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_f + \left(\frac{dP}{dz}\right)_g + \left(\frac{dP}{dz}\right)_m \tag{5.21}$$

$$\left(\frac{dP}{dz}\right)_f = -0.09 \cdot (1 + 2.85 \cdot X^{0.523})^2 \cdot \frac{\mu_v^{0.2} \cdot (G \cdot x)^{1.8}}{\rho_v \cdot D^{1.2}} \tag{5.22}$$

$$\left(\frac{dP}{dz}\right)_g = g \cdot \sin\theta \cdot \left[\left(\frac{1}{1 + \frac{1-x}{x} \cdot \left(\frac{\rho_v}{\rho_l}\right)^{\frac{2}{3}}} \cdot \rho_v \right) + \left(\left(1 - \frac{1}{1 + \frac{1-x}{x} \cdot \left(\frac{\rho_v}{\rho_l}\right)^{\frac{2}{3}}} \right) \cdot \rho_l \right) \right] \tag{5.23}$$

where for horizontal tubes, $\sin\theta = 0$, as result $(dP/dz)_g = 0$.

$$\left(\frac{dP}{dz}\right)_m = -\frac{G^2}{\rho_v} \cdot \frac{dx}{dz} \cdot \left[2 \cdot x + (1 - 2 \cdot x) \cdot \left(\frac{\rho_v}{\rho_l}\right) - (2 - 2 \cdot x) \cdot \left(\frac{\rho_v}{\rho_l}\right) \right] \tag{5.24}$$

5.3.3 Subcooling Process in Condensers

After the saturated vapor is condensed to saturated liquid, subcooling of the refrigerant begins. Similar to superheating subcooling is also sensible and the process takes place in the last few coils of the condenser. The calculation procedure for the subcooling is the same with the superheating.

5.3.4 Air-Side Heat Transfer in Condensers

There are not many studies on the subject of air side heat transfer coefficient of forced-convection in condensers. There are some experimental studies carried out by the refrigeration companies and the one of the experiments done by Arçelik A.Ş. shows that the air side convection coefficient mainly depends on the flow velocity and the orientation of the condenser. The results showed that when the air velocity is about 0.61 m/s and the flow is parallel to fin-wires, the air side convective heat transfer coefficient is about 50 W/m^2.K.

The radiative heat transfer between the outer surface of the condenser and the environment is generally very small compared to the convection heat transfer. Because most of the radiation comes from the condenser itself (due to the geometry of the condenser configuration) thus the temperatures are very close which results in very small radiative heat gains.

5.3.4.1 Air-Side Heat Transfer from Fins

Since the condensers are required to be compact in size there is a requirement to increase the air-side heat transfer area without increasing the volume. This is generally satisfied by using wire fins on the outside of the condenser.

The fins are attached to the outer surface of the condenser, which makes the base temperature T_b of the wire the same as the outer surface temperature of the condenser. The fins will have uniform cross-section and the cross-sectional area will be small compared to the length thus the heat transfer at the tip of the fins can be neglected. With the given assumptions fins will be designed as uniform cross-section fins with adiabatic tips. The total heat transfer can be calculated as [29]:

$$q_f = \sqrt{h \cdot P \cdot k \cdot A_c} \cdot \theta_b \cdot \tanh(mL) \qquad (5.25)$$

where A_c is the cross-sectional area, P is the perimeter, k is the conductivity of the fin and:

$$\theta_b = T_b - T_\infty \qquad (5.26)$$

$$m = \sqrt{\frac{h \cdot P}{k \cdot A_c}} \qquad (5.27)$$

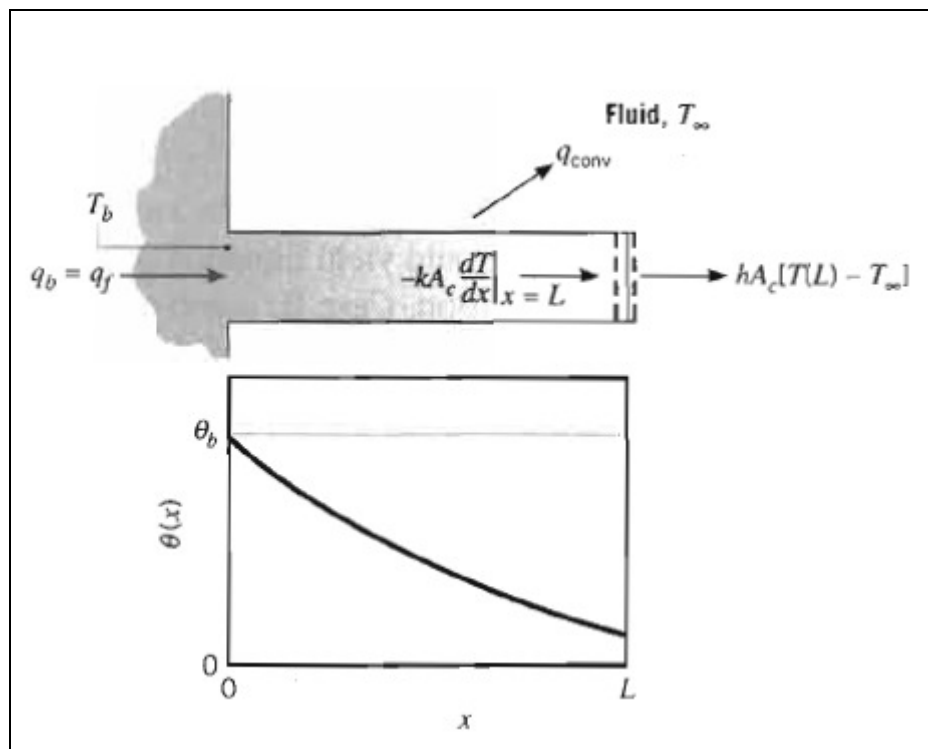Temperature profile of a fin with adiabatic tip and convection on both sides is given in Figure 5.3.

Figure 5.3 Conduction and convection in a fin of uniform cross-section [29].

In this study a vertex type, force cooled, wire on tube type condenser will be designed. For fin efficiency characteristics the condenser will be evaluated as a long tube between two plates. The midpoint between two tubes can be

considered as a fin with an adiabatic tip. The temperature profile of the condenser is given in Figure 5.4.
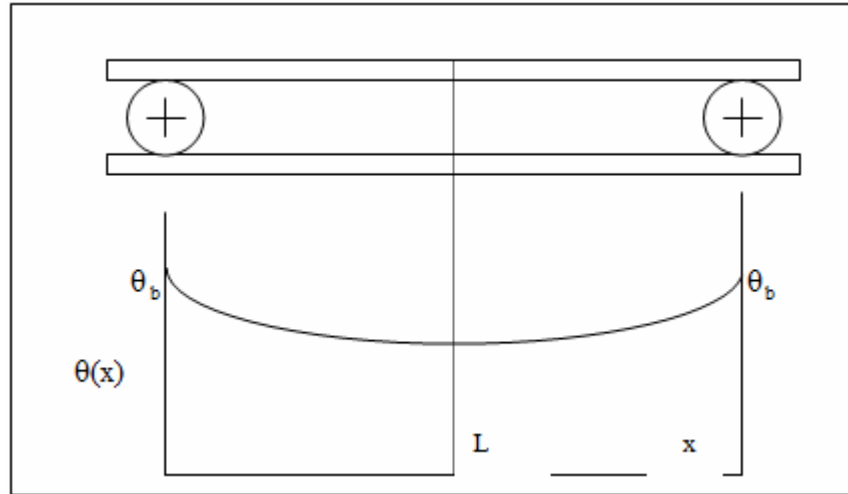

Figure 5.4 Wire fin temperature profile [2].

CHAPTER 6

EXPANSION DEVICE

Third component in the refrigeration cycle is the expansion device. After the superheated vapor is condensed to the subcooled state in the condenser, high pressure liquid is required to be dropped down to the evaporator inlet pressure. Expansion device is basically a mechanism that drops the pressure in a flow by throttling the flow rate.

There are two types of expansion devices:
- Expansion valves
- Capillary tubes

Expansion valves are variable-restriction type devices. They can vary the flow rate by throttling the valve either automatically or thermostatically. Capillary tubes on the other hand are constant-restriction type devices that can throttle the flow only at one ratio. Although expansion valves are beginning to be used in household refrigerators, most of them use capillary tubes as expansion devices. In this study a capillary tube will be designed as the expansion device of the refrigeration system.

6.1 Capillary Tubes

Capillary tube is basically a circular tube with a very small diameter to length ratio. Generally the bore diameter is about 1 mm and the length of the tube is between 1 to 6 m. Since it is only a tube with very small volume requirement it

has a very low cost of manufacturing and a very good choice for household refrigerators.

6.2 Capillary Tube Design

Since it is a very simple mechanism there are only a few factors required to be designed specifically for a refrigeration system. These are; pressure drop, tube diameter and tube length. For a given refrigeration system an iterative design process can be followed by changing the bore diameter and checking whether the required pressure drop is satisfied for the calculated tube length. In this respect the pressure drop and the length of the tube are the only critical parameters required to be calculated for a given diameter.

6.2.1 Pressure Drop Calculation

The pressure drop inside the capillary tube can be calculated by taking small control volumes throughout the tube with length increments ΔL. Then conservation of mass, energy and momentum can be applied to each control volume.

For the conservation of mass:

$$\frac{\dot{m}}{A} = \frac{V_1}{v_1} = \frac{V_2}{v_2} \qquad (6.1)$$

where A is the inner cross-sectional area of the tube in m^2, V is the velocity of the refrigerant in m/s and v is the specific volume of the refrigerant in m^3/kg. The subscripts 1 and 2 denote the refrigerant at the beginning and the end of the control volume respectively.

For the conservation of energy:

$$i_1 + \frac{V_1^{\,2}}{2} = i_2 + \frac{V_2^{\,2}}{2}$$ (6.2)

where i is the specific enthalpy of the refrigerant in J/kg.

And for the conservation of momentum:

$$\left[(P_1 - P_2) - f \cdot \frac{\Delta L}{D} \cdot \frac{V^2}{2 \cdot v} \right] \cdot A = \dot{m} \cdot (V_2 - V_1)$$ (6.3)

where P is pressure in Pa and f is the friction factor. For small Reynolds numbers f will be:

$$f = 0.33 \cdot \mathrm{Re_D}^{-\frac{1}{4}}$$ (6.4)

Throughout the capillary tube the quality of the refrigerant x, will increase from 0 (subcooled at the inlet) to a certain degree, because of the decrease in the saturation temperature as a result of the pressure drop. As a consequence at any point in the tube the specific properties of the refrigerant will be dependent on the quality:

$$i = i_f + x \cdot i_{fg}$$ (6.5)

$$v = v_f + x \cdot v_{fg}$$ (6.6)

$$\mu = \mu_f + x \cdot \mu_{fg}$$ (6.7)

6.2.2 Capillary Tube Length Calculation

As mentioned before the capillary tube length calculation is an iterative process, which begins with alternating the tube diameter until the required pressure drop is satisfied for a limit length. After the tube diameter is determined similar to the pressure drop calculation small increments of pressure drops ΔP should be determined, then the length required for each pressure drop will be calculated and at the end the sum of all incremental lengths will be the total capillary tube length.

By utilizing Eqn.(6.3) and substituting conservation of mass and energy equations the incremental length ΔL for each pressure step ΔP can be found. For the given refrigeration system the known parameters are the mass flow rate $\dot{m}$, cross-sectional area A, pressure and temperature at state 1. Assuming the refrigerant is always at saturated liquid form at the start of the capillary tube, temperature at the end of each pressure step can be found easily (state 2). By knowing pressure and temperature specific volume, enthalpy and viscosity at saturated vapor and liquid states can be found. Knowing that the conservation of energy equation will become:

$$i_1 + \frac{V_1^{\ 2}}{2} = \left(i_{f2} + x_2 \cdot i_{fg2}\right) + \frac{V_2^{\ 2}}{2} \tag{6.8}$$

where

$$V_2 = \left(v_{f2} + x_2 \cdot v_{fg2}\right) \cdot \frac{\dot{m}}{A} \tag{6.9}$$

By substituting Eqn.(6.9) into Eqn.(6.8):

$$i_1 + \frac{V_1^{\,2}}{2} = \left(i_{f2} + x_2 \cdot i_{fg2}\right) + \frac{\left[\left(v_{f2} + x_2 \cdot v_{fg2}\right) \cdot \dfrac{\dot{m}}{A}\right]^2}{2} \tag{6.10}$$

Since the only unknown in Eqn.(6.10) is x_2, it can be found easily by rearranging the terms as a second degree polynomial and solving for x_2. By knowing x_2, velocity, specific enthalpy, viscosity and specific volume at state 2 can be calculated. The parameters on the left hand side of Eqn. (6.3) are mean velocity, friction factor and specific volume. As a result they are required to be calculated first, as given below:

$$V = \frac{V_1 + V_2}{2} \tag{6.11}$$

$$f = \frac{f_1 + f_2}{2} \tag{6.12}$$

$$v = \frac{v_1 + v_2}{2} \tag{6.13}$$

By putting the known parameters into Eqn. (6.3) and rearranging, ΔL will become:

$$\Delta L_{1\text{-}2} = \left[\Delta P_{1-2} - \frac{\left[\dot{m} \cdot (V_2 - V_1)\right]}{A}\right] \cdot \frac{2 \cdot v \cdot D}{V^2 \cdot f} \tag{6.14}$$

Same calculations are applicable if the state of the refrigerant at the capillary tube entrance is subcooled liquid. The total length of the tube is the sum of all incremental tube length.

CHAPTER 7

EVAPORATOR

Evaporator is the last component in the refrigeration system before the whole refrigeration process begins with the compression again. Similar to the condenser, evaporator is a heat exchanger. However its purpose is complete opposite of the condenser. Instead of cooling the refrigerant, the refrigerant in the evaporator is heated in order to satisfy the necessary cooling in the cold space of the refrigerator which is the main purpose of a refrigeration system. The most widely used type of evaporators in refrigerators is called direct-expansion evaporators. Basically the refrigerant is boiled inside the tubes of the coil in this type of evaporators. In this study direct-expansion type evaporators will be designed.

7.1 Heat Transfer in Evaporators

Since evaporators are heat exchangers like condensers basic heat transfer equations are the same with those of condenser design. Eqns.(5.1), (5.2) and (5.3) are valid for overall heat transfer coefficient of evaporators too. There are two main heat transfer mechanisms in an evaporator:

- Heat transfer from solid surface to refrigerant
- Heat transfer from cooled solid surface to cold space air

7.1.1 Heat Transfer from Solid Surface to Refrigerant

In direct-expansion type evaporators boiling of the fluid is a forced convection type boiling. In this type of boiling, a liquid flows by forced convection over a surface kept at a temperature that is higher than the saturation temperature of the liquid. Similar to the condenser (but in reverse order) first the saturated liquid is heated until it becomes saturated vapor. During this process the refrigerant is in a two-phase condition (mixture of liquid and vapor). After the two phase region the refrigerant is further superheated until the input requirements of the compressor is satisfied.

7.1.1.1 Heat Transfer in Two-Phase Region

For forced convection the heat transfer should be divided into two separate regions called boiling and convective heat transfer [30]:

$$q'' = q_b'' + q_c'' \tag{7.1}$$

Similar to the condensers, the convection inside the evaporator will be either annular or stratified due to flow rate. If the quality of the refrigerant is higher than 5% the heat transfer is called high-quality convection, which is the case in refrigerator evaporators.

7.1.1.1.1 High-Quality Convection with Stratified Flow

If the flow rate of the refrigerant is lower than 2 m/s than the convection will be stratified. For the heat transfer coefficient the following equation can be used [31]:

$$h_{TP} = 0.9 \cdot \left[\frac{k_g^2 \cdot \rho_l \cdot \rho_v \cdot i_{fg}'^2}{0.5 \cdot D \cdot (T_w - T_{sat})^2} \right]^{1/4} \tag{7.2}$$

where

$$i'_{fg} = i_{fg} + 0.5 \cdot c_{pg} \cdot (T_w - T_{sat}) \qquad (7.3)$$

In the above equation thermophysical properties of the refrigerant vapor is calculated at the mean film temperature:

$$T_f = \frac{T_w + T_{sat}}{2} \qquad (7.4)$$

7.1.1.1.2 High-Quality Convection with Annular Flow

If the velocity of the refrigerant inside the tube is higher than 2 m/s, than the convection will be annular. For the heat transfer the following correlation can be used [30]:

$$q'' = q_{fc}'' + q_b'' \cdot \left[1 - \left(\frac{\Delta T_{sat,ib}}{\Delta T_{sat}} \right)^3 \right] \qquad (7.5)$$

where

$$\Delta T_{sat,ib} = (T_w - T_{sat})_{incipient} = \frac{1}{\Gamma} \qquad (7.6)$$

and

$$\Gamma = \frac{i_{fg} \cdot k_l}{8 \cdot \sigma \cdot T_{sat} \cdot v_{fg} \cdot h_{fc}} \qquad (7.7)$$

q_{fc}'' can be calculated from:

$$q''_{fc} = \frac{Re_l^{0.9} \cdot Pr_l \cdot k_l \cdot F(X)}{F_2 \cdot D} \cdot \Delta T_{sat} \qquad (7.8)$$

where

$$F(X) = 0.15 \cdot \left[\frac{1}{X} + 2 \cdot \left(\frac{1}{X} \right)^{0.32} \right] \qquad (7.9)$$

where Re_l, X and F_2 are the same terms denoted in Eqns. (5.14), (5.15) and (5.16).

As it comes to q''_b the applicable correlation is:

$$q''_b = 1.89 \cdot 10^{-14} \cdot \frac{g^{0.5} \cdot i_{fg}^{0.125} \cdot k_l^{0.5} \cdot \rho_l^{\frac{17}{8}} \cdot c_{pl}^{\frac{19}{8}} \cdot \rho_v^{0.125}}{\sigma^{\frac{9}{8}} \cdot (\rho_l - \rho_v)^{\frac{5}{8}} \cdot T_{sat}^{0.125}} \cdot (T_w - T_{sat})^3 \qquad (7.10)$$

For the pressure drop calculations equations used in condenser design are applicable, as result Eqns. (5.22), (5.23) and (5.24) can be used.

7.1.1.2 Heat Transfer in Superheated Region

For the superheated region the calculations are similar to those of desuperheating region of condensers. First the flow characteristics will be determined by calculating the Reynolds number from Eqn.(5.5), then the heat transfer coefficient will be calculated from Eqn.(5.6) if the flow is laminar or from Eqn.(5.9) if the flow is turbulent. For the pressure drop calculation Eqn.(5.7) is valid if the flow is laminar whereas for a turbulent flow Eqn.(5.10) can be used.

7.1.2 Heat Transfer to Lower Surface of a Cooled Plate

Contrary to the condensers in evaporators not only convective heat transfer is important but also the radiative heat transfer should be taken into account.

7.1.2.1 Convective Heat Transfer from the Cold Space

Cold space side of the evaporator can be taken as lower surface of a horizontally aligned cold plate. The Nusselt number of a cold plate can be calculated from:

$$
Nu_L = \begin{array}{ll} 0.54 \cdot Ra_L^{\,0.25} & 10^4 \leq Ra_L \leq 10^7 \\ 0.15 \cdot Ra_L^{\,1/3} & 10^7 \leq Ra_L \leq 10^{11} \end{array}
\tag{7.11}
$$

where Ra_L is called the Rayleigh number based on the characteristic length L:

$$
Ra_L = \frac{g \cdot \beta \cdot (T_s - T_\infty) \cdot L^3}{\nu \cdot \alpha}
\tag{7.12}
$$

$$
L = \frac{A_s}{P}
\tag{7.13}
$$

where A_s is the surface area and P is the perimeter of the plate.

7.1.2.2 Radiative Heat Transfer from the Cold Space

The general correlation for calculating the radiative heat transfer between two objects is:

$$
q_{rad} = A_{rad} \cdot \varepsilon \cdot \sigma \cdot \left(T_\infty^4 - T_s^4 \right)
\tag{7.14}
$$

where ε is the emissivity of the surface, σ is the Stefan-Boltzman constant $(5.67 \cdot 10^{-8} \ W/m^2.K^4)$, T_s and T_∞ are the evaporator surface and the cold space

temperature respectively. Since rest of the surfaces of the cold space are assumed to be at the cold space air temperature. A_{rad} is the radiated area and can be calculated as:

$$A_{rad} = F_{1-2} \cdot A_c \tag{7.15}$$

Where F_{1-2} is the view factor and can be taken as 1. Similarly the emissivity of the surface can also be taken as 1 assuming the evaporator surface as a black body. By rearranging Eqn.(7.14) the radiative heat transfer coefficient will be:

$$h_{rad} = \frac{\sigma \cdot \left(T_\infty^4 - T_{sur}^4\right)}{T_\infty - T_{sur}} \tag{7.16}$$

7.1.2.3 Heat Transfer from the Fins

Although the evaporator is considered as a cold plate there are some parts of the evaporator that remains between the fluid flowing tubes. Similar to the fins of the condenser these tubes can also be treated as fins with uniform cross-sectional area and insulated tips. In this case the base temperature will be the surface temperature of the evaporator tube, the fin perimeter will be the length of the tube and temperature difference θ_b will be represented as:

$$\theta_b = T_\infty - T_b \tag{7.17}$$

Knowing these facts Eqn.(5.25) can be used for the fin heat transfer for the evaporator.

In this study evaporator is considered as a round bound evaporator and the flat plate between two tubes are considered. Similar to the condenser the midpoint of each flat part is considered as adiabatic and the fin design will be carried out for fins with adiabatic tips. Since heat transfer is from one side of the evaporator the

other side is considered as adiabatic. Schematic representation of the evaporator and the temperature profile between two evaporator tubes is given in Figures 7.1 and 7.2 respectively.
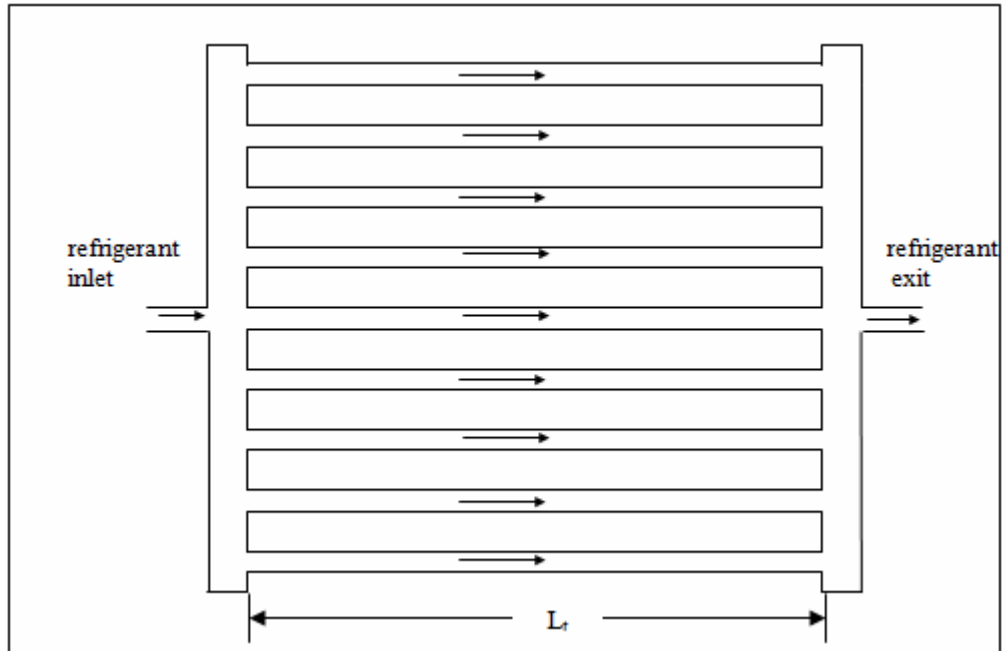
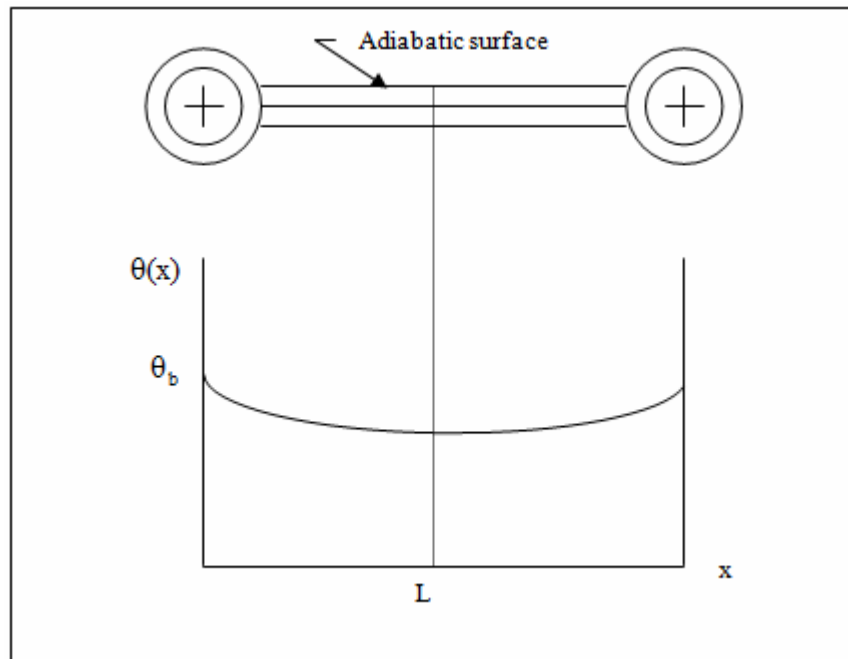Figure 7.1 Schematic representation of the evaporator [2].

Figure 7.2 Temperature profile between two tubes of the evaporator [2].

Where L is calculated by:

$$L = t_p - D_o \qquad (7.18)$$

where

t_p: Tube pitch between centers of each tube

D_o: Outer diameter of each tube

CHAPTER 8

INDIVIDUAL COMPONENT AND SYSTEM DESIGN

Each refrigeration system consists of basically four components; compressor, condenser, expansion device and evaporator. In the previous chapters equations and assumptions required to design each of these components have been defined. By using these equations and some predetermined input parameters each component can be designed separately. In order to achieve this, a computer program has been written for individual component design. In this chapter the input parameters required, the resulting output parameters and the calculation procedure used by the algorithm will be explained.

8.1 Computer Algorithm for Compressor Design

In the design of the compressor some dimensional and mechanical input parameters together with a selection for the operational parameters have been defined. All of the input parameters are called design input parameters:

- Bore Diameter (D)
- Stroke Length (L)
- Number of Cylinders
- Pressure Fraction through the Suction Valve (f_{ps})
- Pressure Fraction through the Discharge Valve (f_{pd})
- Leakage Fraction ($f_{leakage}$)
- Clearance Fraction (C)

- Type of the Compression Process:
  - Isothermal
  - Isentropic
  - Polytropic
- Type of the Re-Expansion Process:
  - Isothermal
  - Isentropic
  - Polytropic
- Higher Working Pressure (P_h)
- Lower Working Pressure (P_l)
- Mass Flow Rate of the Refrigerant ($\dot{m}$)
- Degree of Superheat at the Compressor Inlet

Design procedure of the computer algorithm is:

1- Convert the input data defined by the user to variables of the computer algorithm.

2- Calculate the suction pressure from Eqn.(4.4)

3- Calculate the discharge pressure from Eqn.(4.5)

4- Calculate the swept volume (V_p) from Eqn.(4.1)

5- Calculate the clearance volume (V_c) from Eqn.(4.2)

6- Calculate the clearance volumetric efficiency (η_{cv}) from Eqn.(4.29)

7- Calculate the overall volumetric efficiency (η_v) from Eqn.(4.30)

8- Calculate the suction volume (V_s) from:

$$V_s = V_p \cdot \eta_{cv} \qquad (8.1)$$

9- Calculate the specific work of compression (w_c) from Eqn.(4.31). Please note that for isothermal process n and m will be equal to 1, whereas for isentropic process n and m will be replaced by k.

10- Calculate the work of compression (W_t) from:

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$$W_t = w_c \cdot \dot{m}$$ (8.2)

11- Calculate the pressure, temperature, enthalpy and specific volume at the end of suction, compression, discharge and re-expansion.

12- Calculate the heat transfer from cylinders

13- Calculate the rotational speed from Eqn.(4.33)

14- Calculate the mean piston speed from Eqn.(4.34)

After the calculation procedure is done the displayed output parameters will be:

- Clearance Volumetric Efficiency (η_{cv})
- Overall Volumetric Efficiency (η_v)
- Clearance Volume (V_c)
- Suction Volume (V_s)
- Swept Volume (V_p)
- Rotational Speed (N)
- Mean Piston Speed (C_m)
- Heat Transfer Rate During Compression (Q_c)
- Heat Transfer Rate During Re-Expansion (Q_r)
- Total Power Requirement of the Compressor (W_t)

8.2 Computer Algorithm for Condenser Design

Required input design parameters for the condenser design are:

- Tube Inner Diameter
- Tube Outer Diameter
- Wire Diameter
- Wire Spacing
- Tube Pitch

- Condenser Width

- Condenser Height

- Mass Flow Rate of the Refrigerant

- Condenser Inlet Temperature

- Room Temperature

- Desired Refrigeration Load on the Condenser

- Condenser Pressure

- Thermal Conductivity of the Wire

- Thermal Conductivity of the Tube

For the calculation of the output parameters, the design algorithm is divided into three parts: desuperheating region, two-phase region and subcooling region. For each region the required tube length is calculated and at the end total tube length is found by summing up each length.

8.2.1 Computer Algorithm for Desuperheating Region

Design procedure for the desuperheating region is:

1-   Convert the input data defined by the user to variables of the computer algorithm.

2-   Calculate the saturation pressure of the refrigerant at condensation pressure.

3-   Determine the inlet state enthalpy

At this stage, a for loop is constructed in the computer algorithm. For decremental temperature steps of $\Delta T = 1$ K is chose to decrease the inlet temperature until the refrigerant temperature becomes equal to saturation temperature.

4- $T_2 = T_1 - \Delta T$

5- $P_2 = P_1$

6-   Set tube side mean temperature as the mean of T_1 and T_2

7-   Determine all thermophysical properties of the refrigerant at the mean temperature

8-   Calculate the Reynolds number from Eqn.(5.5) in order to determine the characteristics of the flow

9-   According to the flow characteristics use either Eqn.(5.6) or Eqn.(5.9) in order to calculate the Nusselt number and the heat transfer coefficient

Another loop is constructed this time for the calculation of the tube wall temperature of the condenser. At the beginning of the loop the wall temperature is assumed as the mean of mean tube wall temperature and room temperature.

10-   Calculate the air side heat transfer coefficient from:

$$Nu = \left(0.6 + \frac{0.387 \cdot Ra^{\frac{1}{6}}}{\left(1 + (0.559/Pr)^{\frac{9}{16}}\right)^{\frac{8}{27}}} \right)^2 \qquad (8.3)$$

11-   If the energy balance is satisfied then go to step 13, if not continue to step 12

12-   Calculate new wall temperature from:

$$T_{wall} = T_{room} + \frac{T_m - T_{room}}{\dfrac{h_o \cdot A_o}{h_i \cdot A_i} + 1} \qquad (8.4)$$

and go to step 10. Loop will be terminated when the absolute percentage error of the wall temperature calculation is 0.0001 %.

13-   Calculate overall heat transfer coefficient from Eqn.(5.4)

14-   Calculate the required length for the decremental temperature drop from:

$$\Delta L_{sh} = \frac{\dot{m} \cdot (i_1 - i_2)}{U_o \cdot \Delta T_{lm} \cdot A_c'} \tag{8.5}$$

15-   Calculate ΔP from Eqn.(5.7)

16-   Calculate P_2 as $P_2 = P_1 - \Delta P$

17-   Calculate saturation temperature at P_2

18-   Convert all thermophysical properties of state 2 together with T_2 and P_2 to the properties at state 1.

19-   If the condenser capacity is satisfied or saturation is reached then go to step 20, otherwise go to step 4.

20-   Calculate total desuperheating region tube length by summing up all decremental lengths ΔL_{sh}.

8.2.2 Computer Algorithm for Two-Phase Region

Design procedure for the two-phase region is either for annular or stratified flow depending on the velocity of the refrigerant. The design steps are:

1-   Convert the exit state properties of desuperheating region to the inlet state properties of two-phase region

2-   Calculate the Reynolds number from Eqn.(5.5) in order to determine the characteristics of the flow

3-   If the flow is stratified a single quality decrement Δx will be used, if the flow is annular then a number of quality increments will be used for length calculation and the total length will be calculated by summing up the required region length at each step.

4-   As a first assumption take tube wall temperature as $T_{sat} - 1$

5-   Calculate refrigerant side heat transfer coefficient from either Eqn.(5.11) or Eqn.(5.20) depending on the flow characteristics

6-   Calculate the condensation load from:

$$Q = \dot{m} \cdot (i_g - i_f) \cdot \Delta x \qquad\qquad (8.6)$$

Please note that Δx will be 1 for stratified flow as it is a single step solution.

7-   If calculated capacity is bigger than condenser capacity then calculate the quality of the refrigerant at the exit and terminate the two-phase design.

8-   Calculate air side heat transfer coefficient as in desuperheating region step 10

9-   Calculate the finned area heat transfer from Eqn.(5.25)

10- Repeat the procedure applied in steps 11, 12 and 13 of desuperheating region respectively.

11- Calculate the required length for the decremental quality drop from:

$$\Delta L_{tp} = \frac{\dot{m} \cdot (i_f - i_g) \cdot \Delta x}{U_o \cdot \Delta T_{lm} \cdot A_c'} \qquad\qquad (8.7)$$

Again please note that Δx will be 1 for stratified flow as it is a single step solution.

12- Calculate ΔP from either Eqn.(5.7) or (2.21) according to flow characteristics

13- Calculate P_2 as $P_2 = P_1 - \Delta P$

14- Calculate saturation temperature at P_2

15- If the flow is annular then go to step 4 until the refrigerant is saturated liquid, else continue to step 16.

16- Calculate total two-phase region tube length by summing up all decremental lengths ΔL_{tp}.

8.2.3 Computer Algorithm for Subcooling Region

Design procedure for the subcooling region is very similar to those of desuperheating region:

1-   Convert the exit state properties of two-phase region to the inlet state properties of subcooling region

2-   Set $\Delta T = 0.1$ K

3- $T_2 = T_1 - \Delta T$

4- $P_2 = P_1$

5-   Terminate calculation if both T_1 and T_2 are below room temperature

6-   Calculate condenser load for ΔT decrement

7-   Set tube side mean temperature as the mean of T_1 and T_2

8-   Determine all thermophysical properties of the refrigerant at the mean temperature

9-   Calculate the Reynolds number from Eqn.(5.5) in order to determine the characteristics of the flow

10-   According to the flow characteristics use either Eqn.(5.6) or Eqn.(5.9) in order to calculate the Nusselt number and the heat transfer coefficient

From this step the length calculation is identical to steps 10 to 20 of desuperheating region calculations. Instead of ΔL_{sh}, ΔL_{sc} term is used in order to avoid confusion. At the end, total subcooling region tube length will be calculated by summing up all decremental lengths ΔL_{sc}.

After the calculation of each step is done the total tube length will be calculated from:

$$L = L_{sh} + L_{tp} + L_{sc} \qquad (8.8)$$

The design output parameters for the condenser design are:

- Desuperheating Region Length
- Two-Phase Region Length
- Subcooling Region Length
- Total Condenser Tube Length
- Condenser Length
- Required Condenser Capacity
- Condenser Fin Efficiency
- Condenser Exit Temperature

8.3 Computer Algorithm for Expansion Device Design

As the expansion device as mentioned before a capillary tube will be designed. At the inlet of the capillary tube refrigerant will be either subcooled liquid or two-phase mixture (mostly liquid). If the inlet state is subcooled liquid the calculation will be carried out in one step and for the two-phase mixture which will be after the subcooled state or directly at the inlet there will be a stepwise calculation procedure. Required input design parameters for the capillary tube design are:

- Capillary Tube Diameter
- Mass Flow Rate of the Refrigerant
- Inlet Temperature
- Inlet Pressure
- Number of Pressure Steps for Pressure Calculation Loop
- Required Exit Pressure

Design procedure for capillary tube is:

1- Check whether the inlet state is subcooled or two-phase mixture. If the inlet state is subcooled then go to step 2, else go to step 6

2- Determine the saturation pressure at the inlet using the inlet temperature

3- Calculate the Reynolds number from Eqn.(5.5)

4- Calculate the friction factor from Eqn.(6.4)

5- Calculate the tube length from Eqn.(6.3)

After the design of the subcooled region length is done, the procedure continues with the two phase region calculation:

6- Convert the exit state properties of subcooled region to the inlet state properties of two-phase region

7- Calculate ΔP from:

$$\Delta P = \frac{P_{inlet} - P_{exit}}{\text{Number of Pressure Steps}} \qquad (8.9)$$

8- Set $P_2 = P_1 - \Delta P$

9- Determine the saturation temperatures for P_1 and P_2

10- Determine thermophysical properties at each state

11- Assume $i_2 = i_1$

12- Calculate the quality at the end of each step from:

$$x_2 = \frac{i_2 - i_{f2}}{i_{g2} - i_{f2}} \qquad (8.10)$$

13- Calculate specific volume at state 2 from Eqn.(6.6)

14- Calculate the velocity of the refrigerant at state 2 from Eqn.(6.9)

15- Calculate the specific enthalpy of the refrigerant at state 2 from Eqn.(6.2)

16- Calculate the absolute percentage error between assumed and calculated specific enthalpy. If its higher than 10^{-9} then assume i_2 is equal to the calculated value and go to step 12, else continue to 17

17- Calculate the Reynolds number from Eqn.(5.5)

18- Calculate the friction factor from Eqn.(6.4)

19- Calculate mean velocity from Eqn.(6.11)

20- Calculate mean friction factor from Eqn.(6.12)

21- Calculate ΔL_{1-2} from Eqn.(6.14)

22- Set $x_1 = x_2$ and go to step 6 until the input design exit pressure is reached

23- Calculate the total length of the two-phase region by summing up step lengths ΔL_{1-2}

24- Calculate the total capillary tube length from:

$$L = L_{sc} + L_{tp} \qquad (8.11)$$

The design output parameters for the expansion device design are:

- Total Capillary Tube Length
- Exit State Enthalpy
- Total Pressure Drop inside the Capillary Tube

8.4 Computer Algorithm for Evaporator Design

After the pressure of the system is dropped by the capillary tube the heat transfer between the refrigerant and the cold space is satisfied by the evaporator. As mentioned before the inlet state of the evaporator is two-phase mixture and the exit state is required to be superheated vapor for an evaporator. In this study a multi-tube single pass heat exchanger will be designed as the evaporator.

Required input design parameters for the evaporator design are:

- Tube Inner Diameter
- Tube Outer Diameter
- Tube Pitch

- Number of Tubes
- Inlet State Pressure
- Inlet State Enthalpy
- Mass Flow Rate of the Refrigerant
- Cold Room Temperature
- Evaporator Capacity
- Thermal Conductivity of the Tube

As mentioned before the calculation procedure for the evaporator design will be carried out in two regions; two-phase region and superheating region.

8.4.1 Computer Algorithm for Two-Phase Region

Design procedure for the two-phase region is either for annular or stratified flow depending on the velocity of the refrigerant. The design steps are:

1- Calculate the mass flow rate of the refrigerant through a single tube
2- Assume evaporator character length as 0.1
3- Calculate the inlet state velocity from Eqn.(6.1). If the velocity is lower than 2 m/s then the flow is stratified, else the flow is annular. If the flow is annular then go to step 14, else continue to step 4
4- Calculate the two-phase region load
5- Assume tube wall temperature is 1 K higher than saturation temperature at the inlet state
6- Calculate air side convective heat transfer coefficient from Eqn.(7.11)
7- Calculate air side radiative heat transfer coefficient from Eqn.(7.16)
8- Calculate the heat transfer rate from fins from Eqn.(5.25)
9- Calculate the refrigerant side heat transfer from Eqn.(7.2)
10- Calculate new wall temperature from:

$$T_{wall} = T_{cold\text{-}room} + \frac{T_{cold\text{-}room} - T_{sat}}{\dfrac{h_o \cdot A_o}{h_i \cdot A_i} + 1} \tag{8.12}$$

and go to step 6. Loop will be terminated when the absolute percentage error of the wall temperature calculation is 0.0001 %.

11-   Calculate the overall heat transfer coefficient from Eqn.(5.4)

12-   Calculate the required tube length from Eqn.(8.7)

13-   Calculate the pressure drop from Eqn.(5.7) and terminate the two-phase region calculations

14-   Divide the required quality difference into equal quality increments Δx

15-   Assume tube wall temperature is 1 K higher than saturation temperature at the inlet state

16-   Calculate the boiling convection heat flux from Eqn.(7.11)

17-   Calculate the forced convection heat flux from Eqn.(7.8)

18-   Calculate forced convection heat transfer coefficient from:

$$h_{fc} = \frac{q''_{fc}}{(T_{wall} - T_{sat})} \tag{8.13}$$

19-   Calculate Γ from Eqn.(7.7)

20-   Calculate total heat flux from Eqn.(7.5)

21-   Calculate refrigerant side heat transfer ratio from:

$$h = \frac{q''}{(T_{wall} - T_{sat})} \tag{8.14}$$

22-   Calculate the two-phase region load and go to step 6

8.4.2 Computer Algorithm for Superheating Region

Design procedure for the superheating region is similar to the design procedure of subcooled region in the condenser design. The procedure is:

1- Convert the exit state properties of two-phase region to the inlet state properties of subcooling region

2- Set $\Delta T = 1$ K

3- $T_2 = T_1 + \Delta T$

4- $P_2 = P_1$

5- Calculate heat load from:

$$Q = \dot{m} \cdot (i_2 - i_1) \tag{8.15}$$

6- Set tube side mean temperature as the mean of T_1 and T_2

7- Determine all thermophysical properties of the refrigerant at the mean temperature

8- Calculate the Reynolds number from Eqn.(5.5) in order to determine the characteristics of the flow

9- According to the flow characteristics use either Eqn.(5.6) or Eqn.(5.9) in order to calculate the Nusselt number and the heat transfer coefficient

10- Set the wall temperature is assumed as the mean of mean tube wall temperature and cold room temperature.

11- Calculate the air side heat transfer coefficient from:

$$Nu = \left(0.6 + \frac{0.387 \cdot Ra^{\frac{1}{6}}}{\left(1 + (0.559/Pr)^{\frac{9}{16}}\right)^{\frac{8}{27}}} \right)^2 \tag{8.3}$$

12- If the energy balance is satisfied then go to step 14, if not go to continue to step 13

13- Calculate new wall temperature from:

$$T_{wall} = T_{room} + \cfrac{T_m - T_{room}}{\cfrac{h_o \cdot A_o}{h_i \cdot A_i} + 1}$$ (8.4)

and go to step 10. Loop will be terminated when the absolute percentage error of the wall temperature calculation is 0.0001 %.

14- Calculate overall heat transfer coefficient from Eqn.(5.4)

15- Calculate the required length for the incremental temperature from:

$$\Delta L_{sh} = \frac{\dot{m} \cdot (i_1 - i_2)}{U_o \cdot \Delta T_{lm} \cdot A'_c}$$ (8.5)

16- Calculate ΔP from Eqn.(5.7)

17- Calculate P_2 as $P_2 = P_1 - \Delta P$

18- Calculate saturation temperature at P_2

19- Convert all thermophysical properties of state 2 together with T_2 and P_2 to the properties at state 1.

20- Calculate total superheating region tube length by summing up all incremental lengths ΔL_{sh}.

21- Calculate total tube length from:

$$L = L_{tp} + L_{sh}$$ (8.6)

The design output parameters for the evaporator design are:

- Superheating Region Length
- Two-Phase Region Length
- Total Evaporator Length
- Evaporator Fin Efficiency
- Evaporator Exit Temperature

8.5 Computer Algorithm for Complete Cycle Design

In the design of the complete vapor-compression refrigeration cycle, input parameters mentioned in sections 8.1 to 8.4 are required in a combined manner. However only the evaporator and condenser temperatures are required instead of higher and lower working pressures of the cycle.

Required input design parameters for the complete cycle design are:

- Compressor input parameters:
    - Bore Diameter (D)
    - Stroke Length (L)
    - Number of Cylinders
    - Pressure Fraction through the Suction Valve (f_{ps})
    - Pressure Fraction through the Discharge Valve (f_{pd})
    - Leakage Fraction ($f_{leakage}$)
    - Clearance Fraction (C)
    - Type of the Compression Process:
        - Isothermal
        - Isentropic
        - Polytropic
    - Type of the Re-Expansion Process:
        - Isothermal
        - Isentropic
        - Polytropic

- Condenser input parameters:
    - Tube Inner Diameter
    - Tube Outer Diameter
    - Wire Diameter
    - Wire Spacing

- Tube Pitch

- Condenser Width

- Condenser Height

- Thermal Conductivity of the Wire

- Expansion device input parameters:

  - Capillary Tube Diameter

  - Number of Pressure Steps for Pressure Calculation Loop

- Evaporator input parameters:

  - Tube Inner Diameter

  - Tube Outer Diameter

  - Tube Pitch

  - Number of Tubes

  - Thermal Conductivity of the Tube

- Complete cycle input parameters:

  - Evaporator Inlet Temperature

  - Condenser Exit Temperature

  - Room Temperature (T_{room})

  - Cold-Room Temperature (T_{cold})

  - Refrigeration Load

Design procedure for complete cycle design is:

1- Determine the saturation pressure at the evaporator inlet using the inlet temperature.

2- Determine the saturated vapor enthalpy at the evaporator inlet using the inlet temperature.

3- Determine the saturated liquid enthalpy at the condenser exit using the exit temperature.

4- Determine the superheated vapor enthalpy at the evaporator exit assuming a 5°C of superheat and using the evaporator inlet pressure.

5- Determine a dummy mass flow rate by:

$$\dot{m} = \frac{\text{Refrigeration Load}}{\text{Evaporator Exit Enthalpy - Evaporator Inlet Enthalpy}} \qquad (8.7)$$

Assuming the condenser exit enthalpy is equal to the evaporator inlet enthalpy.

6-   Call the evaporator design module described in section 8.4 using the calculated parameters as the unknown input parameters.

7-   Equate the output parameters of step 6 to unknown input parameters of compressor design module described in section 8.1.

8-   Call the compressor design module described in section 8.4 using the calculated parameters.

9-   Equate the output parameters of step 8 to unknown input parameters of condenser design module described in section 8.2.

10-   Call the condenser design module described in section 8.2 using the calculated parameters.

11-   Equate the output parameters of step 10 to unknown input parameters of expansion device design module described in section 8.3.

12-   Call the expansion device design module described in section 8.3 using the calculated parameters.

13-   Check the reliability of the evaporator inlet enthalpy assumption in step 5 by:

$$\Delta_h = \text{Expansion Device Exit Enthalpy - Evaporator Inlet Enthalpy} \qquad (8.8)$$

If $\Delta_h < 2$ then go to step 14, else go back to step 1 using:

$$\text{Evaporator Inlet Enthalpy} = \frac{(\text{Expansion Device Exit Enthalpy + Evaporator Inlet Enthalpy})}{2} \qquad (8.9)$$

and recalculating the dummy mass flow rate by Eqn.(8.7).

14-   Calculate the compressor irreversibility from:

$$I_{comp} = \dot{m} \cdot T_{room} \cdot (s_{compinlet} - s_{compexit}) - (Q_r + Q_c) \tag{8.10}$$

15- Calculate the condenser irreversibility from:

$$I_{cond} = \dot{m} \cdot T_{room} \cdot (s_{condexit} - s_{condinlet}) - Q_{cond} \tag{8.11}$$

16- Calculate the expansion device irreversibility from:

$$I_{exp} = \dot{m} \cdot T_{room} \cdot (s_{expexit} - s_{expinlet}) \tag{8.12}$$

17- Calculate the evaporator irreversibility from:

$$I_{evap} = \dot{m} \cdot T_{room} \cdot (s_{evapexit} - s_{evapinlet}) - Q_{evap} \cdot \frac{T_{room}}{T_{cold}} \tag{8.13}$$

18- Calculate the total cycle irreversibility from:

$$I_{total} = I_{comp} + I_{cond} + I_{exp} + I_{evapl} \tag{8.14}$$

19- Calculate the coefficient of performance from:

$$COP = \frac{\text{Refrigeration Load}}{\text{Compressor Work}} \tag{8.15}$$

After the calculation procedure is done the displayed output parameters will be:

- Compressor output parameters:
  - Clearance Volumetric Efficiency (η_{cv})

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- Overall Volumetric Efficiency (η_v)
- Rotational Speed (N)
- Total Power Requirement of the Compressor (W_t)
- Temperature, Pressure and Enthalpy at Inlet and Exit States

- Condenser output parameters:
  - Desuperheating Region Length
  - Two-Phase Region Length
  - Subcooling Region Length
  - Total Condenser Tube Length
  - Condenser Length
  - Required Condenser Capacity
  - Condenser Fin Efficiency
  - Temperature, Pressure and Enthalpy at Inlet and Exit States

- Expansion Device output parameters:
  - Capillary tube length
  - Temperature, Pressure and Enthalpy at Inlet and Exit States

- Evaporator output parameters:
  - Superheating Region Length
  - Two-Phase Region Length
  - Total Evaporator Tube Length
  - Evaporator Fin Efficiency
  - Temperature, Pressure and Enthalpy at Inlet and Exit States

- Total cycle output parameters:
  - Mass Flow Rate of the Refrigerant
  - Coefficient of Performance
  - Maximum Cycle Temperature and Pressure
  - Total Cycle Irreversibility

CHAPTER 9

CYCLE DESIGN WITH HYDROCARBON REFRIGERANTS

In order to observe the performance characteristics of hydrocarbons butane (R-600) and isobutane (R-600a) a total refrigeration cycle design is carried out with the design software. Both cycles designs have used the same component parameters and condenser and evaporator temperatures.

9.1 Household Refrigerator Design Input Parameters

In the design of the whole refrigeration cycle, all input parameters mentioned in Chapter 8 is used. For the compressor design the selected input parameters are:

- Bore Diameter: 20 mm
- Stroke Length: 24 mm
- Number of Cylinders: 1
- Pressure Fraction through the Suction Valve (f_{ps}):   98 %
- Pressure Fraction through the Discharge Valve (f_{pd}): 98 %
- Leakage Fraction ($f_{leakage}$):   1 %
- Clearance Fraction (C):   5 %
- Type of the Compression and Re-Expansion Processes: Polytropic

A screenshot of the compressor parameter input screen for whole cycle design is given in Figure 9.1
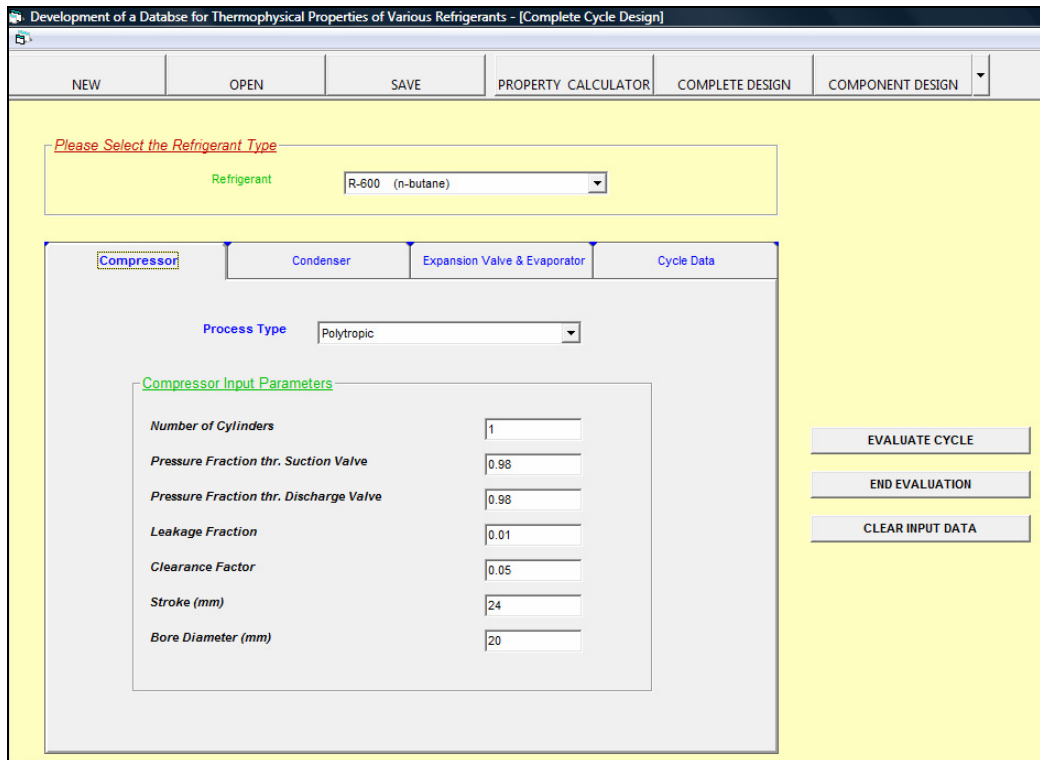
Figure 9.1 Design cycle data input window for compressor

For the condenser design the selected input parameters are:

- Tube Inner Diameter: 3.35 mm
- Tube Outer Diameter: 4.75 mm
- Wire Diameter: 1.1 mm
- Wire Spacing: 6 mm
- Tube Pitch: 30 mm
- Condenser Width: 150 mm
- Condenser Height: 170 mm
- Thermal Conductivity of the Wire: 1000 W/m.K

A screenshot of the condenser parameter input screen for whole cycle design is given in Figure 9.2
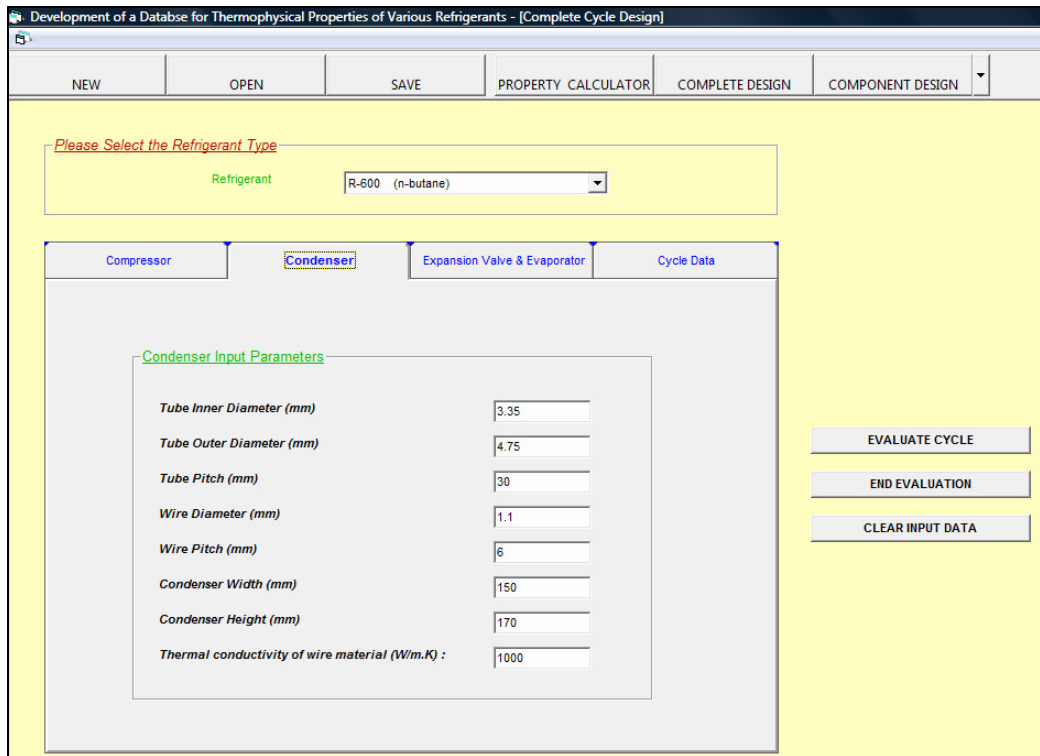
Figure 9.2 Design cycle data input window for condenser

For the expansion device and evaporator design the selected input parameters are:

- Capillary Tube Diameter: 0.87 mm
- Number of Pressure Steps for Pressure Calculation Loop: 20
- Tube Inner Diameter: 10 mm
- Tube Outer Diameter: 12 mm
- Tube Pitch: 24 mm
- Number of Tubes: 23
- Thermal Conductivity of the Tube: 1000 W/m.K

A screenshot of the expansion device and evaporator parameter input screen for whole cycle design is given in Figure 9.3
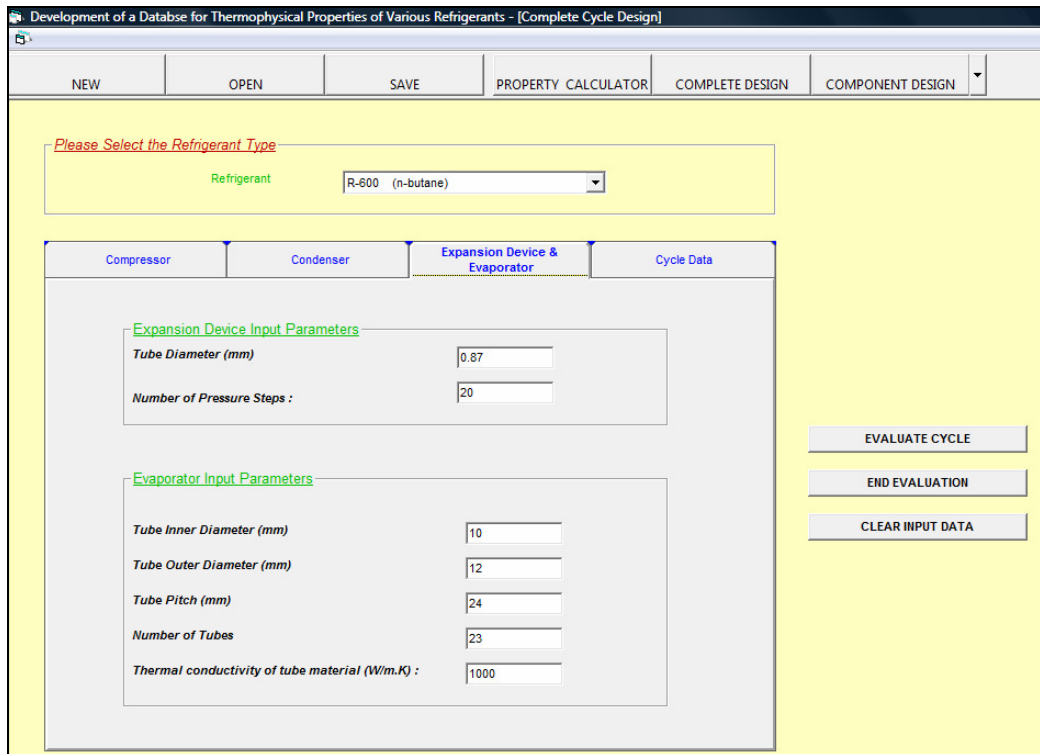
Figure 9.3 Design cycle data input window for expansion device and evaporator

For the complete cycle design, alternating input parameters for R-600 and R-600a are selected. In order to compare the cycle properties of each refrigerant evaporator, condenser and room temperature inputs are changed and different input sets have been formed. Cycle compression and re-expansion types are selected as polytropic and the coefficient k has been calculated for each refrigerant for each input set. Selected input sets are given in Table 9.1. Each set is used for cycle designs with refrigeration loads of 100 W, 125 W and 150 W. The room temperature is changed from 25 °C (298 K) to 40 °C (313 K) with 3 °C temperature increments in order to show the effect of cooling at different climatic conditions. Cold room temperatures are selected as either 0 °C (273 K) or -10 °C (263 K) in order to represent the food and freezer compartments of a refrigerator respectively. The evaporator temperature is selected to be 15 °C lower than the cold room temperature and condenser temperature is selected to be 10 °C higher than the room temperature. An example of the cycle input screen is given in Figure 9.4. Selected example is run with R-600 at a room

temperature of 298 K and cold room temperature of 273 K for a cooling load of 100 W.

Table 9.1 Input parameter sets for refrigerant cycle comparison

Evaporator Temperature (K)	Condenser Temperature (K)	Room Temperature (K)	Cold Room Temperature (K)	Polytropic coefficient for R-600	Polytropic coefficient for R-600a
SET 1					
258	308	298	273	1.1270	1.1425
258	311	301	273	1.1280	1.1440
258	314	304	273	1.1295	1.1465
258	317	307	273	1.1310	1.1480
258	320	310	273	1.1325	1.1505
258	323	313	273	1.1340	1.1530
SET 2					
248	308	298	263	1.1265	1.1415
248	311	301	263	1.1275	1.1430
248	314	304	263	1.1290	1.1455
248	317	307	263	1.1305	1.1470
248	320	310	263	1.1320	1.1495
248	323	313	263	1.1335	1.1520

Figure 9.4Example design cycle data input window for total cycle

9.2 Household Refrigerator Design Output Parameters

After the cycle evaluation is finished, output parameters for each component and the total cycle are represented in the software. In order to represent the results of the cycle evaluation, a cycle design with R-600 at a room temperature of 298 K and cold room temperature of 273 K for a cooling load of 100 W is selected and results are represented in this chapter.

For the compressor the output parameters for the selected cycle are:

- Clearance Volumetric Efficiency (η_{cv}): 78 %
- Overall Volumetric Efficiency (η_v): 69 %
- Rotational Speed (N): 2452 rpm
- Total Power Requirement of the Compressor (W_t): 24.8 W
- Inlet State Data
  - Temperature: 261 K
  - Pressure: 56 kPa
  - Enthalpy: 567.9 kJ/kg
- Exit State Data
  - Temperature: 323.1 K
  - Pressure: 270.5 kPa
  - Enthalpy: 661.8 kJ/kg

A screenshot of the compressor design output parameters are presented in Figure 9.5.

For the condenser the output parameters for the selected cycle are:

- Desuperheating Region Length: 0.3 m
- Two-Phase Region Length: 5.0 m
- Subcooling Region Length: 0.1 m

- Total Condenser Tube Length: 5.4 m

- Condenser Length: 0.134 m

- Required Condenser Capacity: 125 W

- Condenser Fin Efficiency: 99 %

- Inlet State Data

    - Temperature: 323.1 K

    - Pressure: 270.5 kPa

    - Enthalpy: 661.8 kJ/kg

- Exit State Data

    - Temperature: 308.0 K

    - Pressure: 337.3 kPa

    - Enthalpy: 284.1 kJ/kg

A screenshot of the condenser design output parameters are presented in Figure 9.6.

Figure 9.5 Example cycle design output parameters for compressor
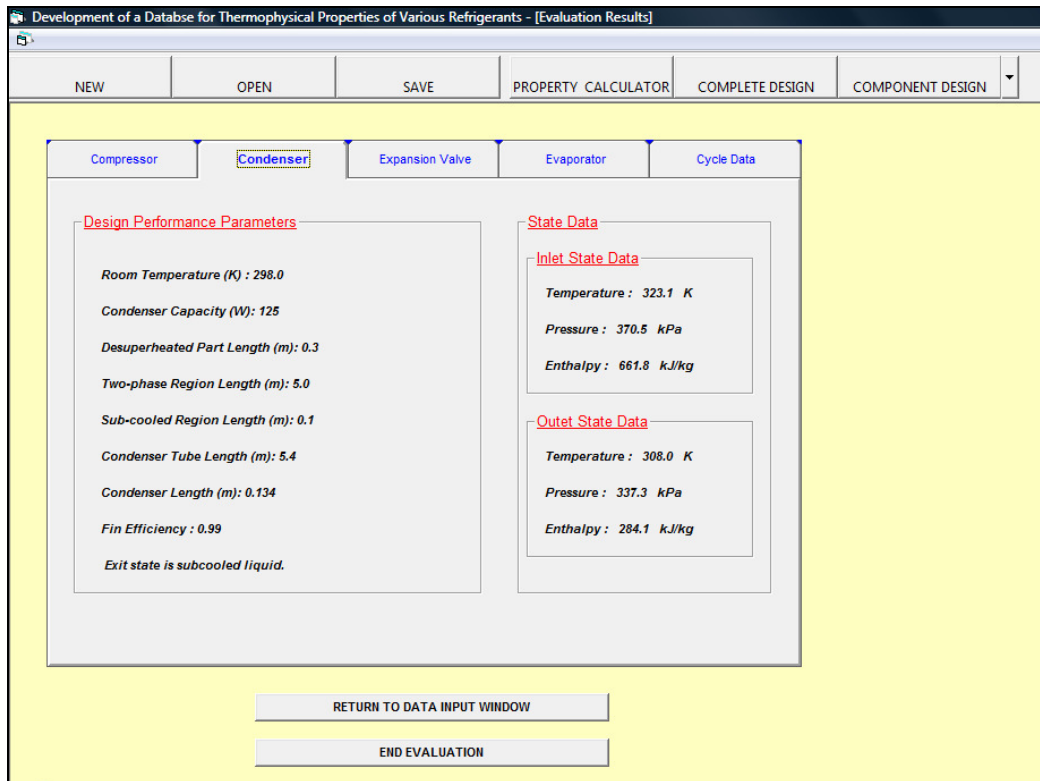
Figure 9.6 Example cycle design output parameters for condenser

For the expansion device the output parameters for the selected cycle are:

- Capillary Tube Length: 3.2 m
- Inlet State Data
    - Temperature: 308.0 K
    - Pressure: 337.3 kPa
    - Enthalpy: 284.1 kJ/kg
- Exit State Data
    - Temperature: 258.0 K
    - Pressure: 56.0 kPa
    - Enthalpy: 278.5 kJ/kg

A screenshot of the expansion device design output parameters are presented in Figure 9.7
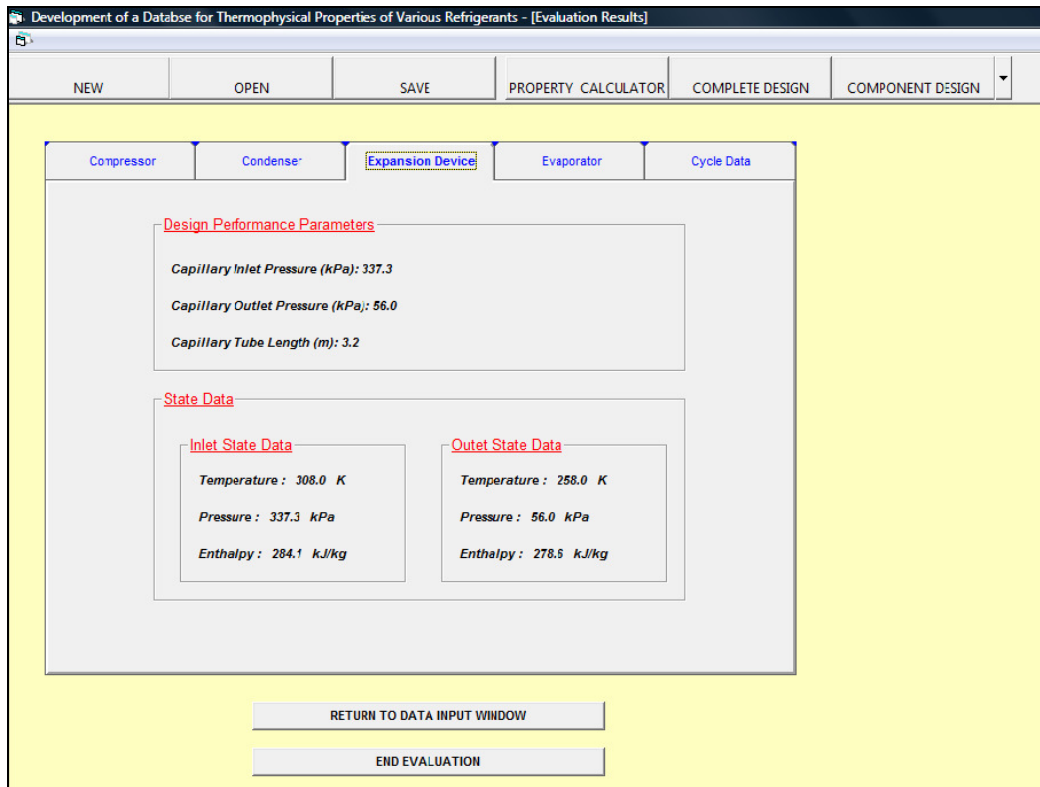
Figure 9.7 Example cycle design output parameters for expansion device

For the evaporator the output parameters for the selected cycle are:

- Superheating Region Length: 0.032 m
- Two-Phase Region Length: 0.702 m
- Total Evaporator Tube Length: 0.735 m
- Evaporator Fin Efficiency: 100 %
- Inlet State Data
    - Temperature: 258.0 K
    - Pressure: 56.0 kPa
    - Enthalpy: 278.5 kJ/kg
- Exit State Data
    - Temperature: 261.0 K
    - Pressure: 56.0 kPa
    - Enthalpy: 567.9 kJ/kg

A screenshot of the evaporator design output parameters are presented in Figure 9.8
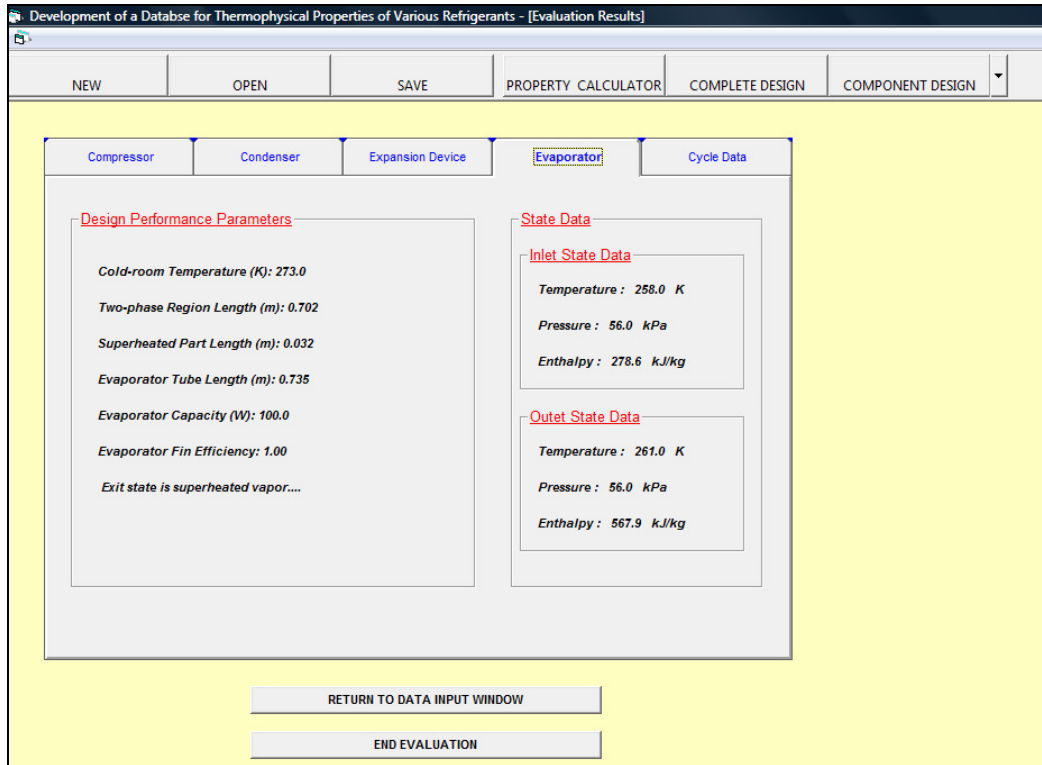
Figure 9.8 Example cycle design output parameters for evaporator

Total cycle design output parameters for the selected cycle are:

- Mass Flow Rate of the Refrigerant: 0.000324 kg/s
- Coefficient of Performance: 4.02
- Maximum Cycle Temperature: 323.26 K
- Maximum Cycle Pressure: 378.06 kPa
- Total Cycle Irreversibility: 14.22 W

A screenshot of the total cycle design output parameters are presented in Figure 9.9
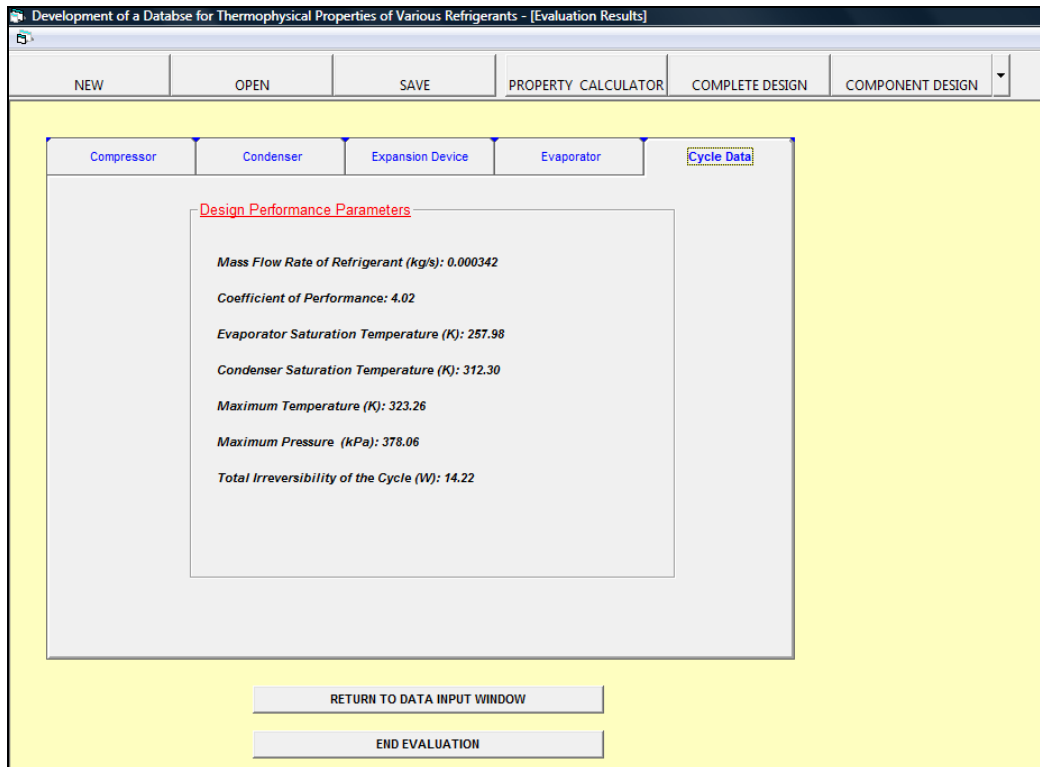
Figure 9.9 Example cycle design output parameters for total cycle

9.3 Hydrocarbon Refrigerant Performance Comparison

In order to compare the performance characteristics of hydrocarbon refrigerants performing in a household refrigerator, each refrigerant is used with the input parameters given in Table 9.1. To compare the efficiency of each refrigerant coefficient of performance (COP), compressor power, evaporator length, condenser length, capillary tube length and irreversibility of the design cycle results are given in Figures 9.10 to 9.15.
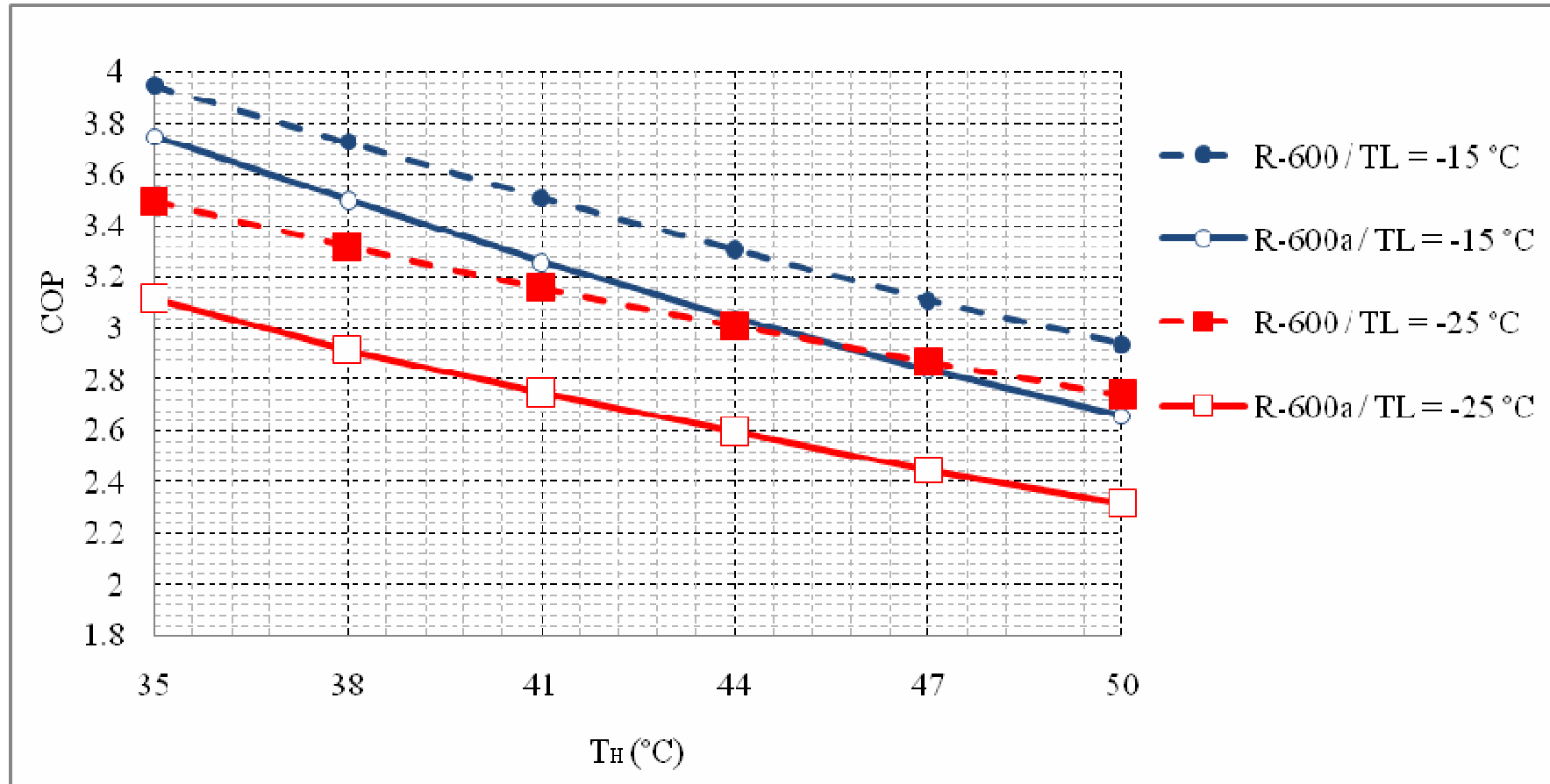
Figure 9.10 COP vs T_H for R-600 and R-600a at different cooling loads
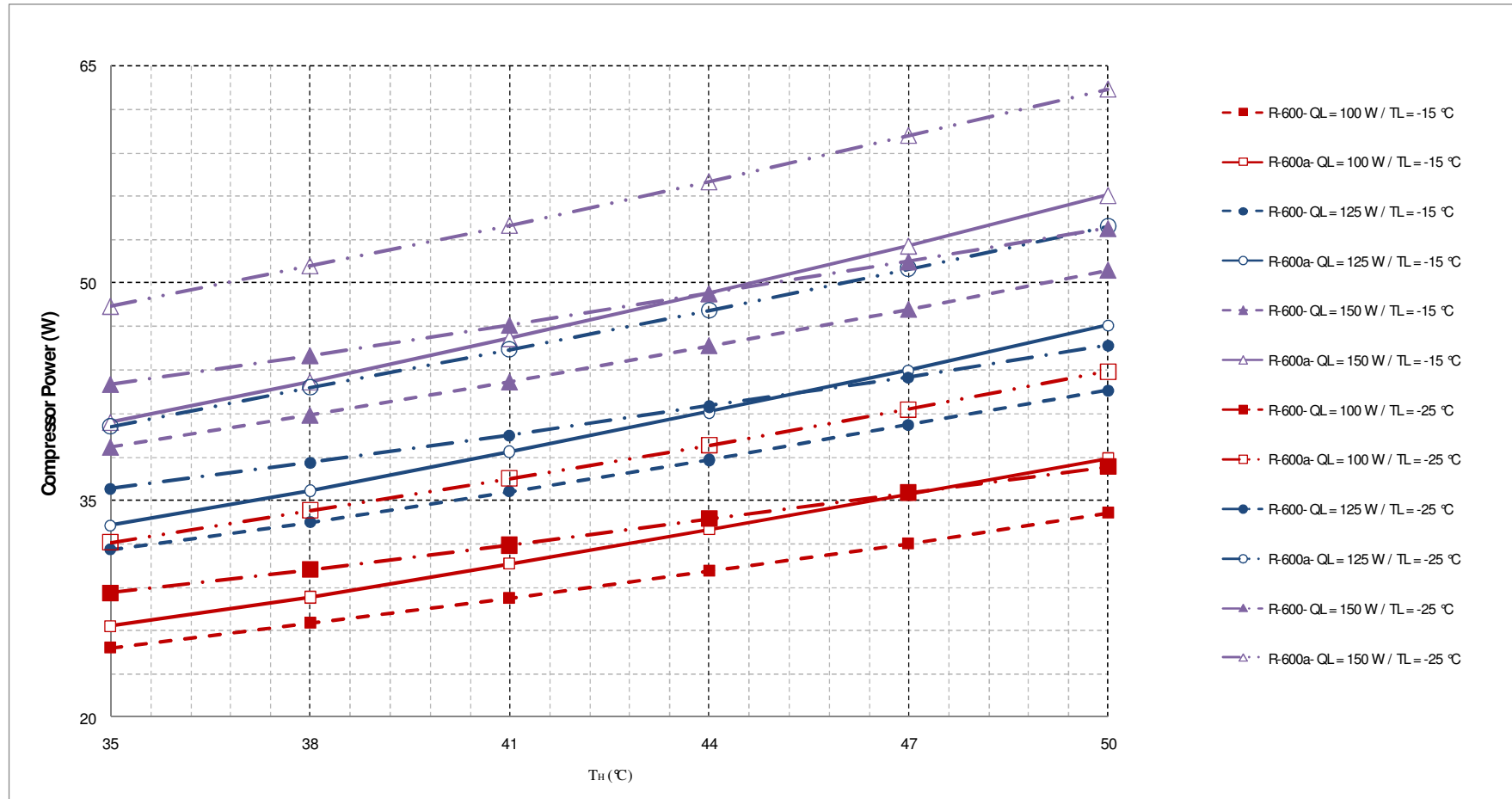
Figure 9.11 Compressor Power vs T$_H$ for R-600 and R-600a at different cooling loads
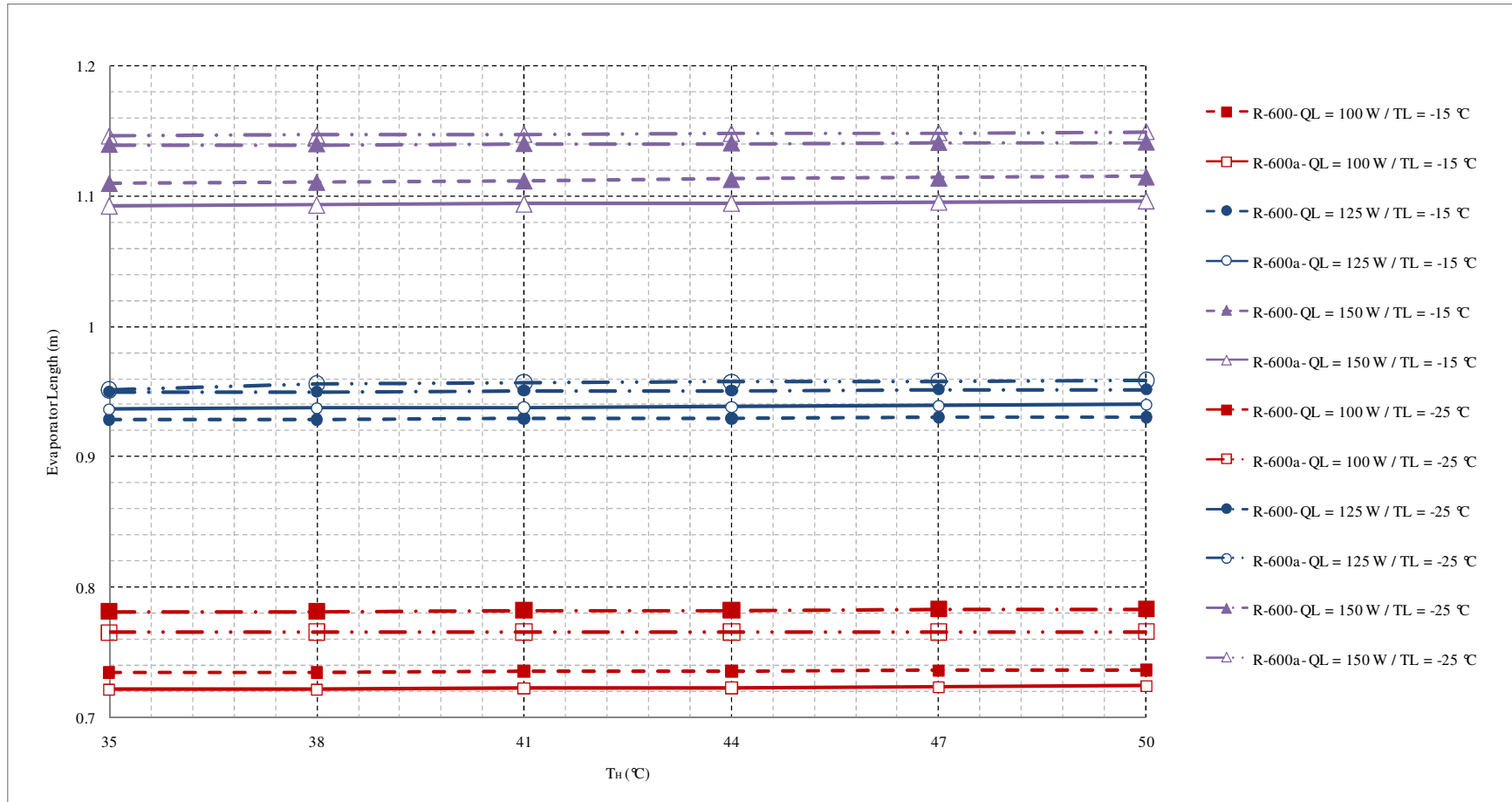
Figure 9.12 Evaporator Length vs T$_H$ for R-600 and R-600a at different cooling loads
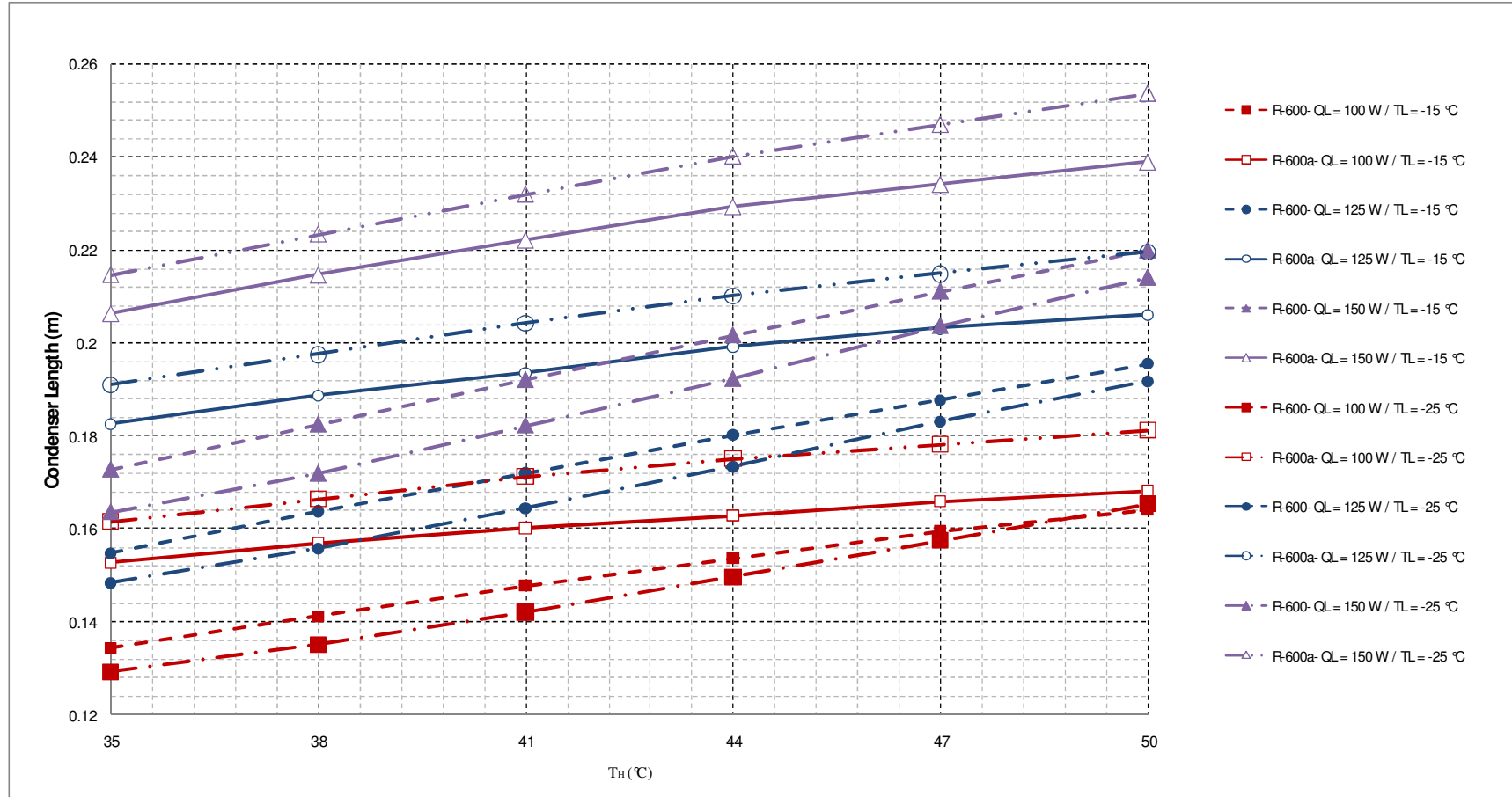
Figure 9.13 Condenser Length vs T_H for R-600 and R-600a at different cooling loads
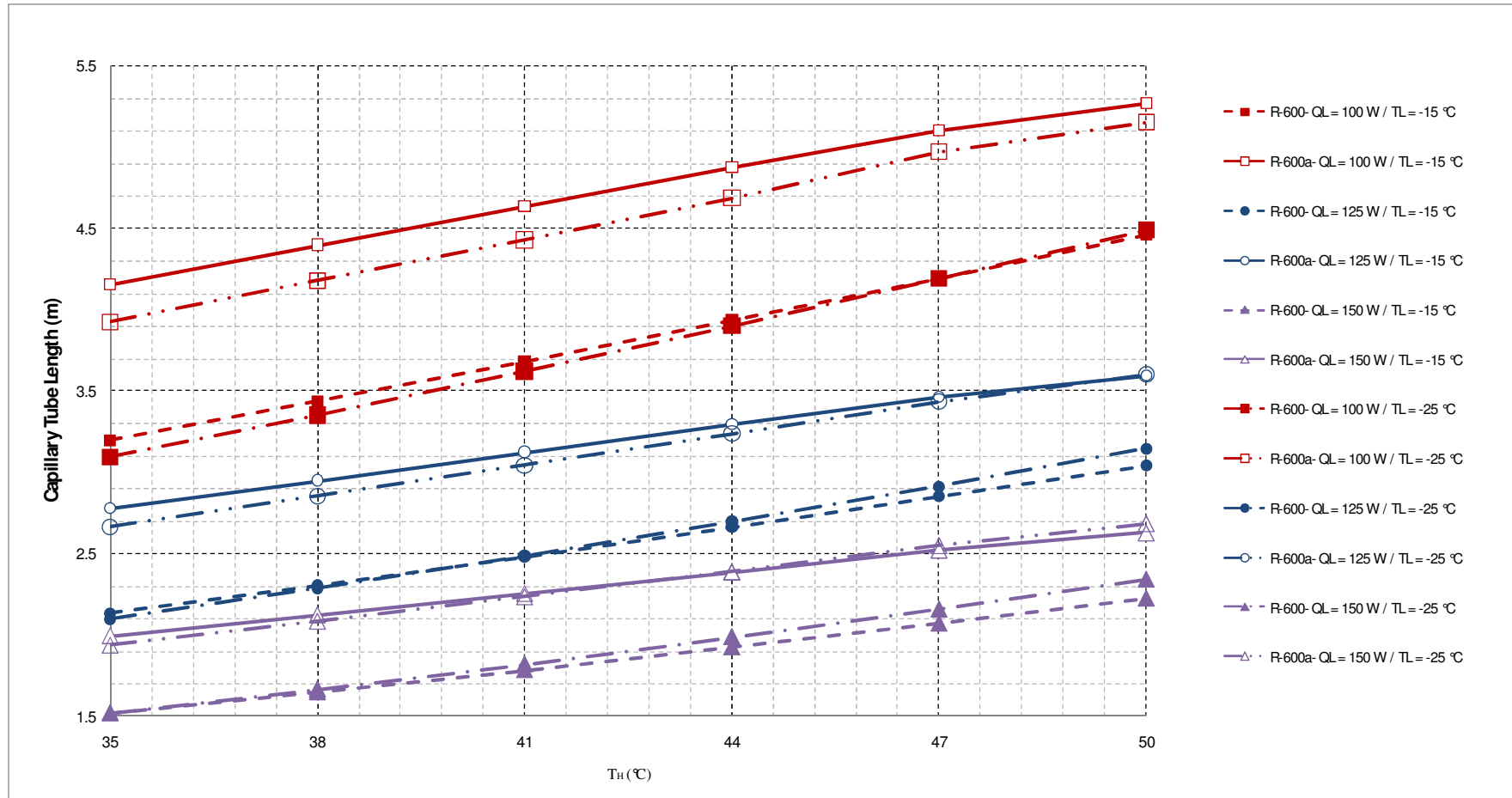
Figure 9.14 Capillary Tube Length vs T_H for R-600 and R-600a at different cooling loads
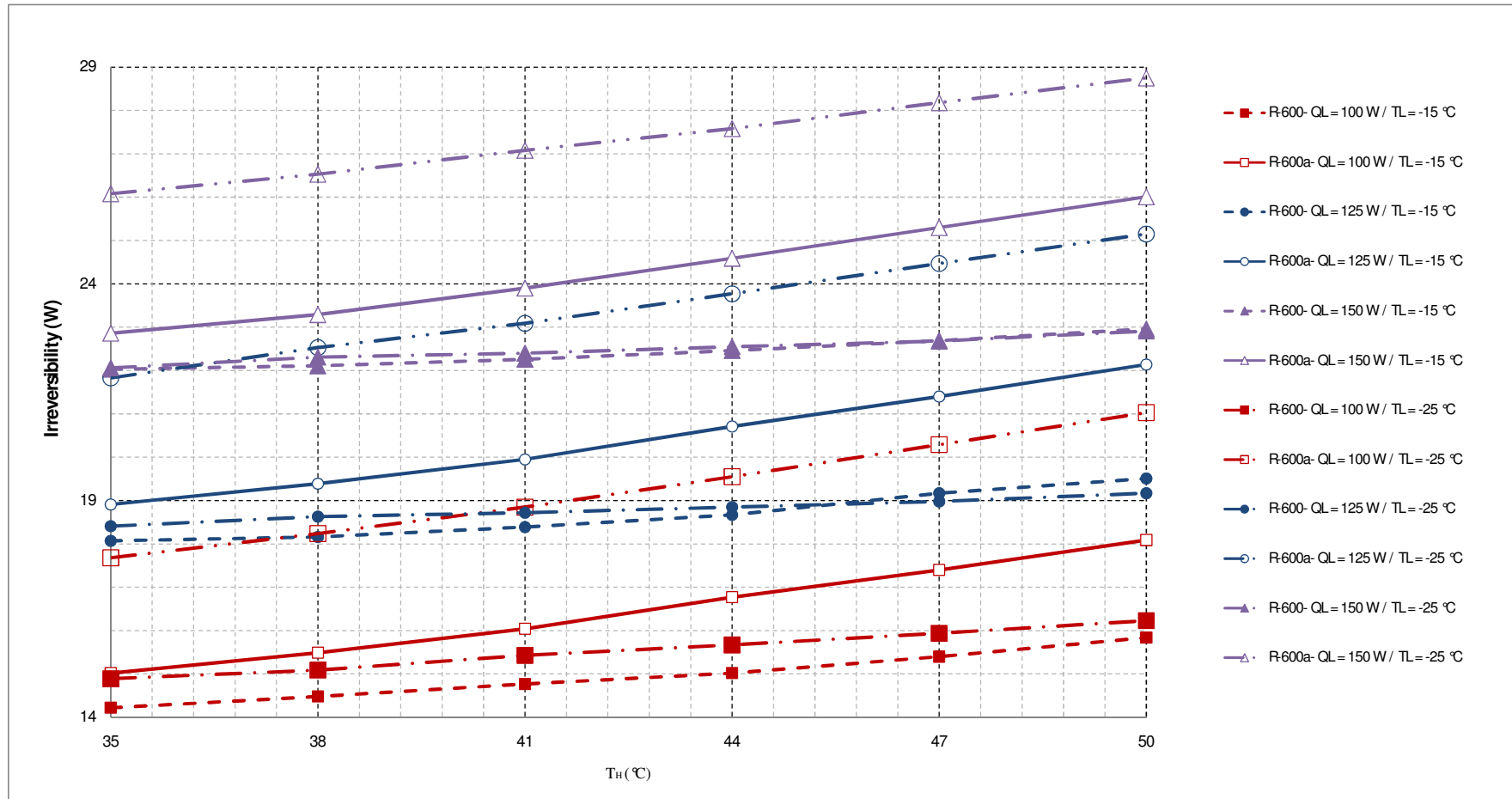
Figure 9.15 Total Cycle Irreversibility vs T_H for R-600 and R-600a at different cooling loads

For the comparison of refrigerant performance the room temperature is alternated between 298 K and 313 K room temperatures with 3 K temperature increments in order to represent different climatic conditions. Cold-room temperature is also varied between 273 K and 263 K representing the food and freezer compartments of the refrigerator. In addition the refrigeration load of the cycle is also varied between 100 W, 125 W and 150 W. COP, compressor power, evaporator length, condenser length, capillary tube length and total cycle irreversibility are used for refrigerant comparison.

Coefficient of performance is not affected from different refrigeration loads, since it does not affect the efficiency of the cycle. However as the condenser temperature increases COP decreases, because of the increased compressor power requirement due to the increase in total condenser load. Similarly as the evaporator and room temperatures decreases the COP also decreases. At the same refrigeration load and condenser temperature, R-600 gives better COP values than those of R-600a. Calculated COP for each run is given in Figure 9.10.

As expected, compressor power of the cycle increases with the increasing refrigeration load. Increase in condenser temperature also increases the compressor power because of the same reason it increases the COP. In agreement with the COP, smaller evaporator temperatures require smaller compressor powers. As it comes to refrigerants, similar to the case in COP, for the same refrigeration load and condenser temperature, R-600 requires smaller compressor power than those of R-600a. Calculated compressor power of each run is given in Figure 9.11.

As the condenser temperature changes the evaporator length remains almost the same for each refrigerant. For small refrigeration loads, at the same evaporator

and condenser temperatures the required evaporator lengths for R-600 are higher than those for R-600a. However for high refrigeration loads, at the same evaporator and condenser temperatures, the required evaporator lengths for R-600 are lower than those for R-600a. Calculated evaporator length of each run is given in Figure 9.12.

Condenser length increases as the condenser temperature and cooling load increases for both R-600 and R-600a. For the same condenser temperature and cooling load, condenser length requirement of the system charged with R-600a is longer than the requirement for R-600. Calculated condenser length of each run is given in Figure 9.13.

Capillary tube length increases as the condenser temperature increases. At the same cooling load and temperature each refrigerant requires almost the same capillary tube length regardless of the evaporator temperature. On the other hand, for the same condenser temperature the required capillary tube length decreases with increasing cooling load. At the same cooling load and condenser temperature, the system that uses R-600a requires a longer capillary tube than R-600. Calculated capillary tube length of each run is given in Figure 9.14.

For the same cooling load and condenser temperature the irreversibility of the system with R-600a is higher than the irreversibility of the system working with R-600 which may be used as a refrigerant selection criterion. Calculated irreversibility of each run is given in Figure 9.15.

CHAPTER 10

CONCLUSION

In this study, a computer program is written in order to perform preliminary design of a vapor compression, household refrigeration cycle and to evaluate its performance characteristics. For fast and accurate calculation of thermophysical properties of the refrigerants at different states, least squares polynomial approximation method is utilized. In order to evaluate the effectiveness of the refrigerants a vapor-compression refrigeration cycle is designed with the program for different temperature ranges.

10.1 Absolute Percentage Deviation of Thermophysical Property Calculations

As mentioned above least squares polynomial approximation is used for calculating the thermophysical properties of the refrigerants at desired states. As the tables given in Section 3 indicate, absolute percentage deviations of the calculated values from the actual tabulated data are very small and acceptable for use in the design software. A maximum absolute percentage deviation of on the order of 10^{-2} % is selected as the limit. As the tables indicate the selected

coefficients satisfy this requirement for each thermophysical property. Deviation on the order of 10^{-13} % was also reached.

For some thermophysical properties like saturation temperature and superheated vapor density, least squares polynomial approximation could not satisfy the maximum allowable percentage deviation of 10^{-2} %. For these properties the natural logarithms of the input parameters are used for the coefficient calculations. This procedure can be applied to any thermophysical property for better percentage deviations. However using natural logarithms will cost computation time and computing power, as a result this approach is avoided if the percentage deviation requirement could be satisfied directly from the input parameters.

For smaller percentage deviations higher order polynomials than the 5th order that is used in this study can be used. Similar to the natural logarithm approach despite better percentage deviation, higher order polynomial approximation will consume much more computation time and power. In order to show the effect of higher order polynomials on thermophysical property calculations, coefficients and percentage deviations of the polynomials of R-600 saturated liquid density calculations at 270 K – 320 K range is given in Table 10.1.

Table 10.1 Coefficients of saturated liquid density equation of R-600 for 5th, 6th and 7th degree polynomial approximations

Polynomial Order	5th	6th	7th
a_0	5.55E+02	7.45E+05	-5.99E+06
a_1	9.93E+00	-1.52E+04	1.45E+05
a_2	-1.12E-01	1.30E+02	-1.51E+03
a_3	5.04E-04	-5.88E-01	8.71E+00
a_4	-1.06E-06	1.50E-03	-3.01E-02
a_5	8.44E-10	-2.04E-06	6.24E-05
a_6	-	1.15E-09	-7.18E-08
a_7	-	-	3.53E-11
% deviation	3.70E-03	3.29E-03	3.23E-03

As it is observed in Table 10.1 higher order polynomials give better percentage deviations, however the order of deviations are the same. After an optimization between the percentage deviations and computing times, selection of 5th order polynomials for thermophysical property calculations will give satisfactory results. Each thermophysical property polynomial used in this study is represented in graphical form with tabulated data for comparison is given in the Appendix.

In conclusion, least squares polynomial approximation is a fast and accurate method for thermophysical property calculation of different refrigerants. This method can be used for development of a database of thermophysical properties for use in vapor-compression refrigeration cycle design software.

10.2 Performance Characteristics of Hydrocarbons in Household Refrigerators

Considering the performance characteristics R-600 could be a better selection than R-600a. However, butane is has a lower boiling point (-2 °C) than isobutane (-12 °C) at 1 atm, which allows isobutane to work at lower temperatures. As a result, despite its slightly lower performance characteristics isobutane is generally preferred to butane in the market.

It is observed that fin efficiency of evaporator is almost 100% for each configuration. As a result for faster computations fin efficiency calculation could be eliminated and evaporator could be designed as a horizontal plate considering the surface areas of the tubes and the flat parts between tubes.

10.3 Suggestions on Future Studies

In this study, a database for thermophysical properties of hydrocarbon refrigerants is formed. Although the current polynomial equations give maximum deviations from the tabulated data on the order of 10^{-2} %, it can be further improved by utilizing higher order polynomials or using natural logarithms of the input parameters. Also other equation derivation methods can be investigated for better deviations and small computation times.

For the performance characteristics evaluation of the refrigerants, a design computer program is written with basic refrigeration components. In future studies these refrigerator components could be changed with other alternative. There is an increasing trend in the market about using expansion valves as

expansion devices instead of capillary tubes. Further software development could focus on this area.

For the refrigeration loads, simply the required capacity is designated as an input for the software. For better understanding of how refrigerators behave at different conditions, an experimental study can be carried out for determining the head load requirement of basic substances stored in household refrigerators.

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APPENDIX

A.   GRAPHS OF REFRIGERANT EQUATIONS
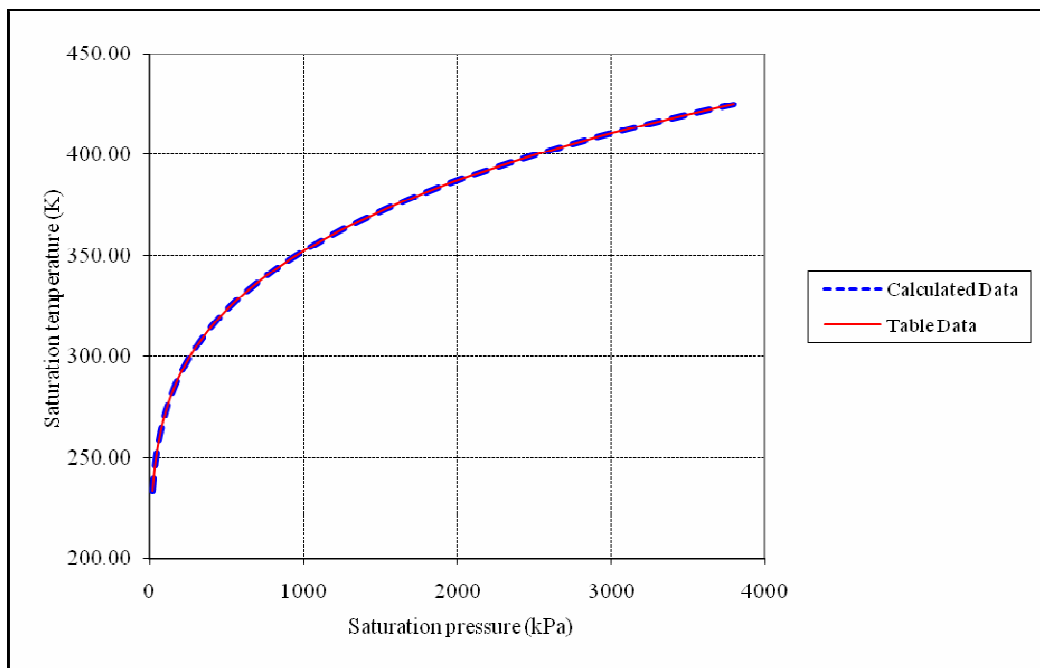
Figure A. 1 Saturation temperature vs. saturation pressure
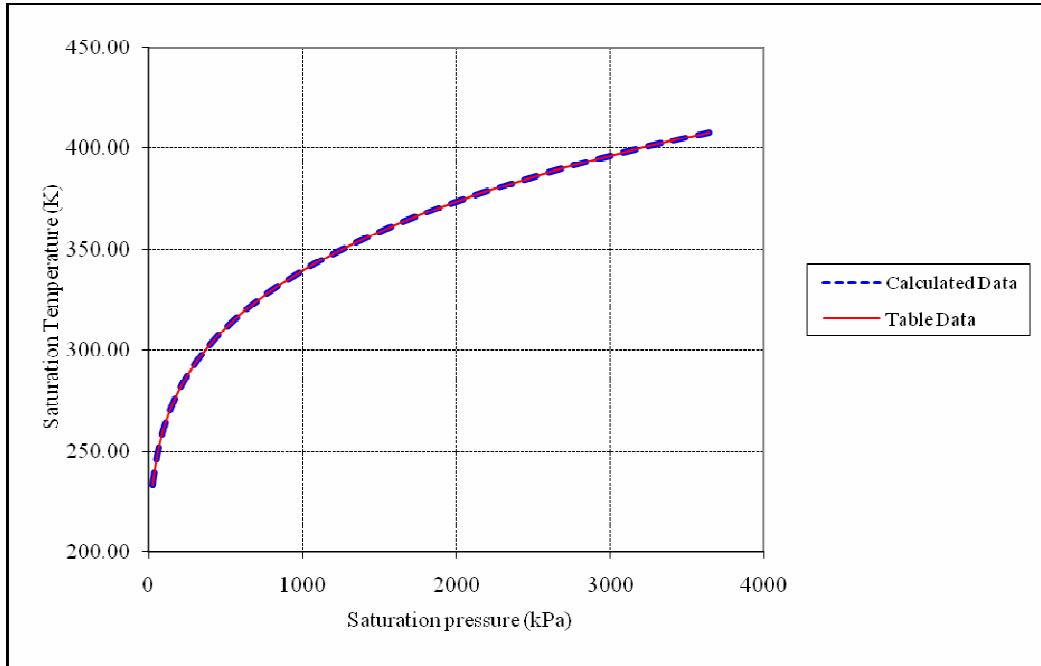
graph for R-600

Figure A. 2 Saturation temperature vs. saturation pressure
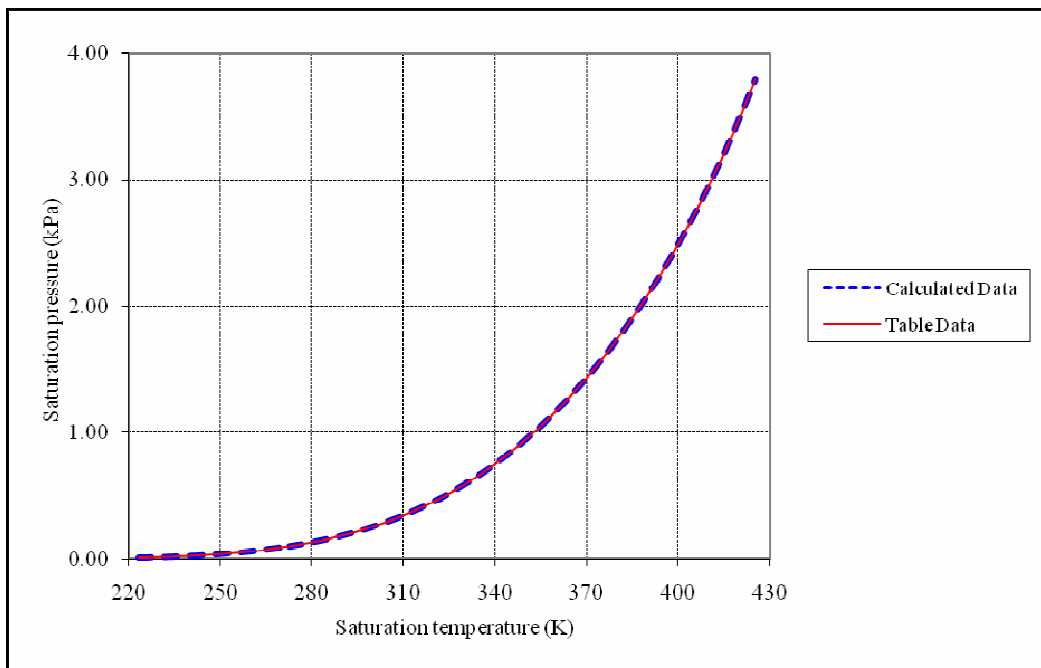
graph for R-600a

Figure A. 3 Saturation pressure vs. saturation temperature

graph for R-600

124
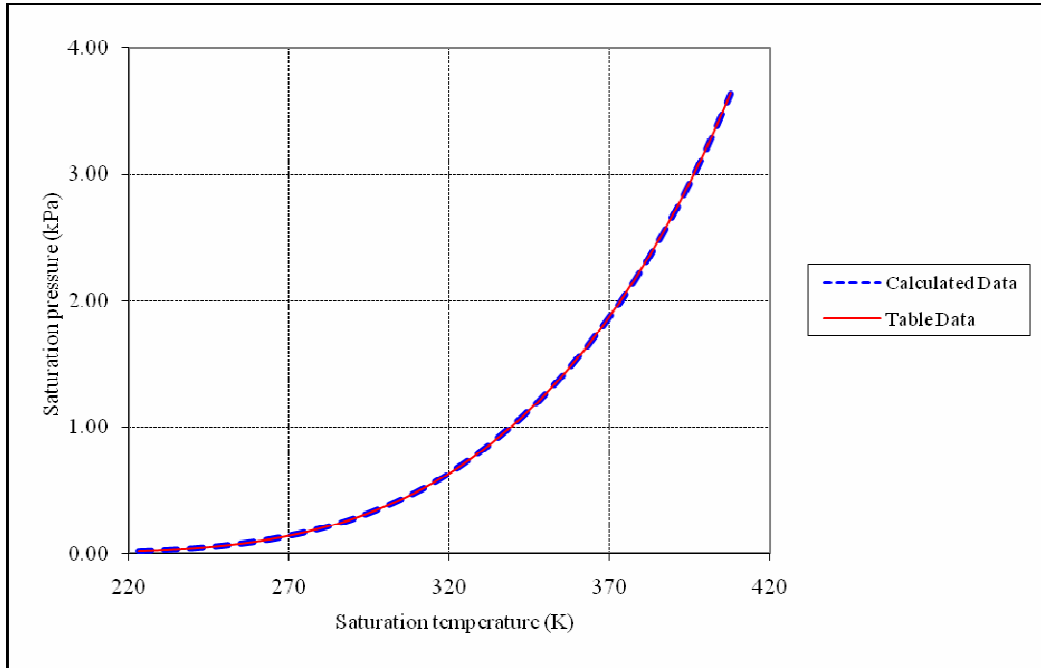
Figure A. 4 Saturation pressure vs. saturation temperature
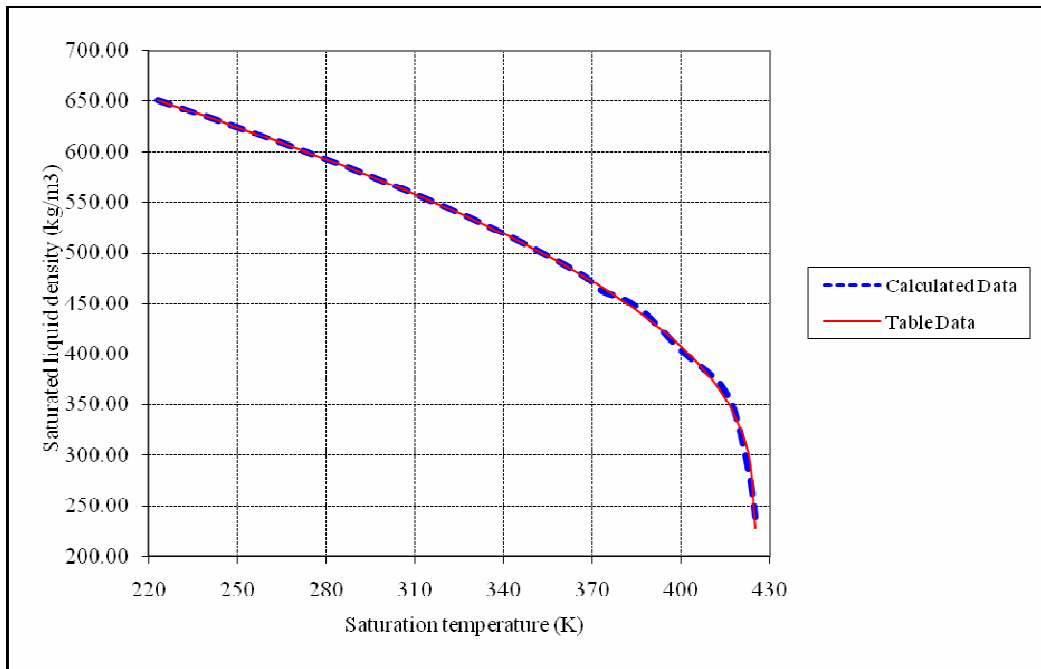
graph for R-600a

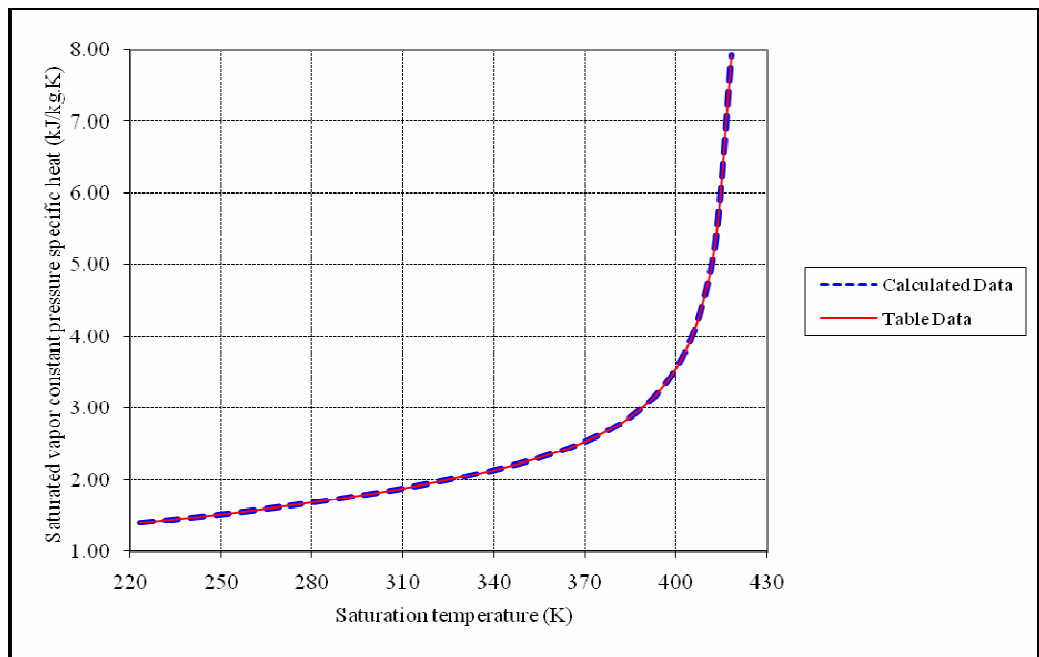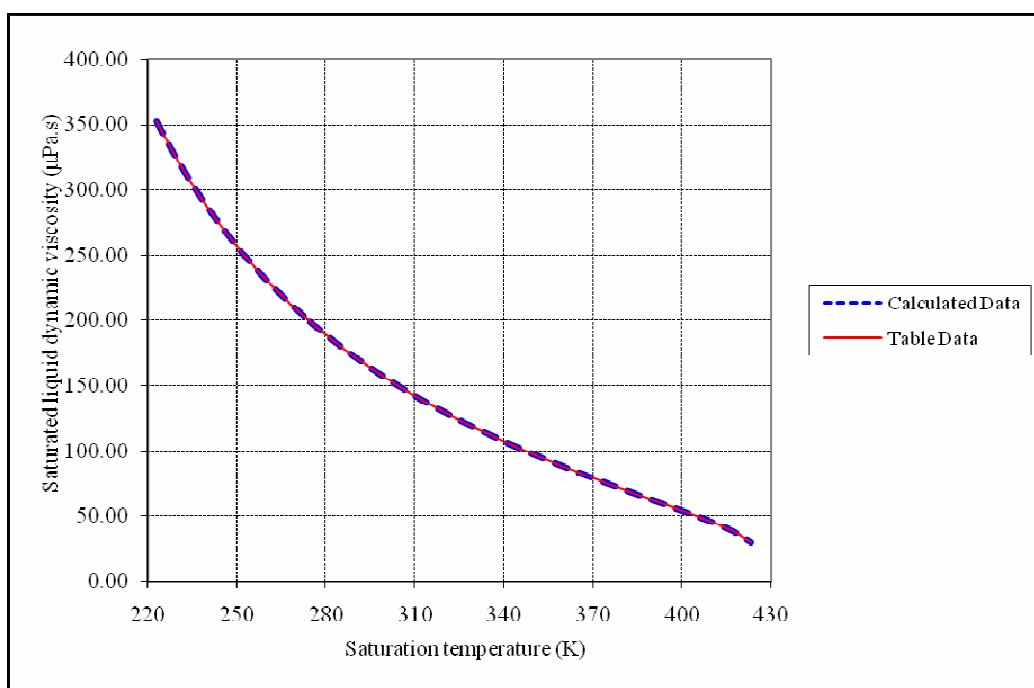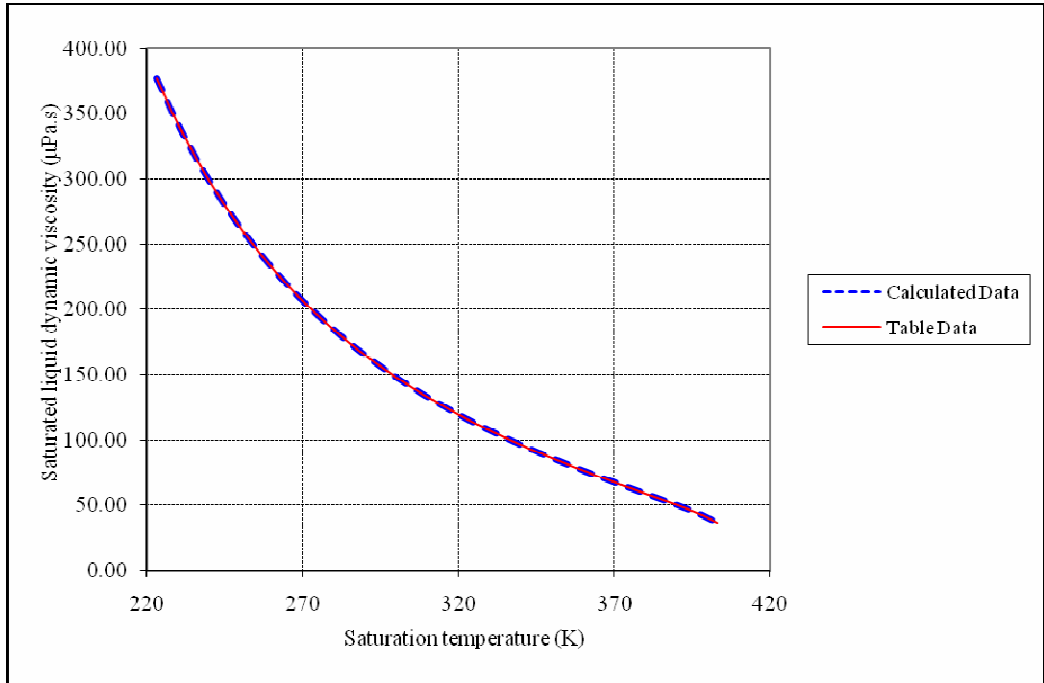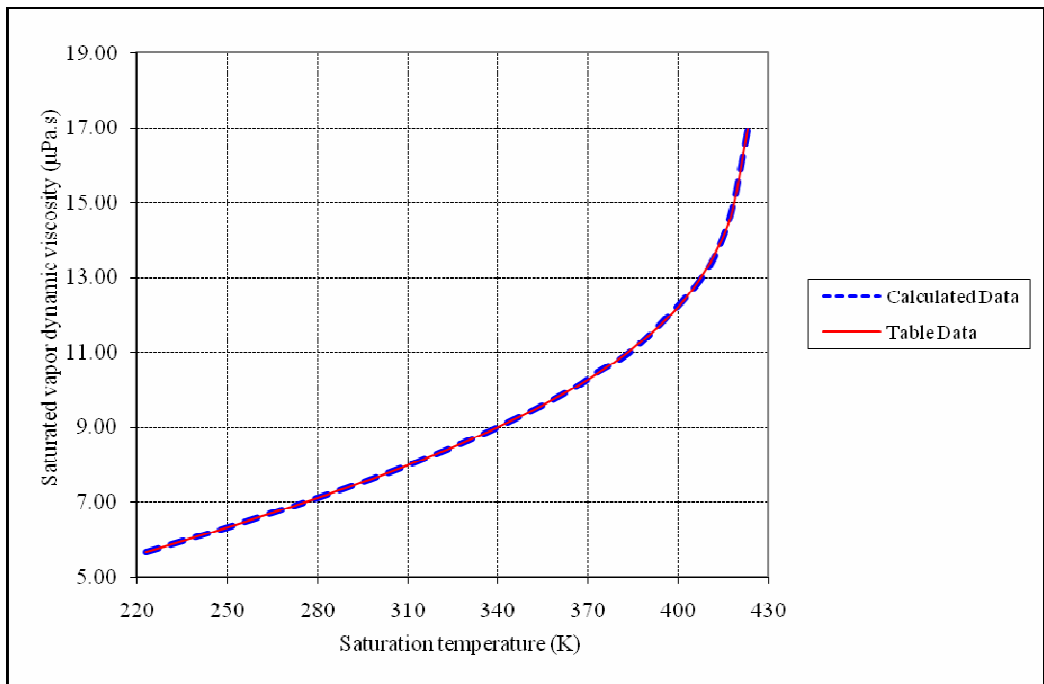Figure A. 5 Saturated liquid density vs. saturation temperature

graph for R-600

125
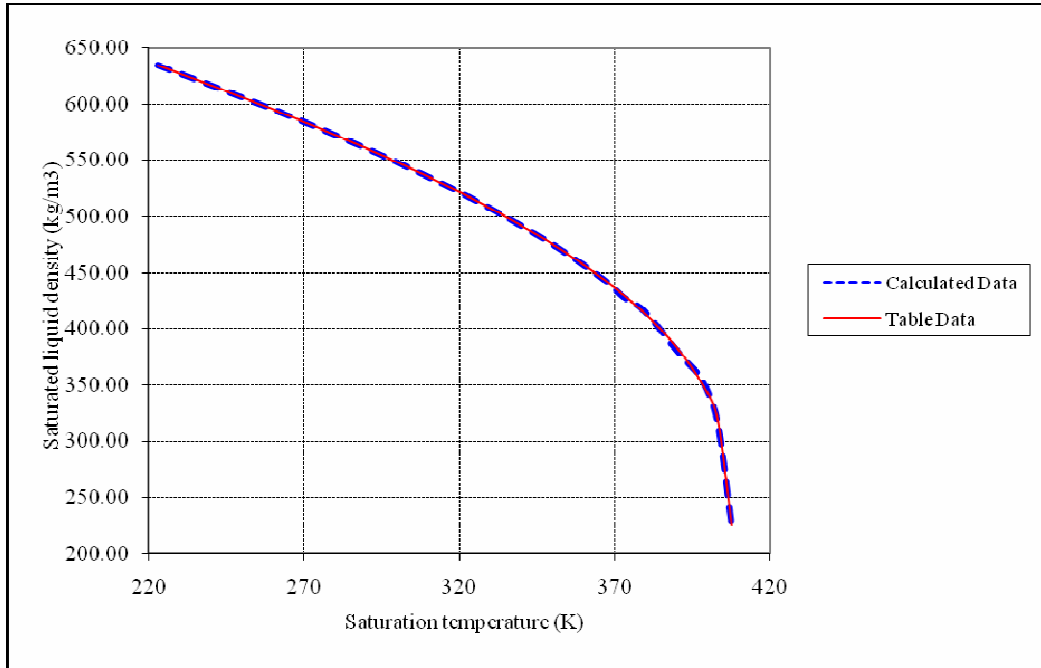
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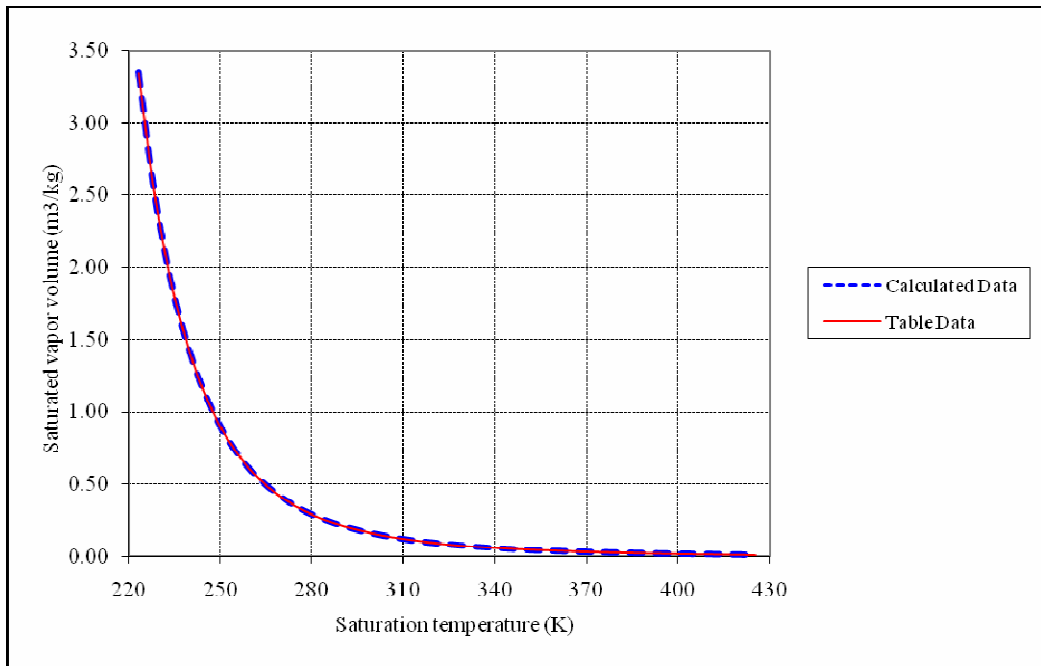
graph for R-600a

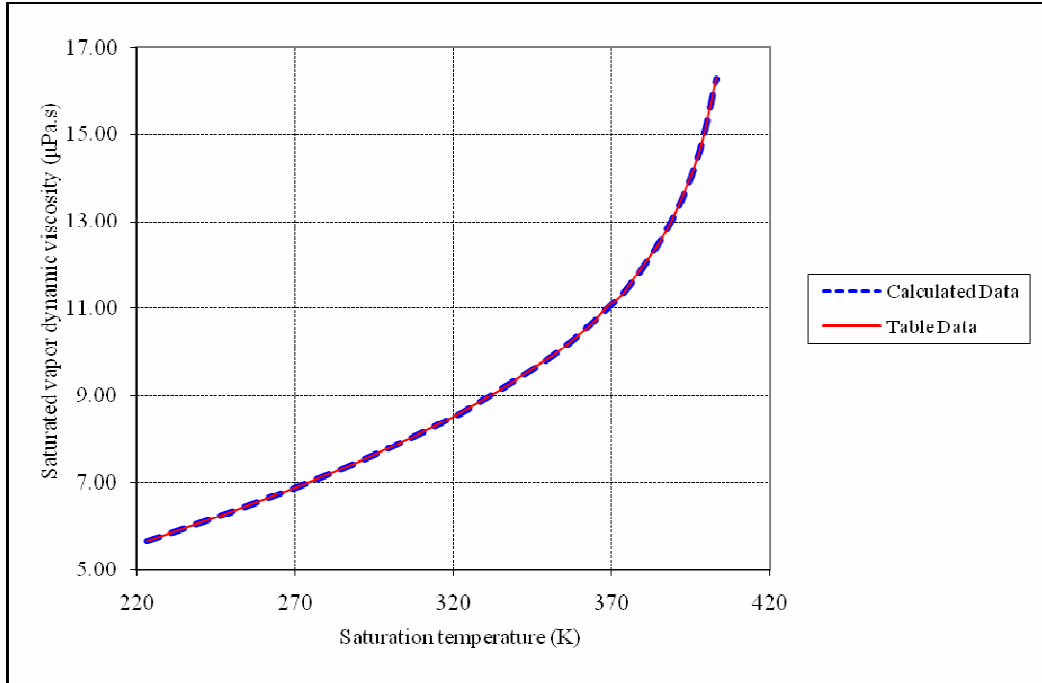Figure A. 7 Saturated vapor density vs. saturation temperature
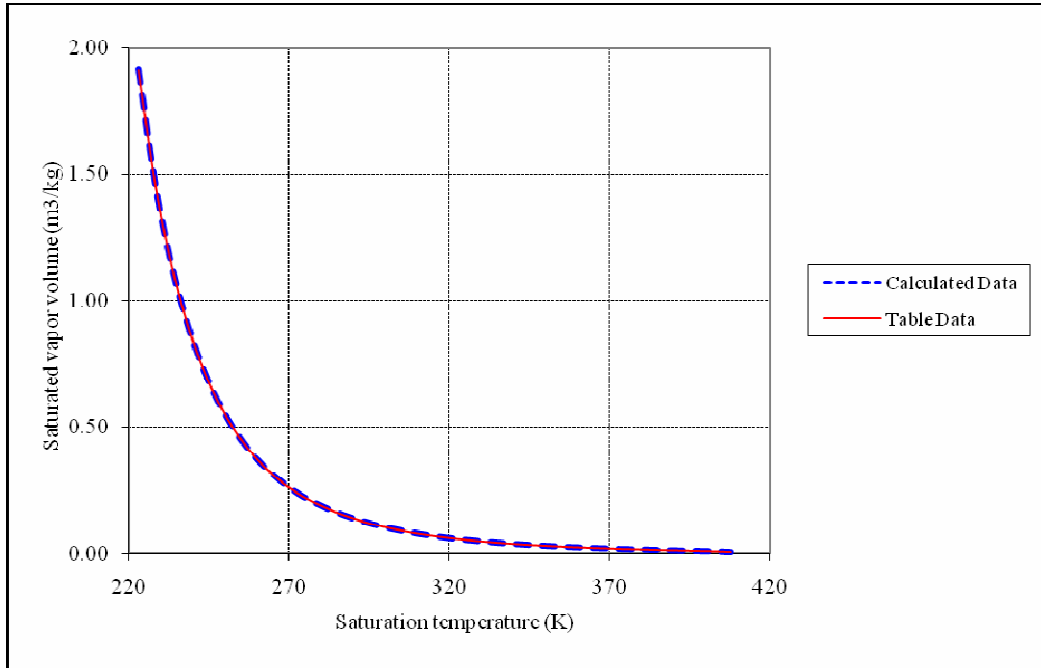
graph for R-600

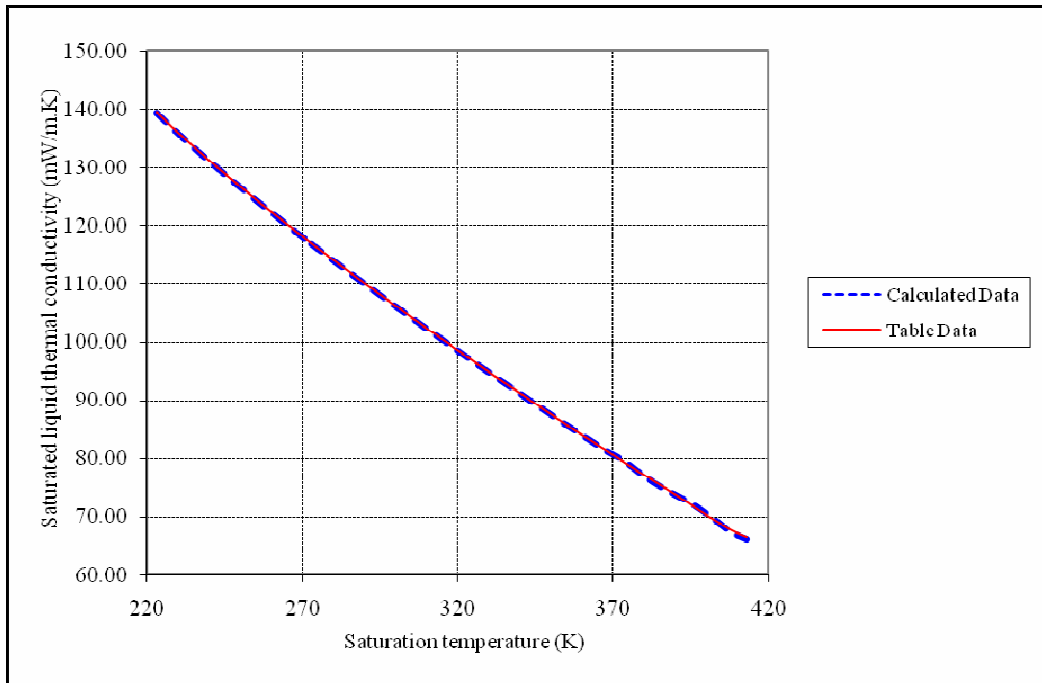Figure A. 8 Saturated vapor density vs. saturation temperature
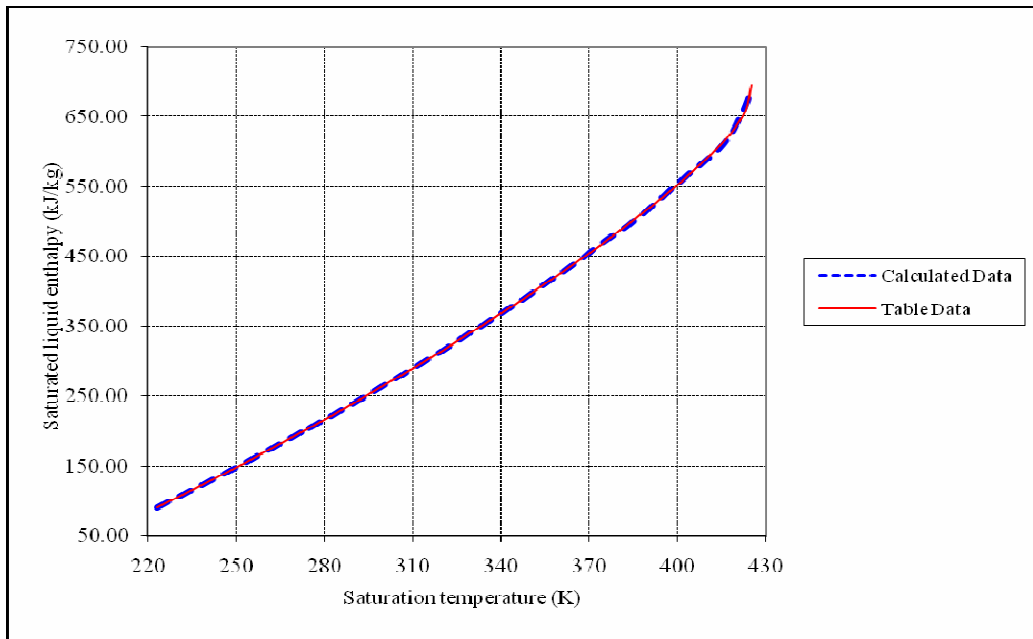
graph for R-600

Figure A. 9 Saturated liquid enthalpy vs. saturation temperature
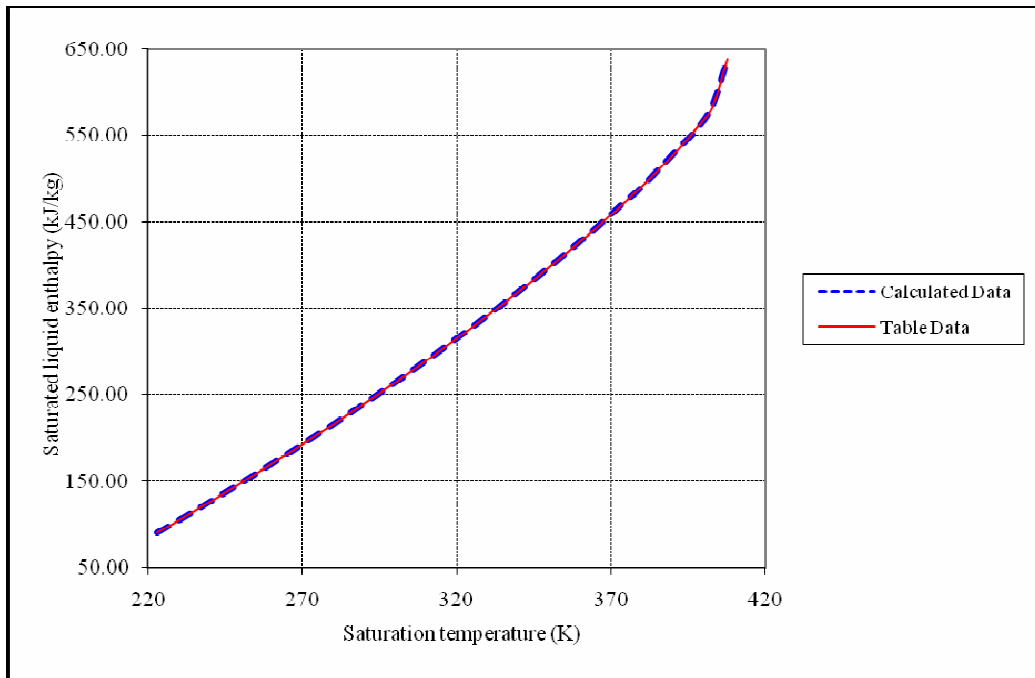
graph for R-600

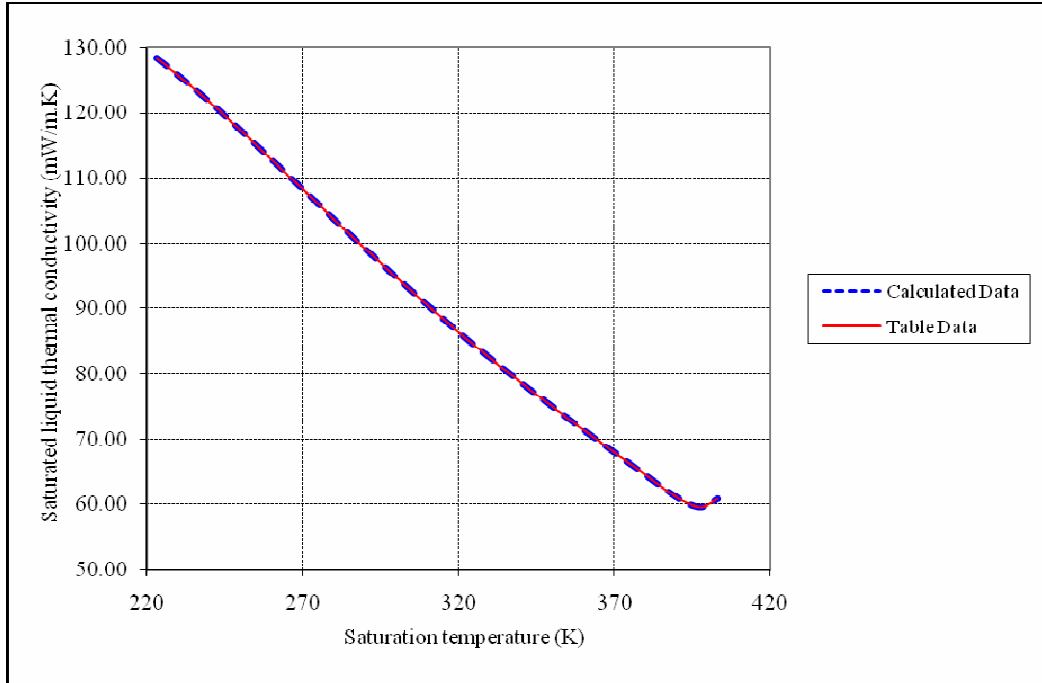Figure A. 10 Saturated liquid enthalpy vs. saturation temperature
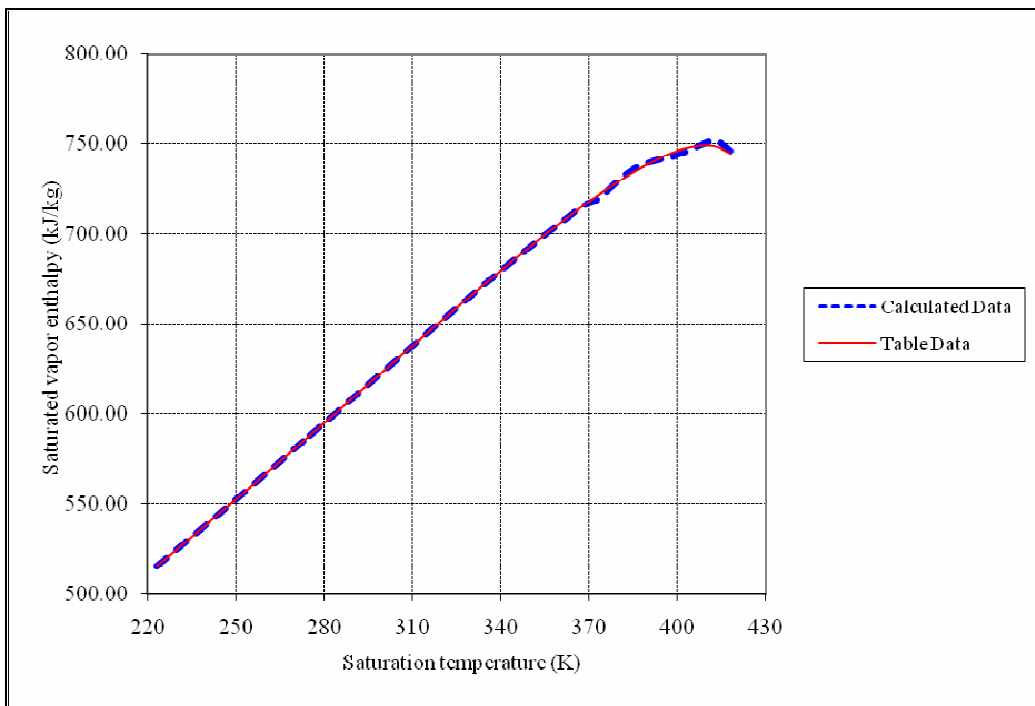
graph for R-600a

Figure A. 11 Saturated vapor enthalpy vs. saturation temperature
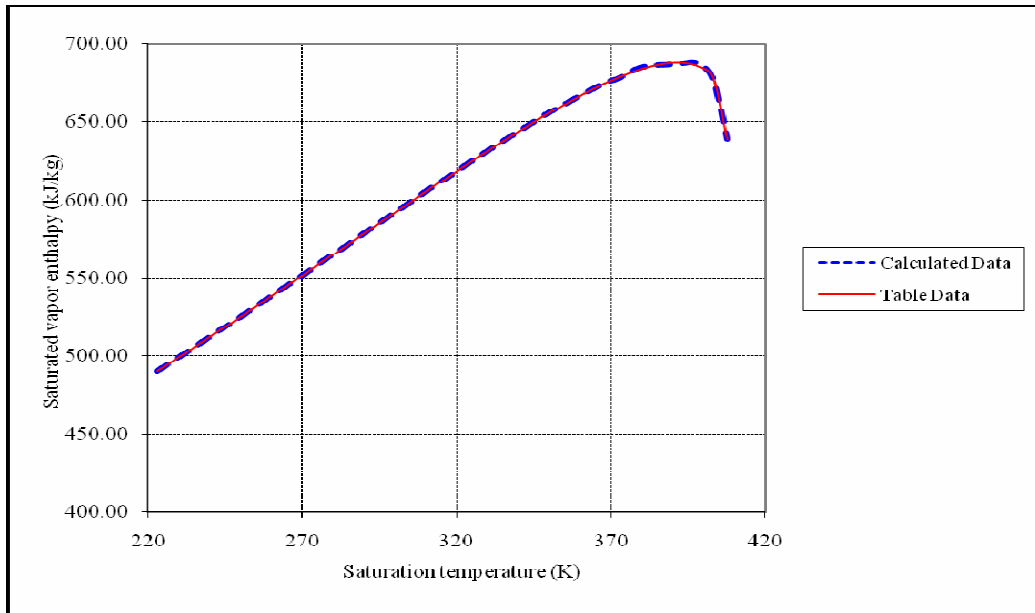
graph for R-600

Figure A. 12 Saturated vapor enthalpy vs. saturation temperature
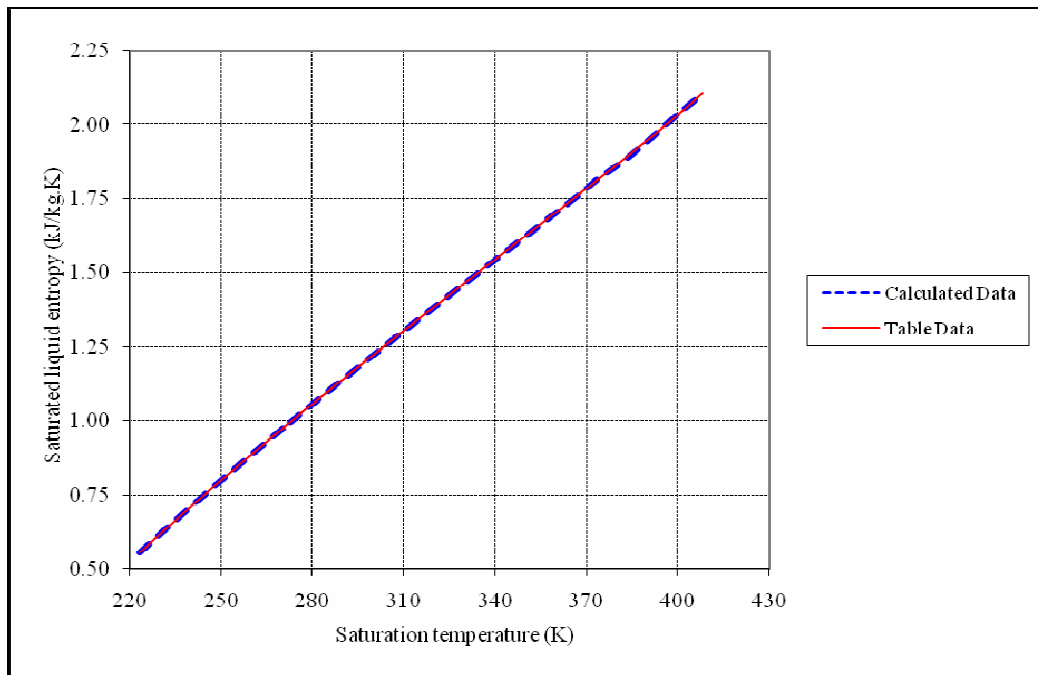
graph for R-600a

Figure A. 13 Saturated liquid entropy vs. saturation temperature
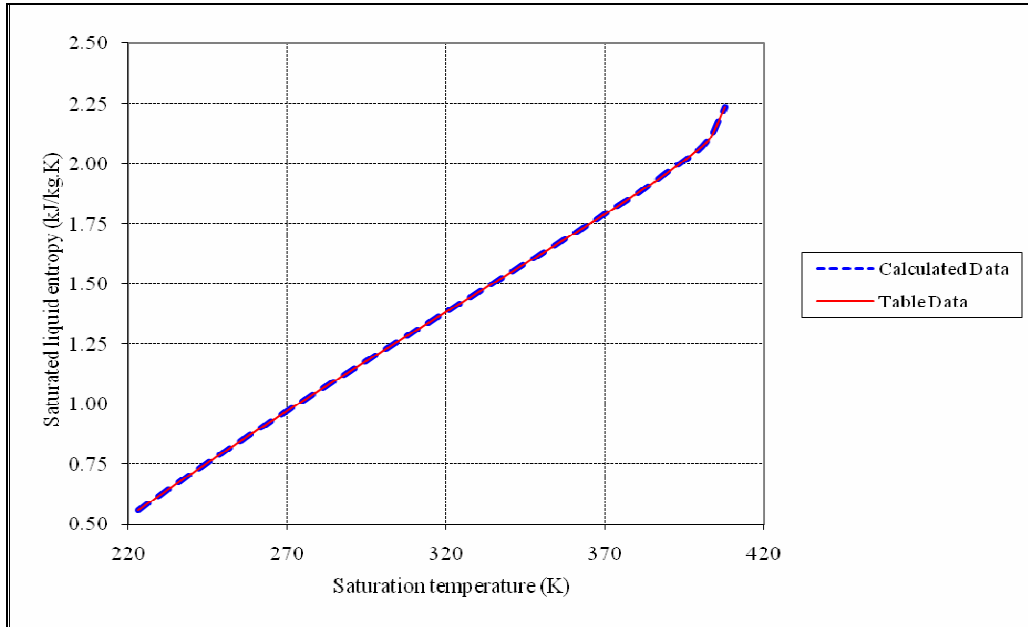
graph for R-600

Figure A. 14 Saturated liquid entropy vs. saturation temperature
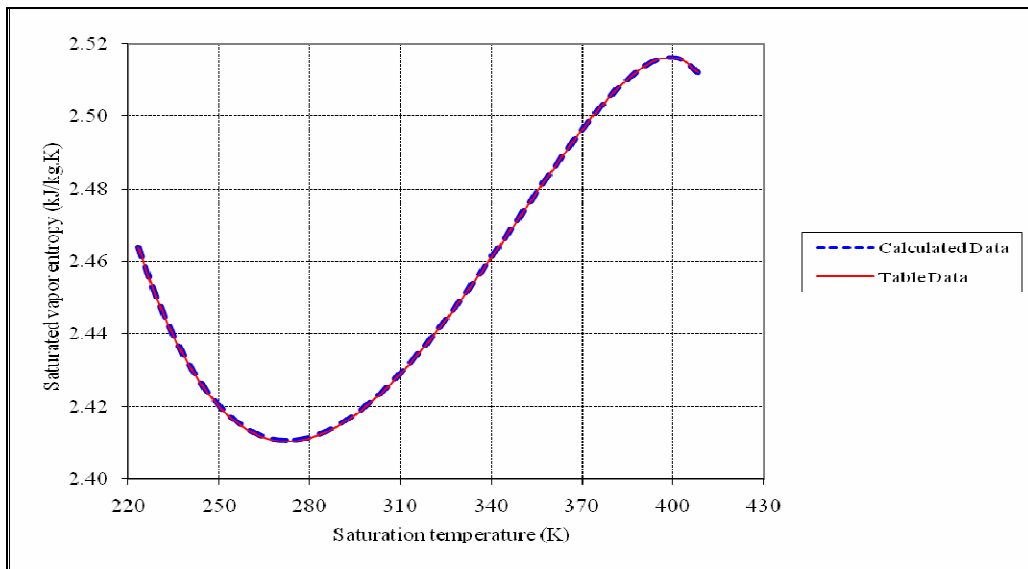
graph for R-600a

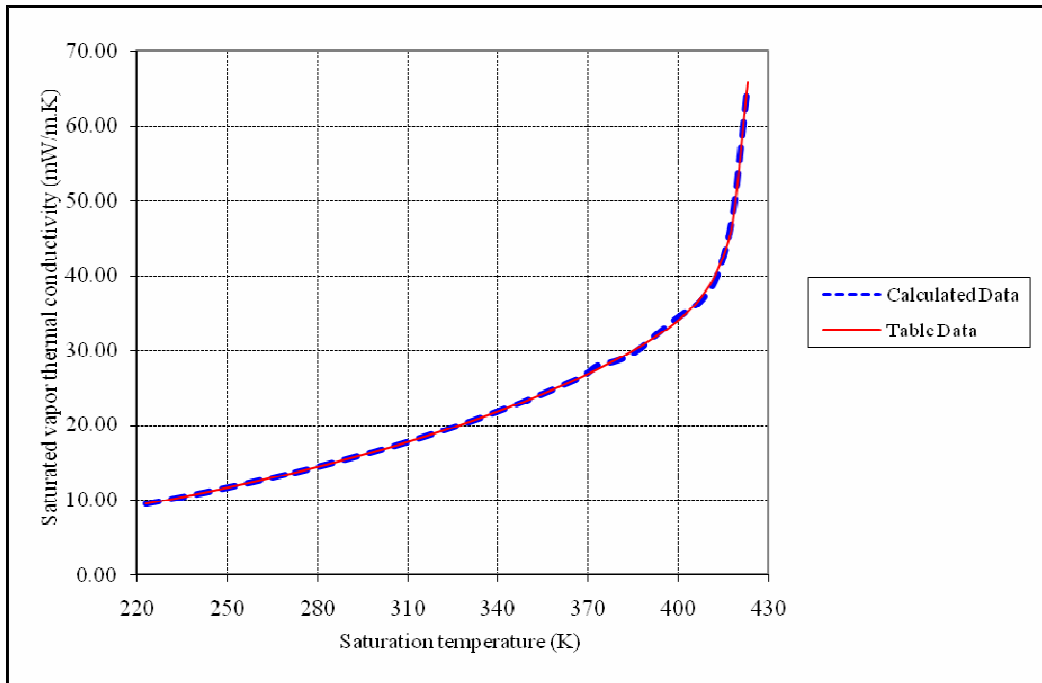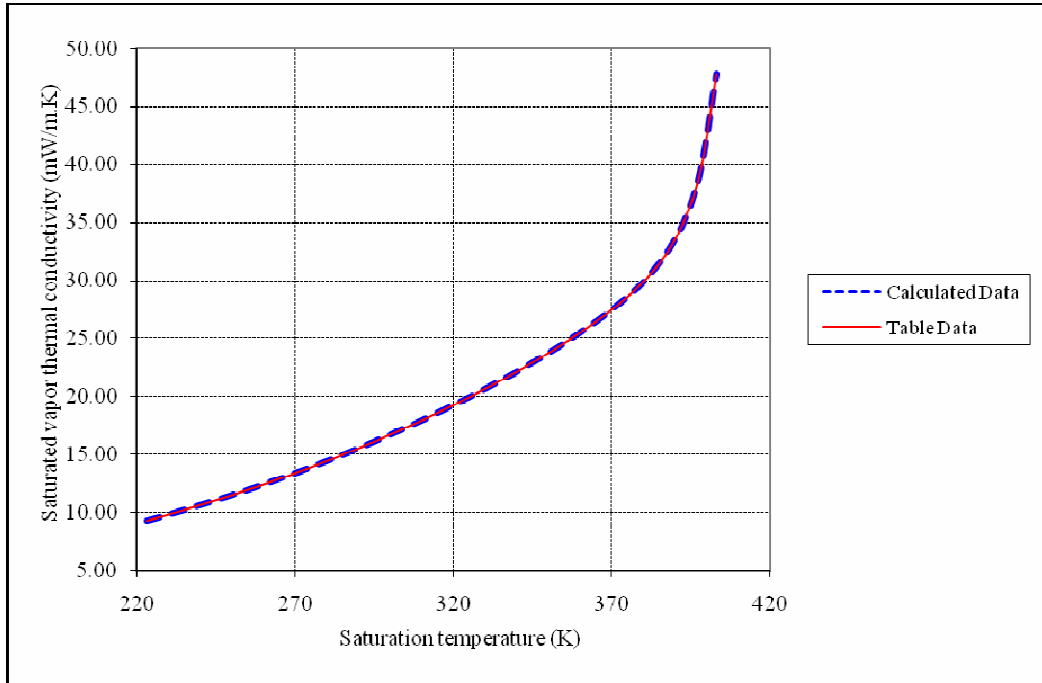Figure A. 15 Saturated vapor entropy vs. saturation temperature
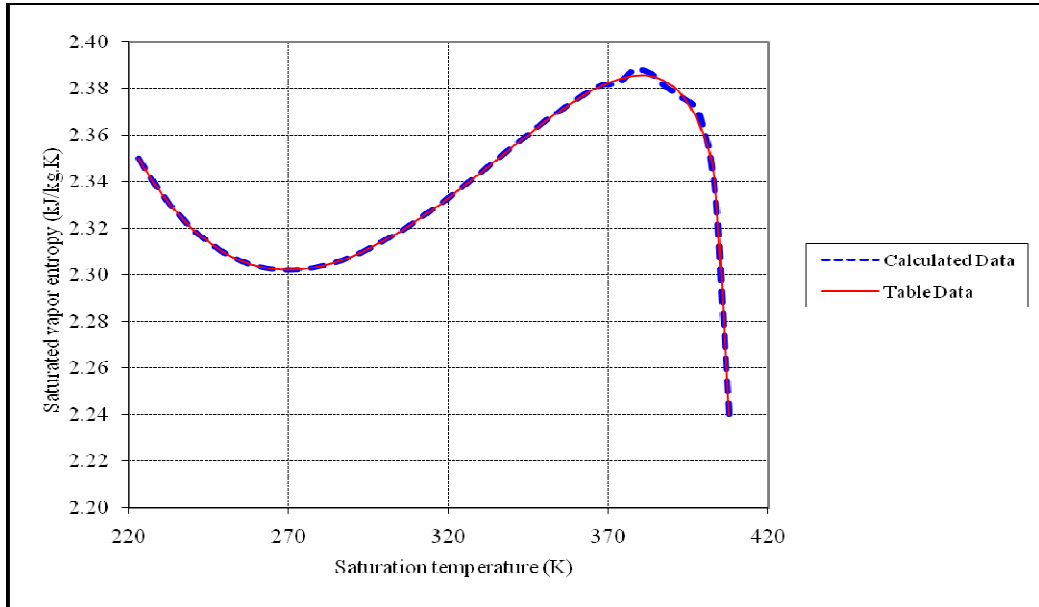
graph for R-600

Figure A. 16 Saturated vapor entropy vs. saturation temperature
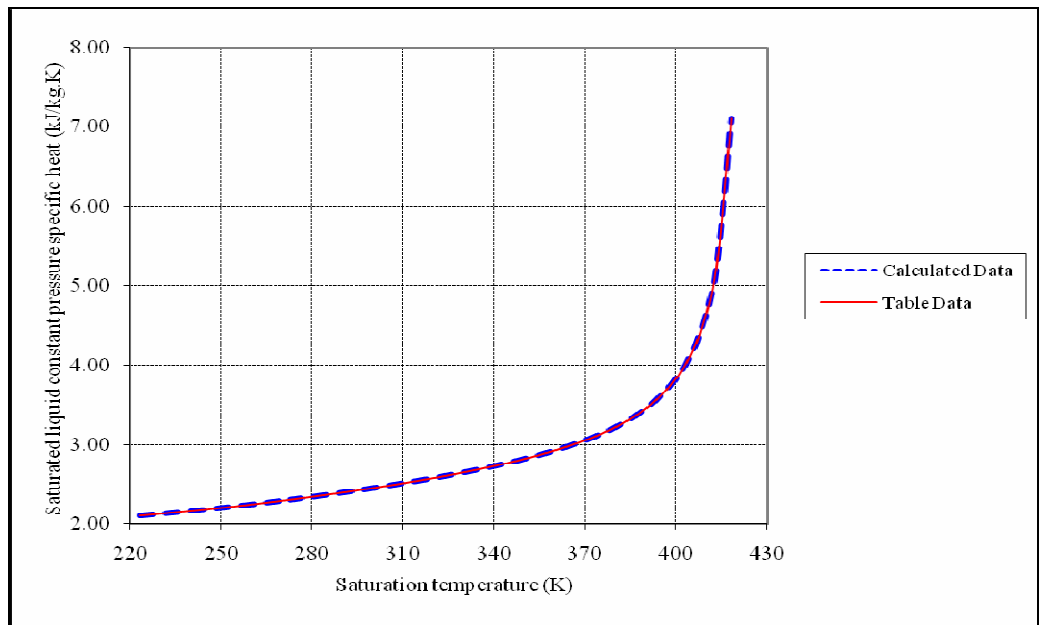
graph for R-600a

Figure A. 17 Saturated liquid constant pressure specific heat vs. saturation

temperature graph for R-600
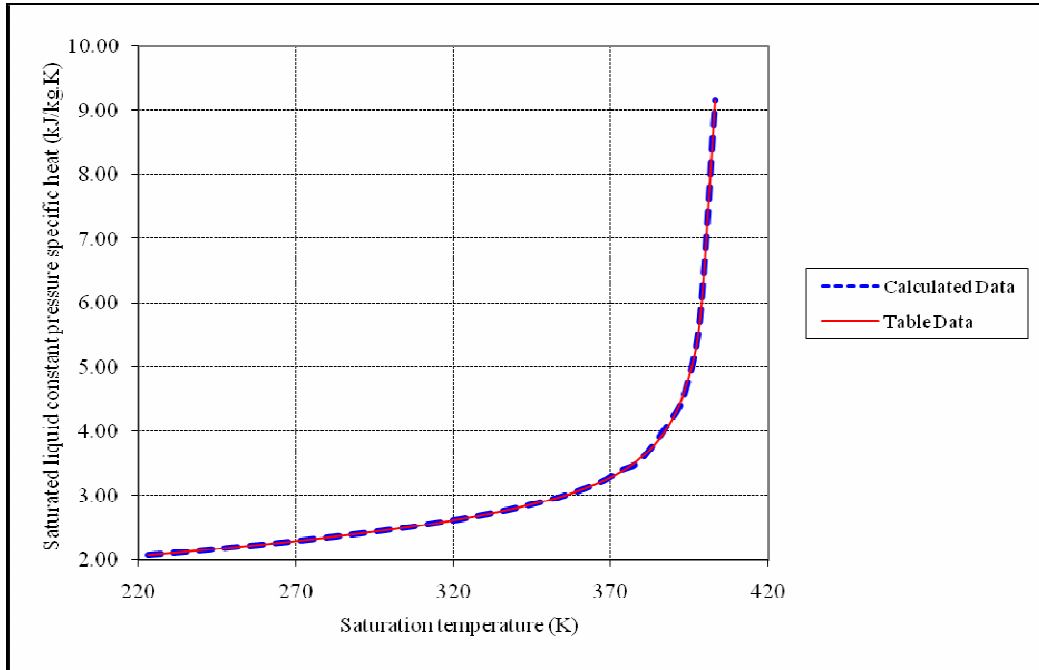
Figure A. 18 Saturated liquid constant pressure specific heat vs. saturation
temperature graph for R-600a

Figure A. 19 Saturated vapor constant pressure specific heat vs. saturation
temperature graph for R-600
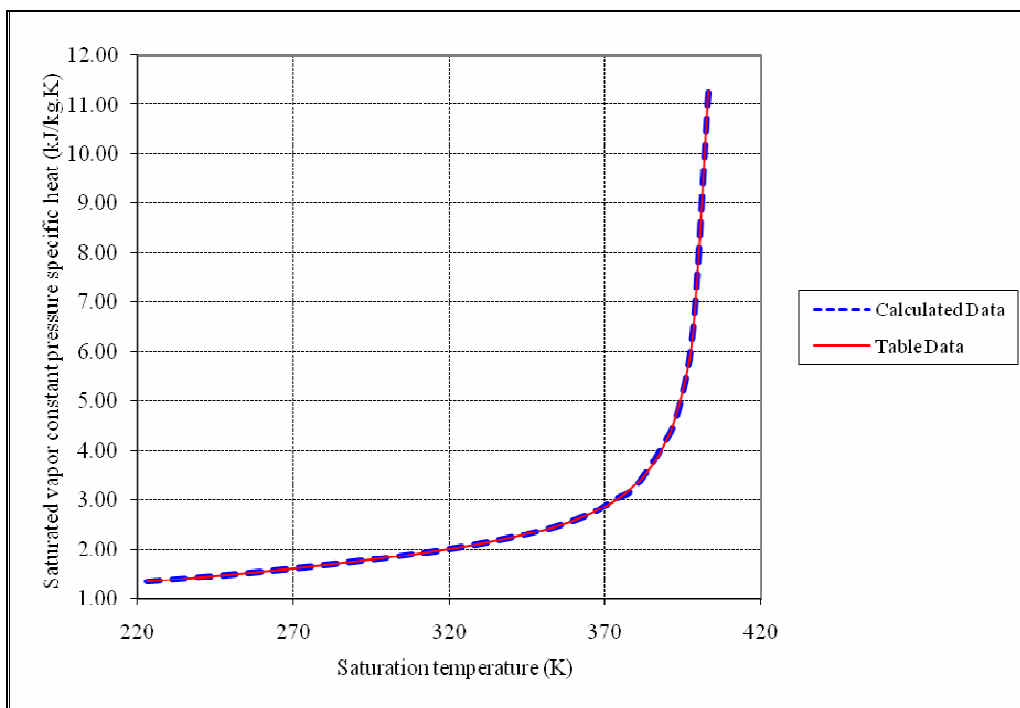
Figure A. 20 Saturated vapor constant pressure specific heat vs. saturation temperature graph for R-600a


Figure A. 21 Saturated liquid dynamic viscosity vs. saturation temperature graph for R-600

Figure A. 22 Saturated liquid dynamic viscosity vs. saturation temperature
graph for R-600a

Figure A. 23 Saturated vapor dynamic viscosity vs. saturation temperature
graph for R-600

Figure A. 24 Saturated vapor dynamic viscosity vs. saturation temperature graph for R-600a

Figure A. 25 Saturated liquid thermal conductivity vs. saturation temperature graph for R-600

Figure A. 26 Saturated liquid thermal conductivity vs. saturation temperature
graph for R-600a

Figure A. 27 Saturated vapor thermal conductivity vs. saturation temperature
graph for R-600

Figure A. 28 Saturated vapor thermal conductivity vs. saturation temperature
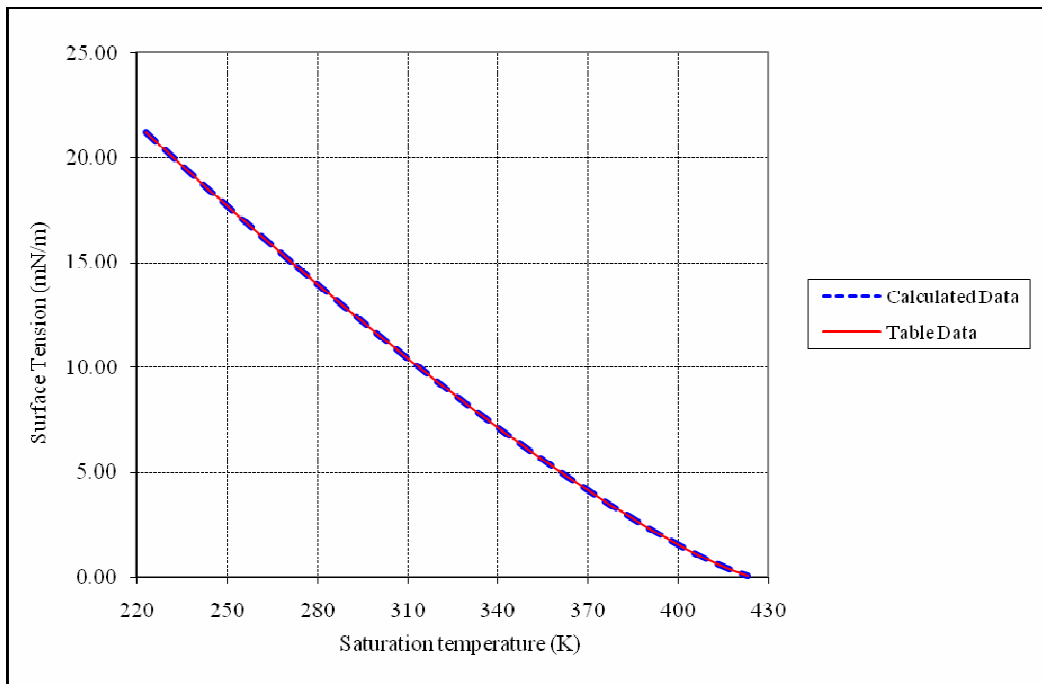
graph for R-600a

Figure A. 29 Surface tension vs. saturation temperature
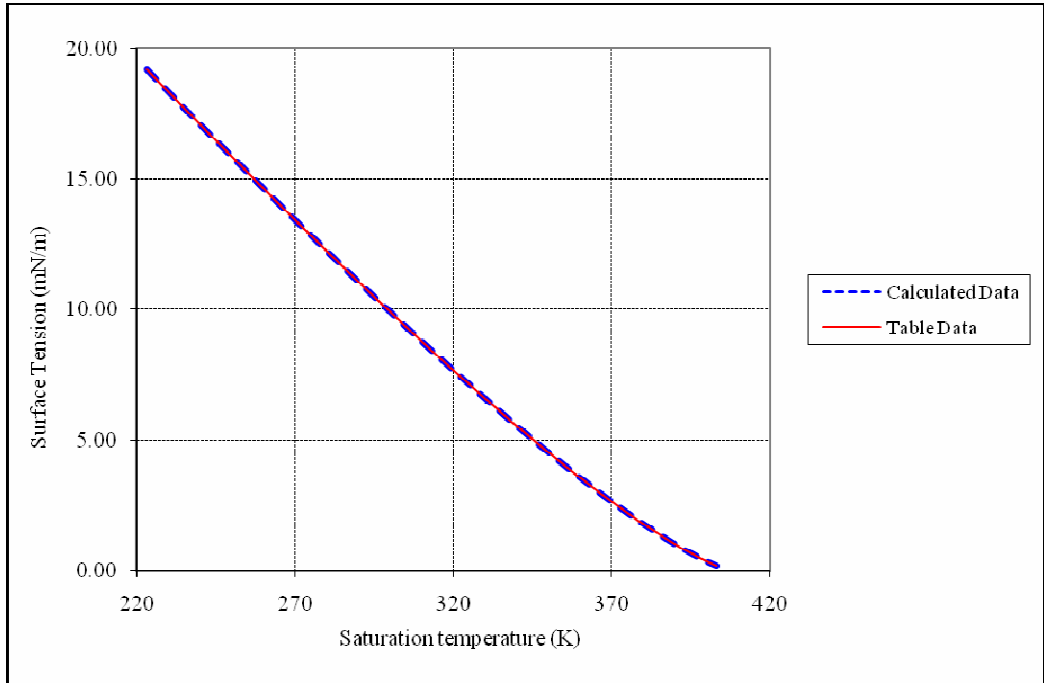
graph for R-600

Figure A. 30 Surface tension vs. saturation temperature
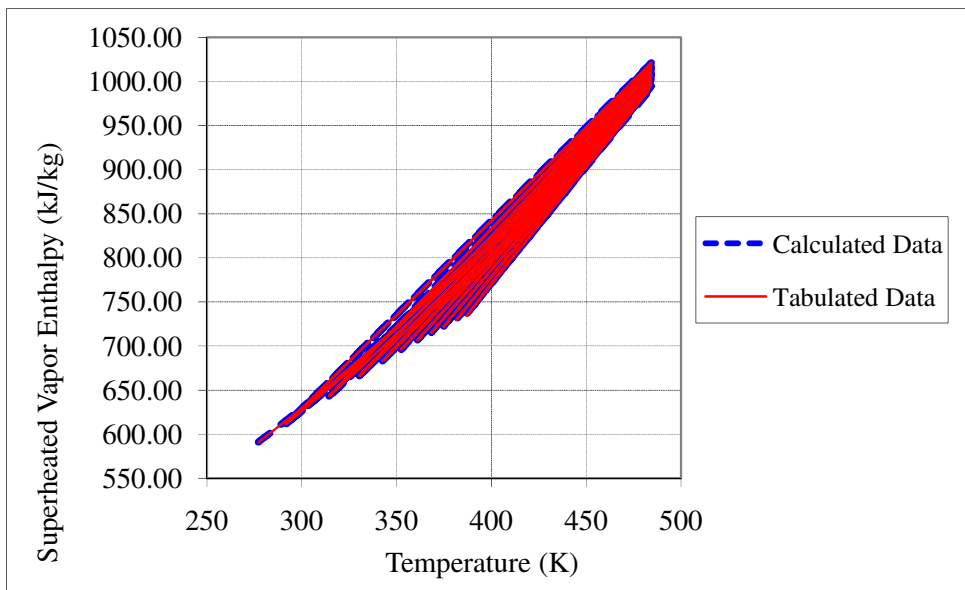
graph for R-600a

Figure A. 31 Enthalpy vs. temperature at constant pressures

graph for superheated R-600
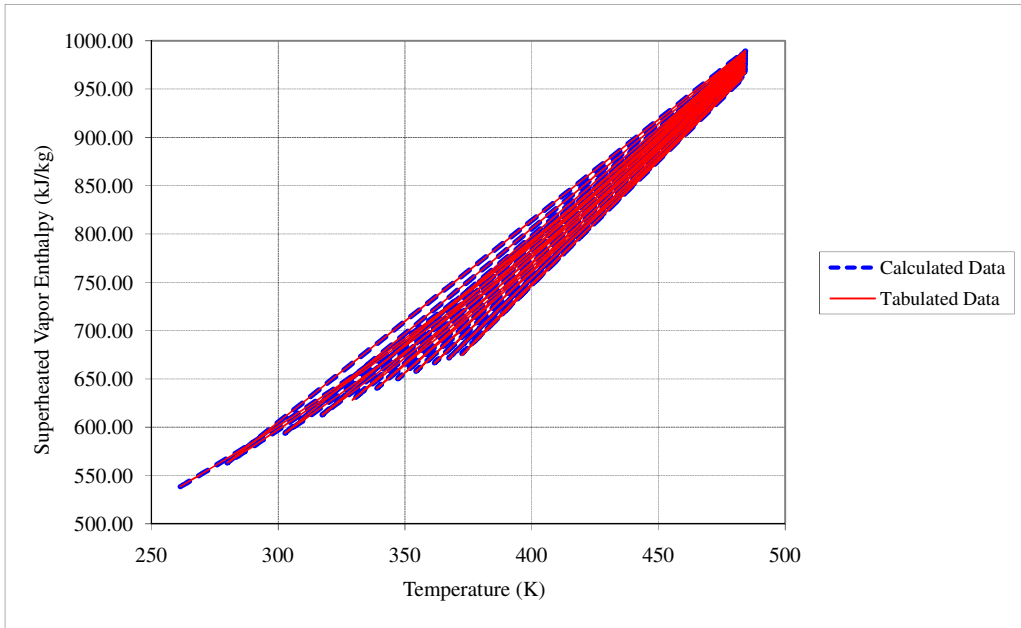
Figure A. 32 Enthalpy vs. temperature at constant pressures
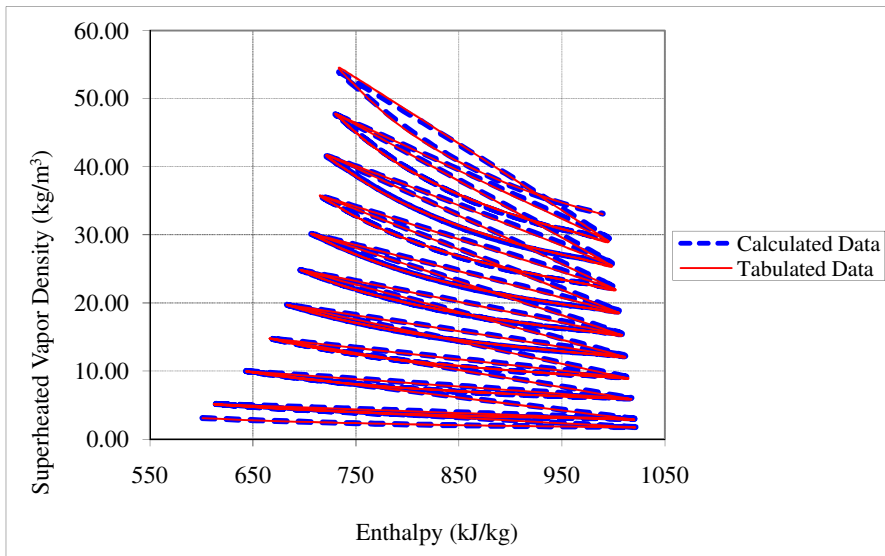
graph for superheated R-600a

Figure A. 33 Density vs. enthalpy at constant pressures
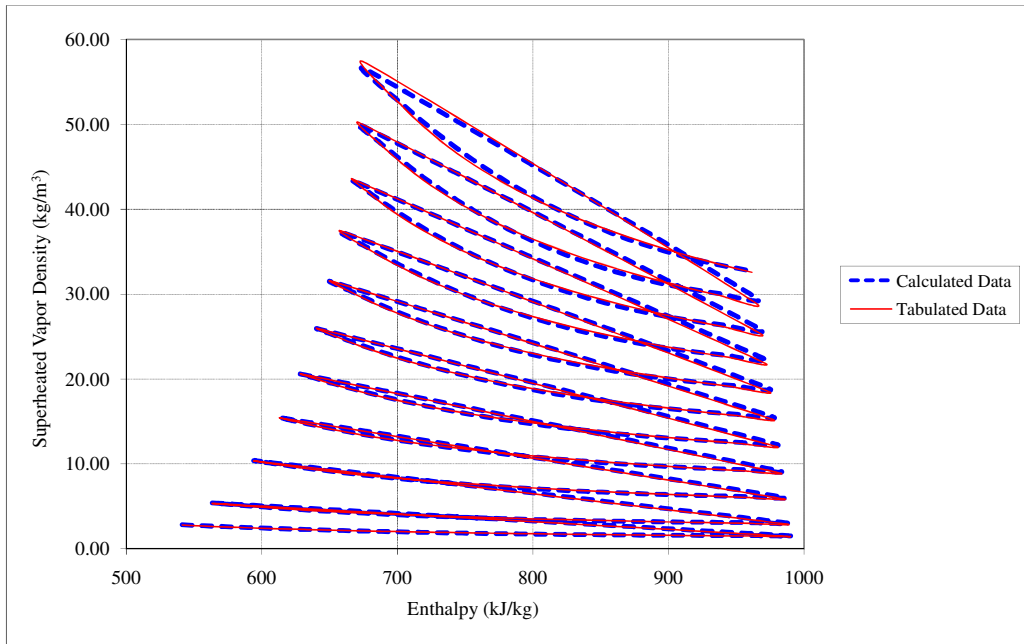
graph for superheated R-600

Figure A. 34 Density vs. enthalpy at constant pressures
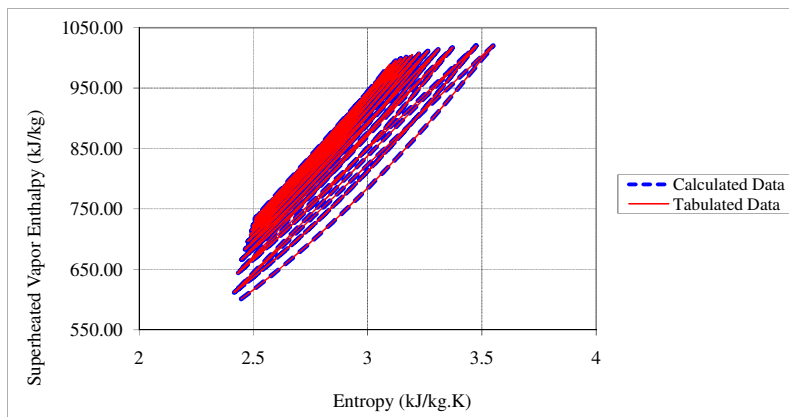
graph for superheated R-600a

Figure A. 35 Enthalpy vs. entropy at constant pressures
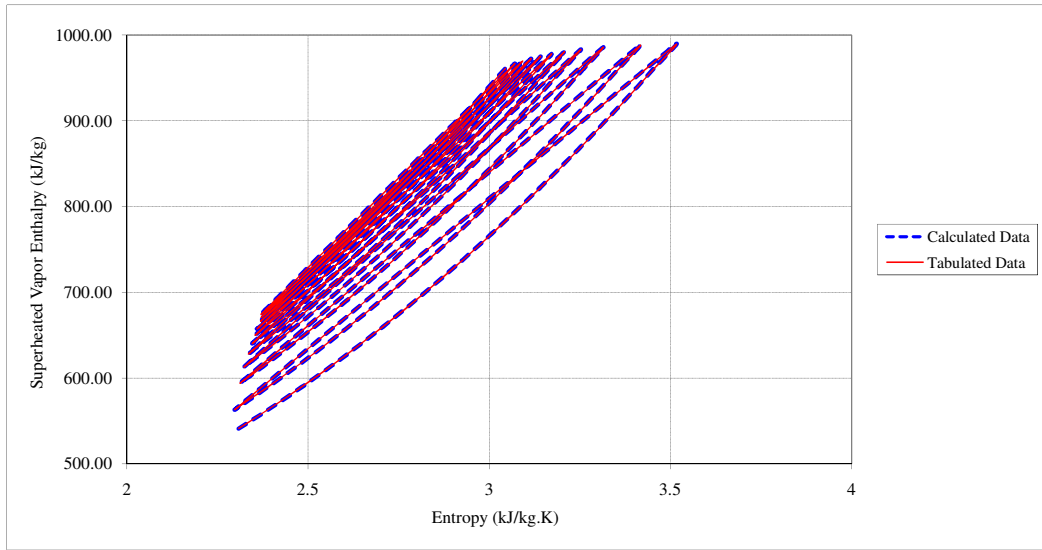
graph for superheated R-600

Figure A. 36 Enthalpy vs. entropy at constant pressures

graph for superheated R-600a