ENERGY, EXERGY AND HEAT TRANSFER PERFORMANCE INVESTIGATION OF A VAPOR COMPRESSION AIR CONDITIONING SYSTEM

JAMAL UDDIN AHAMED

THESIS SUBMITTED IN FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

FACULTY OF ENGINEERING
UNIVERSITY OF MALAYA
50603, KUALA LUMPUR

2012

ORIGINAL LITERARY WORK DECLARATION

Name of the candidate: **Jamal Uddin Ahamed** I.C/Passport No.

Registration/Matric No: **KHA 080057**

Name of the Degree: Doctor of Philosophy (PhD) in Mechanical Engineering

Title of Project Paper/ Research Report/ Dissertation / Thesis ("this work"): **Energy, Exergy and Heat Transfer Performance Investigation of a Vapor Compression Air Conditioning System**

Field Of Study: **Energy**

I do solemnly and sincerely declare that:

a) I am the sole writer /author of this work;
b) This work is original;
c) Any use of any work in which copyright exists was done by way of fair dealings and any expert or extract from, or reference to or reproduction of any copyright work has been disclosed expressly and sufficiently and the title of the Work and its authorship has been acknowledged in this Work;
d) I do not have any actual knowledge nor do I ought reasonably to know that the making of this work constitutes an infringement of any copyright work;
e) I, hereby assign all and every rights in the copyrights to this Work to the University of Malaya (UM), who henceforth shall be owner of the copyright in this Work and that any reproduction or use in any form or by any means whatsoever is prohibited without the written consent of UM having been first had and obtained actual knowledge;
f) I am fully aware that if in the course of making this Work I have infringed any copyright whether internationally or otherwise, I may be subject to legal action or any other action as may be determined by UM.

Candidate's Signature Date:

Subscribed and solemnly declared before,

Witness Signature Date:

Name:

Designation:

ABSTRACT

In this thesis, the energy, exergy and heat transfer performance of a split type air-conditioning unit are evaluated and the benefits of changing refrigerant (R22) with an alternative mixture (M1) are investigated. In this regard, a test rig consisting of a split air-conditioner was designed to work with R22. The test was conducted for two refrigerants; the first one was R22, the second one was a mixture of R22 and R290 with a mass ratio of 3:1.

The inlet and outlet temperature and pressure fluctuations at the compressor, evaporator, condenser and expansion device were measured by a digital data logger. The compressor power consumption was also carefully monitored by a digital power meter.

The results obtained from the two different refrigerants were compared. It was found that the coefficient of performance of the mixture was about 10-15% higher than that of R22 for the ambient temperature, T_a= 27°C. The result reveals that the coefficient of performances of the refrigerants increased with the increase of evaporator temperature and decreased with the increase of condenser temperature. The seasonal energy-efficiency ratio was found to be higher at lower ambient temperature. It was found that the compressor energy usage was increased up to 10-17% with the increase of condenser temperature. The test results show that by using M1, energy consumption can be saved up to 5.22% in the vapor compression air-conditioning system.

Exergy efficiency analysis was conducted for all the components of the vapor compression system at different evaporator temperature. This analysis also deals with the effect of evaporator, condenser and ambient temperatures on exergy losses. The result shows that exergy losses decreased with the increase of evaporator temperature and ambient temperature. However, the exergy losses increased with the increase of condenser temperature. It was found that about 60-

65% of the exergy losses were taken place at the condenser. Exergy efficiency of the system for the mixture was found to be higher than that of R22. The refrigerating effect of the air-conditioner was increased up to 2 to 6% while using the mixture (M1) instead of R22 at different evaporator temperatures. The heat transfer coefficient of M1 was measured higher than that of R22 at all working conditions and it was increased with the increase of mean evaporating temperature. At the working conditions, cooling capacity of M1 was higher than that of R22. It was found that the heat rejection ratio varied with different operating temperatures. It was also found that by increasing the set point by 4°C (from 20°C to 24 °C); the energy consumption can be saved up to 24.40%.

Finally, a correlation for the coefficient of performance of the system based on different condenser temperatures, ambient temperatures and evaporator temperatures have been developed. Based on the results of this work, it can be concluded that the mixture (M1) is the suitable alternative for the replacement of R22 based on energy, exergy and heat transfer performance.

ABSTRAK

Dalam konteks tenaga, *exergy* dan pemindahan haba bagi unit penyaman udara yang berjenis split type windowtelah dianalisa dan faedah daripada penggantian bahan penyejuk dengan menggunakanbahan campuran alternatif telah diselidik. Dalam kerja penyelidikan ini, rig ujikaji yang terdiri daripada unit penyaman udara berjenissplityang telah direkabentuk untuk bekerja dengan bahan penyejuk R22 telah digunakan. Ujikaji telah dijalankan dengan menggunakan dua jenis bahan penyejuk yang berbeza, bahan penyejuk yang pertama adalah R22 dan bahan penyejuk yang kedua adalah M1. Bahan penyejuk M1 telah dihasilkan daripada campuran bahan penyajuk R22 dan R290 dengan nisbah jisim 3:1. Nilai kemasukan dan pengeluaran suhu dan nilai perubahan tekanan bagi setiap alat pemampat, penyejat, pemeluwap dan pengembangan telah disukat dengan menggunakan alat sensor dan data bagi nilai-nilai tersebut telah direkodkan dengan menggunakan log data digital. Meter berkuasa digital telah digunakan untuk memantau dan meneliti nilai penggunaan kuasa pemampat. Alat-alat penyelidikan yang telah digunakan dalam kerja penyelidikan ini telah diletakkan di dalam bilik kebuk yang terkawal.

Keputusan kajian yang telah diperolehi daripada hasil pengunaan dua bahan penyejuk yang berbeza telah dibandingkan. Hasil kajian menunjukkan bahawa, pada keadaan suhu ambien T_a=27°C,nilai pekali prestasi bagi M1 adalah 10-15% lebih tinggi daripada nilai pekali prestasi bagi R22. Hasil kajian juga menunjukkan bahawa nilai pekali prestasi bagi bahan penyejuk meningkat dengan peningkatan nilai suhu penyejat manakala menyusut dengan peningkatan suhu pemeluwap. Nilai nisbah kecekapan tenaga bermusim yang diperolehi pada keadaan T_a = 27°C adalah lebih tinggi berbanding pada keadaan T_a = 31 °C. Kajian mendapati bahawa nilai penggunaan tenaga pemampat meningkat sehingga 10-17% dengan peningkatan nilai suhu

pemeluwap. Keputusan kajian juga menunjukkan bahawa dengan menggunakan bahan penyejuk M1 di Malaysia, penggunaan tenaga dalam penghawa dingin dapat dikurangkan sebanyak 5.22% berbanding dengan penggunaan R22.

Kajian terhadap kecekapan exergy bagi setiap komponen telah dilaksanakan untuk mengkaji kesan penyejat, pemeluwap dan suhu ambien terhadap kerugian exergypada nilai suhu penyejat yang berbeza. Hasil kajian menunjukkan bahawa nilai kerugian exergy menurun dengan peningkatan nilai suhu penyejat dan penurunan nilai suhu ambien. Secara purata 60-65% daripada kerugian exergy berlaku pada alat pemeluwap. Nilai kecekapan exergy sistem untuk bahan penyejuk campuran M1 adalah didapati lebih tinggi daripadabahan penyejuk R22. Kapasiti penyejukan penghawa dingin telah meningkat sehingga 2-6% apabila menggunakan campuran (M1) pada suhu penyejat yang berbeza. Nilai pekali pemindahan haba untuk bahan penyejuk campuran M1 telah mencatatkan nilai yang lebih lebih tinggi daripada R22 pada semua keadaan. Nilai pekali pemindahan haba juga telah meningkat dengan peningkatan nilai purata suhu penyejatan. Pada keadaan kerja yang diberikan, nilai kapasiti penyejukan adalah lebih tinggi untuk campuran M1 berbanding R22. Hasil kajian menunjukkan bahawa nisbah penolakan haba berubah pada setipa suhu operasi yang berbeza. Hasil kajian juga mendapati bahawapenggunaan tenaga dapat dijimatkan dengan peningkatan titik set sebanyak 4°C sehingga 24.40%,

Akhir sekali, korelasi bagi pekali prestasi sistem telah dibangunkan berdasarkan nilai suhu ambien, suhu penyejat dan suhu pemeluwap yang berbeza. Kesimpulannya, berdasarkan kepada prestasi tenaga, exergy dan pemindahan haba, campuran bahan penyejukM1 adalah alternatif bahan penyejuk yang sesuai untuk menggantikan R22.

ACKNOWLEDGEMENT

The author would like to mention with gratitude to Almighty Allah’s continual kindness without which no work would reach its goal. The author is highly grateful and obliged to the supervisors, Professor Dr. Masjuki Hj Hassan and Associate Professor Dr. Saidur Rahman, Department of Mechanical Engineering, University of Malaya, Malaysia, for their continuous guidance, patience and enthusiastic encouragement throughout this research work.

The author would like to thank to Dr. Mustafizur Rahman, a Post-Doctoral Research Fellow, UMPEDAC Research Centre, University of Malaya, for his continuous inspiration and motivation in research. Author is thankful to Mohammad Rafiqul Islam, an M.Eng.Sc student in the Mechanical Engineering Department and Md. Shouquat Hossain, an M.Eng.Sc, student under Electrical and Electronics Engineering Department, UM, Malaysia. Author is also grateful to all friends in the Department of Mechanical Engineering, University of Malaya and all members of Energy lab for their inspiration and constructive advice.

Author is grateful to the Ministry of Education, Malaysia to approve the High Impact Research Grant (UM.C/625/1/HIR/022 and UM.C/HIR/MOHE/ENG/07) and the FRGS research fund (FP 060-2008C) for financial support to complete my research. Author is very much grateful to the UPGP for providing IPPP project (PS 090-2009B) for financial support.

Finally, the author likes to express his sincere thanks to all other academicians and members of the Mechanical Engineering Department, UM, Welding Shop of UM and Energy Lab for their cooperation and help in the successful completion of the work.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF SYMBOLS AND ABBREVIATIONS

Abbreviations

AES Average energy savings (%)

AEU Average energy use (kWh)

AIS Average exergy savings (%)

CFC Chloro-flouro-carbon

COP Coefficient of performance

ED Exergy destruction (kJ/kg)

EER Energy efficiency ratio

EI Environmental impact

EST Energy savings at set point temperature, T (°C)

GWP Global warming potential

h_c Heat transfer coefficient, $kW/m^2.K$

HC Hydrocarbon

HCFC Hydro chloroflouro carbon

HFC Hydroflouro carbon

HRR Heat rejection ratio

HVAC Heating, ventilating air conditioning

MO Mineral oil

ODP Ozone depletion potential

OECD Organization for Economic Co-operation and Development

POE Polyol ester

Pr Pressure ratio

RAC Refrigeration and air conditioning

RE Refrigerating capacity (kJ/kg)

SEER Seasonal energy efficiency ratio

TEC Total energy consumption

Symbols

C_p Specific heat capacity at constant pressure, kJ/kg.K

G Mass flux (kg/m^2)

h Specific enthalpy of the refrigerant (kJ/kg)

I Exergy losses or destruction (kJ/kg)

m Mass flow rate (kg/s)

M1 Mixture of R22 and R290 with mass ratio, 3:1)

T Temperature ($^{\circ}$C)

$\bar{T}$ Average Temperature

P Pressure in bar (kN/m^2)

q Heat removal rate (kJ/kg)

Q Condenser duty (kJ/kg)

s Entropy of the refrigerant (kJ/K)

U Uncertainty in the measurands

W Work of compression (kJ/kg)

W_u Total uncertainty error in the variables

wx_1, wx_2, wx_3, … Errors in the individual variables y Dependent variable

η Energy efficiency (%)

μ Viscosity (Pa.s)

σ Surface tension (N/m)

ρ Density of fluid (kg/m^3)

φ Volume fraction (%).

Ψ Specific exergy (%)

Subscripts

0 Atmospheric

1, 2, 3, 4 State of the processes

a Ambient condition

ac Air conditioner

c Compressor

cond Condenser

el Electrical

evap/ev Evaporator

ex Exergy

in Inlet condition

mev Mean evaporator

$mevT_1$ Initial mean evaporator temperature

$mevT_2$ Final mean evaporator temperature

n Nano- particles

out	Outlet condition
r	Refrigerant
r, L	Refrigerant in liquid state
r, L, n	Refrigerant in liquid state with nano particles
s	Saturation state of the refrigerant
wi	wall
x	Dryness fraction

CHAPTER 1: INTRODUCTION

1.1 Research background

Refrigeration and air-conditioning systems are the main sectors of the vapor compression system. In early date, the performance of the air conditioning system and refrigerator were measured based on only the energy performance of the system. Refrigerants were selected based on the high energetic performance. Early refrigerants were ammonia, carbon di oxide, etc. At present, Ammonia is used in the air conditioner for commercial purposes but R22 is used in the air conditioner for domestic purposes. In some of the regions, R407C is being used as refrigerant. Nowadays, not only energy but also exergy analysis of the air-conditioning system has been expected remarkably as it has been found in some literatures (Carl, 2008; Gong et al., 2008; Jiangtao et al., 2009; Kim et al., 2002; Wing, 2004; Zhentao, 2006).

However, due to industrialization and social changes, usages of energy are increasing day by day in each of the energy sectors. It has been observed that total energy consumption has been increased by 49 % from 1990 to 2035 or average increased by 1.4 % per year (EIA, 2011). Total energy usage in the year 2007 was 521.90 quadrillions kJ and in the year 2035, it will be 779.16 quadrillions kJ. In the IEO 2010 projections, total world consumption of marketed energy from 1990 to 2035 is shown in Figure 1.1.

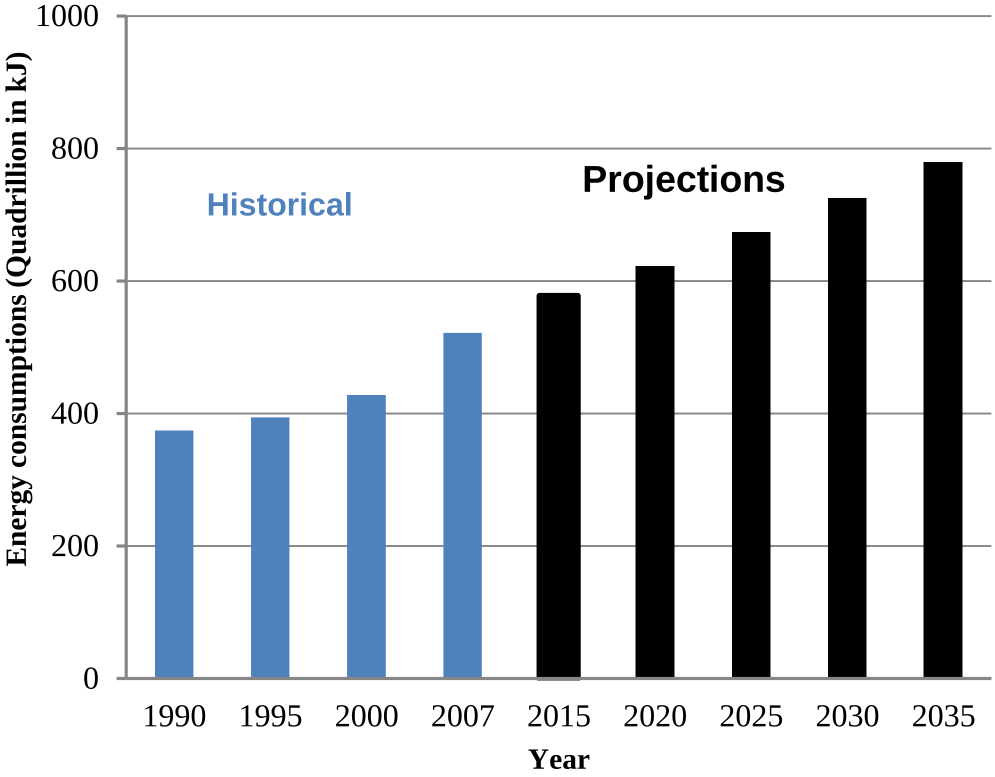


Figure 1. 1: World energy consumption projections for the period 1990 to 2035. (EIA, 2011)

Annual market demand in the Europe for air conditioner and refrigerator is assumed to be increased about 1 to 2% per year from 2002 to 2015. But the annual market in the sector of air conditioning is growing about 4% per year. In the United States of America, annual market growth is similar to the Europe but the growth in domestic and commercial refrigeration has become doubled. The market demands of the comercial and industrial refrigerator of different regions of the world in the different sectors are presented in Table 1.1.

Table 1.1: Expectations for the annual market growth of different types of refrigeration systems.

(IPCC, 2006; Wang, et al., 2005)

Areas/sectors	Annual market demand growth 2002-2015 (%/yr)			
	Europe	USA	Japan	Developing countries
Refrigeration in domestic sector	1	2.2	1.6	2-4.8
Refrigeration in commercial sector	1.8	2.7	1.8	2.6-5.2
Refrigeration in industrial sector	1	1	1	3.6-4.0
Refrigeration in transport sector	2	3	1	3.3-5.2
Stationary A/C	3.8	3	1	5.4-6
Mobile A/C	4	4	1	6-8

The highest increase in the projected values in energy demand is found for the non-OECD economical countries. In Malaysia, final energy demand in 2007 was increased by 9.8% to register at 44,268 ktoe compared to the previous year (NEB, 2007; Oh & Chua, 2010). The refrigeration and air-conditioning sectors consumed about 15% of all electricity consumed all over the world (Rogstam, 1996). This figure illustrates the significance of achieving optimal energy efficiency for the refrigeration and air conditioner.

In Malaysia, overall energy demand per year increases by 6% (Hasanuzzaman et al., 2008; Saidur et al., 2009). Malaysia is a warm and hot climatic country. Hence, the use of air-conditioner is increasing very fast. Total energy utilized in both the residential and commercial sectors was about 51% of the total energy consumption (Saidur et al., 2009). Air conditioner consumes about 65% of commercial energy usage and 57% of the office buildings. Energy demand in the air conditioners at home is increasing. Air conditioning appliances at home is consuming about 50% of the energy consumed in the house and 20% of the residential sectors (Jafar, 2010). Energy usages by the air conditioner depends on the operating hours, refrigerant types, environmental condition such as ambient temperature and room temperature, relative humidity, etc.

Globally, the demand for air conditioners (RAC's) is accelerating. The market of air conditioner in the southern and eastern areas in Europe is expanding (Adnot et al., 2000). In the United States, about 27% of households owned by RACs, and demand increased by an average of 9.2% annually from 2000-2006 (DOE, 2007). From the pattern seen in the United States, RAC's represent a first opportunity for HVAC ownership for low- to mid- income households. Throughout Asia, rapid industrialization is being increased affluence in an area of largely and equatorial climates; Malaysia alone anticipates growth from 0.25 million units in 1991 to over 1.5 million units by 2020 (Mahlia et al., 2002). Low labor cost relative to Western Europe, North America, and Japan has been resulted in rapid industrial expansion in the last two decades, including sourcing of appliances such as RAC's (Dupont & Vauvert, 2004).

A lot of research has been conducted every aspect of the energy sector in the whole world. Saidur (2010) investigated the total energy used in the field of the air conditioning system in Malaysia. Author has found that equipment used in the air conditioner played a key role of the energy consumption in Malaysia. Author has also compared the breakdown of energy utilization by the air conditioners for a number of selected countries.

Since the world energy crisis in the early 1970s, scientists and researchers have focused to recuperate energy usage, exploit renewable energy resources and diminish the impacts of energy use on the environment (Cengel, 2007). A lot of researches have been found that the changes in global climate are mainly caused by greenhouse-gas emissions. In recent decades, the concept of sustainability has entered into the engineering world, and it appears to be connected every aspect of an engineer's profession. So, refrigerant selection also became one of the criteria for the air conditioner to save the environment from greenhouse-gas emission. It is seen from the statistics that the carbon di oxide emission in the developed countries has been raised tremendously. It is also the prime issue for the energy sector to use emission-free gases in the systems. Table 1.2 shows the picture of emission of carbon di oxide from the used refrigerants in Germany (Schwarz, 2005).

Table 1.2: Emissions from the refrigeration and A/C sectors in Germany in 2002.

Category	Refrigerants	Emissions (ton/yr)
Refrigerated vehicles	R134a, R404. R410A, R152a, R218	52
Heat pumps	R134a, R407C, R404A, R410A	22
Domestic Freeze	R134a	1
Stationary A/C	R134a/ R407C	116
Passengers cars	R134a	1405
Mobile A/C (without cars)	R134a	173
Commercial refrigerator/freezer	R134a, R407C, R404A, R423, R125, R152, R116, R218	814

So, from Table 1.2, it can be concluded that the vapor compression systems discharge huge amount of emission in the atmosphere. It is essential to rethink about refrigerants usages in the vapor compression systems. It is essential to minimize the use of energy with efficient utilization of the system.

But energy analysis alone cannot help to reduce the energy usage in the energy sectors. Energy analysis deals with only the first law of thermodynamics. This analysis cannot be able to find how much energy is potential to convert into useful work. To identify and improve a system, a new technique has been recently introduced known as exergy analysis. It is based on both 1st and 2nd law of thermodynamics. Exergy analysis detects the degradation of any system, and demonstrates the performance of that system. It is defined as the availability of maximum work potential of a substance or a form of energy corresponding to the surrounding conditions (Yilmaz et al., 2001). Alsaad and Hammad (1998) compared the thermal performance of R12 to that of the hydrocarbon (R290 and R600) mixtures used in the domestic refrigerators. These mixtures showed better performance for the domestic refrigerator. The cost of refrigerant and power consumption has been decreased. Somchai and Nares (2005) found that propane and butane mixture with mass ratio of 3:2 is the most suitable substitute for R134a. Sekhar *et al.*(2005) examined the refrigerant mixtures of R134a and HC; utilized in the domestic refrigerator, deep freezer and vending machine as well as the walk in cooler.

Heat transfer also helps to increase energy performance of the vapor compression air conditioning system. In order to reduce power consumption in the vapor compression refrigeration system, many systems are developed. There are some active and passive methods available to reduce the power consumption or increase the heat transfer in condenser and the evaporator. One of the popular methods is the usage of fin in the condenser and usage of fan in the evaporator. Another method is using inserts inside the tubes. Sometimes nanofluid also helps to increase the heat absorption in the evaporator and heat rejection in the condenser. Some literatures showed that using proper refrigerant and lubricant, heat transfer enhancement is possible. By increasing condenser duty and heat rejection ratio, the system can be improved. This will help to reduce the power consumption by a compressor as well as reduce the condenser size. Hence, the cost of condenser and evaporator will be decreased.

Considering the crisis of energy in the whole world as well as in Malaysia, the government of Malaysia has emphasized on the energy efficient techniques for the energy sector especially in the domestic energy utilization sectors. Malaysian government has already started some energy efficient initiatives to reduce energy utilization. Based on the observation of the importance of the study in mind, this thesis has been focused on energy, exergy and heat transfer performance analyses of the vapor compressor system for domestic air conditioning system. In this thesis, the analysis based on the existing refrigerant R22 and the selection of alternative refrigerants to the existing refrigerant R22 acquiring less energy usage, less exergy destructions as well as high heat transfer rate have been studied. Following sections will provide the importance of energy, exergy and heat transfer performance analyses for the vapor compression system in the air conditioner. However, the energy, exergy and heat transfers are strongly influenced by the temperature in the evaporator, condenser and ambiance.

1.2 Importance of energy, exergy and heat transfer analysis in the vapor compression air conditioning system

Energy analysis deals with the first law of thermodynamics. It measures the coefficient of performance of the vapor compression system which is used in the refrigerator and air conditioner. Sometimes in the developed countries, performance of air conditioner is treated by the term energy efficiency ratio (EER). Though CFC and HCFC played as good performers as refrigerants but their high ODP made them phase out after Montreal Protocol. This is because in the stratosphere layer, ozone absorbs the ultra-violate rays from the sun. The free chlorine atoms come from the CFC and HCFC may cause to deplete the ozone layer. As a result, the world becomes warm day by day. This should be harmful for the living beings in the world. As CFC and HCFC have many appropriate properties such as non-flammability, low toxicity, and material compatibility, hence it becomes very popular to be used widely in the air conditioners and refrigerators. There are millions of domestic refrigerators and air conditioners around the world. It is urgently needed to mitigate the environmental problem (global warming) for the welfare of human being (Radermacher & Kim, 1996).

The GWP, ODP and COP are to be considered for selecting suitable refrigerant. Heat transfer also varies for different refrigerant and its lubricant. The heat transfer performance is also a vital factor for minimizing energy consumption and selecting refrigerant-lubricant mixtures. Thermal properties such as thermal conductivity, heat transfer coefficient, viscosity and surface tension also play a major role for high heat transfer for the air conditioner.

In case of refrigerators and domestic air conditioners, scientists and researchers are probing an environmental caring refrigerant (Sattar et al., 2008). R290 has zero ODP but it is flammable. However, if it is possible to eliminate the leakages while manufacturing the refrigerating & air conditioning system, then this refrigerant can be used safely. In some cases, R134a is considered as a refrigerant. As it has zero ODP and it is miscible with lubricant polyol ester. However, it is also required to be used bigger compressor. Moreover, Polyol ester is not cheaper and not available in the market. R134a is hygroscopic in nature. Hence, R134a is not suitable as refrigerant. Some of the hydrocarbons like propane, iso- butane etc. are mixable with R134a by the suitable ratio and this can be used as a refrigerant. Thus, the mixture will be miscible with the mineral oil and it becomes a refrigerant with low GWP and low flammability. R22 is suitable and already is used in the air conditioner and heat pumps. Its performance is remarkable. By mixing hydrocarbons like R1270, R290, etc. and R22 can be used as an enriched mixture. This mixture will cause less GWP and less flammability. The performance of the air conditioner, heat transfer rate and energy consumption can be compared for different cases. In some cases, changes in the hardware configuration of the equipment are not required (United Nations Development Programme, 1999). Hydrocarbon refrigerants were accepted before the introduction of CFC and HCFC-fluids (Granryd, 2001). Hydrocarbons have GWP closer to zero. Moreover, hydrocarbons provide zero ODP, less toxicity and higher chemical stability, with suitable thermodynamic, physical and chemical properties (Teng et al., 2012). The only problem is flammability occurs for careless uses and leakages. So, in smaller application (Maclaine-Cross, 2004), it is safer. However, mixing with R134a in the proper ratio, it may reduce also the flammability and hence increase the thermodynamic performance of a small air conditioner.

Hydrocarbons are environmental friendly and found to be of '0'ODP and low GWP. Hydrocarbons are cheaper than R134a and R22. These refrigerants are available in the nature. Most of the hydrocarbons are well miscible with mineral oils and compatible with the materials of the vapor compression systems. The thermal properties of hydrocarbons are very close to those of CFC refrigerants but these are non-toxic and environmental friendly (Sattar et al., 2008). Tashtoush*et al.,(2002)*, and Sekhar and Lal, (2005) reported that ODPs of hydrocarbons are very low ($<5.10^{-4}$) and the GWPs are quite high. Thus, still inconsistent reports on GWP of hydrocarbons are obtained.

There are many studies performed by many researchers using HCs mixture as refrigerants. Fatouh and Kafafy (2006) assessed the prospect of using HCs mixtures as refrigerant in order to exchange with R134a in the domestic refrigerators. The authors analyzed the performance of domestic refrigerators using different refrigerants such as R134a, propane, butane and propane/isobutene/n-butane mixtures with different mass fractions. Authors observed that pure butane experienced low COP but the HCs mixtures showed high COP and can be replaced with R134a in the air conditioners used in the cars. Authors found that R290/R600/R600a with 50/40/10% would be the best alternative to R134a (Kabul et al., 2008). This mixture showed the best performance of all other mixtures. Arcaklioglu et *al*. (2005), Jung et *al*.(2000) and Arcaklioglu (2004) reported that using mixtures, there is a change in temperatures during phase change at constant pressure. This is called temperature gliding. This causes disadvantages of the systems. However, it is easy to eliminate this problem using pure hydrocarbons as refrigerants (Bayrakci & Ozgur, 2009).

The direct effect of refrigerants is one of the main points of concerns regarding environmental effects of HVAC systems; another one is the contribution to global warming from carbon dioxide produced due to the electrical energy consumption of air conditioning. Air conditioners account for about two percent of residential energy utilization in the United States. RAC's are the least efficient devices for providing cooling, compared to central air conditioning system (Rosenquist, 1998). The U.S. has imposed a legislation requiring improvement of appliance energy efficiency, last updated in 2000. Further improvements are desired as part of the United States' national energy plan (Grant et al., 2001).

For domestic air conditioner, a mixture (Jiangtao et al., 2009) of R152a, R125 and R32 were used as refrigerant to measure the performance as a potential alternative of R 22. They made a comparative study about COP and other parameters for R 407C, a blend of R152a/R125/R32 and R22. The blend is zoetrope mixture and found as very close COP to R22 but less flammable and it is safer than others. However, most of the researches are based on energy performance and computational. Park and Jung (2008) studied the replacement of R22 with R290 andR1270. They studied the energy performance, work of compressions of R22, R290 and R1270. They found that R290 has the best performance in a small capacity plant. But they consider about the environmental issue.

Energy analysis for a vapor compression refrigeration system indicates the measure of some parameters. These are coefficient of performance, energy efficiency ratio, heat transfer by condenser and heat transfer by evaporator. In most of the cases, coefficient of performance is the indication of performance of the vapor compression system. So, improvement of performance of the air conditioner or refrigeration system means the increase of coefficient of performance (COP). Higher COP indicates saving of energy also. But performance parameter depends on some factors. Akintunde et al. (2011) found that overall performance of the system is strongly depends on operational parameters. From the literature, factors that affect the COP of the system are listed below:

- Evaporator mean temperature
- Condensation temperature
- Ambient temperature
- Sub cooling and superheating temperature
- Latent heat of vaporization of the refrigerants
- Miscibility of refrigerant with the lubricant
- Pressure ratio of the compressor

Miscibility of the refrigerants with the lubricants is established already. Hydrocarbons have high heat of vaporization. Mixture of hydrocarbon with R22 also has higher latent heat of vaporization compared to that of R22. Generally, all air conditioners are designed for 2-5 °C superheat and sub cooling for getting better performance. Higher compression ratio indicates higher compression work, hence it causes less performance. Energy efficiency ratio (EER) is used also instead of the coefficient of performance.

Only the difference is the unit. In some countries, EER is used for air conditioner and COP is used for the refrigerator-freezer. Many researchers found that temperature has a vital effect on the performance. It is clear from the definition of COP that it is the ratio of heat transfer in the evaporator to the electric power consumed by the compressor. Heat transfer in the evaporator depends on the difference between the enthalpy of the inlet and outlet refrigerant. But enthalpy is the function of temperature, pressure and latent heat of vaporization of the substances. The compressor power also depends on compressor efficiency, discharge pressure, irreversibility of the compression process and pressure ratio of the compressor. Hence, the performance of vapor compression system is related to many parameters.

Over the past 30 years, many advanced technologies have been developed and used in the HVAC industry. Accordingly, the efficiency of major energy based components of HVAC systems, such as chillers, fans, pumps and heat exchangers have been dramatically raised. Some optical control strategies have been promoted to make the HVAC operations more effective. Most of the studies are performed based on the first law of thermodynamics, which is called energy analysis. It is used to evaluate the thermodynamic performance of HVAC systems. Energy analysis is based on the quantity of energy but it is not involved with the quality of energy. It is realized as the most important aspect of energy. On the contrary, second law analysis is based on exergy analysis. Exergy analysis deals with the quality of energy, capacity of energy to do work i.e. useful energy. It reveals the consumption of energy that means the degradation of energy. It introduces an index of energy degradation. Exergy analysis indicates the more inefficient sites or location or components of a system and helps for modification (Ahamed et al., 2011; Saidur et al., 2010). Exergy analysis is more meaningful and appropriate compared to energy analysis when applied for systems with more than one type of energy sources. Since 1970s, exergy analysis has been developed in the case of intensive energy i.e. power generation plants. Recently, it is used in the vapor compression analysis. But most studies are based on the theoretical analysis not experimental.

It is stated that the vapor compression system discharges enormous amount of heat to the surroundings. Due to the differences in temperatures between any system and its surroundings, irreversibility is taking place. This irreversibility degrades the performances of the system components. In order to increase the performance of any system, it is essential to measure the thermodynamic losses in the individual components of that system. For measuring the efficiency of the vapor compression system, the most universally used term is the COP. This analysis is sometimes treated as 1st law of energy analysis. But the first law of energy analysis DOEs not make any difference between heat and work. First law of energy analysis cannot detect the basis of thermodynamic losses in the cycle. It cannot find the information on how, where and how much of the system performance is degraded. On the other hand, second law of thermodynamics is capable to identify the thermodynamic losses in the cycle. Using the idea of irreversibility, thermodynamic losses (i.e. exergy losses) in the vapor compression cycle can be calculated. Precisely, the term ‘exergy’ is defined as the maximum amount of potential of work which can be formed by a system or flow of matter or energy when the system becomes equilibrium with a reference environment (Edgerton, 1992 ; Kotas, 1985; Moran, 1989; Moran & Sciubba, 1994; Rosen et al., 2008; Szargut, 1980 ; Szargut et al., 1988). Exergy indicates the potential of any system or flow to cause changes relative to the reference environment. In a particular time, a certain amount of hot fluid contains higher exergy during the cold season rather than in the hot season. Exergy is destroyed when energy losses its quality. Exergy is the portion of energy which has economic value and is useful. Aim of the exergy analysis is to investigate the performance of the system. Exergy analysis is usually aimed to determine the maximum performance of the system and identify the exergy destruction in the different sites. From this analysis, main components with maximum exergy destruction can be found. Hence, potential improvement can be done in that site (Kanoglu, 2002).

There is a limited analysis about exergy for vapor compression refrigeration using pure hydrocarbons whereas in many investigations, hydrocarbons were found as an acceptable refrigerant alternative to R22. Liedenfrost (1980) investigated exergy of a refrigeration cycle using Freon as refrigerant. Bejan (1989) showed that the exergy efficiencies decrease with the decrease of refrigerating temperature. Author presented two models for clarifying the trend. Thermodynamic imperfections were explained largely by the heat transfer irreversibility in the models.

Exergy creates a link between the second law of thermodynamics and environmental impact (EI). Exergy measures the deviation of any state from the environment (Edgerton, 1992; Szargut, 1980). The exergy efficiency is mostly influenced by the system and the states of the environment. The decrease in EI of any process shows the increase of exergy efficiency. Exergy analysis is used to improve the sustainability of the system. Cornelissen (1997) suggested that exergy losses can be minimized to obtain sustainable development using non-renewable energy forms. The author also found that EI is connected to the emissions and resource depletion which can be defined with exergy (Rosen et al., 2008). Figure 1.2, shows a relationship between EI and sustainability with exergy efficiency.

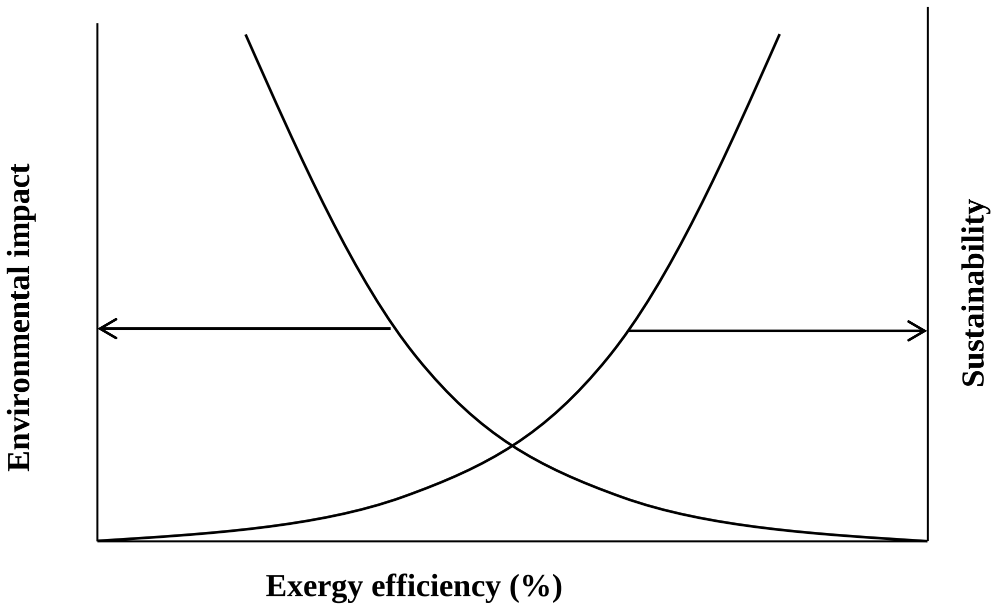


Figure 1. 2: Qualitative example of a correlation between the environmental impact and sustainability of a process with its exergy efficiency.

(Rosen et al., 2008)

From Figure 1.2, it is clear that when any system complies with 100% exergy efficiency, its environmental impact will be zero. It means that in that system, exergy will be converted into work without any loss. It can also be seen that at that point the sustainability of the system will be infinite. The emissions and the sustainability indices are also related with exergy efficiency.

From the above discussion, it is observed that exergy analyses deals with some parameters. These are exergy destruction in the whole system and in the components, exergetic efficiency, exergy defects, etc. These parameters are not fixed in any condition. The factors those are liable to change these factors are as follows:

- Deviation of any system temperature (evaporator and condenser) from ambient
- Environmental/surroundings temperature

- Refrigerant properties such as enthalpy, entropy, viscosity, etc.
- Pressure ratio of the compressor and irreversibility of the system
- Processes (heating, cooling or compression or expansion)
- Pressure losses in the system

Heat transfer analysis gives the information or technique how suitably heat is transferred in the different section of the system. Heat transfer analysis helps to reduce the cost of equipment, increase the performance of the equipment as a heat exchanger. Nowadays, various heat transfer augmentation techniques are applied for heat transfer enhancement. In the vapor compression system condenser and evaporator are the heat exchanger. Performance of these parts causes to reduce the mass flow rate of refrigerants as well as the cost and compression work. Heat is rejected in the condenser and added in the evaporation. Using fins and inserts, heat transfer can be enhanced. This system has been established. But another method to increase heat transfer performance is the proper selection of refrigerant. Heat transfer rate can be increased by using refrigerant of high latent heat of vaporization. Hydrocarbon has high latent heat of vaporization. By using it, the heat removal or heat addition in the devices can be improved. It's a new technique that nano lubricant can increase the heat transfer coefficient. Hence the heat transfer rate will be increased. The mass and energy flow diagram of condenser is shown in Figure 1.3.

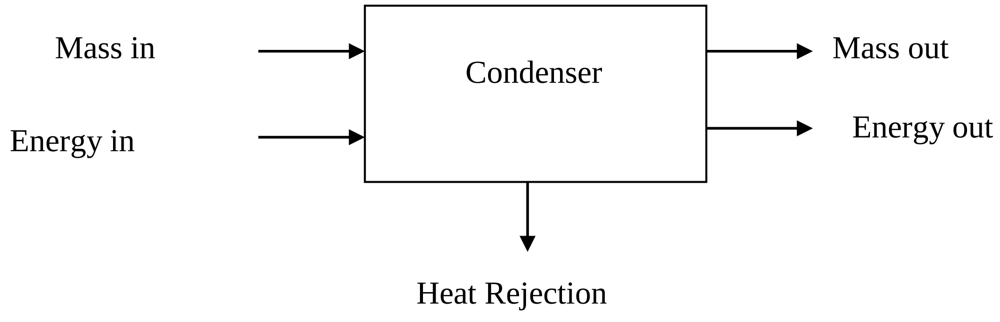


Figure 1. 3: Schematic diagram for mass and energy flow of condenser.

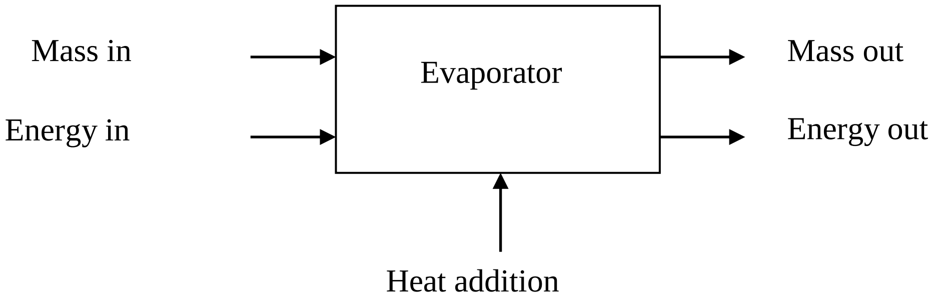


Figure 1.4: Schematic diagram for mass and energy flow of evaporator.

In the condenser, energy balance is as follows:

$$Q_{in} = Q_{out} + Q_{removal} \quad (1.1)$$

From the energy flow through the evaporator shown in Figure 1.4, energy balance can be written as:

$$Q_{in} = Q_{out} + Q_{addition} \quad (1.2)$$

Considering refrigerant flow rate as m_r, the Equation (1.2) can be rewritten as:

$$Q_{addition} = m_r \times (h_{in} - h_{out}) \quad (1.3)$$

Heat transfer analysis deals with some parameters such as cooling capacity of the system, convective heat transfer coefficient in the evaporator (h), etc. The parameter ‘h’ is varied for different refrigerant at different temperature. Higher heat transfer coefficient indicates cooling with higher rates. Faster cooling reduces the running time of the compressor hence the energy usage by the compressor. Sometimes, heat flux is also used as a parameter for measuring or understanding heat transfer performance in the air conditioning system. Some research shows that using inserts and additives, the heat transfer rate of any system can be improved. Nowadays, nano-lubricants also are in consideration for enhancing heat transfer

Nano lubricant is a mixture of lubricant- mineral oil with a small amount of CuO. This causes a reduction of wear and friction in the matting parts of the compressor and hence reduces the exergy losses due to friction and irreversibility. Some metals like Cu and Al have high thermal conductivity. By mixing the nanoparticles with lubricant, total thermal conductivity will be increased. Fluid thermal properties also have an effect in heat transfer. Al_2O_3, TiO_2 are also nanofluid used to enhance the heat transfer. These nanofluids can be used with mineral oil. Nanofluid helps to increase heat transfer by two ways (Bi et al., 2008). Some of the particles remain in the compressor and helps to increase lubrication characteristics and some of them flow with the refrigerant and increase heat transfer in the evaporator. So, heat transfers properties are related with the refrigerant properties, lubrication, nanofluids and thermal properties of the refrigerant. Following thermal properties are also liable for increasing heat transfer:

- Thermal conductivity of the refrigerant or fluid
- Viscosity of the refrigerant
- Surface tension of the refrigerant
- Density of the fluid
- Latent heat of vaporization of the fluid

High viscous fluid causes frictional losses to flow; hence, it reduces the heat transfer. High thermal conductive materials cause to increase the heat transfer rate. High surface tension creates less heat transfer in the evaporator during boiling. However, surface tension has influence on condensation heat transfer. High surface tension creates a film on fins surface of a condenser and cause a high-heat transfer (Pelletier & Palm, 1996). All these parameters are varied with temperature of the fluid, especially with the evaporator temperature.

From the literature, it is found that limited researches have been performed on exergy analysis of the individual components of a vapor compression refrigeration system as well as domestic air conditioner. It is found that HCs have greater advantages on the basis of energy and other environmental impacts. So, it has become an essential to know the exergy performance as well as energy performance of the mixture of propane (R290) and R22 comparing with the existing refrigerant R22. It is also found that a few analyses have been carried out on the effect of operating temperatures i.e. condenser and evaporator temperatures on exergetic performances of vapor compression air conditioner. However, exergetic performances of R22 and its mixture with R290 were not studied and examined by experiments for the domestic air conditioner. In this study, energy and exergy performances of an air conditioner using R22 were considered as a baseline and then the performances of the air conditioner with a mixture of R22 and R290 with a ratio of 3:1 (by mass) were estimated and compared with existing ones (R22). Exergy losses in the individual components were also investigated at different evaporating temperatures and condensing temperatures.

There is no correlation developed for the variations of COP with the changes of the evaporator, ambient and condenser temperatures. But in some researches, it has been found that coefficient of performance varies on these operating temperatures. So, novelty of this research can be summarized as below:

- Comparative study of Mixture of HC (R290) with R22 based on thermal performance
- Exergy analysis
- Experimental investigation in Malaysian atmosphere
- Development of correlation based on different operating temperature with COP
- Obtained low GWP refrigerant

- Analytical analysis using nano fluid.

1.3 Objectives and scope of the study

In the domestic air conditioner, generally R22 is used as refrigerant. Its energy analysis is already developed. But its exergy analysis is not yet established. Besides, R22 is to be phased out according to Montreal's protocol by 2030. Therefore, it is essential to study the energy, exergy and heat transfer performance of the existing refrigerant and its alternative, and proposed some refrigerants or their mixture. Being an environmental friendly refrigerant, hydrocarbon draws an important attention among the scientists and researchers. Considering the thermodynamic properties of different hydrocarbons, it can be chosen as a refrigerant. Latent heat of vaporization of the refrigeration should have as large as possible. As from the literature, R290 and its mixture with R22 are the candidates for analysis with R22. Pressure ratio of the refrigerant should be small with high volumetric efficiency and low power consumption.

1.3.1 Objectives of the study

The objectives of this thesis are:

a) To evaluate the effect of refrigerants R22 and M1 on the thermal performance of the vapor compression air-conditioning system.
b) To investigate the effect of evaporator, condenser and ambient temperatures on exergy losses, exergy efficiency and coefficient of performance.
c) To compare the exergy performances of the vapor compression system using different refrigerants.
d) To improve the heat transfer performance of a vapor compression air conditioning system for different refrigerants.
e) To develop a correlation for COP with different operating temperatures.

f) To estimate the energy and exergy savings with the application of different energy savings techniques.

1.3.2 The scope of the study

This study can be applicable to the domestic and commercial cooling systems. For the industrial sector, we need to use the proper refrigerant with a leak proof condition. Energy analysis makes the system acceptable as it reduces the energy usages in the air conditioning and cooling sectors. Exergy analysis makes it acceptable based on second law of thermodynamics.

The scope of this study includes:

- Preliminary investigation of the thermal performance of R22 using mineral oil as lubricant.
- Retrofitting the air conditioner with hydrocarbon and mixture of hydrocarbons and R22 using mineral oil as lubricant.
- Monitoring the pressures and temperatures at the inlet and outlet of the compressor and evaporator.
- Investigating the energy, exergy and heat transfer enhancement of hydrocarbon (R290) and its mixture of with R22 as well as of R22.
- Comparing the thermal performance of HCs and their blends with R22.

1.4 Limitations of the study

For getting better result, it needs to collect the data for the different mass ratio of the mixtures (R22 and R290) at different evaporator and condenser temperatures. However, it was quite difficult to make more variations in the mass ratio because of using hermetic sealed compressor with fixed capillary tube. It was also not possible to make large variations in the evaporator and condenser temperatures. Based on the daylight and weather conditions, the atmospheric temperatures were changed and hence the evaporator temperature was also changed. It was tough to join at many locations in the gas lines. There was a chance to cause leakage of gas through the joining. Author did not set a thermostat in the flow pipeline. Variation in the refrigerant flow was not controlled with any valve. Because in this experiment, flow was changing automatically and it was measured with the help of flow meter. Naturally, the flow conditions were changed. As it was impossible to change the compressor size, so the experimental set up could not be used for different mass of refrigerant for checking.

1.5 Organization of the thesis

This thesis comprises five chapters. The contents of the individual chapters are described briefly here.

Chapter 1: The world energy scenario, energy usage in the refrigerating and air conditioning sectors in the world as well as in Malaysia are described. Importance of the research in the air-conditioning sector for reduction of energy uses is described. The importance of exergy analysis, heat transfer analysis in the vapor compression system is also discussed. Objectives, importance of this study, novelty of the research, limitations and background information are described in this chapter.

Chapter 2: This chapter shows the review of the literature on energy, exergy and heat transfer of the vapor compression system for air-conditioning system in brief. In this chapter, thermal properties of different refrigerants used in the vapor compression air conditioning system are discussed. Environmental problems of the present refrigerants are also tabulated. Using software, REFPROF 7, heat-transfer parameters of R290, R407C, mixtures of propane and R22 are compared with the existing refrigerant R22.

Chapter 3: This chapter describes the information about the experimental set up and procedure. This chapter designates the methods applied to collect data and necessary mathematical formulations to calculate the parameters related to energy, exergy and heat transfer analysis. Here, the short description of the equipment is also presented. In this chapter method of charging refrigerant and control of their weighs are described. Specifications of the machines used in this study for taking data are also described.

Chapter 4: In this chapter, the results obtained according to the objectives are described and presented in graphs. These results are explained with appropriate causes and consequences. Energy and exergy parameters are analyzed sequentially and compared with others' works. Heat transfer parameters such as condenser duty, heat transfer rate in the evaporator and condenser, heat transfer coefficient, etc. are calculated and described the cause and effect of temperature on these parameters. Some correlations are developed for exergy destruction and COP with ambient temperature and atmospheric temperatures.

Chapter 5: In this chapter, the summary of the results/outcomes in this study about the energy and exergy efficiency are summarized according to the objectives. Some recommendations are given here for future research. Limitations of the present work are also presented here.

CHAPTER 2: LITERATURE REVIEW

2.1 Overview

This chapter gives an overview of the pertinent literatures on energy, exergy and heat transfer performance of an air-conditioning system where vapor compression systems are used. Related journal articles, reports, conference papers, U.S. energy reports, PhD thesis, Master's dissertations, web sources and books were collected from different sources for this study. About 80% of the most relevant and high indexed peer reviewed journals such as Energy, Applied Energy, Journal of Refrigeration, Energy Policy, Energy and Building, Exergy- an International Journal, Energy Conversion and Management, Applied Thermal Engineering, International Journal of Energy Research, Journal of Heat Transfer collected and studied. Some of the high quality PhD thesis and Master's dissertation such as by Zhentao (2006), Philip (2001), Carl (2008), Khan (1992), Qureshi (2004), Saidur (2008), Mansouri (1996), Jahansson (2003), Wing (2004), Kenny (2003) are considered as a strong support for this study. Besides, sizeable amount of pertinent information has been collected by personal communication with the potential and key researchers regarding this field of area from the whole research world. Some computational works are also collected to select refrigerants for best performance. Thus, the thesis may become strong support for the future world in this area.

2.2 Reviews on energy efficiency and its uses in the vapor compression air conditioning system

Recently, energy and environmental issues are caused a vital effect on the massive usage of the air conditioner. In the energy sector, refrigeration system is assumed to be treated as the prime part of the energy consumption. Some of the researchers studied on the performance of the vapor compression refrigeration system. Thermal performance of the vapor compression system was the best concern for designing and selecting refrigerant from the very beginning of the air conditioning system. The indication of a good performance means the highest coefficient of performance of the refrigeration system. It shows the high energy efficiency also. This is based on the first law of thermodynamics analysis. Rapid economic growth during the last few decades in the world has been accompanied with more offices, house buildings and more energy and environmental problems. Especially in the developed and developing countries, more energy is consumed in the air conditioning systems for their developed works and taking comfort life. In Malaysia, final energy demand in year 2007, increased by 9.8% to register at 44,268 ktoe compared to the previous year (NEB, 2007). According to the trend of the developed countries, energy consumption will inevitably increase to 35% or so (Gong et al., 2008; Zhu & Lin, 2004).

The industrialization caused to increase the utilization of the new technological products in our daily life after the second half of the twentieth century. Hence, more energy consumption was happened. Usage of energy is the inseparable part of our daily life. Moreover, the rate of energy consumption per capita has become an indication of success in development of any country. To supply energy eternally, securely and adequately to the developed sectors, it needs to increase the productivity and activity in energy consuming and producing sectors. Currently, there are three policy themes are identified and are considered in designing the energy sectors. These are (Brown, 1996) as follows:

- the traditional energy policy agenda relating to security of energy supply;
- the environmental impact of energy, its production, transformation and use;
- the trend towards liberalization and enhancement of competition in energy markets, notably in the electricity and gas sectors.

This situation led the scientists to develop the cycling ways of the energy and to get more efficiency from the energy sector. In the last decade, significant effort has been made to increase the rate of heat transfer in order to improve overall performance of air conditioner. Improvement in the performance may result in the reduction of the cost and energy savings for the air conditioner unit. It is also necessary to think about the wear occurred in the compressor. Proper lubricant and additives may reduce the wear in the mating parts and hence the life of the compressor will be increased.

Many researchers have been studying the subject from the beginning of twentieth century. The performance can be improved by using various augmentation techniques such as finned surfaces, integral roughness and insert devices. Room air conditioners account for about 2% of residential energy use in the United States, RACs are the least efficient devices for providing cooling, compared to central air conditioning system (Rosenquist, 1998). The U.S. Energy Department has forced for legislation requiring improvement of appliance energy efficiency, last updated in the year 2000. Further improvements are desired as the part of the United States' national energy plan (Grant et al., 2001). Another technique is to choose right refrigerant with higher heating value/enthalpy content to cause fewer mass of refrigerant charges. Refrigerant higher enthalpy refrigerant causes lower compression outlet temperature. Thus, the energy consumption by the compressor will be reduced. Now a days, R134a is used in the domestic refrigerator and R22 is used in the domestic air conditioner for vapor compression system. So, researchers are searching alternative refrigerants with high performance as well as reduced cost.

The energy balance is a basic method of any process investigation. It makes the energy analysis possible, points at the needs to improve the process, is the key to optimization and is the basis to developing the exergy balance. Analysis of the energy balance results would disclose the efficiency of energy utilization, in particular, parts of the process and allow comparing the efficiency and the process parameters with the currently achievable values in the most modern installations. They will establish also the priority of the processes requiring consideration, either because of their particularly low efficiency.

Alsaad and Hammad (1998) compared the thermal performance of a CFC12 in a domestic refrigerator with the new refrigerant propane/butane mixtures. The authors found that these mixtures showed better performance for the domestic refrigerator. Cost of refrigerant as well as power consumption also is reduced. Somchai and Nares (2005) investigated the application of mixtures of propane, butane and isobutene in a domestic HFC134a based refrigerator. Among the different proportion of the mixture, authors found that propane and butane mixture (60:40) is the most appropriate and alternative refrigerant to R134a. Devotta *et al.* ((2001) selected HFC134a, R 290, R407C, R410A and the blends of HFC 32, HFC-134a and HFC125 and found R 290 as a potential candidate. Another study was performed based on R134a, R600 and R600a in the domestic refrigerator and found that R600a has the highest performance among the refrigerants (Sattar et al., 2008). Devotta et al.(2005; 2005b) also found that for air conditioner, 12.4 to 13.5% energy consumption is reduced using R290 instead of R22. Sekhar *et al.*(2005) investigated refrigerant mixtures of HFC-134a/HC in domestic refrigerator and deep freezer and vending machine and walk in cooler. Sekhar *et al.*(2005) also found that of R290 and R600 as a mixture caused to reduce energy consumption by 4-11%. Mohanraj et al.(2007) studied about mixture of R290 with R600a and found that 4% energy consumption is reduced compared to R134a. To be accepted as a refrigerant for domestic and light commercial use, it should be non-corrosive, chemically stable, boiling temperature lower than ambient temperature, and critical temperature higher than ambient temperature (Ahamed et al., 2011; Saidur et al., 2010).

2.3 Review on exergy analysis in the vapor compression air conditioning system

Generally, based on the first law of thermodynamics, energy performance of HVAC system is determined. However, the exergy analysis can show accurately the location of inefficiencies of a system. The method of exergy analysis is somewhat an innovative practice in which the technique of estimating the thermodynamic losses maintains the second law instead of the first law of thermodynamics. The outcomes of exergy analysis are utilized to measure and optimize the HVAC system. Saidur et al. (2007) found that major exergy losses are occurred in the refrigerator followed by air conditioner, washing machine, fan and rice cooker and, about 21% of total losses are occurred in the refrigerator and 12% are the air conditioner. It is observed that a major portion of exergy losses among the energy sector is happened in the refrigerator and air conditioner (about 33%). Using irreversibility term, it is more supportive to determine the optimum operating conditions (Bejan, 1997; Hepbasli, 2007, 2008). Furthermore, integrating energy analysis, entropy and exergy analysis are able to show a whole image of the system performance.

Nowadays, it is recommended that R290/R600 is used in the domestic refrigerator as the refrigerant instead of R134a. There are a number of reasons for choosing this refrigerant. Firstly, the refrigerator in the experiment using R290/R600 as refrigerants consumes less power than that of the refrigerator using R134a. This is because the saturation temperatures of R290/R600 are lower, and the latent heats of vaporization of the refrigerants are higher than those of R134a. As hydrocarbons have higher latent heat of vaporization and lower evaporator temperature, so these cause less exergy destruction. Based on second law efficiency i.e. exergy efficiency, hydrocarbons are appropriate refrigerants. Alsaad and Hammad (1998) have examined the performance of a domestic refrigerator using propane/butane mixture for a possible replacement of R12. A comparative study is presented for different pure HCs R290, R600, R600a, R1270 and also R22 and R134a with theoretical analysis. This was a comparative study about energetic and exergy performance (Bayrakci & Ozgur, 2009). The energetic and exergy efficiency reaches to the maximum for R1270 at all working conditions. However, the same efficiencies were obtained with R600. Some researches related to exergy and energy analysis on vapor compression systems are shown in Table 2.1.

Table 2. 1: Effect of refrigerant on exergy losses in case of vapor compression system.

Objectives	**Comparison**	**Results**	**Reference**	**Comments**
To perform second law of thermodynamics	R22 and R407	Irreversibility in the compressor is high.	(Aprea & Greco, 2002)	Based on exergy efficiency, R407 would not be a substitute of R22.
Energy and exergy analysis of vapor compression system	R 502, R404 and R507	Exergy efficiency of R507 is better compared to the other refrigerants.	(Arora & Kaushik, 2008)	Theoretical analysis with condenser temperature range 40 to 55 °C.
Energy and energy analysis of pure hydrocarbons	R290, R600, R600a, R1270 , R22 and R134a	R1270 showed better energy and exergy efficiency, R600 also same efficiency	(Bayrakci & Ozgur, 2009)	Theoretical analysis based on EES package program
Thermo-dynamic analysis of vapor compression analysis	R134a, R12, R502	R 134a showed better performance on the basis of 2nd law efficiency and inter-staging is better than single staging	(Khan, 1992)	Research work made comparison with different refrigerants
To know the COP, exergy efficiency and its destruction	R22, R407C and R410A	COP and exergetic efficiency of R22 is higher	(Arora et al., 2007)	Based on computational program

		compared to others.		
Exergy based Refrigerant selection	Three mixtures: R23/R290, R23/R600, R125/R600	Less exergy loss with R23/R290 in comparison with R125/R600 and R23/R600 mixtures.	(Somasundaram et al., 2004)	It is an auto simulation process and applicable for low temperature. Difference in Boiling points of the mixtures is high.

Coefficient of performance (COP), Exergy destruction (ED) and exergy efficiency of refrigerants R22, R407C and R410A have been expected with the help of a developed computational model (Arora et al., 2007). Their investigations have been carried out for evaporator and condenser temperatures within the ranges of -38°C to 7°C and 40°C to 60°C, respectively. The results indicate that COP and exergy efficiency for R22 are higher than those for R407C and R410A. The optimum evaporator temperature with the minimum ED ratio has been evaluated at different condenser temperatures. Experimentally, Aprea and Greco (2002) found that R 407C would not be a substitute of R22 for air conditioning system. Kilicarslan and Hosoz (2010) studied the energy and irreversibility parameters of a cascade refrigeration system employing various refrigerant couples, namely R152a-R23, R290-R23, R507-R23, R234a-R23, R717-R23 and R404a-R23. From all cases, the refrigerant couple R717a-R23 has the highest COP and lowest irreversibility. However, there are some limitations while natural refrigerants are used as refrigerants. Hence, the couple R152a-R23 is the optimum solution.

In natural gas liquefaction plant, exergy losses for refrigeration cycle using propane, ethane and methane as refrigerants were studied by Mehrpooya*et al*. (2006). The results showed that largest exergy loss was occurred when ethane was used as a refrigerant. From their study, it is found that the exergy analysis of a NGL (Natural gas liquids) in the propane based refrigeration cycle can be applied to the other actual cycles. At a lower temperature, Somasundaram *et al*. (2004) found that ARC system with R23/R290 mixture, showed less exergy loss in comparison to R125/R600 and R23/R600 mixtures. Authors selected the refrigerants based on higher differences in boiling points among the refrigerants in order to have effective phase separation.

Shilliday *et al*. (2009) tried to make an analysis for energy and exergy using R744, R404A and R290 refrigeration cycles. To date, it is generally accepted that a transcritical R744 system has shown lower COP compared to an R404A system. In their study, a comparative analysis of exergy and energy has been performed on trans-critical R744 cycles for commercial refrigeration use and compared to R404A and R290. In that analysis, R290 displayed the lowest exergy destruction ratio (EDR), whereas, R744 displayed the highest exergy destruction ratio. Venkataramanamurthy and Kumar (2010) investigated the energy, exergy flow and 2^{nd} law efficiency of vapor compression refrigeration system for refrigerants R22 and R436b. Exergy efficiency and COP of R436b (58% of R290, 42% of R600a) were found higher than those of R22 in all the ranges of temperatures. Another thing is that R 436b has low GWP and zero ODP. An exergy analysis was conducted for a domestic refrigerator between the evaporation and condensation temperatures ranges from -15^{0}C and 40°C using R12 and R413 refrigerants (Padilla et al., 2010). Authors found that the overall exergy performance of R413a was working better than R12. System working with R413a required less power and less irreversibility than the others.

2.4 Heat transfer performance of refrigerants

Generally, the working fluid has low thermal conductivity. Recently, in order to enhance the heat transfer rate from the fluid, the U.S. National Nanotechnology Initiative (NNI) has investigated liquids with dispersed nano sized particles called nanofluids. These can be used to improve the working fluid thermal properties due to their special properties which have been attracted much attention throughout the world.

At first, Choi *et al*.(1996) offered for forming nanofluid by suspending high thermal nanoscale metallic or nonmetallic particles in the base fluid like lubricant with small percentage. These nanofluids have shown that they have superior heat transfer capabilities. Many efforts have been made to investigate the effective thermal conductivity, the convective heat transfer and heat transfer capability during the changes of phase of nanofluifds. Wen et al.(2005) studied the pool boiling heat transfer characteristics of Al_2O_3–H_2O nanofluids and showed that alumina nanofluids can significantly enhance boiling heat transfer. The enhancement increased with increasing particle concentration and reached 40% at a particle loading of 1.25% by weight. Recently, some investigations have been performed using nanoparticles in refrigeration systems and found that these were advantageous to enhance the energy efficiency and reliability of the refrigerators. For example, Wang et al. (2003) found that TiO_2 nanoparticles can be used as additives to enhance the solubility of the mineral oil in the hydro fluorocarbon (HFC) refrigerant. In addition, refrigeration systems using a mixture of R134a and mineral oil with TiO_2 nanoparticles appear to give better performance by returning more lubricant oil back to the compressor with similar performance to systems using R134a and POE oil. Bi et al.(2008) investigated the reliability and performance of a domestic refrigerator with nanoparticles in the working fluids. They used $TiO_2$with mineral oil in the HFC134a instead of polyol-ester. The energy consumption of the R134a refrigerant using mineral oil and nanoparticles mixture as lubricant was saved with 26.1% less energy consumption used with 0.1% mass fraction TiO2 nanoparticles compared to the R134a and POE oil system, which is quite significant for a domestic refrigerator (Bi et al., 2008).

Kedzierski and Gong (2007) found a boiling heat transfer enhancement with nanofluids for refrigerant/lubricant mixtures by using lubricant based nanofluid. The study obtained 50% to 275% improvement in the boiling heat transfer for 4% of volume occupied by CuO having 30 nm diameters. Kedzierski and Gong (2007) further studied about the effect of concentration of CuO nanoparticles on R134a/luibricant mixtures during pool boiling heat transfer. The measurement showed that use of the 2% CuO volume fraction results in smaller pool boiling heat transfer than that of the 4% CuO volume fraction with R134a/polyol ester. But it should also be emphasized not only on enhancement of heat transfer but also on tribological effect in the compressor. Polyol ester (POE) with HFC (Mizuhara et al., 1994) causes wears in the compressors. For the hermetic compressor, there are a number of areas where wear can occur. But the friction characteristics are lower for POE with HFC in comparison to mineral oil. Some additives may help to increase the heat transfer characteristics and as well as reduce the wear behavior also.

In the present study, different types of nanolubricant mixtures are studied (analytically) and find out the high performed mixtures for more heat transfer in the air-conditioning system. Frictional losses are also in an account in those cases. It is the target of this study is to achieve a refrigerant mixture for the air-conditioning system which can provide more thermal and exergy efficiency and more sustainability of the whole system. Wear characteristics will also be in consideration for the hermetic compressor. As uses of air conditioner are increased day by day, so it will give us a quality air conditioner with reliability. The significance of the study is to assist industry and related research works in the optimal combination refrigerant lubricant mixtures for the air conditioner. The output of this research will help air conditioner industry in many ways. The result of material compatibility of compressor with the HFC and the lubricant of this research will help to get high heat transfer rate with minimum frictional losses in the compressor. The miscibility of the MO and POE lubricant has been tested.

2.5 Nature of refrigerants and their properties

Hydrocarbons can be used in the existing refrigeration equipment as a replacement for the original refrigerant and are fully compatible with mineral or synthetic oil. In some cases, no changes in the hardware configuration of the equipment are needed (Undp, 1991). Hydrocarbon refrigerants were accepted before the introduction of CFC and HCFC-fluids (Granryd, 2001). The short atmospheric life time of hydrocarbons takes their GWPs close to zero. Moreover, hydrocarbons provide zero ozone depletion potential (ODP), low toxicity and chemical stability, together with suitable thermodynamic, physical and chemical properties (Maclaine-Cross,2004). Lee and Su (2002) conducted an experiment using isobutene as a refrigerant in the domestic refrigerator and found that its performance was comparable with those of R12 and R22. Akash and Said (2003) studied the performance of LPG from the local market as an alternative refrigerant for R12 in the domestic refrigerator (Sattar et al., 2008). Alsaad and Hammad (1998) investigated the performance of a R12 domestic refrigerator working with a propane/butane mixtures successfully. Somchai and Nares (2005) investigated the application of mixtures of propane, butane and isobutene in a domestic R134a refrigerator and found that propane/butane 60%/40% is the most appropriate alternative refrigerant. Devotta *et al.*(2001) selected R134a, R290, R407C, R410A and the blends of R32, R134a and R125 and found R290 as a potential candidate. Sekhar *et al.* (2005) investigated refrigerant mixtures of R134a/HC in domestic refrigerator and deep freezer and vending machine and walk in cooler. Properties of the refrigerants (especially hydrocarbons) used in the vapor compression refrigeration system are shown in Table 2.2 which are available in nature.

Table 2. 2: Properties of refrigerant for domestic refrigeration

(Maclaine-Cross & Leornadi, 1996; Palm, 2008).

Refrigerant	Triple Point (^{o}C)	Boiling Point (^{o}C)	Critical point (^{o}C)
Propane	-189	-42.08	96.70
Isobutene	-145	-11.76	134.70
Normal butane	-138	-0.54	152.01

Acceptable performance and life for refrigerants in domestic and light commercial use require those to be the non-corrosive, chemically stable, boil below at temperatures and have a critical temperature above the ambient temperature. Tables 2.2 and 2.3 show the naturally occurring refrigerants which satisfy the criteria of refrigerants usable in the small vapor compression system (Maclaine-Cross & Leornadi, 1996).

Table 2.3: Thermal and chemical properties of the refrigerants used in the vapor compression system

(ACRIB, 2001; Hychill, 2010).

Refrigerant	Mol. Mass, g/mol	N. B. P.(°C)	Critical temperature (°C)	Latent heat (kJ/kg)	Freezing point (°C)
R 134a	102.03	-26.15	101.06	222.5	-96.6
R 600a	58.13	-11.73	135.0	367.7	-159.6
R 600	58.10	-0.5	153.0	385.6	-135
R 22	86.48	-40.8	96.02	233.2	-160
R 290	44.1	-42.1	96.8	424.3	-187.1

Hydrocarbons are the most extensively used alternative refrigerant in the vapor compression cooling systems such as domestic refrigerators and air conditioners (Bi et al., 2008). The uses of chlorofluro carbons (CFCs) is a principal basis of depletion in the upper atmosphere, and then it is phased out and replaced by hydro-fluorocarbons

(HFCs) which have zero ODP. But it mixed with lubricant in the compressor causes bulging equipment that chokes the flow and severe friction in the compressor.

2.5.1 Global warming potential on air conditioner

Global warming potential (GWP) is a relative value, used to compare the impact of an emitted gas on the climate and its contribution to climate change. The standard equation for GWP (100) is derived on the basis of the ratio of time integrated radioactive forcing for one kg of any matter or substance, comparative to that for the same amount of CO_2 over a period of 100 years (Honaka et al., 2002; IPCC, 2006). The global warming depends on the density of gas and the atmospheric lifetime (Good et al., 1998; Highwood & Shine, 2000; IPCC, 2006). GWP (100) values can be changed if the radioactive efficiency or the lifetime of the gas is changed (IPCC, 2006).

$$GWP_X(TH) = \frac{\int_0^{TH} \Delta F_X \, dt}{\int_0^{TH} \Delta F dt} \qquad (2.1)$$

To calculate the GWP, Equation (2.1) is valid for sources of gas with long lifetimes. HFC refrigerants are used as a refrigerant for their high thermodynamic performance. However, HFCs showed high GWP on the environment. The CFCs were familiarized at the end of the 1920s as a safe substitute to the existing flammable and poisonous substances. After that, about 65 years later, civilization had understood the alarms of CFCs. When the two researchers were awarded the Nobel Prize for their novel findings, CFCs were forbidden in many countries (Palm, 2008).

For design, repair and services of equipment used in the air conditioner and refrigerators, a series rules or corrections as an international standards are available considering safety. These rules are also applicable for flammable refrigerants. These corrections are providing a structure to develop future technology introducing hydrocarbon as refrigerant. The European Parliament has recently sanctioned a new regulation and directive that banned fluorinated gases with GWP over 150 in all new car with air-conditioner as of 2014, with a phase-out period starting in 2011 (Corberan et al., 2008; Jose et al., 2008).

During the last 15 years, many of the halogenated hydrocarbon refrigerants have been proven to deplete the earth's stratospheric ozone layer. Within the Montreal Protocol (Mp, 1987) the production, distribution and use of CFCs, HCFCs, halogens and methyl bromide are described and the developed countries have phased out the production and uses of these substances (Dosat & Haron, 2002). The thermo-physical properties of HFC-134a are very similar to those of R12, and it is non-toxic, environmentally safe, and its ozone depletion potentials (ODPs) are very low ($<5.10^{-4}$), but the global warming potentials (GWPs) are high (GWP 1300) (Sattar et al., 2008; Sekhar & Lal, 2005; Tashtoush et al., 2002). R22 is used as a refrigerant in the domestic refrigerant which has zero ODP but GWP value is 1700. So, it is a common anouncement in the Europe and rest of the world that R22 should be pahsed out by 2030 (Mp, 1987). R290 has less GWP (value of GWP is 3) and zero ODP. So, by mixing R290 with R22 can be a siutable alterative for R22 to reduce the environmental impacts.. So, there creates a great interest in Europe and elsewhere to be used of hydrocarbons as refrigerants.

The mixture of R22 and R290 can be a candiate in replacement of R22 for air conditioning system which will provide less GWP. If the proposed mixture is R22: R290 (3:1), the GWP will be changed. For the mixture (M1), GWP can be calculated as follows:

$$GWP(M1) = 0.75 \times GWP(R22) + 0.25 \times GWP(R290)$$

$$\Rightarrow GWP(M1) = 0.75 \times 1400 + 0.25 \times 0$$

$$\Rightarrow GWP(M1) = 1050 + 0$$

$$\Rightarrow GWP(M1) = 1050.0$$

So the mixture reduces the GWP compared to that of R22. If the ratio of R290 increases, the GWP will be reduced. If the mass ratio will incur 50:50, then the GWP will be 851.5. But this mixture will be flammable and safety concern. For the mixture, by mass ratio15:85, GWP will be 1190.

2.5.2 Ozone depletion potential (ODP) on air conditioners

Ozone in the Earth's stratosphere layer acts as a shield to prevent most of the harmful ultraviolet light passing through the atmosphere. Ozone in the stratosphere layer has two special properties. Firstly, it devours a relatively small chemical lifetime. Thus, it cannot be distributed over the atmosphere uniformly. Its mixer in the atmosphere is controlled by the complex dynamical and chemical processes (IPCC, 2006). Its ability to absorb ultraviolet leads to an increase in the stratosphere temperature with altitude. Besides, stratosphere ozone layer is essential because it helps to protect the life of the surface living beings from the harmful ultraviolet radiation. From 1974, the devastation of stratospheric ozone layer has been observed. Molina and Rowland (1974) discovered that CFC molecules were stable enough to remain in the atmosphere until they migrates to the stratosphere ozone layer. Then, the molecules will release a chlorine atom with the presence of ultraviolet radiation. Thus, the atomic chlorine will destroy the ozone layer. Consequently, the depletion of ozone layer will allow more harmful ultraviolet to reach the Earth's surface and cause genetic damage to living organisms. The Montreal Protocol (MP, 1987) was called to find the solution for the ozone depletion. At the meeting, all the members agreed that the phase-out of CFCs would be by 2000. Moreover, in the 1992 meeting in Copenhagen, the participants of the protocol agreed to bring forward the phase out date to 1996. Meanwhile, HCFCs are expected to be completely phased out by 2030. ODP of a compound is the relative amount of the ozone degradation with trichloro-fluoromethane (R11) whose ODP is 1.0 by definition.

Recently, scientists have paid much attention to the so-called ‘‘natural fluids’’, which are considered to be more eco-friendly than the synthetic fluids (Alsaad & Hammad, 1998; Beyerlein et al., 1993; Hammad & Alsaad, 1999; Jung, et al., 1996; Liu et al., 1995; Lorentzen, 1995; Purkayastha & Bansal, 1998). These are the hydrocarbons and mixtures of hydrocarbons. These were recommended to be used as refrigerants in the domestic refrigerators to be replaced potentially as ozone depleting fluids. While hydrocarbons are used in large amounts, these will be very inexpensive and available. They are environmentally friendly with zero ODP, and they do not cause any greenhouse effect (Akash & Said, 2003).

2.6 Alternative refrigerants

Early refrigerants were toxic or flammable or both. Early refrigerators leaked refrigerants rapidly mainly through the seals on the compressor drive shaft, creating a fire and health risk (Maclaine-Cross & Leornadi, 1996). Ammonia and carbon dioxide were the only refrigerants available for refrigerations and air conditioners. Ammonia is a poisonous gas when it inhales in large quantity. The search for a completely safe refrigerant having good thermal properties led to be replaced these by halocarbon refrigerant in late 1920s. So, halogenated hydrocarbon refrigerants have been used over the last five or six decades. R22 was used in residential air conditioner and heat pumps due to its energy efficient property. It contains ozone depleting Chlorine and hence the Montreal Protocol decided to phase out R22 eventually they proposed to stop the production of HCFC from 1996 (UNEP, 1987). Halogenated hydrocarbons such as CFC, HCFC and HFC are used for depletion and global warming potential (Dosat &

Haron, 2002). Within the Montreal Protocol (MP) and its amendments, the production, distribution and use of CFCs, HCFCs, halons and methyl bromide are described. Following the MP, the developed countries have phased out the production and use of these substances. The MP approves the use of R134a, a HFC as the alternative of refrigerant CFC12, but did not summarize the future application or policy for this refrigerant. The International Panel on Climate Change (IPCC) describes the specifications of those refrigerants which are not given by the MP (1987). So, the IPCC DOEs not specify about outlines of CFCs but DOEs inform about HFCs, PFKs etc.(United Nations Development Programme, 1999).

Within the IPCC, the environmental effects of HFCs including R134a are deliberated upon. Research has shown that R22 DOEs not have any ODP like CFCs but has a considerable Global Warming Potential (GWP). This problem has proved a worldwide discussion on the need and possibilities of HFCs as refrigerant. It is reasonable to assume that a substitute for CFCs and HFCs need to be found. For the adverse effect on the environment, the uses of the above mentioned refrigerants will be terminated within 2030 (Lee & Su, 2002). Many corporates are forced to find alternatives. Researchers are trying to find a suitable refrigerant having zero ODP and GWP. From the aspect of thermodynamic and environmental safety, hydrocarbons are eco-freindly with zero ODP and negligible global warming potential.

In this case, HFCs can be considered as transitional substances. From few years, various alternative refrigerants have been proposed (Cavallini, 1996; Radermacher & Jung, 1993) and analyzed (Jung et al., 2000) within the limitations of Montreal Protocol. From the literature, it is found that the performance of R22, R290 and R1270 was comparable. R290 is identified as direct substitute of R22. Park and Jung (2006) studied about the performance of R290 and R1270 with different proportions. They found that except R1270, all the refrigerant mixtures showed higher COP and similar refrigeration capacity. In some cases, R407C was considered as an alternative to R22 as it has roughly similar vapor pressure to R22. But it is non-azeotropic mixture. Hence, it may cause the performance degradation of the system in future. R410 is adopted in the refrigerator as an alternative.

Park and Jung (2008) studied about R22, R1270 and R290. They found R290 is the best alternative to R22. Their research was based on small capacity plant/system. But this hydrocarbon has flammable property. So, it is necessary to think about nonflammable refrigerant, having zero GWP with high efficiency.

2.7 Comparisons of different refrigerants used in vapor compression systems

There are three types of refrigerants; synthetic, natural and mixtures of these two. Synthetic refrigerants were used for refrigeration, cold storage and air conditioner. These are: R11, R12, R22 (HCFC 22), R502 (CFC12+HCFC22), etc. They have to be phased out due to their high ODP and to be replaced by R134a and blends of HFCs. Generally, synthetic refrigerants are non-toxic and non-flammable. However, compared to natural refrigerants, the synthetic refrigerants offer lower performance and also have higher GWP. As a result, synthetic refrigerant faces an uncertain future. The most widely used and oldest natural refrigerant is ammonia. It has good thermodynamic, thermo-physical and environmental properties. However, it is toxic and not compatible with some of the common materials of construction such as copper, etc. So, opinions differed on replacements for conventional refrigerants. Natural refrigerants are essentially comeback; one advantage of using them is that they are familiar in terms of their strengths and weakness. They are also completely environment friendly. It is also possible to use blends of various HFCS with HCs with required properties to suit specific applications. Another thing is that it is important to take precautions from leakage, as this will change the composition of the mixture. Table 2.4 shows a list of refrigerants being replaced and their replacements.

Table 2.4**:** Refrigerants, their applications and substitutes

(Lesson 26, 2008)

Refrigerants and their properties	Application	Suggested substitutes R for Retrofit /N for New
R11, NBP=23.7^{0}C, T_{cr}= 197.98^{0}C, h_{fg} at NBP = 182.5 kJ/kg, C_p/C_v = 1.13, ODP = 1.0 GWP = 3500	Large in air conditioning systems, industrial heat pumps as foam blowing agent	R 123(R,N), R 141b(N), R 245fa (N), n-pentane(R,N)
R 12, NBP=-29.8^{0}C, T_{cr}= 112.04°C, h_{fg} at NBP = 165.8 kJ/kg, C_p/C_v = 1.126, ODP = 1.0, GWP = 7300	Small air conditioners, domestic refrigerators, Water coolers and small cold storages.	R 22 (R,N), R 134ab (R,N), R 227ea (N), R401A and R-401B (R,N), R 411A and R411B (R,N), R 717(N).
R 22, NBP=-40.8^{0}C, T_{cr}= 96.02°C, h_{fg} at NBP = 233.2 kJ/kg, C_p/C_v = 1.166, ODP = 0.05 GWP = 1500	Air conditioning systems and cold storages.	R 417A(R,N), R 290, R407C(R,N), R 404A (R,N),R 507 and R 507A (R,N), R 410A and R-410B (N), R 717(N).
R 134^a, NBP=-26.15^{0}C, T_{cr}= 101.06^{0}C, h_{fg} at NBP = 222.05 kJ/kg, C_p/C_v = 1.102, ODP = 1.0, GWP = 1200	Used as replacement for R12 in domestic refrigerators, water coolers, automobiles, A/Cs, etc.	No substitute required: Immiscible in mineral oils Highly hygroscopic.
R 717(NH_3) NBP=-33.35^{0}C, T_{cr}= 133.0^{0}C h_{fg} at NBP = 1368.9 kJ/kg C_p/C_v = 1.31, ODP = 0 GWP = 0	Food processing, Frozen food cabinets, Ice plants and cold storages.	No substitute required Toxic and flammable Incompatible with copper Highly efficient Inexpensive and available.
R 600 (iso-butane) NBP=-11.73^{0}C, T_{cr}= 135.0^{0}C, h_{fg} at NBP = 367.7 kJ/kg, C_p/C_v = 1.086, ODP = 0, GWP = 3	Replacement for R12 Domestic refrigerators, Water coolers	No substitute required Flammable Eco- friendly.

2.8 Justification for selection of refrigerants

Generally, R22 is used in the air conditioner as it has high energy efficiency and energy performance. But many researchers are searching alternative refrigerant to match their thermodynamic properties. Among the properties, vapor pressure is also a vital criterion for selecting refrigerant alternative to R22 (Jiangtao et al., 2009; Venkataramanamurthy & Kumar, 2010; Wongwises & Chimres, 2005). Most available refrigerants are R290, R134a, R410a and R407C. Also binary mixtures can be made as an alternative to R22. Mixing R290 with R22 can be considered as alternative. As R22 has GWP 1700 and R290 has flammability problems so mixing with proper proportions GWP and flammability can be reduced. Using REFPROF 7 software, it is found out the vapor pressure of the selected refrigerants. Hydrocarbons alone cannot be used as an alternative due to its flammability problem. But some of the researchers suggest that for small capacity air conditioner used in the small offices or houses, it is tolerable (upto 2.5 kg) (ASHRAE 34, 1997). Figure 2.1 shows the vapor pressure of the refrigerants with evaporator temperature within a certain range. In this figure, R290 shows very close vapor pressure to R22. R134a and R407C also show close to R22 for a given temperature ranges. A mixture (M1) of R290 and R22 (ratio 3:1 by mass) also shows very close vapor pressure to that of R22 and R290. Only R290 also can be used as an alternative refrigerant to R22 for domestic or small official houses. M1 also can be used instead of R22 in the domestic and official purposes. This mixture is the improvement of refrigerant R22. It has low GWP and low ODP. Analytically, it is clear that R290 shows the best alternative to R22 because the saturation pressure of R290 is very close to R22. It is cheaper, available in nature, recommended for small houses and domestic purposes. It is flammable but tolerable for small uses.

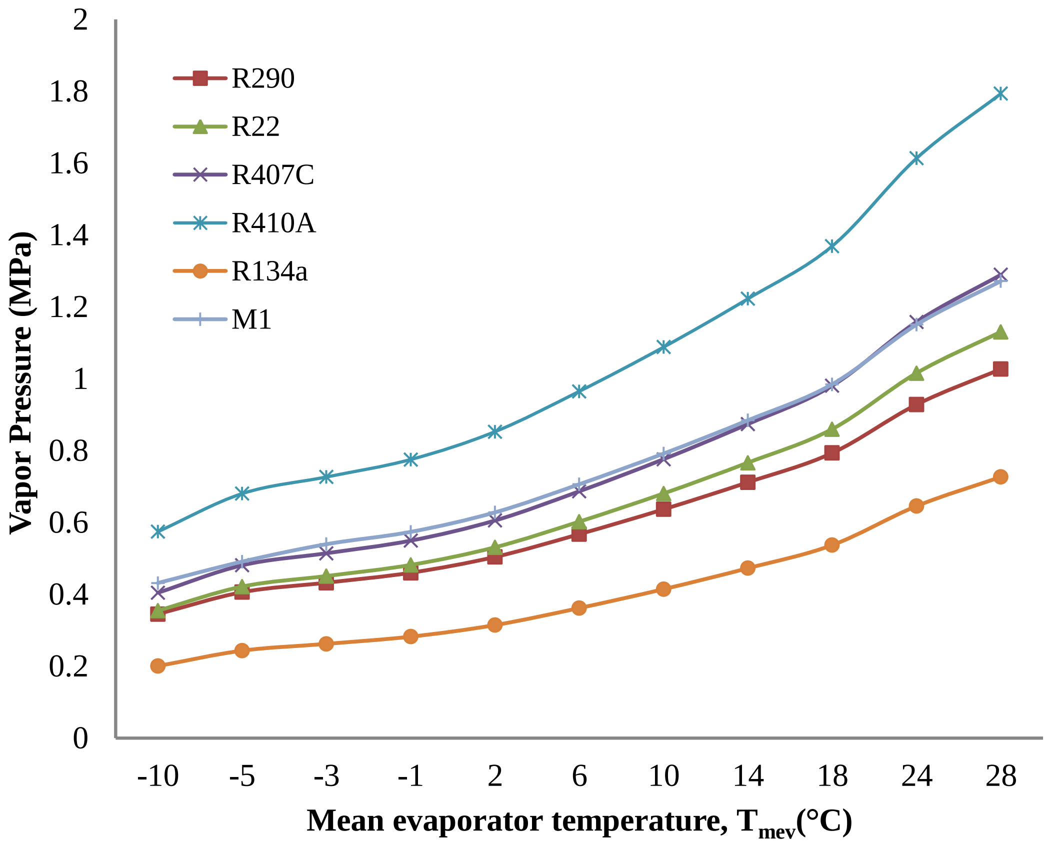


Figure 2.1: Comparison of saturation vapor pressure of different refrigerants with at different temperatures

Pressure ratio of the refrigerants is also the factor for choosing alternative refrigerant. Pressure ratio is the ratio of condenser pressure to the evaporator pressure. It indicates the required size of the compressor also. Figure 2.2 represents the comparison of pressure of R22 and its substitute's candidatures at varying evaporating temperature. From the Figure 2.2, it is clear that from temperature -10°C to 28°C, pressure ratio of R290, M1 and R22 are become very close. Figure 2.2 also shows that pressure ratio for R407C and R22 are almost same at all the evaporator temperatures. But the pressure ratio of R134a is always higher than that of R22 (upto 18%). The pressure ratio of R290 and M1 are 2 to 6% lower than that of R22 in the range of mean evaporator temperatures from -1 to 28°C. So, it means that for R134a heavy compressor and higher compressor work are necessary than R22.

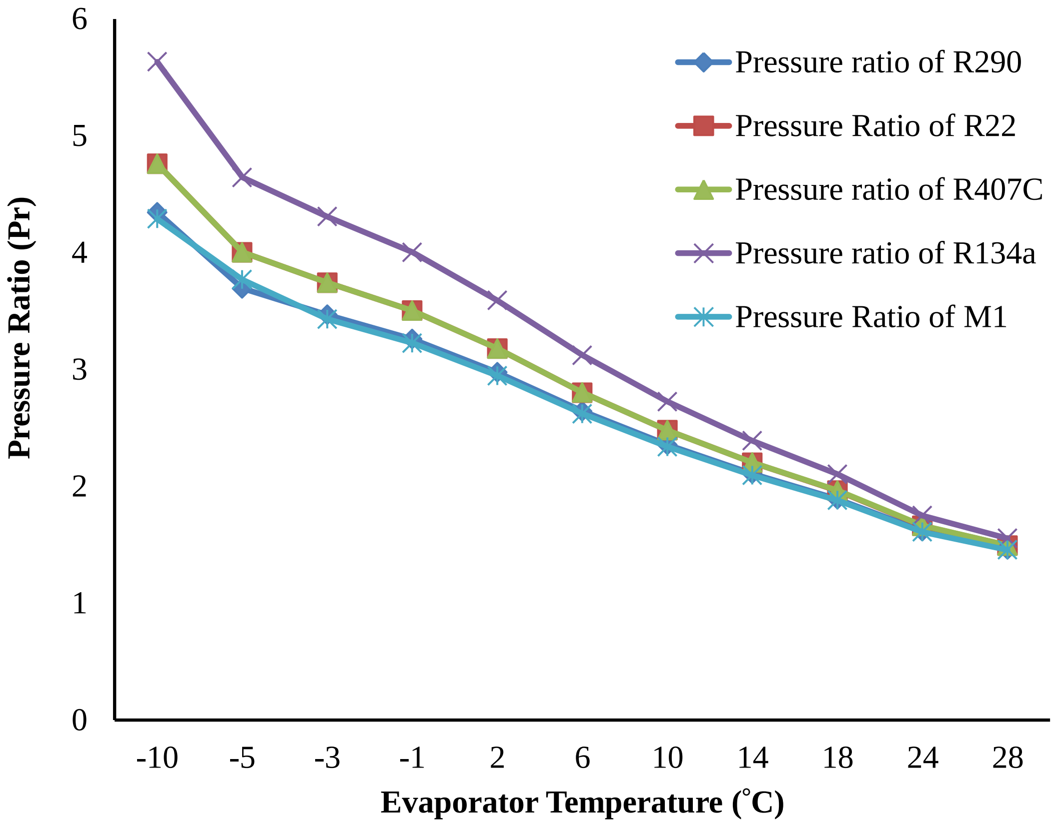


Figure 2.2: Pressure ratio for different refrigerants at different evaporator temperatures.

From the Figure 2.2, it is clear that R407C, M1 and R290 shows the similar trend of pressure ratio to that of R22. Their variations of pressure rise with evaporator temperatures are similar to that of R22. So, refrigerant R290 can be mixed with R22 which will cause no ODP and less GWP and hence causes to achieve higher COP. Mixing R290 with R22 with a ratio of 25: 75 (M1) by mass may cause the reduction of flammability. To increase the heat transfer performance of the air conditioner, nano fluids having high thermal conductivity and fully miscible with the HC mixture and lubricant can be added with small proportion. But the mixtures of nano fluids should be compatible with the existing equipment or hardware configuration of the air conditioner unit. Considering the flammability, the amount of hydrocarbon should be less. Safety precaution should be taken during charging of refrigerant in the system. R290 is a hydrocarbon having high latent heat of vaporization. Thus the heat transfer also will increase while R290 is being used. As R290 has high latent heat vaporization, less amount of R290 will be capable to cool a room. Hence, the total exergy losses also will be reduced.

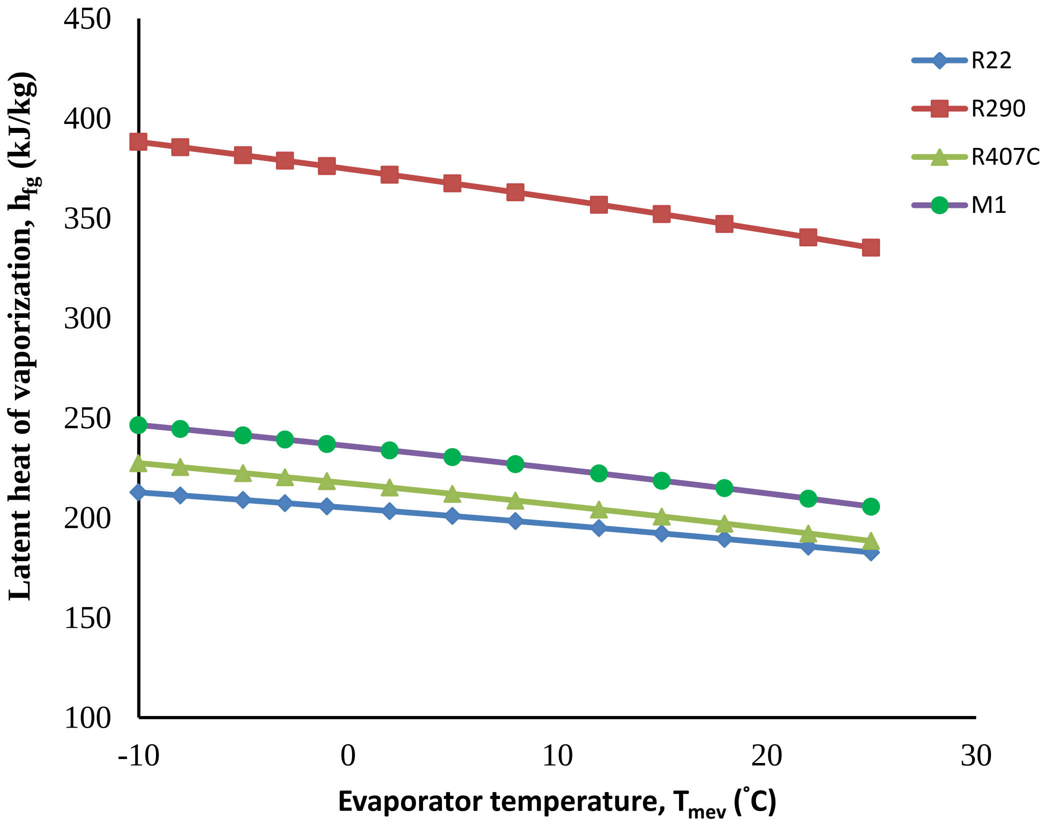


Figure 2. 3: Variation of heat of vaporization with evaporator temperatures for different refrigerants.

From Figure 2.3, it is clear that propane has higher heat of vaporization than that of R22 and the others. The mixture (M1) also shows higher heat of vaporization than that of R22. R290 shows similar heat of vaporization to R22 but M1 and R290 show higher heat of vaporization than R22 at every evaporator temperatures.

Thermal conductivity is important in heat transfer in both single and two phase flows. If the thermal conductivity is higher the heat transfer also will be higher. In the Figure 2.4, it is observed that thermal conductivity of propane is higher than that of the others. Thermal conductivity of the mixture (M1) in both cases (single and 2 phase flows) is higher than that of R22 at every evaporator temperatures. Higher thermal conductivity will influence the heat transfer. But the thermal conductivity in liquid state is decreasing with increasing the evaporator temperature. The thermal conductivity of the vapor DOEs not increase/decrease with temperature changes (Pelletier & Palm, 1996).

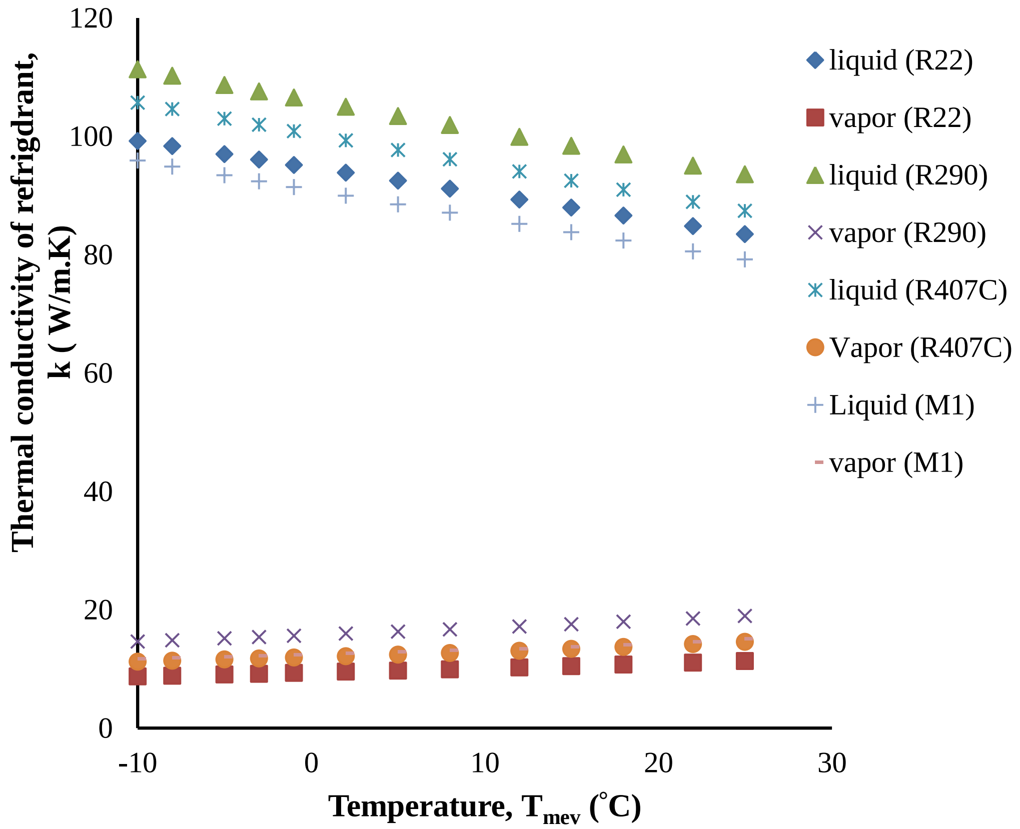


Figure 2.4: Variation of thermal conductivity of the fluid with evaporator temperature.

Viscosity of fluid is important in heat transfer. Figure 2.5 shows that viscosity decreases with increase of temperatures. Viscosity of R22 is higher than all other refrigerants presented in the figure. Viscosity of the mixture M1 is lower than that of R22 at every temperature (Pelletier & Palm, 1996).

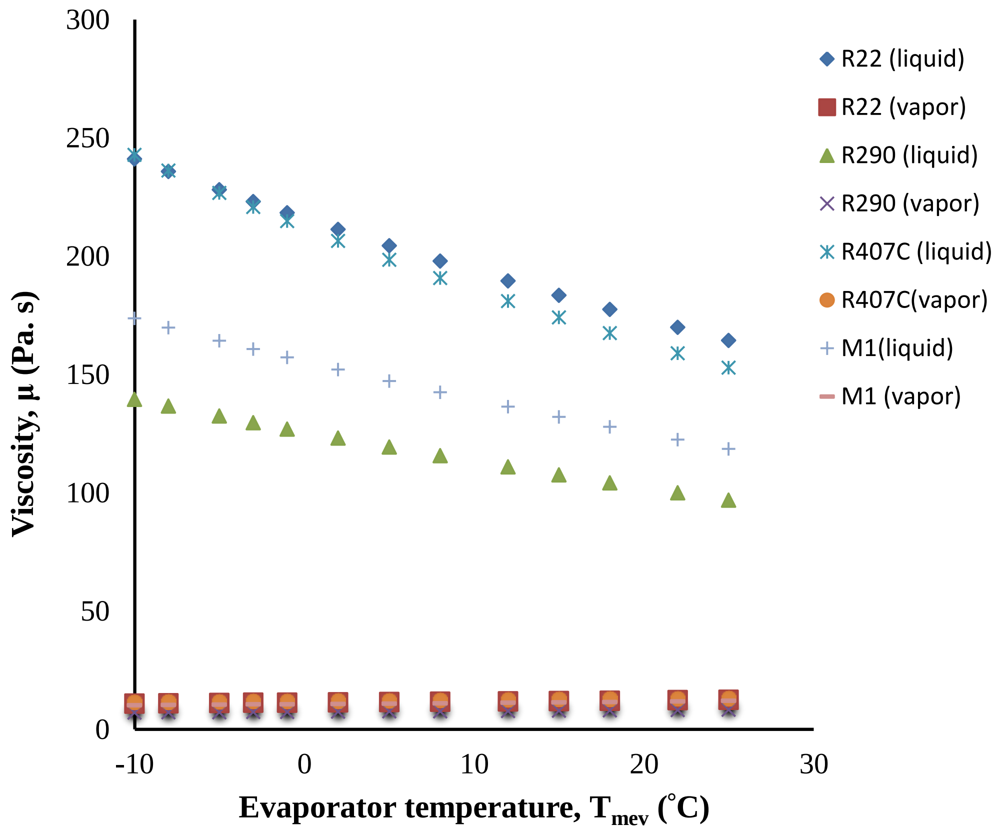


Figure2.5: Variation of viscosity of the fluid with evaporator temperature.

The surface tension of mixture M1 is lower than all other refrigerants shown in Figure 2.6. Higher surface tension indicates lower heat transfer in the evaporator in case of nucleate boiling. It creates a film over the tube wall and decrease heat transfer from the air side to the refrigerant. Boiling heat transfer coefficient is expected to be decreased with increasing surface tension. It is also observed that surface tension is decreasing with increasing evaporator temperature. Hence, the heat transfer performance is higher at higher evaporator temperature. In the condenser, higher surface tension causes an effect on the fins' surfaces to increase the heat transfer (Pelletier & Palm, 1996).

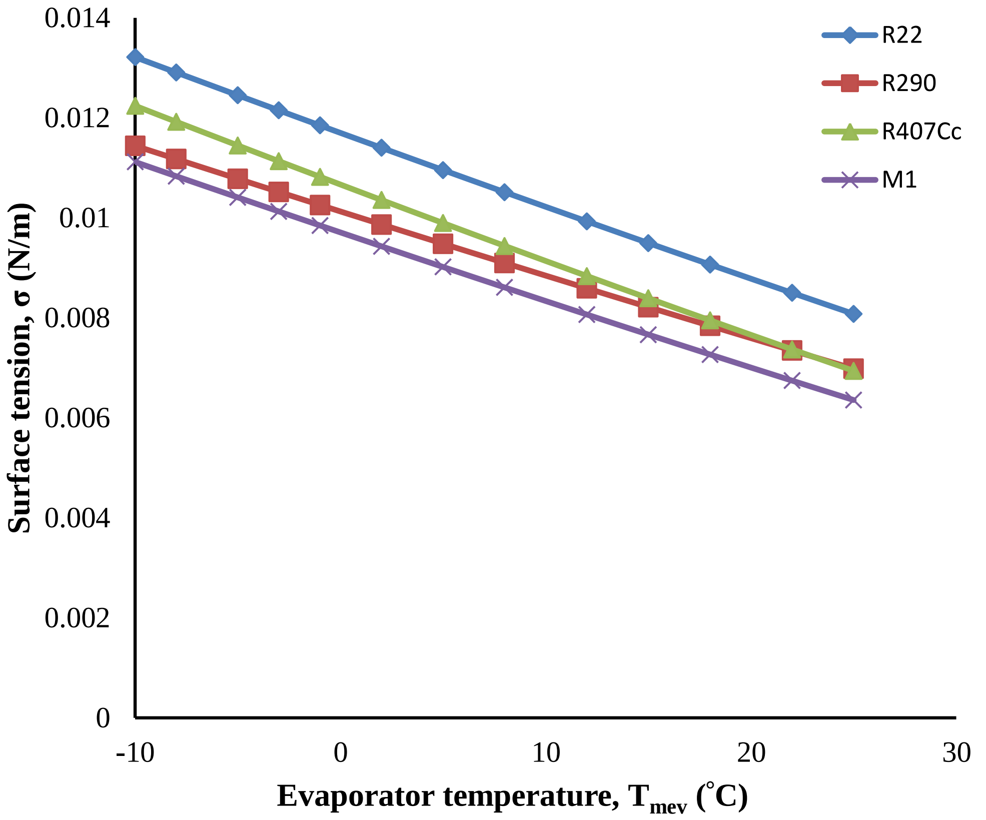


Figure 2.6:Variation of surface tension of the refrigerants with evaporator temperature changes at constant condensing temperature.

From the above discussion, it is clear that Mixture M1 has high potentiality to sustain as a refrigerant in the domestic air conditioner system. It has low viscosity, high thermal conductivity, low surface tension which helps to increase heat transfer rate in the evaporator. It also shows low pressure ratio compared to other refrigerants which indicates less power is required to run the compressor.

CHAPTER3: METHODOLOGY

3.1 Introduction

The experimental set up for room air conditioner was built up to measure energy, exergy and heat transfer parameters by using refrigerant R22 and blend of HC (R290) at Energy lab, Faculty of Engineering, University of Malaya. Thermo-physical properties of the refrigerants were obtained using the commercial software REFPROF 7 at different operating conditions. Flow rates of the refrigerants through the evaporator were determined. Temperatures of the refrigerants in inlet and outlet of the evaporator, compressor, condenser and expansion valve as well as inlet and outlet pressures of the compressor were measured for different environmental conditions. A split type window air conditioner was used for the investigation, and the model is Panasonic CU-PC18DKH (2 HP). The compressor was hermetic sealed type. Specifications of the evaporator, condenser and capillary tube are shown in Appendix A. For measuring the necessary data, some apparatuses were used, and all are described briefly in this chapter. Data was collected using the equipment such as power meter, three pressure transducers, four thermocouples (K-type), a magnetic flow meter and a data logger connecting to a PC. For energy analysis, coefficient of performance (COP), energy usage, work of compression, refrigeration effect and condenser duty were calculated with the measured data of temperatures, pressures and flow rates. For exergy analysis, exergy efficiency and exergy destruction in the components were measured for the refrigerants R22 and M1 at different operating conditions. For heat transfer analysis, the heat rejection rate in the condenser, cooling capacity of evaporator and heat transfer coefficients in the evaporator were calculated with the measured data using some mathematical formulae. The size of the room used for testing was 4X3 m^2. During the experimental works, relative humidity of the room was found to be 45%-53%.

3.2 Experimental facilities

This section provides a brief description of the facilities used for conducting experimental work on the air conditioner. Refrigerants were selected based on the justification described in the literature review. The techniques of charging and recovery of refrigerant into/from the air conditioner are discussed here. Details list and explanation of the apparatuses are presented in section 3.2.1. The apparatuses used in the experiments are listed in Table 3.1.

Table 3. 1: List of aparatuses and their functions used in this experimental work.

Apparatus	Purpose	Apparatus	Purpose
Vacuum pump	To discharge and evacuate the system and make moisture free	Thermocouples K-type	To measure the temperature
Digital electronic charging scale	To measure weight of refrigerants	Digital power meter	To measure power consumption
Data logger	To collect and transfer data to PC for further analysis	Charging hose and pipes	To help for charging gas from cylinder
Flow meter	To measure the refrigerant	Room air conditioning unit	To comparative study with refrigerant
Pressure Transducer	To measure the pressure at desired location	Hygrometer	To measure the humidity of the room
REFPROF 7	To find the properties of the refrigerants		

(a)

(b)

Figure 3. 1: (a) Indoor parts of the air conditioning unit used in the experiment;(b) Outdoor parts of the air conditioning used in the experiment.

The indoor and outdoor parts of the air conditioner used in the experimental works are presented in Figure 3.1 (a, b). Indoor unit contains evaporator only. Whereas, in the outdoor unit, there is a condenser, throttling device and compressor with a hermetic sealed motor.

3.2.1 Experimental apparatuses

3.2.1. (a) Vacuum pump

A vacuum pump was used to evacuate the system as no ingredients or moisture particle could remain in the system. Different types of refrigerant and their mixture were poured in different rates. Before charging new refrigerant or replacing refrigerant, it is essential to clean the inside of the pipes and tubes of the whole air conditioning system. Sometimes, moisture happened to retain in the compressor or in the pipe lines with the refrigerants. As a result, it may damage the valve seals, bearing journals, cylinder wall and other surfaces in the system. The system must be cleaned and evacuated before charging the gas because the impurities and remaining amount of the refrigerant will affect the performance of the system during experimental investigations.

Two different types of refrigerants were used in the experimental investigations. It was required to change the refrigerants for investigation and comparisons. Normally, in the vapor compression air conditioning system, R22 is used as a refrigerant. This refrigerant is the basis for comparisons. At first, R22 was charged into the cylinder. When the experimental investigations were finished, then it was needed to discharge the existing refrigerant from the system. A Yellow jacket of 4 cfm vacuum pump was used to evacuate the system (Figure 3.2) and its specifications are given in Table 3.2. From the service port, one port was connected to the vacuum pump and another one was connected with the compressor inlet port. The vacuum pump was powered connecting with a main power supply. At first, the discharge line was opened and made it vacuum. But to clean the lines from the impurities, vacuum cleaner was used. Vacuum pressure was enough to pull the moisture from the system.

Figure 3.2: Vacuum pump for clearing the gas from the cylinder.

Table 3.2: Specifications of the Yellow jacket supervac vacuum pump

Specifications	**Range/values**
Model	C55JXKCH-4835
Type	2 stages 4 cfm: Thermally overloaded
H/P	1/2HP
Capacity	95 L/m, 114L/m
Frequency	50/60Hz
RPM	1425/1725
Voltage	115/230
Ampere	7.4/3.7, 5.6/2.8
Part number	93544
Serial number	P 303184
Compressor	Rotary vane oil sealed type
Company	Ritchie Engineering Company Inc., Bloomington, USA

3.2.1. (b) Digital electrical charging scale

This device was used to charge the refrigerant/ mixture of refrigerants into the system. With the help of Yellow Jacket digital electrical charging scale, refrigerant was charged. This device was manufactured by Ritchie Engineering Company Inc. USA. It is a metric programmable charging scale. It is an automatic digital charging system by which different amount of refrigerant can be charged. First of all, R22 was charged up to a pressure limit 4.83 bar recommended by the manufacturer. The amount of R 22 charged was measured with the electrical charging scale and found as 850 gm. There was an automatic system to set digitally in the electrical charging system to control the charging of refrigerant also. After that the gas cylinder was removed from the input service port. While the outlet port was tightened then the experiment was performed.

Once data was taken then the system was evacuated with the help of the Yellow jacket pump. Then another refrigerant M1 was being chargded. The cylinder was placed on the digital scale. The facility used for charging is shown in Figure 3.3. Air conditioning and refrigerating board of UK has concluded that hydrocarbon charge will be 45-55% of the CFC or HCFC gas. In this experimental investigation, R22 and mixture of R290 with R22 were used as refrigerant. The charging system consists of a platform, an LCD, an electronic controlled valve and a charging hose (Figure 3.3). One hose was connected from gas cylinder to control electronic valve and another hose was connected from control panel to the service port. This hose was connected to the compressor inlet section.

Figure 3.3: Digital electronic charging scale for charging refrigerant in the system.

The LCD display was connected with the weighing platform and it was also connected to the power supply. The LCD display has two functions. Firstly, it displayed the weight of the cylinder placed on the platform. Secondly, it controlled the electronic valve which commanded the charging of refrigerants.

3.2.1. (c) Thermocouples

Thermocouples were used to measure the temperature at the inlet of the evaporator, compressor, condenser and expansion valve. K-type thermocouples were used in this testing. These types of thermocouples are suitable to be used in room temperature condition, because this experiment was performed at room temperature. These thermocouples sensed the temperature of refrigerant in the piping, and transmit the output to the data logger to be stored in the computer. And then these were connected to the piping by using T-socket. There are four thermocouples set at the beginning of evaporator, condenser, compressor and expansion valve. The photographic view of the thermocouples and T-socket are shown in Figures 3.4 and 3.5, respectively. Two thermocouples were set at the outside of the room. One was set at the outlet of the compressor and another one was set at the outlet of condenser. These thermocouples were connected to the data logger via the channels 1 and 2.

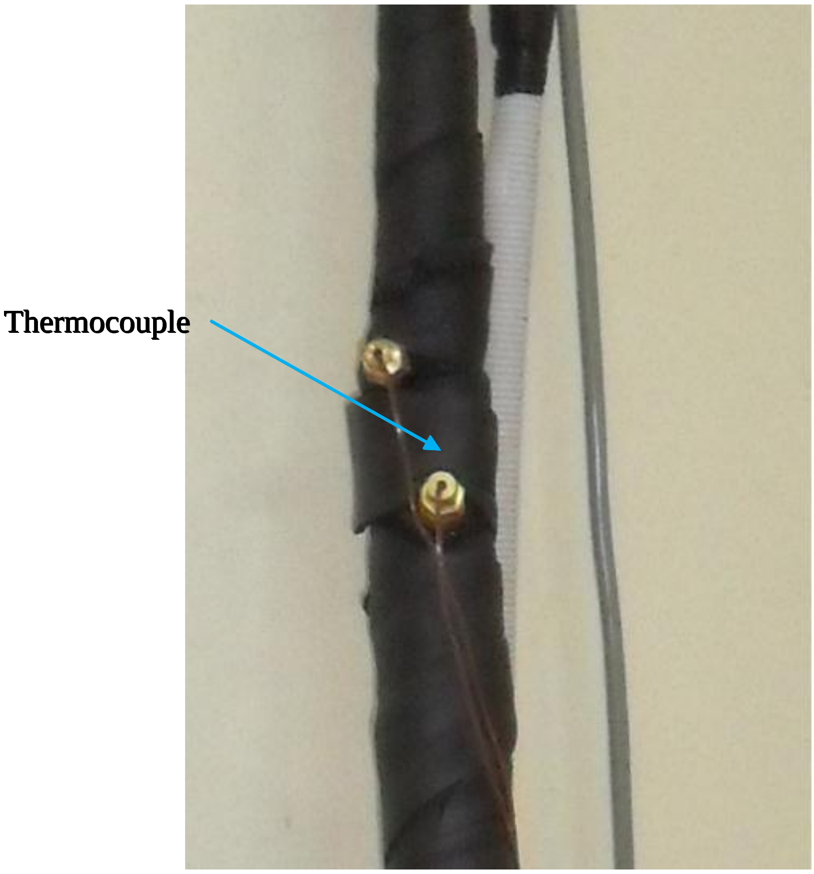


Figure 3.4: K-type thermocouples used in the experiment.

Figure 3. 5: T-socket used to join the thermocouples with the tubes.

To record the dry bulb and wet bulb temperature of indoor and outdoor air temperature, one thermometer called hygrometer was used. This thermometer was used to measure the temperature of the room.

3.2.1. (d) Data logger

A data logger was used to store data in the computer. In the experiment, data logger was used to detect the input data from the thermocouples and to transmit the data to the computer. The data logger (DT80 Type) was used in the experimental work. But it was interfaced with the computer and displayed as text form. Data logger interfacing with PC can show the output data. Thermocouples were directly connected with the data logger into the different channels. The data was displayed in the computer. However, the data logger itself has also a display system. Four thermocouples were connected with two channels (1 and 2). Pressure transducers could not be connected directly to the data logger. A voltage regulator was used to reduce the supplied voltage from 220 to 25 dc voltage. Through adapter the pressure transducers were connected with the data logger into the channels 3 and 4.

Figure 3.6: Data logger used to detect pressure and temperature.

A data logger was used for power meter, pressure transducer and thermocouples. Thermocouples were set through different ports to the data logger. At first, it was necessary to define the channels for temperature readings. Pressure channels were also selected for different ports. Ports 1 and 2 were selected for temperatures and ports 3, 4 and 5 were selected for pressures transducer. Specifications of the data logger are shown in Appendix A (Table A3).

3.2.1. (e) Two phase flow magnetic type flow meter

The flow meter was used to measure the flow rate of the refrigerant that flows through the evaporator. In this experimental investigation, the ABB type of flow meter was used to measure the flow rate of two phase refrigerant flow. The flow meter was connected to piping lines of refrigerant flow with the help of flange. The specification of the flow meter is DN 25 meter size, with digital LCD display and 4 to 20 mA signal output, 24Vdc power supply. With an extension and reducer, the flow meter was connected to the pipeline before the evaporator. This flow meter can measure the flow of refrigerant having different densities with different properties. It can also measure the flow rate of two phase flow. Necessary properties were supplied by REFPROF 7 software for setting the thermal properties of the refrigerant in the flow meter. Hence, the flow rates of the refrigerant were detected in liter /minute or gm/s according to the flow condition. Specifications of the flow meter are given in Appendix A (Table A5).

Figure 3. 7: Flow meter settings in the air conditioner unit.

3.2.1. (f) Pressure transducer

Pressure transducers were used to measure the pressure of the flowing refrigerant at different locations of the systems (shown in Figure 3.8). Pressure transducer was used to detect the refrigerant pressure in desired unit. The range of pressure in this transducer is 0-50 BarG. Output will be in mA and the range will be 4.00-19.994 mA. Operating temperature range is -40^0 C to 125 ^{0} C. The model is 132F, BCM sensor technologies. The output voltage from the pressure transducer is converted into voltage by adding resistance of 250 Ώ. Therefore, scaling is necessary for making a relationship between the voltages and pressure.

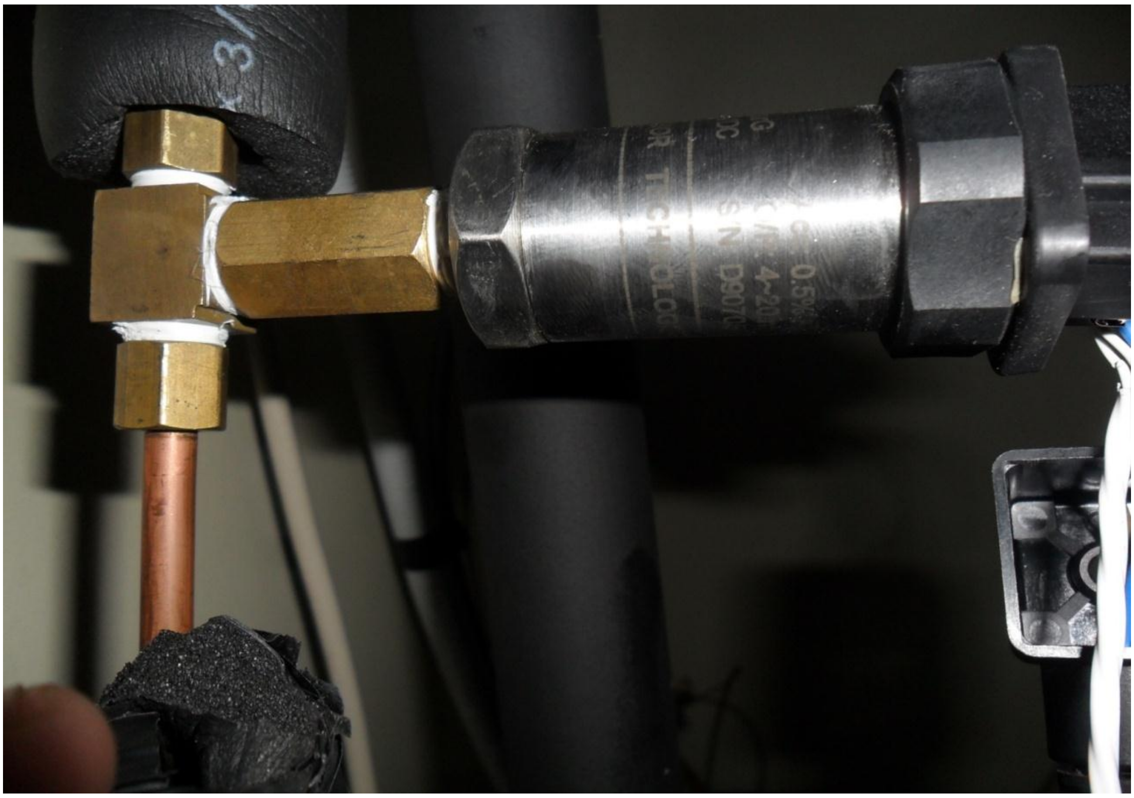
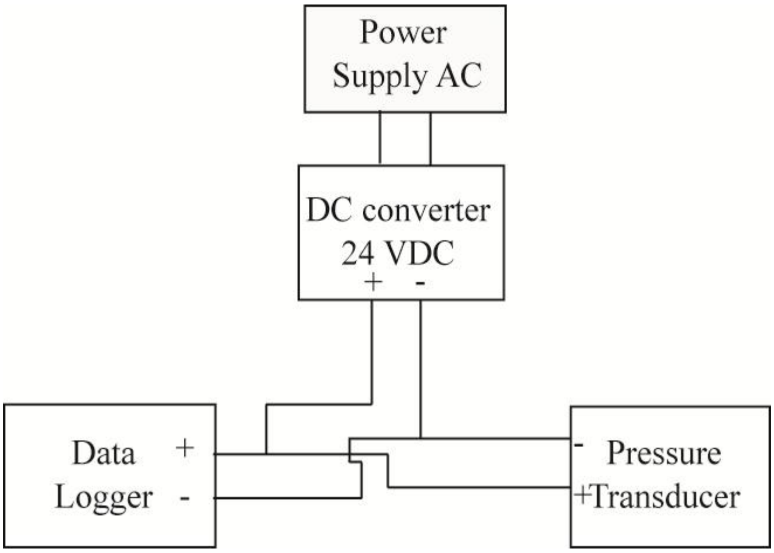
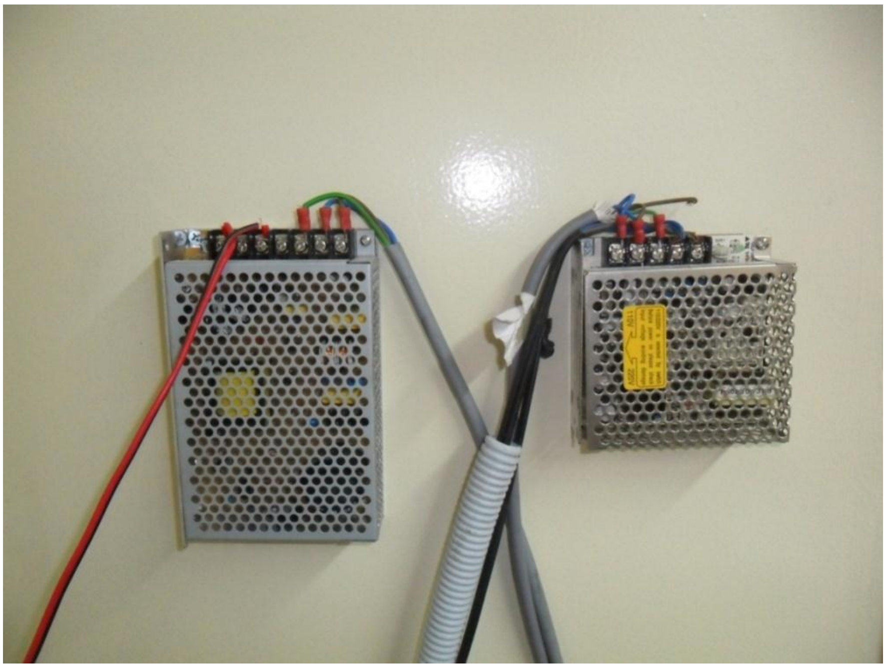


Figure 3.8: Pressure transducer attached with the line through T-socket.

Voltage range for 0-50 bar pressure is 1V -5V. 1V is for 0 bar pressure and 5V is for 50 bars. There is a linear relationship between the pressure and the output voltage. Pressure transducer needs power supply with voltage ranges from 0 to 2 4 volt (dc). Pressure transducers were connected with data logger via voltage adapter. Circuit diagram of the power supply to the pressure transducer is shown in Figure 3.9. The specification of the pressure transducer has been given in Appendix A (Table A4).



(a)



(b)

Figure 3. 9: (a) Electrical connection of pressure transducer to data logger via power supply, (b) Voltage regulator and DC supplier.

3.2.1. (g) Digital power meter

A digital power meter was used to measure the power/energy consumption by the compressor. Power meter was used to read the voltage (volt), currents (amp), and also the power (kW) that has been consumed by the air-conditioner unit during running condition of the air conditioner different refrigerants. The model of the power meter that has been used in this experiment is YOKOGAYA WT130 type (Figure 3.10).

Figure 3.10: Digital power meter used to measure power consumption by the compressor.

The specification of the digital power meter has been given in Appendix A (Table A2).

3.2.1. (h) Charging hose and valves

A Yellow Jacket type charging hose was used in this experimental investigation (shown in Figure 3.11). This hose was used to charge the refrigerant during the installation, and also it was used for recharging of different refrigerant replacing the existing ones. Valves are used to control the amount of refrigerants that flows to outdoor unit during charging the gas. There was a small pressure gauge which indicates the amount of pressure increase due to charging refrigerants. It was recommended by the manufacturer for this set up that charging refrigerant is limited up to 4.48 bars.

Figure 3.11:Charging hose and valve to charge and discharge the refrigerant.

3.2.1. (i) REFPROF 7 software

The most powerful and authentic software REFPROF 7 has been used to measure the thermal properties of the refrigerants. For analytical and experimental analysis, it is essential to know the thermal properties. Properties of any kind of mixtures with any proportions with the refrigerants are easy to find or determine. It is easy to find boiling point of new mixture, latent heat of vaporization, critical pressure etc. for the mixture and can plot graphical presentation. Any output from the data logger or power meter stored in the computer. So, to make sure there have a connection between the computer and this device, the computer was connected properly with the power meter software, data logger software, and also REFROP 7 software. REFROP 7 software, are the software that can calculate the enthalpy and entropy of refrigerant. Visual Studio 6 also needs to install to visualize the properties of refrigerants from REFROP 7.

3.2.1.(k) Room air conditioning unit

The air conditioner has been used in this experiment is Panasonic, PC18DKH. This air conditioner has capacity of 2 hp. The air conditioner is a non-inverter type which is a standard model. The specifications of the air conditioner used in this research are presented in Table 3.3.

Table 3. 3: Specifications of the air conditioner used in the experiment.

Specifications	**Value or range**
Type	Split type air conditioner
Model No.	Panasonic: CU-PC18DKH
High Pressure	2.7 MPa
Low Pressure	1.6 MPa
Refrigerant	R22
Charge limit for R22	980 g
Power input	1.92-1.98 kW
Current	8.9-8.6 mA
Capacity	19010-19080 kJ/h

3.3.1 (l) Hygrometer for relative humidity

Hygrometer was used to measure the humidity of the room. It was also used to measure the dry valve and wet valve temperature of the controlled room. This device is used to measure the room temperature also (Figure 3.12). Due to weather changes and atmospheric condition, humidity was changed. This device is calibrated by technical specialist through university authority. It was found that error in room temperature was approximately 0.5°C and in relative humidity was around 0.5%.

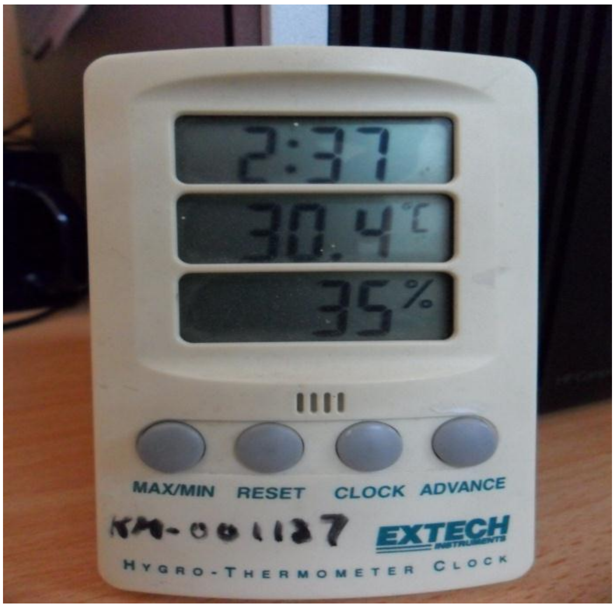


Figure 3.12: Hygrometer used to measure the humidity ratio, temperature in the controlled room condition.

3.2.2 Experimental set up and procedures

Experimental facility consists of thermocouples, pressure transducers, power meter and flow meter. At first, the main parts of the split type air conditioner are installed by the experts. After that experimental apparatuses were connected to the main system. Hygrometer is used to measure the room temperature, relative humidity i.e. the dry bulb and wet bulb temperatures.

3.2.2(a**)** Experimental set up

In this work, a split-type-air conditioner provided by the Institute of air conditioner research unit is to be adopted as the refrigerant unit and some parts of it will be modified for operation convenience. The experimental set up consists of four T-type thermocouples, three pressure transducers in the different positions and an ABB type flow meter. To know the condition of the room, a hygrometer is set in the room to detect the atmospheric condition i.e. relative humidity, dry bulb and wet bulb temperature. Flow meter is of magnetic type and can detect the flow rate through the evaporator. It can measure both the flow rate as mass flow rate and volume flow rate according to the selection of unit. In this experiment, mas flow rate was measured in g/sec. It is necessary to define the other properties of the refrigerants using the REPROF7 software.

All the thermocouples and pressure transducers are connected with the PC through a data logger. Figure 3.13 shows the schematic diagram of the experimental system, where the left part of the Figure shows the indoor appliances while the right part is outdoor ones. First time it necessary to evacuate the compressor before charging refrigerant in to compressor. With the help of vacuum pump, the refrigerant inside the pipes and other ingredients are discharged. This supervac system pulls the contaminants to create a vacuum up to 15-17 micron. Then placing the cylinder on the scale and connecting to the service port and control panel refrigerant is charged up to 4.26 bar pressure. The amounts of charging refrigerants are different for different refrigerants. During refrigerant charging the compressor is on. The desired room temperature is set. The data logger and flow meter is observed. After sometimes the condition of controlled room will be steady state and temperature reading will be steady. Then the temperatures and pressures reading are recorded and at the same time dry bulb and wet bulb temperatures are also recorded.

In the similar way, seven readings are taken for each refrigerant. Simillar procedures have been done refrigerant M1. From these data, energy analysis, exergy analysis and heat transfer parameters are calculated.

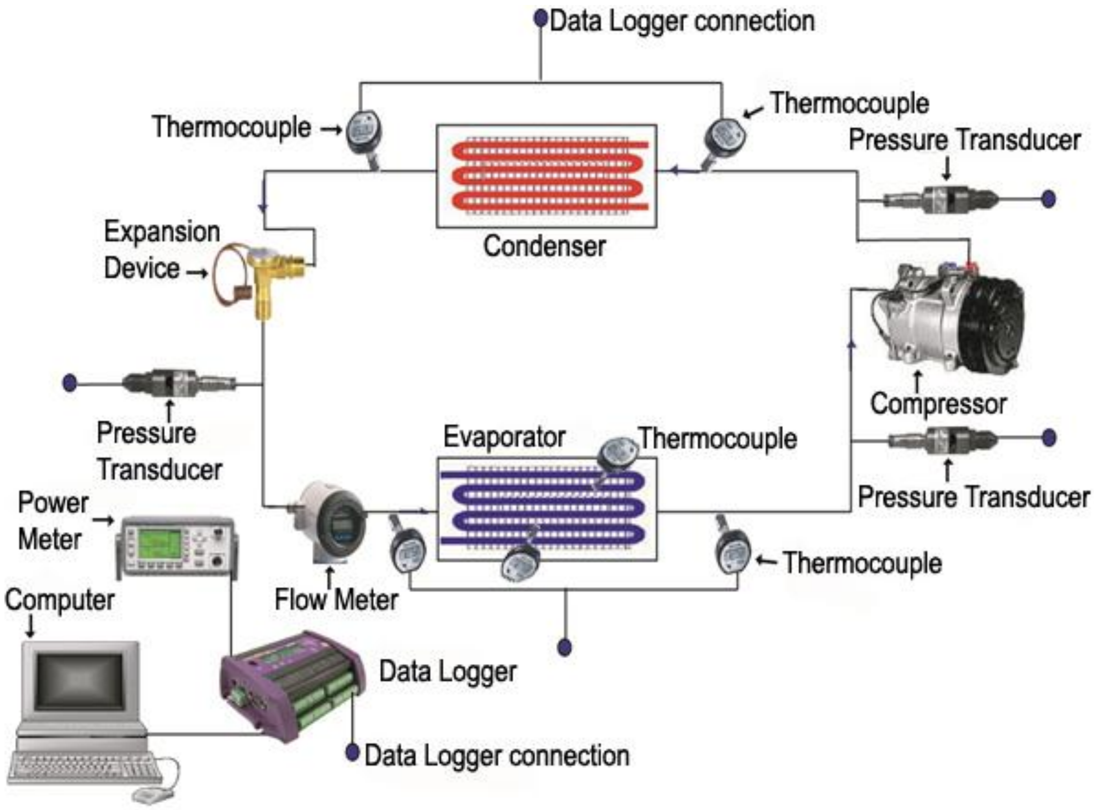


Figure 3.13: Schematic diagram of the experimental setup used in the Energy Lab, University of Malaya, Malaysia.

Enthalpy of the refrigerants at the different temperatures and pressures are obtained from the REFPROF 7. The refrigerating capacity could be obtained by multiplying the enthalpy difference with the air volume. There is a rectifier and a nozzle for measuring the air flow rate; thermocouples are set far from the nozzle to record the dry bulb and wet bulb temperatures of the inlet air. The pressure transducers were set on the hole of the nozzle to measure the pressures, pressure difference and static pressures at different locations. The thermocouples are also located on the outlet position to measure the dry bulb and wet bulb temperature of the outlet air.

Each of the measurements was recorded seven times continuously at a time step of 3 minutes interval when the working conditions are to be stable for 30 minutes. The recorded data averaged. All of the data were taken at standard air conditioning working conditions.

Moisture combining with the refrigerants may cause reaction with the lubricants and other materials in the system. It may cause a deterioration of the lubricants and make sludge on the valves and produce other effects which will reduce the life of the system. Hence, initially the system was cleaned with trichloroethylene to eliminate unwanted traces of dirt or any other foreign materials inside the system that may influence the accuracy of the test results.

3.2.2 (b) Experimental procedure

Refrigerant should be charged through the inlet port. Yellow jacket digital electronic scale was used to charge the system. It is an automatic digital charging device charges gas accurately. At first, the system was to be charged with R22 then the experiment will be performed. Amount of charging depends on refrigerant types. For R 22, charging was around 850 gm. It was monitored or controlled by service port also. After the data was taken, the refrigerant was recovered/removed from the system. Then the HCs mixture with different proportions was poured. Air conditioning and refrigeration board of UK has recommended that the equivalent hydrocarbon charge of a CFC or HCFC system should be approximately 40-50% by mass (ACRIB, 2001). In this experimental investigation, equivalent hydrocarbon charge of HFC 22 was considered approximately 25% by mass. The amount of charging R22 was 850 gm. For the refrigerant M1, the amount of R22 was 637.5 gm and the amount of R290 was 212.5 gm. The amount of refrigerant was adjusted by knowing the pressure inside the service port. The refrigerant was poured upto 4.26 bar pressure and observing whether there was sufficient cooling occurred in the room or not. During experimental work, the room temperature was maintained at approximately 19-20°C.

3.4 Determination of the properties of the refrigerants

Generally, R22 is used as refrigerant in the air conditioner for the domestic purposes. Due to global warming problems, it has been found from the literature that R22 would be replaced by R290. R290 can be used as a substitute of R22 which occupies zero GWP and zero ODP. From the literature and using REFPROF 7 software, some properties (thermal and chemical) of the proposed refrigerants are tabulated in Table 3.4. Critical temperature of R290 is much closer to that of R22. But molecular weight of R290 is about half of the R22. It indicates that R290 occupies a half of the volume of the compressor cylinder. If 50% of R290 gas is poured it will cause flammability problem. In order to reduce the flammability effect, M1 was selected by mixing 75% of R22 with 25% of R290. M 1 shows that critical temperature is very much closed to that of R22 but the molecular mass is about 80.63% of R22. Among the three mixtures, density of M1 is more close and comparable to that of R22.

Table 3. 4: Properties of the refrigerants used in the vapor compression system.

Refrigerant	Chemical Formula	Critical Temp °C	Critical Pressure MPa	Critical density Kg/m^3	Molar mass g/mol	Normal Boiling Point (°C)
R290	$CH_3CH_2CH_3$	96.675	4.2471	218.54	44.096	
R22	$CHClF_2$	96.145	4.99	523.84	86.468	-40.8
R134a	CF_3CH_2F	101.06	4.0593	311.9	102.03	-26.074
R410A	Mixture of R32 and R125(50:50)	71.358	4.9026	459.53	72.585	
R407C	Mixture of R32, R125,R134a(23:25:52)	86.034	4.6298	484.23	86.204	
Mixture 1	R22/290 (3:1)	86.02	4.5334	383.27	69.72	
Mixture 2	R22/R290(50:50)	87.4	4.3995	304.385	58.406	
Mixture 3	R134a/R290	82.875	4.056	306.56	61.579	

3.5 Mathematical formulations

Energy and exergy analyses need some mathematical formulations for the simple vapor compression cycle. In the vapor compression system, there are four major components: evaporator, compressor, condenser and expansion device. Heat rejection and heat addition will be different for different refrigerants and hence it will cause a change in energy efficiency for the refrigerants. Based on the exergy analysis, exergy losses in the different components of the system will not always same. Exergy is consumed or destroyed due to the entropy changes and irreversibility of the system depending on the associated processes (Sahin et al., 2005). To specify the exergy losses and destructions in the system, thermodynamic analysis is to be made. In this study, the following assumptions are made:

1. Steady state conditions are remained in all the components.
2. Pressure loses in the pipelines are neglected.
3. Heat gains and heat losses from the system or to the system are not considered.
4. Kinetic as well as potential energy and exergy losses are not considered.

Schematic diagram of vapor compression refrigeration system is shown in Figure 3.14 and the relevant T-S diagram of the system is shown in Figure 3.15.

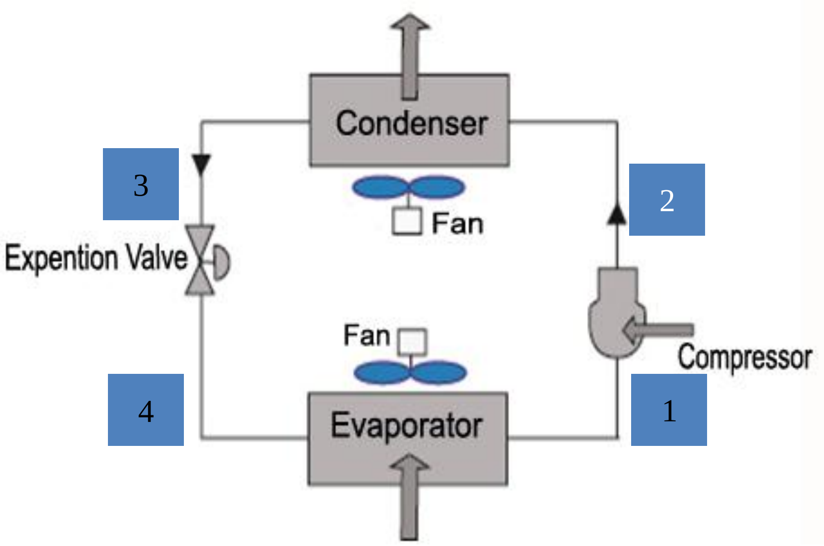


Figure 3.14: Schematic diagram of vapor compression refrigeration system

(Shilliday et al., 2009; Yumrutas et al., 2002).

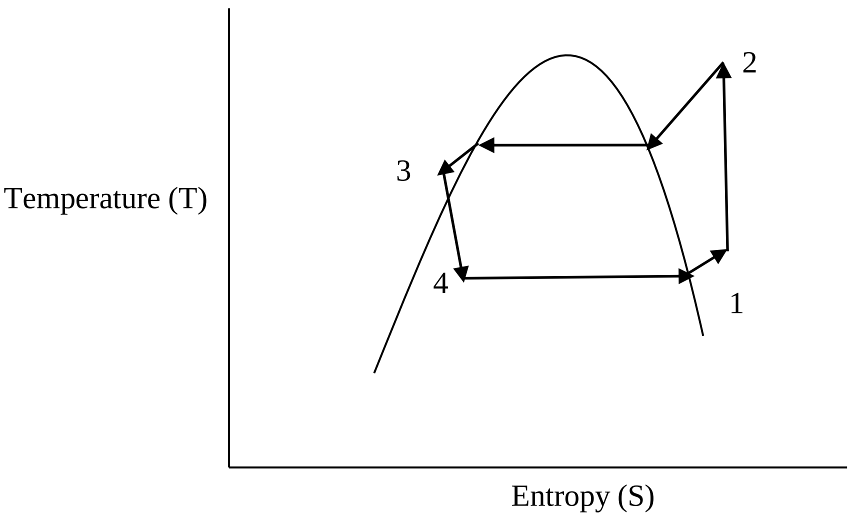


Figure 3.15: T-S diagram of ideal vapor compression refrigeration and air conditioning system.

(Shilliday et al., 2009).

The pressure transducers and thermocouples were to be connected with the data logger. The output of these sensors was in the form of voltage or current. The data logger converts this voltage or current to pressure in MPa and temperature in °C or °F according to the setting. The data logger can read the thermocouples of types J, K and T. In this experimental work; K-types of thermocouples were used. The data logger was interfaced with the computer, and software was to be installed to operate the data logger. Power meter software (WT130 power meter) was installed in the computer which stores the power consumption of the air conditioner at an interval of one minute within 24 hours. The pressures and temperatures from the data logger were used to determine the enthalpy of the refrigerants at different states using REFROP 7 software. Values of enthalpies were used to obtain the performance of the air conditioner.

The performance parameters are refrigeration capacity, the energy consumption of the compressor, the co-efficient of performance (COP), condenser duty, heat rejection ratio, discharge pressure and temperature. The second law analysis deals with exergy efficiency, exergy destructions, etc. in the components of the systems at different operating conditions for both refrigerants.

3. 5.1 Energy analysis of the air conditioning system

Air conditioner uses vapor compression refrigeration system in the thermodynamic cycle. Refrigerant is used as a working fluid in the vapor compression system. Energy changes in each components of vapor compression air conditioning system. Four components are evaporator, compressor, condenser and expansion device. In the evaporator, heat is absorbed from the controlled room or chamber and thus the refrigerant becomes vapor from the liquid states. The capacity of the refrigerant to absorb heat per kg of refrigerant from the surrounding is called the refrigeration capacity. It depends on the properties of refrigerant flow and vapor pressure as well as atmospheric temperature. The compressor needs external power supply to compress the vapor with constant entropy. But it has some frictional losses and the efficiency of compressor is generally about 75%. In the condenser, heat is rejected from the high temperature refrigerant to the outside. Heat rejection is enhanced by using fin in the condenser. The throttle valve permits the fluid to expand and thus causes pressure drop with constant enthalpy, no loss of energy. Mathematical formulae for energy analysis are analyzed below according to the components of vapor compression system.

3.5.1(a) Evaporator

Basically, main purpose of installing the air conditioning system is performed by the evaporator. Heat is absorbed from the room and thus the room is being cooled. The energy and mass balance of the evaporator is shown in Figure 3.16. Mass of refrigerant entered into the evaporator $m_{r,in}$ is equal to mass flow out of the evaporator ($m_{r,,out}$). Q_{ev} is the total heat absorbed by the evaporation process.

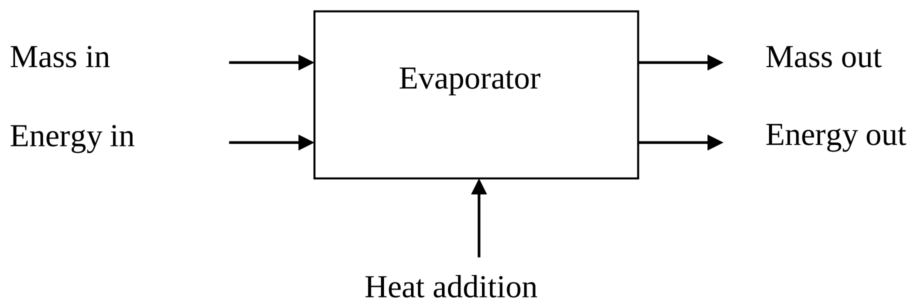


Figure 3.16: Schematic diagram of mass and energy flow for evaporator.

The total heat addition from cold space which is called refrigerating effect (Q_{ev}). This can be easily obtained from the energy balance of evaporator shown in Figure 3.16 by using Equation (3.1). From Figure 3.16, energy balance and mass balance can be shown in the following way,

Mass balance, $m_{rin} = m_{rout} = m_r$

Energy Balance,

$$m_{r,in}\, h_4 + Q_{ev} = m_{r,out} h_1$$

$$\Rightarrow Q_{ev} = m_r (h_1 - h_4) \quad (3.1)$$

Here, Q_{ev} also sometimes called total heat absorption in the evaporator. It can be expressed as the evaporator capacity to extract heat from the cold room is denoted by q_e. This capacity is called refrigeration capacity of the evaporator is defined by

$$q_e = (h_1 - h_4) \quad (3.2)$$

From the Equation (3.2), it is observed that q_e is the function of enthalpies at the inlet and outlet of the evaporator. These enthalpies also varied with pressure, temperature and refrigerant properties such as viscosity, surface tension, density, etc.

3.5.1(b) Compressor

The compressor is hermetically sealed rotary compressor. Ideally, the work of compression (W_{cs}) is considered as isentropic. This can be expressed as:

$$W_{cs} = m_r (h_{2s} - h_1) \quad (3.3)$$

Actual work of compression (Wc) is defined as

$$W_c = \frac{W_{cs}}{\eta_{is}} \quad (3.4)$$

Or the actual compressor work is directly measured the power meter reading. Thus the relation between enthalpy and actual work of compression is as follows:

$$W_c = m_r (h_2 - h_1) \quad (3.5)$$

For per kg of refrigerant flow, the actual work of compression will be

$$q_c = (h_2 - h_1) \quad (3.6)$$

Where,

h_1 = Enthalpy at inlet of compressor, kJ/kg

h_2 = Enthalpy at outlet of compressor, kJ/kg

3.5.1 (c) Condenser act as an heat exchanger

In the condenser, heat is rejected as the fliud inside the tube have higher temperature compared to that of in the hot atmosphere. The hot vapor condensed to liquid state with constant pressure. Free convection heat transfer is happened here. The mass flow rate through out the whole arrangement is same and it is m_r. Energy balance of the condenser is shown in Figure 3.17.

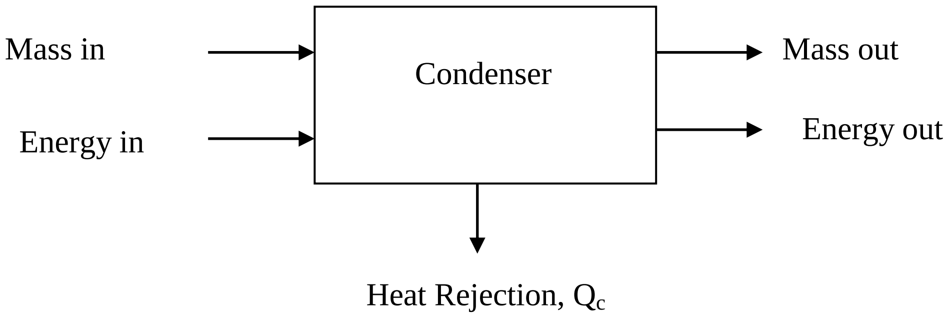


Figure 3.17: Schematic diagram for mass and energy flow of thecondenser

From the Figure 3.17, mass balance can expressed as follows: $m_{rin} = m_{rout} = m_r$

Energy Balance can be expresssed as follows:

$$m_{r\ in} h_2 - Q_c = m_{r\ out} h_3$$

$$\Rightarrow Q_c = m_r (h_2 - h_3) \tag{3.7}$$

Heat rejection per kg of refrigerant flow is called consdenser duty. Heat rejected by the condenser to the atmsphere is given by

$$q_c = (h_2 - h_3) \tag{3.8}$$

Where,

h_2 = Enthalpy at inlet of the condenser, kJ/kg

h_3 = Enthalpy at outlet of the condenser, kJ/kg

3.5.1(d) Throttle valve

In the throttle valve, enthalpy remains constant, only the pressure is changed. No heat is absorbed or rejected during throttling process. Sometimes capillary tube is used as throttle valve. But pressure drop depends on diameter and length of the capillary tube. For the inverter type air conditioner, the tube is adjusted with the load.

First law analysis is sometimes called energy analysis. The performance of vapor compression air conditioning system is measured by the coefficient of performance (COP) and is defined as the ratio of net refrigeration effect to the work of compression. High COP indicates how well the cooling of the space is performed by the air conditioner. It can be expressed as:

$$COP = \frac{Q_e}{W_{el}}$$

$$COP = \frac{m_r(h_1 - h_4)}{W_{el}} \tag{3.9}$$

From Equation (3.9), it is clear that coefficient of performance of a vapor compression system depends on the heat transfer in the evaporator and the compressor power. Heat transfer in the evaporator depends on the difference in enthalpy of the inlet and outlet refrigerant. But enthalpy is the function of temperature, pressure and latent heat of vaporization of the refrigerant. The compressor power also affects the compressor efficiency, discharge pressure and pressure ratio of the compressor. It also affects the irreversibility of the compression system. Another thing is that for higher heat of vaporization of hydrocarbons, the mass flow rate is decreased. Hence the compressor power to cause the pressure high is decreased. So, the compressor power is reduced and thus the COP increased.

The COP is not maintained constant for all the running time of the air conditioner. As the ambient temperature is changing with day time, so the COP also changes. This COP is called seasonal COP as the ambient temperature is changed with time. It is easy to define COP as average value for a range of ambient temperature.

3.5.1 (e) Pressure ratio

It is the ratio of compressor outlet pressure to the inlet pressure. In another words, it's the ratio of condenser pressure (Pc) to evaporator pressure (Pe). High compression ratio indicates compressor has to work more to lift the pressure for a given mass flow rate. It can be expressed as

$$\mathrm{Pr} = \frac{P_c}{P_e} \tag{3.10}$$

High pressure in the condenser needs larger size of the condenser and long capillary tube. It is also related to cost.

3.5.1 (f) Energy efficiency ratio

The term energy efficiency ratio (EER) is very similar to COP. It is the ratio of the cooling or refrigeration capacity of an air conditioner in British thermal units (Btu) per hour, to the total electrical power input (in watts) under certain specified condition. Air conditioner with EER ratings higher than 10 are considered as most cost effective. When the EER ratio is higher, the unit cost to operate will be lower. Mathematically,

$$EER = \frac{Q_e \ in\ Btu\ in\ hr}{W_{el} \ in\ W} \tag{3.11}$$

This energy efficiency ratio sometimes varies with period or season or time. This is termed as seasonal energy efficiency ratio (SEER).

3.5.2 Exergy analysis

Mathematical formulation for exergy analysis in different components can be arranged in the following way (Bayrakci & Ozgur, 2009):

Specific exergy in any state, $\Psi_i = (h_i - h_0) - T_0(s_i - s_0)$ (3.12)

3.5.2 (a) Evaporator

Heat addition in evaporator, $Q_{ev} = \dot{m}(h_1 - h_4)$ (3.13)

Exergy destruction, $I_{ev} = \dot{m}(\psi_4 - \psi_1) + Q_{ev}\left(1 - \frac{T_0}{T_{ev}}\right)$

$$I_{ev} = \dot{m}[(h_4 - h_1) - T_0(s_4 - s_1)] + Q_{ev}\left(1 - \frac{T_0}{T_{ev}}\right) \quad (3.14)$$

3.5.2 (b) Compressor

Compressor work, $W_c = \dot{m}(h_2 - h_1)$ (3.15)

For non-isentropic compression, $h_c = \dfrac{h_{2s} - h_2}{\eta_c}$ (3.16)

Electrical power, $W_{el} = \dfrac{W_c}{\eta_{mech} \times \eta_{el}}$ (3.17)

So, exergy loss, $I_{comp} = \dot{m}(\psi_1 - \psi_2) + W_{el}$

$$= \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2)] + W_{el} \quad (3.18)$$

3.5.2 (c) Condenser

$$Q_{cond} = \dot{m}(h_2 - h_3) \quad (3.19)$$

Exergy loss,

$$I_{cond} = \dot{m}(\psi_2 - \psi_3) - Q_{cond}(1 - \frac{T_o}{T_{cond}})$$

$$= \dot{m}(h_2 - h_4) - T_0(s_2 - s_3) - Q_{cond}(1 - \frac{T_o}{T_{cond}}) \qquad (3.20)$$

3.5.2 (d) Expansion device

Exergy destruction, $I_{\exp} = \dot{m}(\psi_4 - \psi_3)$

$$= \dot{m}(s_4 - s_3) \text{ [Throttling, } h_4 = h_1\text{]} \qquad (3.21)$$

Total exergy destruction in the system

$$I_{total} = I_{cond} + I_{\exp} + I_{comp} + I_{evap} \qquad (3.22)$$

Exergy efficiency,

$$\psi_x = \frac{\psi_1 - \psi_4}{W_{el}} \qquad (3.23)$$

With reference to cited literatures, it is assumed that mechanical efficiency of the compressor is 90% and the electrical efficiency of the motor is 90%.

3.5.3 Heat transfer analysis

Heat transfer is occurred by convection through the condenser and evaporator. Heat gain is happened in the evaporator and heat rejection is occurred in the condenser. Cold refrigerant entered into the evaporator and it gets heat from the air inside the room. So, the room is being cooled at a desired temperature. After getting from the compressor, hot fluid goes through the condenser and it rejects heat to the atmospheric air. Heat transfer analysis deals with the cooling rate of the room air and heat rejection rate in the condenser. Heat rejection ratio is also the parameter for heat transfer analysis of the air conditioning system. Setting two thermocouples at the inlet of condenser and outlet of the condenser, temperature inlet and outlet can be measured. Setting thermocouples at the beginning of the evaporator and outlet of the evaporator cooling capacity can be measured. Using REFPROF 7 software, enthalpy of the refrigerant at that state can be measured. With this measured data heat rejection ratio can be obtained. Cooling capacity denotes how fast the room is being cooled. It also denotes high heat transfer performance and low cost. From the selected refrigerants, cooling rate and heat rejection rate can be measured and observed. From these results it can be concluded which refrigerant will be better based on performance. Some of the parameters are indication of heat transfer analysis.

3.5.3 (a) Heat transfer coefficient

Heat transfer occurs in the evaporator as well as in the condenser. Refrigerant enters the evaporators as liquid phase but it receives/absorbs heat from the room though the wall of the tube. From wall of the tube heat is transferred by convection to the fluid. Then it becomes vapor. Heat gained from the air of the room air is the changes of enthalpy of refrigerant through the pipe. Due to pressure drop in the evaporator, temperature also changes. Mean temperatures are taken as saturation temperature of the refrigerant gas (Ts). Heat is entered to the refrigerant through the tube. Tube wall temperatures are changed. There are four thermocouples set over the tube. The inside wall temperatures are measured as Tw_i. Heat absorption by the refrigerant is Q_{ev}. Then the radial heat flux q, for the evaporator is to be measured as follows:

$$q = \frac{Q_{ev}}{\pi d_i L} \tag{3.24}$$

The average inside tube wall temperature ($\bar{T}wi$) will be measured as follows:

$$\bar{T}wi = \frac{\sum_{1}^{4} Tw_i}{4.0} \qquad (3.25)$$

The mean static pressure in the evaporator has been taken as the mean pressure using the outlet and inlet pressures. The mean temperature of the refrigerant, T_s is to be taken as saturation temperature of the refrigerant at mean pressure. The coefficient of heat transfer of the evaporator for the refrigerant was calculated by the following equation:

$$h_c = \frac{q}{(\bar{T}w_i - Ts)} \qquad (3.26)$$

The values of "h_c" are changing with refrigerants changes. It will be different for R22 and for the M1. For M1, its value is high compared to that of R22 at every evaporator temperatures. High values of h indicate that heat transfer is high. Some cases q is increased with mixing refrigerant with high conductive materials such as nano fluid. It is in found that nowadays CuO and Al_2O_3 are used in research for increasing heat transfer rate. The term q also depends on the viscosity, surface tension, density of the fluid, and these parameters are related to temperature.

3.5.3(b) Heat rejection ratio

Heat is rejected in the condenser and absorbed in the evaporator. The rate of heat transfer in condenser is the heat rejection. It depends on the refrigeration capacity of the evaporator and heat added by the compression process. Heat rejection ratio is the ratio of heat rejection from the condenser to the heat addition in the evaporator. Condenser should be designed such a way that heat rejection must be higher than the heat addition in the evaporator. However, this should not be so large. Higher values of HRR indicate more heat is rejected in the condenser, but sufficient heat is not added from the room to the evaporator. Thus, the room is not sufficiently cooled. On the other hand, heat is added by the friction in the compressor. This is avoidable. Higher HRR also indicates over design of the condenser which indicates a high cost.

3.5.3 (c) Condenser duty

A condenser is an important device in the vapor compression air conditioning system. Hot vapor passes through the condenser which enters the condenser as a discharged from the outlet of the compressor. Condenser and evaporators are the heat exchanger of the system. In the condenser, there are some fins used to enhance the heat transfer to the atmospheric in the outdoor parts. Condenser duty is the heat transfer or rejection to the atmosphere. The rate of heat transfer is the function of evaporator temperature and the condensing medium. It also depends on the refrigeration capacity of the system. Condenser duty depends on the condensing pressure and temperature of the fluid. Proper condensation is necessary to condense the refrigerant and to make return it the evaporator. It is assumed for ideal case that the pressure remains constant through the condensation but in practical case there is a pressure loss in the condenser. Thus it causes irreversibility of the system.

The mathematical formulation for condenser duty of the system is q_{cond} and can be expressed as (Refer to Figures 3.14 and 3.15)

$$q_{cond} = h_2 - h_3 \tag{3.27}$$

The enthalpy at states 3 and 4 shown in Figure 3.17 can be found using REFPROF 7 for different refrigerants. It is also denoted as heat rejection capacity of the condenser.

3.5.3 (d) Cooling capacity

Literature shows that uses of hydrocarbons with mineral oils as lubricant cause to enhance the heat transfer rate in the evaporator. High heat transfer causes also less exergy destruction hence it reduces the power consumption as well as high useful work. At first, heat is transferred from the refrigerant to the inner side of the condenser pipe by convection. Then from the inner side by conduction, heat goes to outer side of the pipe and by convection again it is transferred to the atmosphere. Hydrocarbon mixed with R22 also causes high latent heat of vaporization. High cooling capacity indicates faster cooling of the room. It reduces also the refrigerant mass and compressor running time. Hence, the power consumption is reduced. It also reduces the discharge temperature of the compressor. Thus, the life of the compressor is increased.

$$\text{Cooling capacity, } Q_{cool} = m_r(h_1 - h_4) \tag{3.28}$$

Sometimes, cooling capacity can be treated as ton of refrigeration (TR). It is obtained by multiplying 3.51 to the values in kW.

3.5.4 Energy savings techniques

By increasing energy performances of the system, a large amount of energy can be saved. After all, energy savings is one the most important criteria for all researches. Malaysia is a hot climatic country. Usage of air conditioner is increasing day by day. Malaysian Government has taken some regulations called energy regulations to control the energy usage in different sectors. They urged that in the domestic air conditioner thermostat temperature for the control room should be increased from 22°C to 24°C. Some techniques can be adopted to reduce the energy usage. These are as follows:

- Using hydrocarbon mixture as a refrigerant
- Increasing thermostat temperature in the room
- Changing operating temperatures

3.5.4 (a) Hydrocarbon mixture as a refrigerant

Generally, the refrigerant used in the air conditioner is R22 for domestic purposes. But hydrocarbon R290 can be used mixing with R22 to increase the performance. Thus, the energy usage will also be reduced. Energy savings can be treated as follows:

$$AES\% = \frac{AEU_{R22} - AEU_{M1}}{AEU_{R22}} \times 100\% \tag{3.29}$$

3.5.4 (b) Changing thermostat temperature

Generally, the indoor temperature set point has a great effect on energy usage by the compressor of the air conditioner. Increasing indoor set point temperature will save energy more (Kongkiatumpai, 1999; Yamtraipat et al., 2003). There are some literatures relating to that. Yamtraipat et al. (2004) studied about the indoor set point temperature from 22 to 28°C. In this experiment, the indoor set point temperature was around 20 °C. Recently, Malaysian Government announced a regulation for all the office buildings, universities and organizations that the set point for their air conditioner should be 24°C to save the electrical energy. Consider the average energy used per day for indoor set point temperature at 20°C is AET_{20} in kWh. Average energy used per day (considering running time of air conditioner unit is 8hr/day) for any temperature, T is assumed as AET_T. Based on the energy usage at set point temperature 20 °C, energy savings in percentage for other set point temperatures will be as follows:

$$EST\%_{20-T} = \frac{AET_{20} - AET_T}{AET_{20}} \times 100\% \tag{3.30}$$

Malaysian Government declared for setting indoor set point temperature as 24°C. If the annual energy consumption (TEC_{ac}) of air conditioning systems in Malaysia is found then the total average energy savings in Malaysia for air conditioner can be easily calculated by using Equations (3.30) and Equation (3.31) as follows (Kongkiatumpai, 1999; Yamtraipat et al., 2003):

$$AES = TEC_{ac} \times \%EST_{20-24} \tag{3.31}$$

Due to changes of set point temperature, energy usage can be reduced. Thus, annual electrical bill will be saved. Annual Bill savings in Malaysia will be as follows:

$$Bill \quad Savings = AES \times Unit \ \mathrm{Pr}ice \tag{3.32}$$

In Malaysia, unit price of electricity is 0.218 RM/ kWh.

3.5.4 (d) Changing operating temperatures

Operating temperatures for air conditioner are evaporator, condenser and ambient temperatures. These temperatures have great effect on coefficient of performance as well as power consumption. Evaporator and condenser temperatures depend on refrigerant types, pressure ratio, etc. Ambient temperature cannot be changed easily. But it is changing due to changes of environmental conditions. Its incremental rate can be decreased by proper tree plantation, making green the environment, reducing the usage of high GWP substances and refrigerants. Power consumptions of the system at different ambient conditions were found and from this data energy savings can be calculated.

3.5.5 Exergy savings or loss reductions

Exergy savings indicates the reduction of exergy losses. It indirectly causes to increase the exergy efficiency of the system. Exergy loss reduction helps to achieve higher performance to convert energy into useful work. Exergy loss causes to degrade the performance of the system. It will cause to increase power consumption also. Nowadays researchers are trying to reduce the exergy losses. Condenser temperature, evaporator temperature and ambient temperature have a great effect on exergy reductions. Using proper additives, lubricants, refrigerant with high latent heat of vaporization and low pressure ratio will cause to reduce the exergy losses of the system. Changing thermostat indoor temperature will cause higher evaporator temperature and hence it reduces the exergy losses.

3.5.5 (a) Replacement of refrigerant R22 with M1

In the household and small capacity air conditioner, R22 is used as a refrigerant. Based on the exergy and energy analysis, hydrocarbon mixture with R22 has been proposed to be used as an alternative. The mixture (M1) was used in this experimental investigation for savings energy and exergy. Exergy loss for R22 at given operating temperatures is Ix_{R22}, and exergy loss for M1 at that condition is Ix_{M1}. Reduction of average exergy losses or exergy savings rate can be determined as follows:

$$AIS_{refrigerant} = \frac{Ix_{R22} - Ix_{M1}}{Ix_{R22}} \times 100\% \tag{3.33}$$

3.5.5 (b) Increasing mean evaporator temperature

For a given condenser temperature range, when mean evaporator temperature increases, exergy lose will be decreased. It also depends on ambient temperature. Consider initial mean evaporating temperature as T_{mevT1} and increased mean evaporating temperature as T_{mevT2}. Exergy losses at that temperature are Ix_{mevT1} and Ix_{mevT2}. So, average exergy savings can be treated as follows:

$$AIS_{mevT2} = \frac{Ix_{mevT1} - Ix_{mevT2}}{Ix_{mevT1}} \times 100\% \tag{3.34}$$

3.5.5 (c) Decreasing mean condensation temperature

Many literatures stated that exergy losses varied due to changes in evaporating temperature. Generally, the condensation temperature is considered as 45°C. So, exergy loss at this condensation temperature is considered as base exergy loss ($Ix_{Tc=45}$). For any condensation temperature Tc (°C) (lower than 45°C), exergy loss can be treated as Ix_{Tcond}. Then the exergy savings can be calculated as follows:

$$AES_{cond} = \frac{Ix_{Tc=45} - Ix_{Tcond}}{Ix_{Tc=45}} \times 100\% \tag{3.35}$$

3.6 Uncertainty analysis

In the real researches, almost all researchers of the world deal with data resultant from experiments or tests. The connection between primary measurements and the results is always a mathematical function of some kind i.e. if U is a result and X_1, X_2 X_3 are primary measure and of them. Uncertainty (U) in the measurements can be calculated as

$$U = f(X_1, X_2\ X_3 \ldots\ldots\ldots\ldots\ldots\ldots) \tag{3.36}$$

In the experiment, it is necessary to calculate the uncertainty in the results from the estimates of uncertainty in the measurands. This calculation process is called "propagation of uncertainty". According to Kline and McClintock (1953), the uncertainty U resulting from the uncertainties in X_1, X_2 X_3 …………can be expressed as follows:

$$W_U = \left[\left(\frac{\delta R}{\delta X_1} w_{x1} \right)^2 + \left(\frac{\delta R}{\delta X_2} w_{x2} \right)^2 + ------ \left(\frac{\delta R}{\delta X_m} w_{xm} \right)^2 \right]^{\frac{1}{2}} \tag{3.37}$$

Where wx1, wx2, wx3-------are the uncertainties in the individual variable X_1, X_2, X_3 -----.

In this experiment primary measurements were T_a, T_{mev}, T_c, P_{in}, P_{out}, W_c and m_r,. The results of uncertainty analysis of the primary measurements are presented in Table 3.5, while the uncertainties of the calculated quantities are presented in Table 3.6.

Table 3. 5:Uncertainties in measurands

Item measured	Bias limit, B	Precision limit, P	Total limit, W(%)
Temperature , T	0.4	0	0.4
Pressure, P	.5	0	0.5
Mass flow rate, m_r	.05	.02	0.053
Power Consumption	.0034	.01	0.0156
Enthalpy, h	.05	0	0.05
Entropy, s	.05	0	0.05

Table 3. 6: Uncertainties in calculated quantities

Quantity	Total uncertainty (%)
COP	0.80
R.E.	0.17-.15
Condense duty	0.15-.12
Exergy loss	2.25-3.5
Exergy efficiency	1.5-2.0

The calculations of the above values are given in Appendix B.

CHAPTER 4: RESULTS AND DISCUSSIONS

4.1 Introduction

This chapter covers the variations in COP, compressor power, energy consumption for different refrigerants when changing the operating conditions such as evaporator temperature, ambient temperature and condensation temperature. Here refrigerants are R22 and mixture of R22 and R290 with a mass ratio of 3:1. The energy consumption and performance of air conditioner have been evaluated based on the refrigerant R22. Results of this study are also compared with those of the other researchers for the similar sectors. Exergy parameters such as exergy efficiency, total exergy destruction and exergy losses in different components were calculated with the measured data at various ambient, evaporator and condensing temperatures. Heat transfer parameters such as heat transfer rate, heat rejection ratio, cooling rate and condenser duty and heat transfer coefficient were calculated for both refrigerants and the results were compared for analysis at different conditions.

4.2 Actual vapor compression cycles for the refrigerants

Ideal vapor compression cycle consists of two isentropic processes and two constant pressure processes. But the actual vapor compression cycle was different from the ideal one. Entropy was not maintained constant in the compression process. Due to the friction occurred and other causes, the compressor compress the refrigerant with non-isentropic process. When refrigerant flows through the evaporator and condenser, there is a frictional loss in the pipe lines. Thus, the pressures were not maintained constant. The pressure loss inside the evaporator and condenser caused a loss or decrement of performance of the system. This caused an increase in the irreversibility of the system as well as the exergy losses. The capacity of the system to perform is well when cooling of the room was degraded due to pressure losses. Figure 4.1 shows actual T-S cycle of the vapor compression system. Pressure losses also depend on evaporator temperature and refrigerant types. Hydrocarbon mixture, M1 has low pressure losses compared to the refrigerant R22. This mixture has high exergy performance also.

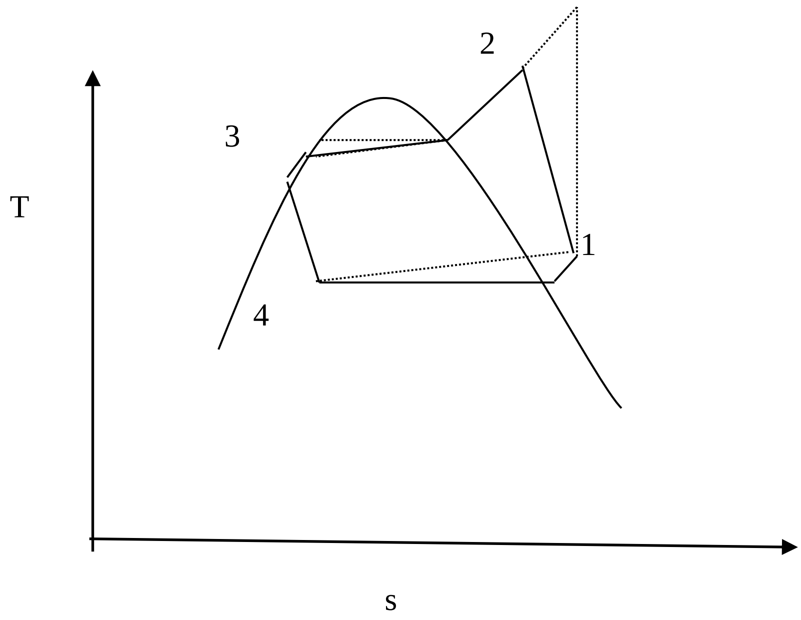


Figure 4.1: Actual T-S cycle used in the analysis.

The enthalpies of R22, and the mixture of R290 and R22 were to be calculated with the help of REFPROP 7 software. The enthalpies at the inlet and outlet of the compressor, condenser, expansion device and evaporator for the above mentioned refrigerants at specified temperatures and pressures were also calculated. Using Equations (3.1)-(3.8), condenser duty, refrigerating effect, work of compression, coefficient of performance and heat rejection ratio were easily calculated and the results for different refrigerants are compared with each other. The results are discussed in the following 4.3 to 4.4 sections.

4.3 Energy analysis, and its parametric variations

In energy analysis, the usable parameters for vapor compression refrigeration systems are coefficient of performance (COP), energy efficiency ratio (EER), evaporation capacity or refrigeration effect and condenser duty or heat transfer by condenser. All the parameters were varied with evaporator temperature, condenser temperature, ambient temperature and refrigerant types.

4.3.1 Effects of evaporator temperature on coefficient of performance

Figures 4.2 - 4.4 show the variations in coefficient of performance of the air conditioner with the variations in mean evaporator temperature for three distinct ambient temperatures. The COP of the vapor compression air conditioning system with R22 is assumed to be a benchmark because the air conditioner was originally designed for R22. When the mean evaporator temperature in the evaporator increases, the COP also increases remarkably for different ambient conditions. Its variations also depend on ambient temperatures.

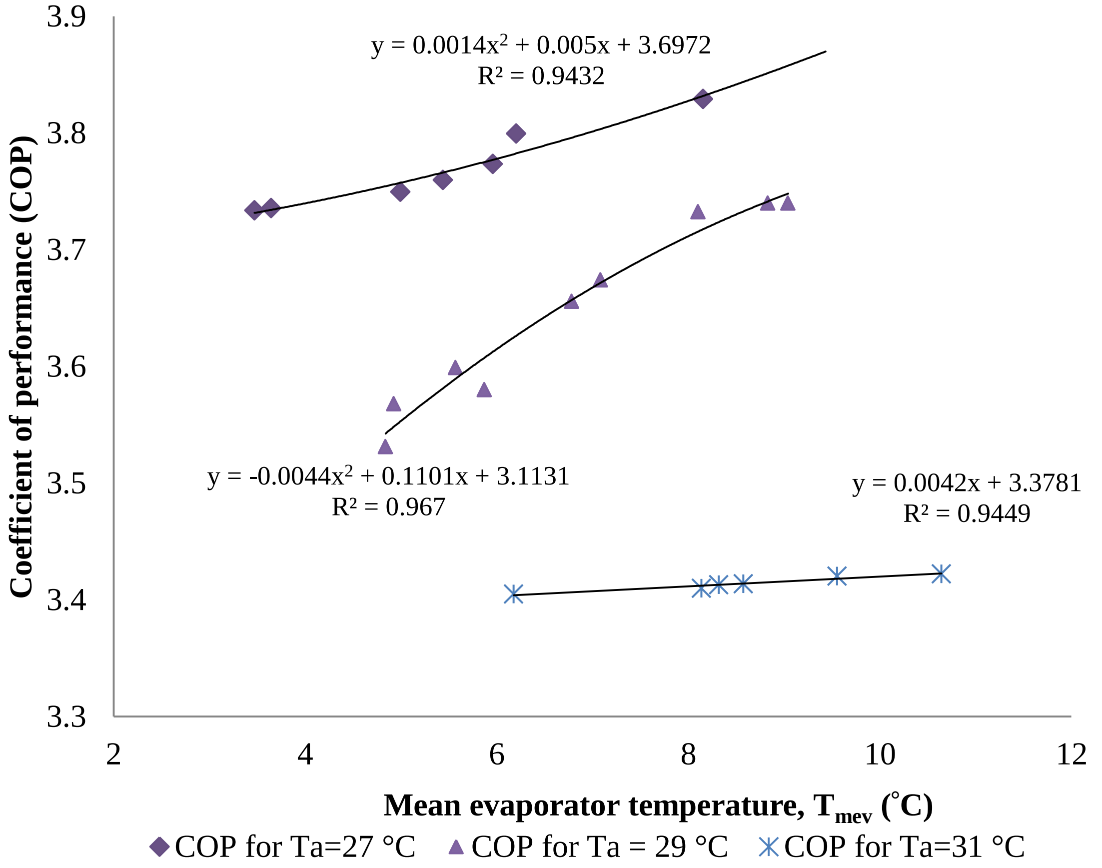


Figure 4.2: Variation of coefficient of performance (COP) with mean evaporator temperature (T_{mev}) at T_a=31°C and Ta = 31°C for refrigerant R22.

Figure 4.2 shows the variations of COP with respect to mean evaporator temperature at ambient temperatures 27° C, 29° C and 31° C. At lower ambient temperatures, COP has higher value for the same evaporator temperature. When the ambient temperature is decreased, the difference between the ambient and evaporator chamber temperature is reduced. Hence, the exergy losses are reduced and thermal performance of the system is increased. Thus, the COP is increased. The COP would be increased when the condenser temperature decreases. Similar trends are found in the studies conducted by Kilicarslan and Hosoz (2010), Sattar (2008), Bayrakci & Ozgur (2009), Radermacher & Jung (1993), Jung, Devotta et al.(1996), Park and Jung (2008), etc.

The clear idea for the variation of COP with mean evaporator temperature and ambient temperature can be found from Figure 4.2. Coefficient of performance at T_a =27°C is 15-20% higher than that at T_a=31 °C for the given mean evaporator temperature. The COP increases at all ambient conditions with the increase of mean evaporator temperature. The increase in the coefficient of performance of the system are varied with different mean evaporator as well as different ambient temperatures. The increments of COP at three ambient temperatures of 27, 29 and 31°C are shown in Figure 4.2 where it can be noted that these increments are not similar for all ambient temperatures. At T_a = 27 °C, the coefficient of performance is higher than of that at T_a=29 °C and T_a= 31° C, respectively.

From Figure 4.2, correlations based on mean evaporator temperature were derived. There are three corelations for COP with mean evaporator temperature for three ambient temperatures. The correlation for COP at ambient temperature T_a= 31°C can be written as

$$COP = 0.004 \times Tmev + 3.375 \tag{4.1}$$

The R^2 value for this equation is found to be 0.9449.

The correlation for coefficient of performance at T_a= 27 °C is shown in Figure 4.2 as

$$COP = 0.0014T_{evp}{}^{2} - .005T_{evp} + 3.6972 \tag{4.2}$$

The R^2 value of this equation is found to be 0.943.

COP at ambient temperature, T_a = 29 °C is found as

$$COP = 0.0044T_{evp}{}^{2} + .01101T_{evp} + 3.31131 \tag{4.3}$$

The R^2 value for the equation is found to be 0.967.

From the Equations (4.1)-(4.3), it is observed that there is a higher constant term at ambient temperature, T_a= 27 °C. It indicates that higher COP is occurred at the ambient temperature, T_a = 27°C compared to the other ambient temperatures. For measuring COP with the help of Equations (4.1)-(4.3), the mean evaporator temperature, T_{evp} will be used as °C.

4.3.2 Effect of hydrocarbon mixture on coefficient of performance

From Figure 4.3, it can be seen that mixture M1 has higher coefficient performance than that of R22. Refrigerating effect of pure R290 is higher than that of R 22. Thus, refrigerating effect of the mixture of pure R290 and R22 (M1) become higher than that of R22. Consequently, the coefficient performance of the mixture (M1) is higher than that of R22 at different evaporator temperatures. The mixture has higher latent heat of vaporization than that of R22. Hence, it absorbed more heat from the controlled room.

Park and Jung (2008) studied that COP had been varied with the types of the refrigerants and their properties. The authors studied about the energy performance of R22, R290 and R1270. Authors found that R290 had the best energy performance.

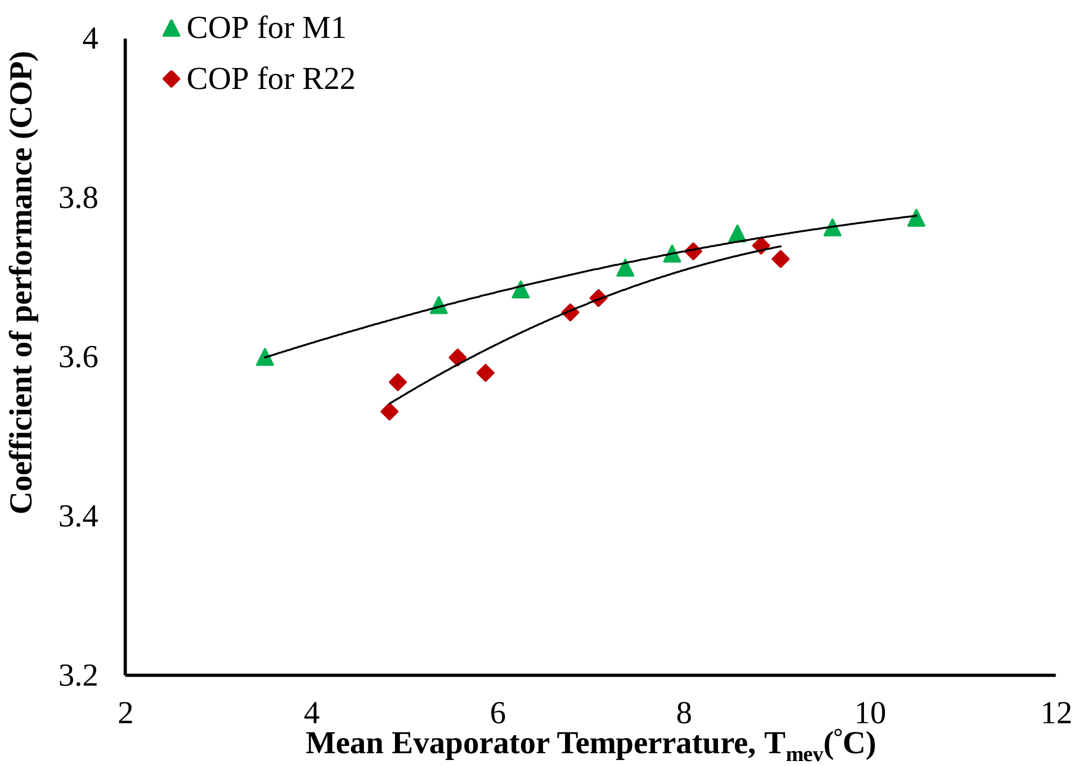


Figure 4. 3: Variation of coefficient of performance on mean evaporator temperature at T_a= 29 °C for M1 and R22 as refrigerant.

Sattar (2008) also found that hydrocarbon (iso- butane) had shown higher COP than that of R134a. Author proposed to use hydrocarbon and their mixtures, like iso-butane (R600a) and mixture of R600 and R600a at some particular ratios for better performance of the domestic refrigerator. Figure 4.4 shows that COPs of the mixtures at all the evaporating temperatures are higher than that of R22 at ambient temperature, T_a= 27°C. It has been found that COP increases with the increase of evaporator temperature because at higher evaporating temperature, the performance of the compressor increases and irreversibility decreases. Thus, the work of compressor decreases as well as the value of seasonal COP increases.

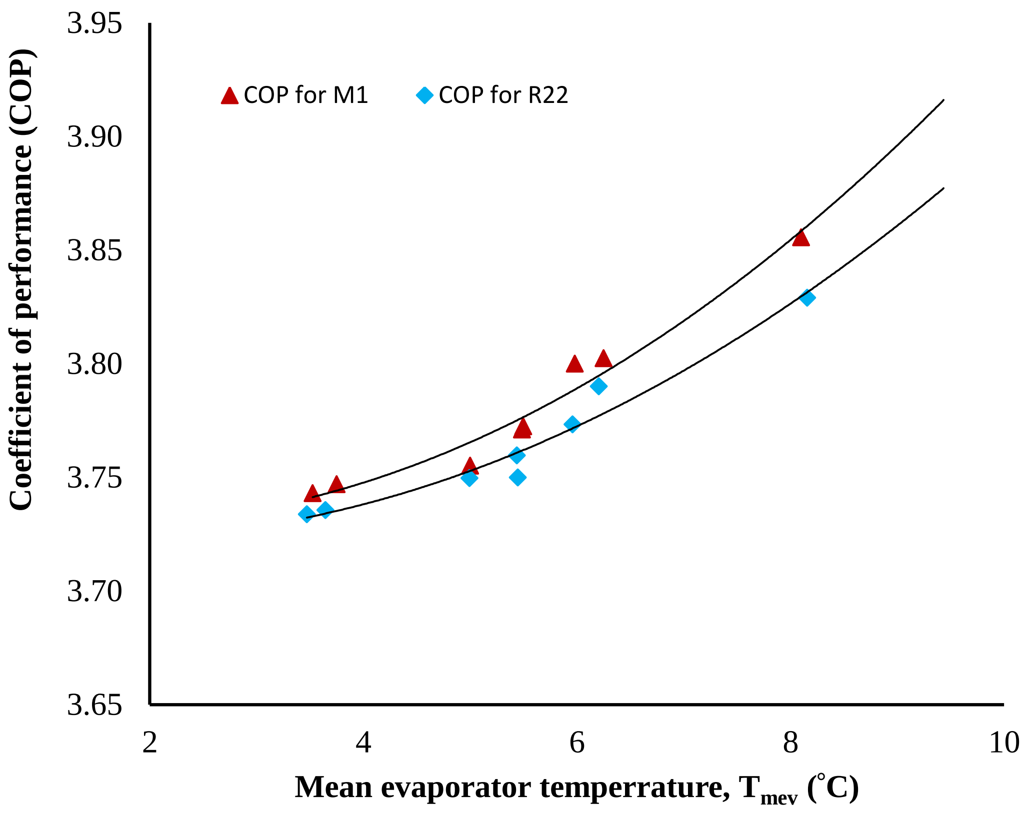


Figure 4.4: Variation of coefficient of performance with different mean evaporator temperaturse at ambient temperature, T_a= 27 °C for M1 and R22.

4.3.3 Effect of condenser temperature on coefficient of performance

It is observed in Figure 4.5 that coefficient performance decreases with the increase of condenser temperature. At higher condenser temperature, the system needs more power to do more work by the compressor. The work of compression has to be increased. Thus, the coefficient of performance is decreased. Similar trends are found from the studies conducted by Kilicarslan and Hosoz (2010), Sattar (2008), Bayrakci & Ozgur (2009), Radermacher & Jung (1993), Jung, et al (1996), Park and Jung (2008), etc.

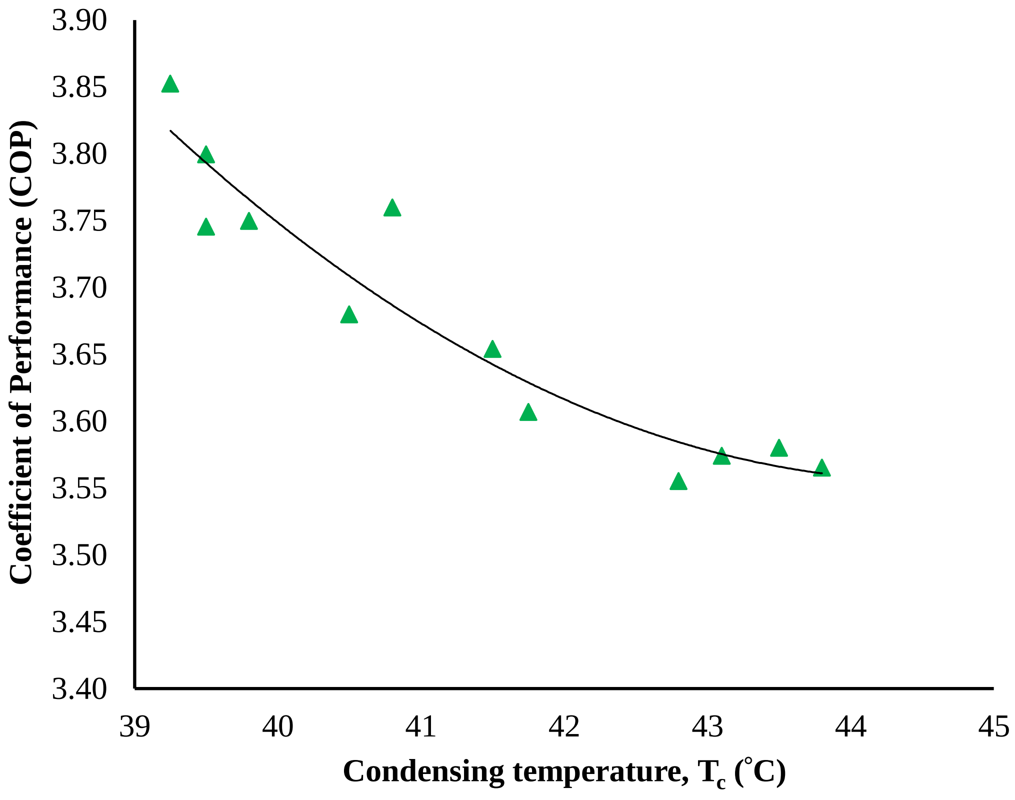


Figure 4.5: Variation of coefficient of performance with the variation of condensing temperature for a given evaporator temperature.

It is found that coefficient of performance decreases with the increase of condenser temperature. The reason is that at higher condenser temperature, outlet temperatures of the compressor are higher, and thus the works of compression are increased. Hence, irreversibility of the compressor is increased and the performance is decreased at higher condenser temperature.

4.3.4 Effect of ambient temperature on the coefficient of performance

Performance of the air conditioning system depends on the ambient temperature of the surrounding. It is found in Figure 4.6 that with the increase in ambient temperature, the coefficient of performance decreases with a very small amount. Though, there is a fluctuation in the deceasing order, it is always decreasing. This fluctuation occurs due to evaporator temperature and condensing temperature variations.

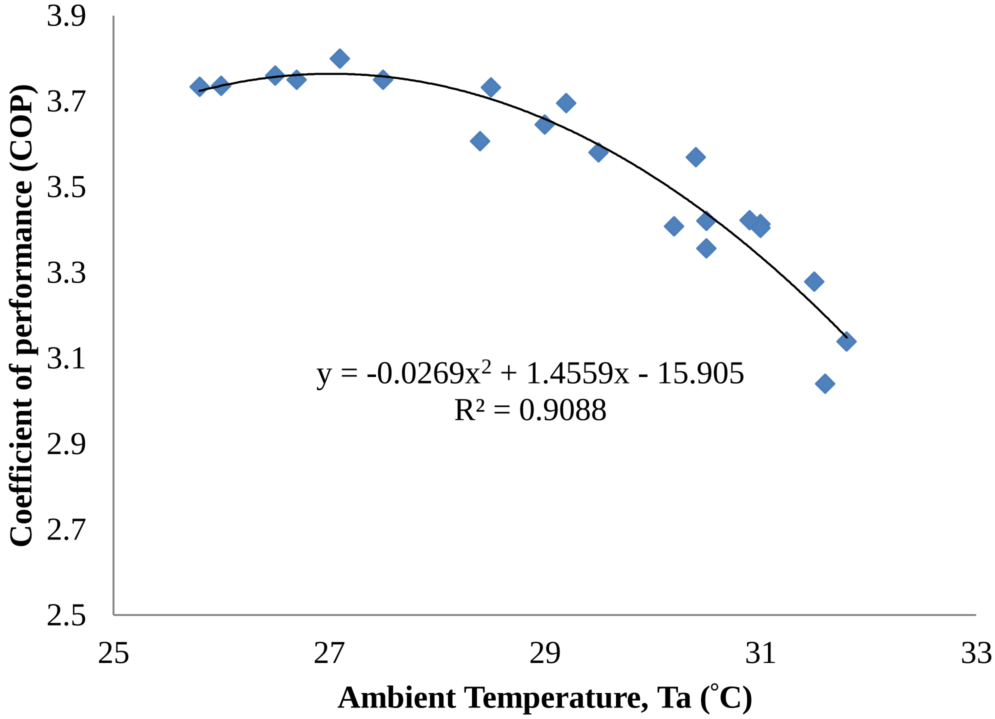


Figure 4.6: Variation of coefficient of performance with ambient temperature for a given range of mean evaporator temperature for R22.

For a constant evaporator temperature, COP decreased with a certain rate with the increase of ambient temperature. If the ambient temperature increases, the difference between the room temperature and ambient temperature will increase. Deviation of the system causes the less performance. Hence, the degradation of the system is increased. Thus the performance of the system is reduced. In some literature, it is observed that the performance of the system is considered in the atmospheric temperature 25 °C. But in this experiment, environmental temperature increased in the day time. In the morning, it was 25.8 °C but in the mid time it was 31.5 °C. Sometimes, it was 32.5 °C. Malaysia is a hot tropical country. It is well known that system performance depends on the temperature variation of any system. When the ambient temperature increases, the variation of room temperature and ambient temperature are become higher. Hence, system performance is degraded. For this experimental work, with a range of mean evaporator temperature ranging from 5.5 °C to 10.0 °C, it is found a correlation of coefficient of performance (COP) with ambient temperature. About 85% data covers this correlation.

4.3.5 Effect of pressure ratio on coefficient of performance

Pressure ratio means the ratio of the outlet pressure of the compressor to the inlet pressure. It is considered that the outlet or discharge pressure of the compressor is equal to the condenser pressure and the inlet pressure of the compressor is taken as equal to the evaporation pressure. Hence, the pressure ratio of any vapor compression is considered as the ratio of condenser pressure to evaporator pressure. Higher pressure ratio indicates that the pressure of compressor is high compared to that in the evaporator chamber.

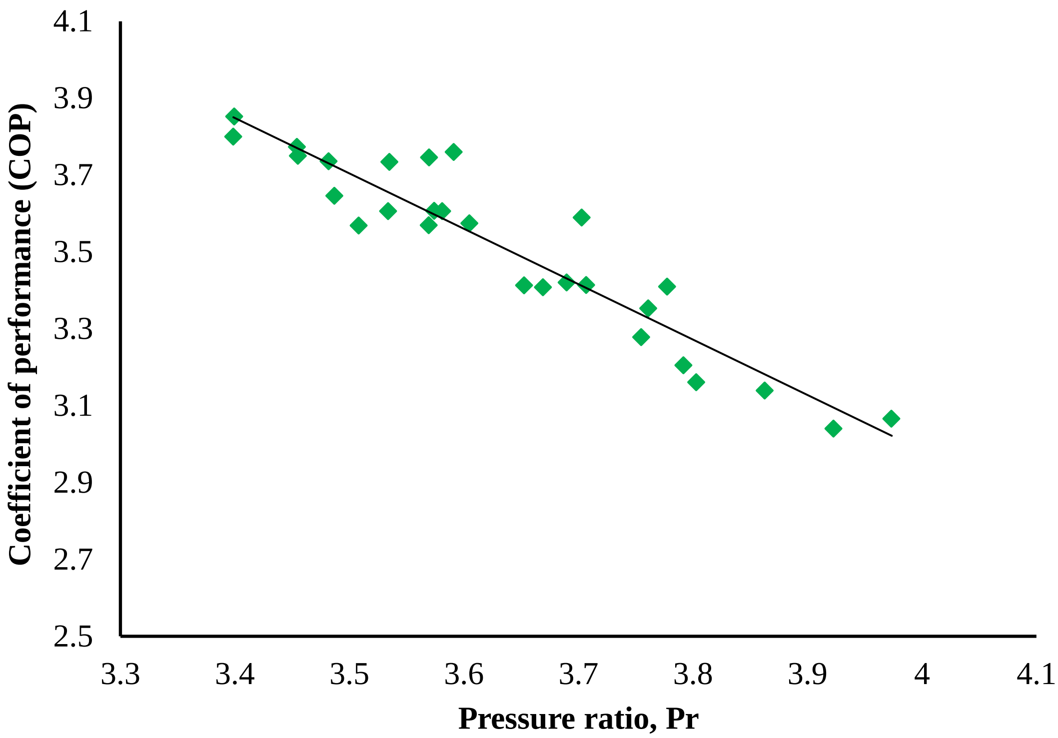


Figure 4.7 Variation of coefficient of performance with pressure ratio (Pr).

Figure 4.7 shows the variation of coefficient of performance with different pressure ratio. Coefficient of performance is decreased with the increase of pressure ratio. Higher pressure ratio means the higher compression work. Hence the performance is decreased.

Jiangtao et al.(2009) calculated that COP of R22 at evaporator pressure 625 kPa was 3.68, COP of R407C at evaporator pressure 649 kPa was 3.55. Some of the companies who are engaged in manufacturing and supplying air conditioner unit found that for practical air-conditioning system, temperature rises from 5 °C to 30°C, the COP will be 6 (AZR, 2011). Park and Jung (2008) found that coefficient of performance of R290 was 3.80 at pressure ratio 2.37 whereas, the COP of R22 was 3.40 at pressure ratio 2.74. The COP of R22 was 2.05 at pressure ratio 4.78 and COP of R290 was found 2.28 at pressure ratio 4.62. The COP was decreasing with the increase of the pressure ratio.

4.3.6 Effect of refrigerants on the energy-efficiency ratio

Figure 4.8 shows that energy efficiency ratio of the refrigerants with different mean evaporator temperatures. This figure shows that EER changes with the changes of refrigerant. Previous study shows that energy efficiency ratio varied with evaporator temperatures. But this figure compares the EER of R22 and the mixture M1 at different evaporator temperatures. The mixture M1 is the mixture of 25% R290 and 75% R22 by mass. This small addition of R290 with R22 causes an increase in EER at all mean evaporator temperatures. R290 has higher cooling capacity as well as higher latent heat of vaporization. Hence, mixture shows higher EER than that of R22. The average increase of EER for using the mixture M1 is about 2.0% compared to that of R22.

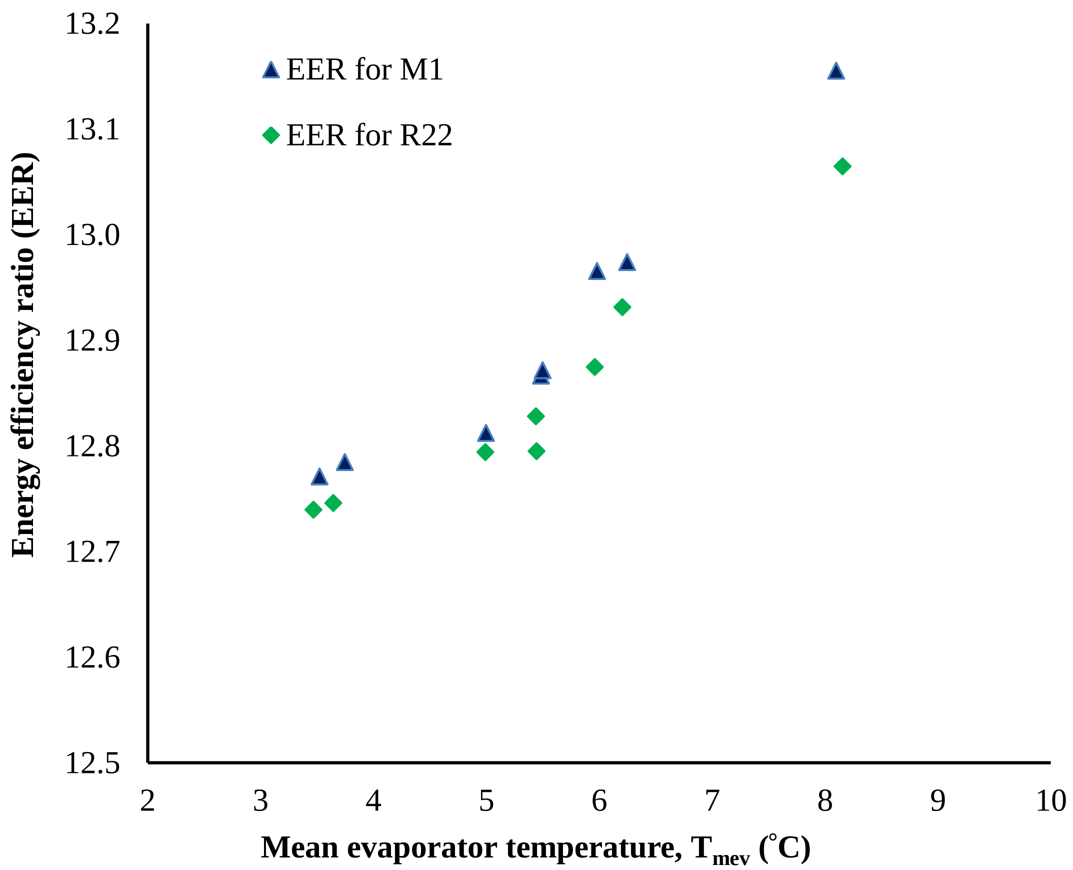


Figure 4.8:Variation of energy effeciency ratio with mean evaporator temperatures

4.3.7 Effect of evaporator temperature on energy efficiency ratio

Figure 4.9 shows the variations of energy efficiency ratio with different mean evaporator temperatures at three ambient conditions. From Figure 4.8, it is clear that like seasonal COP, energy-efficiency- ratio also changes with the changes of evaporator mean temperatures. It also changes with the changes of ambient temperatures. Nowadays, it is recommended that EER should be greater than 10 (Usde, 2011). In this experiment, EER is found higher than 10 at all evaporator temperatures. EER is higher at lower ambient temperatures and it increases with the increase of evaporator temperature. At different time and season, EER is also changed and at that time, it is termed as seasonal energy efficiency ratio (SEER).

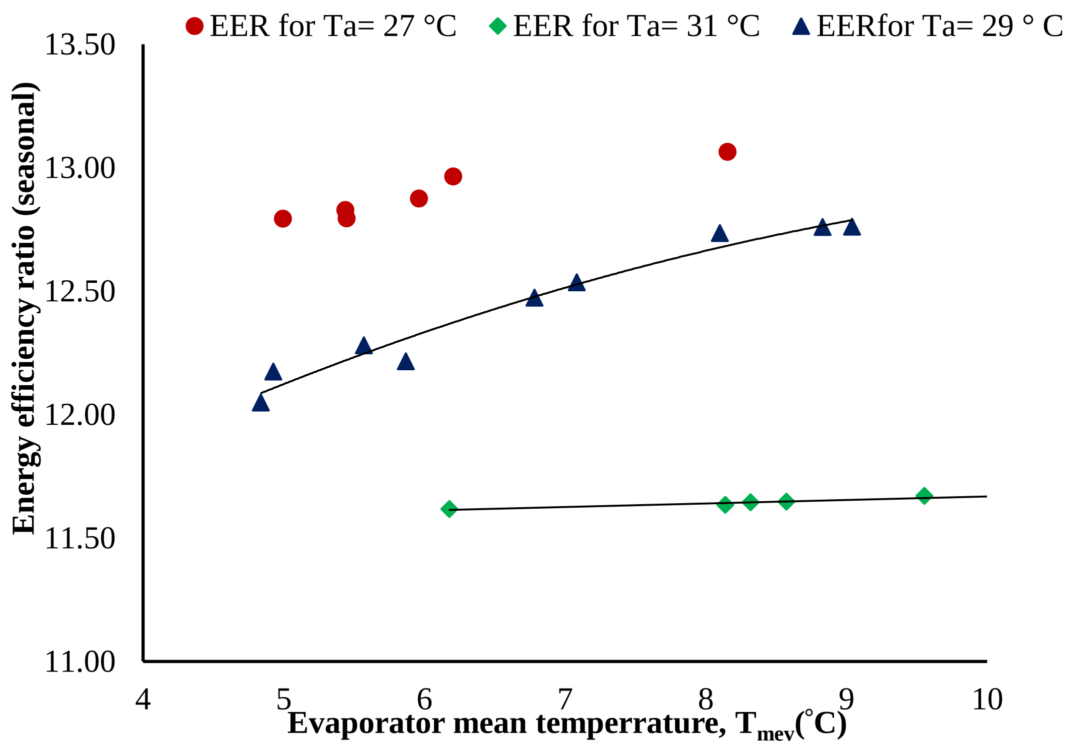


Figure 4.9: Variation of energy efficiency ratio on mean evaporator temperature for R22.

4.3.8 Effect of ambient temperature on energy efficiency ratio

Figure 4.10 shows the variations of energy efficiency ratio at different ambient temperature. Energy efficiency ratio depends on ambient condition. Increase of ambient temperature causes a decrease of thermal performance i.e. energy efficiency ratio. Higher ambient temperature causes higher difference in temperatures of the conditioned room (control room) and the atmosphere. This difference indicates the deviations in the system's performance. Hence, the performance of evaporator is reduced. More work of compressor was necessary to cool the room upto the desired temperatures. Figure 4.10 shows that with the increase of ambient temperature, the EER decreases. But this decrement is very small in quantity. This is also proved in section 4.4 when the correlation is found depending on three operating temperatures such as evaporation temperature, condenser temperature and ambient temperatures.

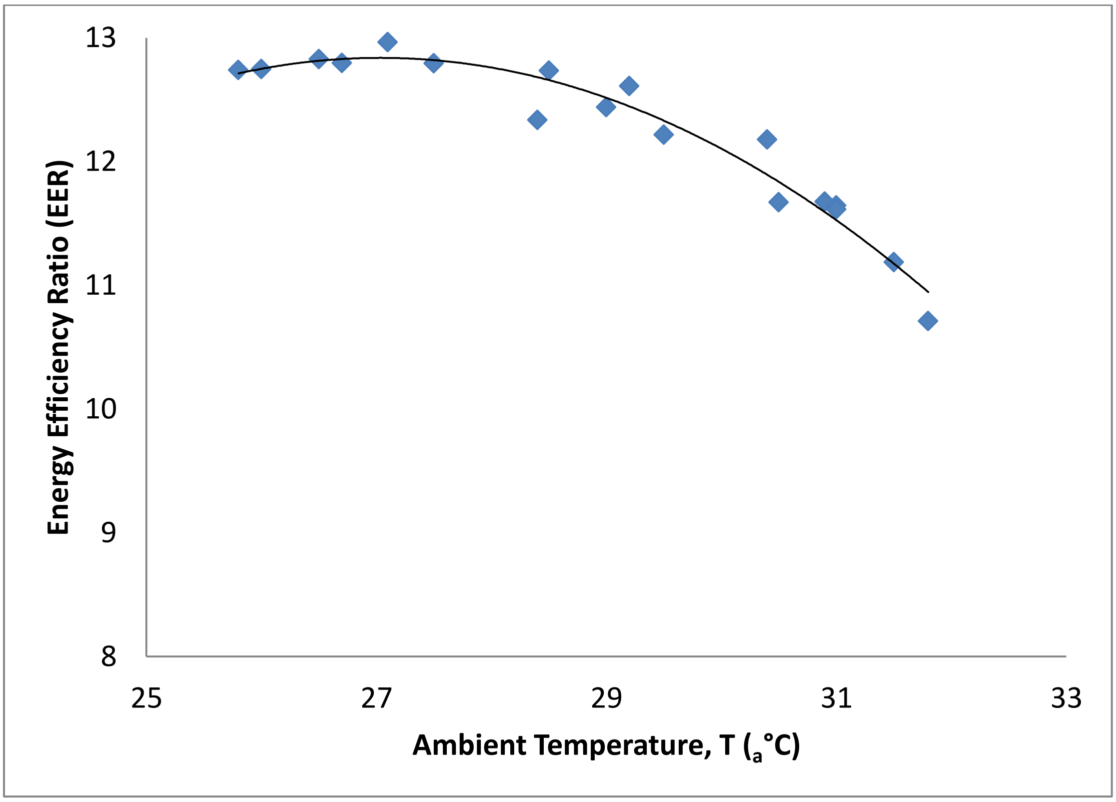


Figure 4.10: Variation of energy efficiency ratio with increase of ambient temperatures.

4.3.9 Effect of condenser temperature on energy efficiency ratio

Figure 4.11 shows the variations of energy efficiency ratio with different condenser temperature. Condenser temperature has a vital effect on energy efficiency ratio. It is shown in Figure 4.11 that energy efficiency ratio decreases with the increases of condenser temperature while the other parameters are remained unchanged or with a small variation. High condenser temperature indicates higher discharge pressure of the compressor. Hence, the compressor work is increased. Thus, the EER is decreased. EER is changing with respect to day time. So, it is called seasonal EER (SEER). Average EER can be obtained for a particular model by analyzing one year data.

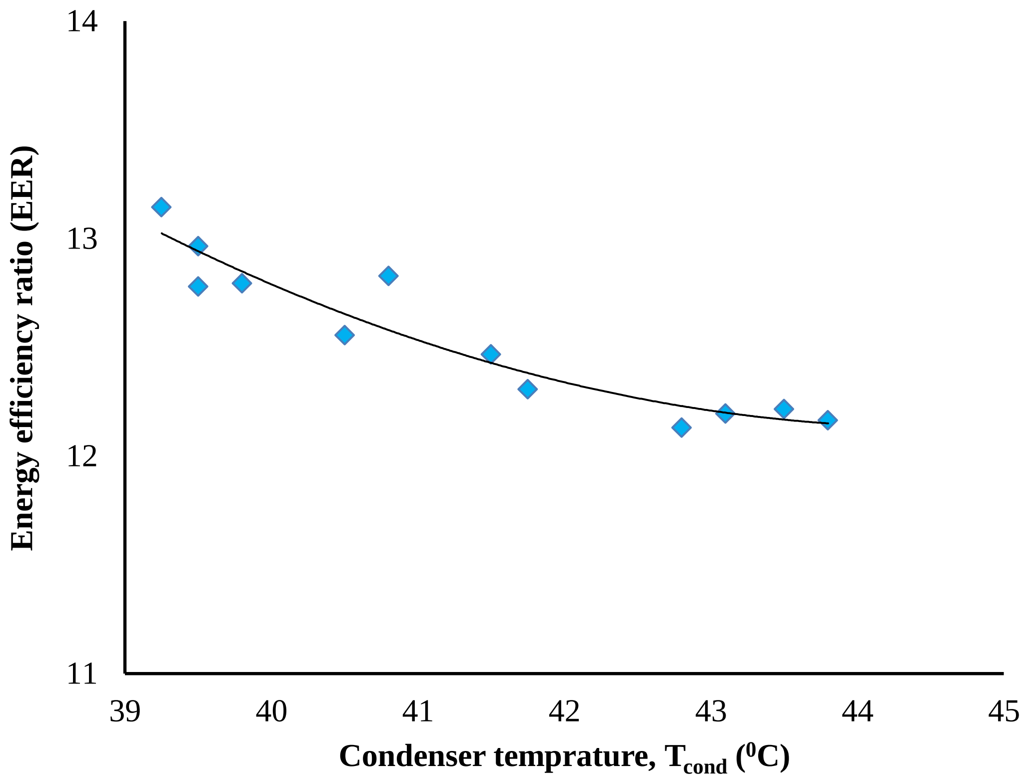


Figure 4.11: Variation of energy efficiency ratio (EER) with increase of condensing temperature.

4.3.10 Effect of evaporator temperatures on refrigerating effect

Figure 4.12 shows the variation of refrigeration effect in the evaporator with different evaporator mean temperatures at different ambient conditions. At higher evaporator temperature, the enthalpy of the refrigerant is higher. So the refrigerating effect in the evaporator is higher. At the same time, it is not possible to maintain constant condenser temperature; so, the trends are not in straight line. At all ambient temperatures, refrigerant effects are increased with the increase of evaporator temperatures. But the increase is not similar for all ambient temperatures. At ambient temperature T_a= 27 °C, the trend is higher compared to that of other ambient conditions. At higher ambient temperature, the system needs more refrigerant to cool at desired temperature. Hence, the capacity of per kg refrigerant is decreased.

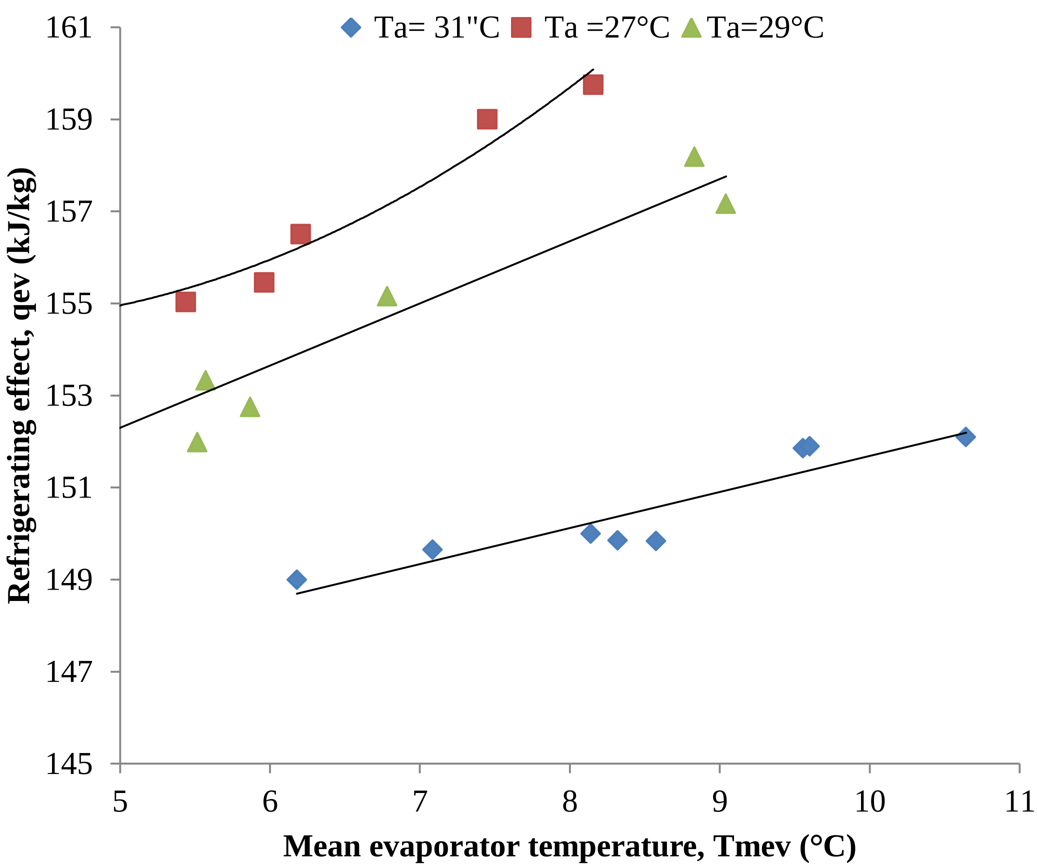


Figure 4.12: Variation of refrigerating effect in the evaporator at different evaporator temperatures for R22 at T_a= 27°C, 29° C and 31°C.

4.3.11 Effect of refrigerants on refrigerating effect

Figure 4.13 shows the variation of refrigerating effect with evaporating for refrigerant R22 and the mixture M1. Refrigerating effect in the evaporator depends not only on evaporator temperature but also on refrigerant types. It is observed that the mixture M1 shows higher refrigerating effect than that of R22 at every evaporating temperature. As the mixture contains higher latent heat of vaporization hence it creates higher refrigerating effect. Cooling capacity also increases with uses of hydrocarbons. Higher refrigerating effect indicates higher cooling capacity of the refrigerant. It also enhances the energy performance of the vapor compression system. Comparing with Figures 4.11 and 4.12, it is observed that at lower ambient temperature, the refrigerating effect is higher.

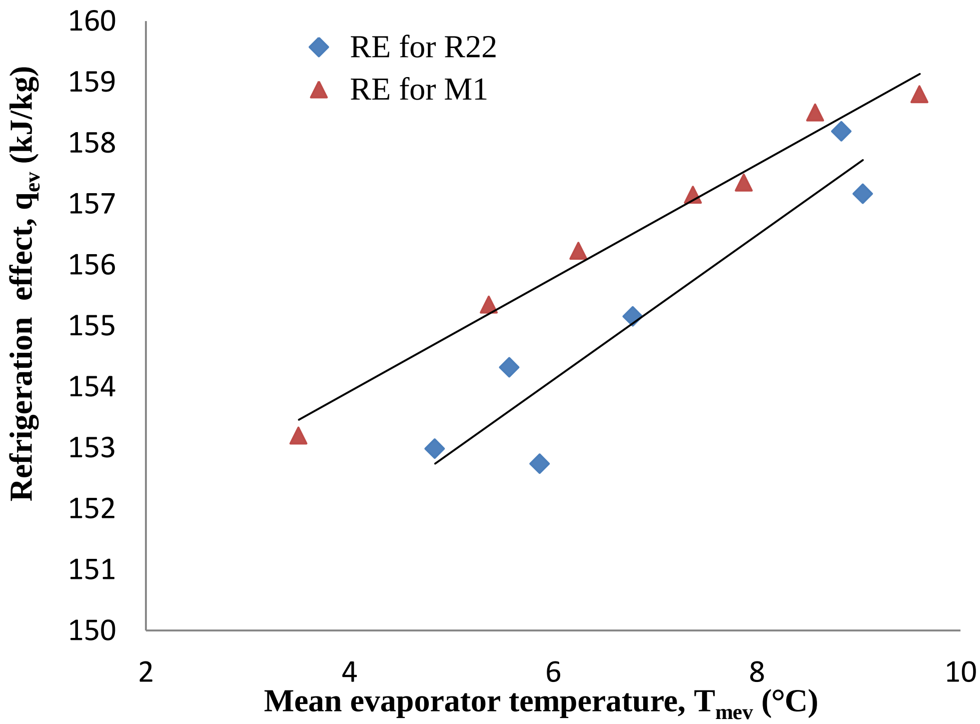


Figure 4.13: Variation in refrigerating effect at different evaporator temperatures for different refrigerants at ambient temperature, T_a= 29 $^{\circ}$C.

4.4 Energy consumption by the compressor at different day time

Figure 4.14 shows the variations of power consumption with different ambient conditions. The data regarding energy consumptions by using data logger and pressure sensors has been described in the experimental facility section. At first, data has been taken for the refrigerant R22. The power consumption and the readings of the thermocouples and pressure transducers are taken within some intervals during day time from 10.00 am to 6.00 pm and stored in the computer with the help of data logger. Power consumption is varied with day time because the outside or ambient temperature is increased with time up to noon. In the morning it is lower and in the afternoon, the atmospheric temperature was decreased. So, power consumption as well as ambient temperature is increased after 9.00 am to 1.30 pm. After that, the ambient temperature and power consumption is decreased. Hence, the power consumption is decreased which is shown in Figure. 4.14. So, it is clear that the power consumption depends on atmospheric temperature or dead state temperature. In this experiment, dead state or ambient temperature is not controlled manually but it is changing with nature. Figure 4.14, shows that power consumption depends on ambient or dead state temperature. If reduced. Total power consumption for the period 10.00 am to 6.00 pm (8 hours) in day is around 7.15 to 7.74 kWh. It also varies for refrigerants types and their charges. May be the power consumption for R134a will be minimum compared to the HC blends. The power consumption also can be plotted in graph for different refrigerants.

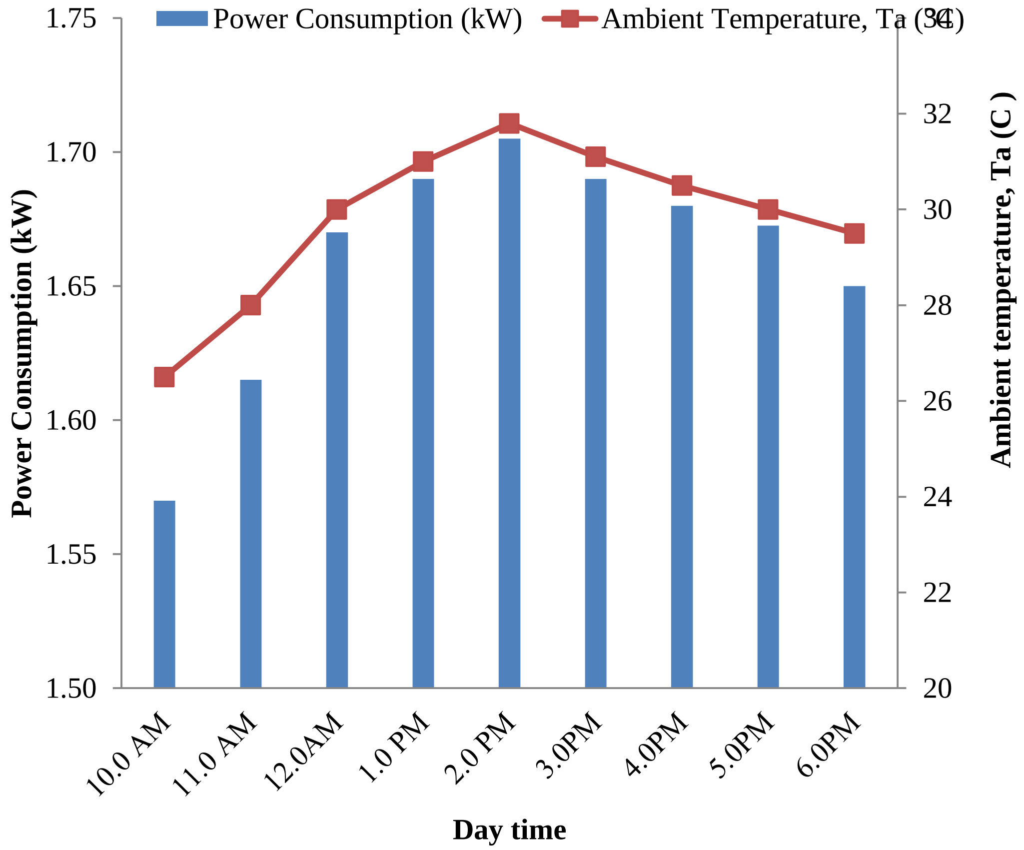


Figure 4.14:Power consumption at day time for R22.

4.4.1 Variations of power consumptions with pressure ratio

Figure 4.15 shows the variation of power consumption with the variation of pressure ratio. Power consumption by the compressor depends on the pressure at the outlet of the compressor. Pressure ratio means the ratio of outlet pressure to inlet pressure of the compressor. It is shown in the Figure 4.15 that with the increase of the pressure ratio (P_{co}/P_{ci}), the power consumption is also increased. Some of the data are fluctuating. It indicates that the variations in the ambient temperatures and evaporator temperatures have some effects. High pressure ratio means higher discharge pressure corresponding to the inlet pressure which indicates that compressor has to perform more work.

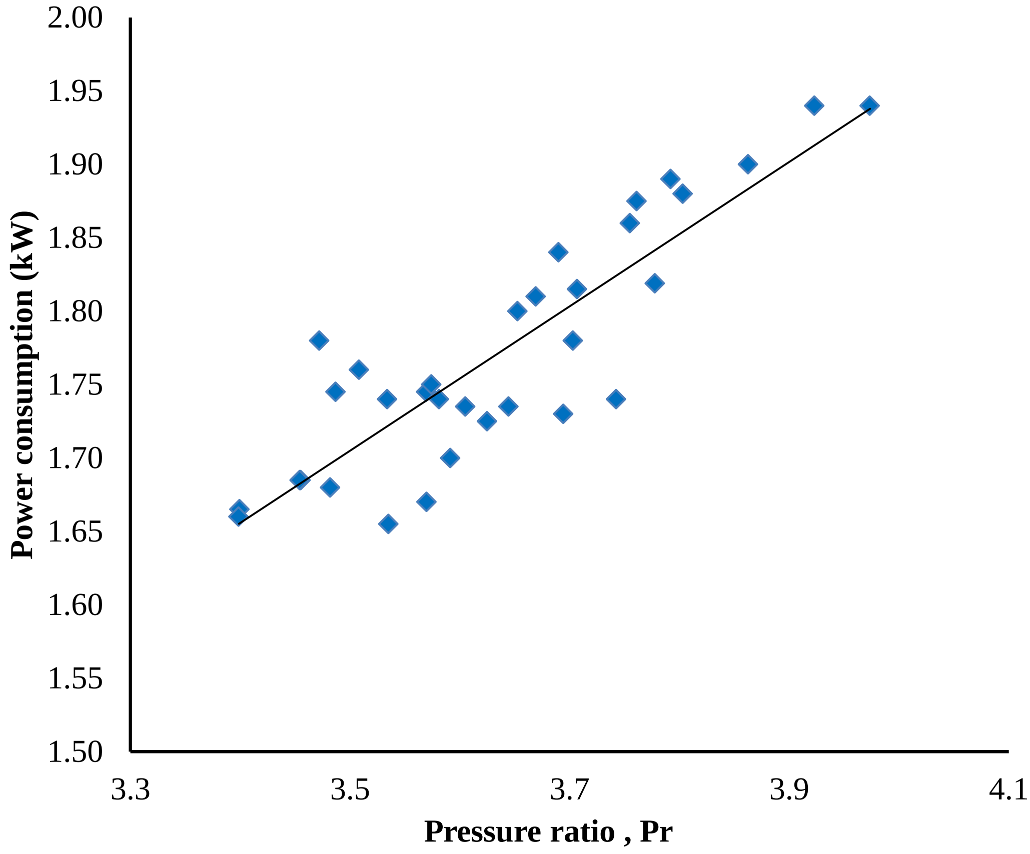


Figure 4.15:Variation of power consumption with pressure ratio (Pr) for refrigerant R22.

Total energy consumed by the compressor while using R22 and mixture (M1) is measured by multiplying the power with the compressor running time. It is shown in the Table 4.1 that power consumption per day reduces from 7.50 kWh to 7.10 kWh while using the mixture M1 instead of R22. The performance based on power consumption of M1 is better than that of R22. Latent heat of vaporization of the mixture M1 is higher than that of R22. Thus, the mass flow required to cool at a given room condition is reduced for the mixture.

Table 4.1: Energy consumption of different refrigerants used in the experiment during the working hours (from 10.00 am to 6.00 pm).

Refrigerant	Energy Consumption in kWh/day (from10.00 am- 6.00 pm)	Work of Compression (kW)	Energy Consumption decreased (%)
R22	7.50-7.80	1.61-1.76	-
M1	7.10-7.40	1.57-1.60	4.5-5.5

4.4.2 Variation of work of compression with ambient temperature

Normally, in Malaysia, it is summer always, but the ambient temperature is not always remains constant. In a particular day, also the ambient temperature is changing with sunlight. For this reason, power consumption by the compressor in the air conditioner unit is changing. Figure 4.16 shows that work of compression by the compressor depends on ambient temperature. Work of compression increases with the increase of ambient temperature. This is occurred because the difference between the controlled room temperature and the ambient temperature is increased.

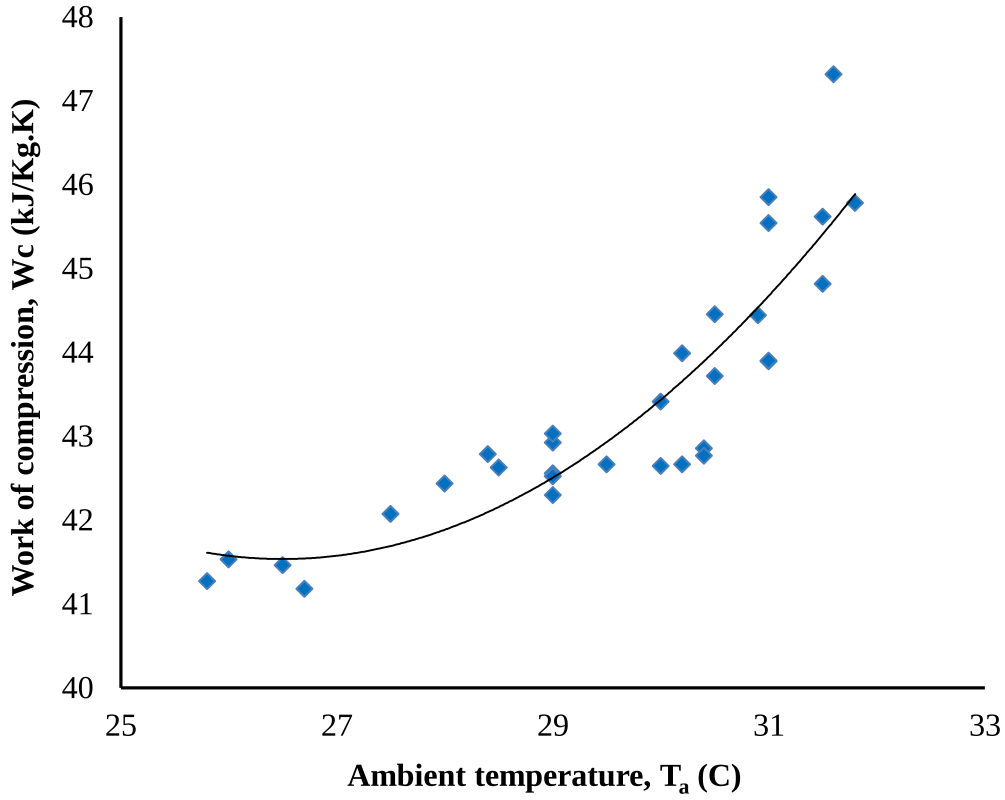


Figure 4.16: Variation of work compression with different ambient temperature, T_a ($^{\circ}C$).

Thus, the necessary work done by the compressor is increased to cool the controlled room. However, in the low ambient temperature ranges, its effect is slow but in the higher ambient temperatures, its effect is high. At the mid-day, the ambient/atmospheric temperature becomes higher compared to that of at morning and afternoon. In order to maintain a constant room temperature, it is required to supply higher work by the compressor because cooling load and the compressor work are increased.

4.4.3 Variation in work of compression with condensing temperature

It is observed in the Figure 4.17 that work of compression increases with the increase of condenser temperature. Higher condenser temperature indicates higher outlet pressure of the compressor. From Figures 4.17 and 4.18, it is concluded that work of compression has strong effect on condenser temperature when the ambient temperature is not considered. For a given ambient temperature, work of compression has less effect on the variation of condenser temperature. At the ambient temperature 29 °C, work of compression increases up to 3% for condenser temperature from 42.5 to 44.5 °C.

In the Figure 4.18, it is observed that work of compression incresaed upto 17% for the ranges of condenser temperature from 37 to 51°C. This is not only due to condenser temperature but also evaporator and ambient temperature changes. For large variation of condenser temperartture, it is observed that work of compression increased highly where the ambient temperature varied from 26.5 to 31.8 °C shown in Figure 4.18.

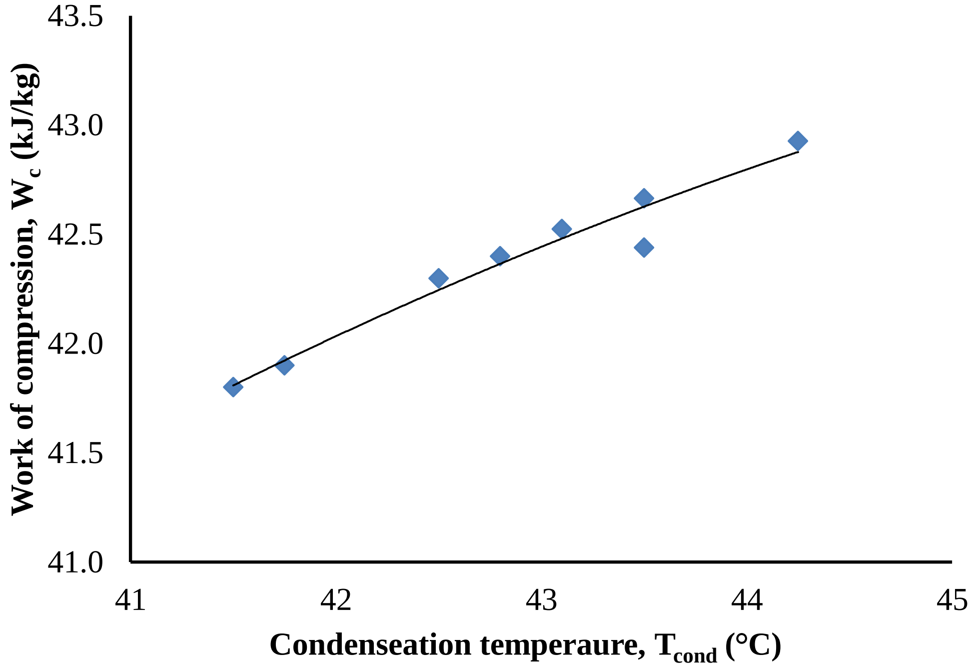


Figure 4.17:Variation of work of compression with condensing temperature, T_{cond} (°C) at ambient temperature, T_a= 29 °C.

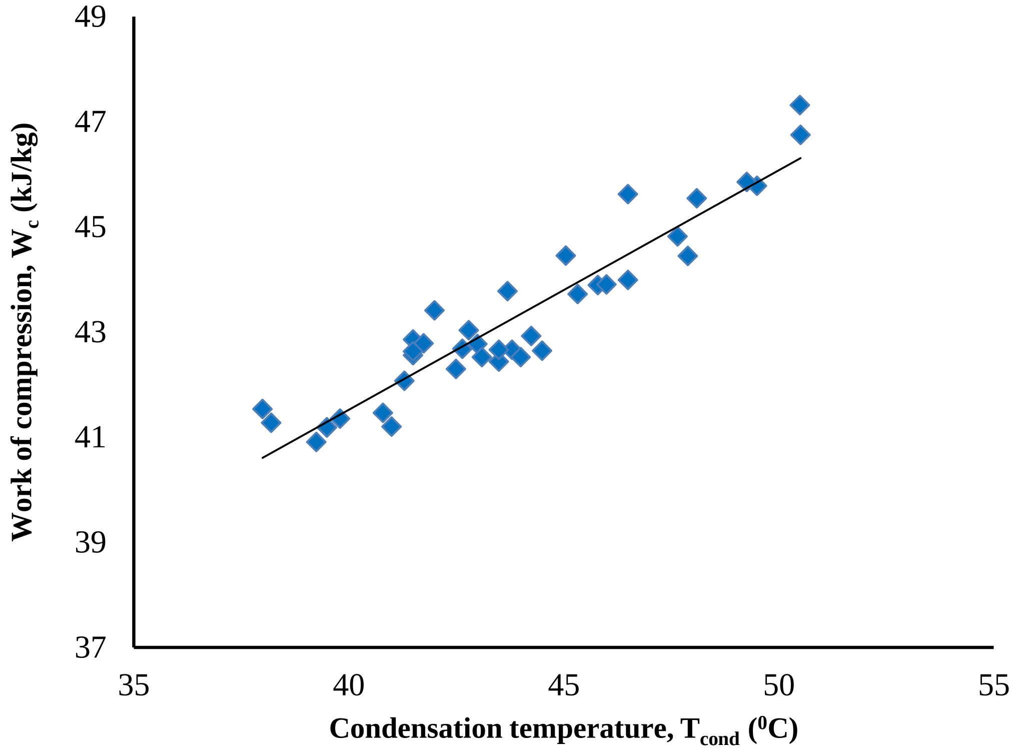


Figure 4.18:Variation of work of compression with the variation of condensing temperature, T_{cond} ($^{\circ}$C) for ambient temperatures range (26.5-31.8 $^{\circ}$C).

4.4.4 Variation of work of compression with mean evaporator temperature

Figure 4.19 shows the variations in work of compression with different mean evaporating temperature at two distinct ambient conditions. Work of compression varies with the variation of mean evaporator temperature. It is shown in Figure 4.19 that work of compression has the inverse relation with mean evaporator temperature. It decreases with the increase of mean evaporator temperature.

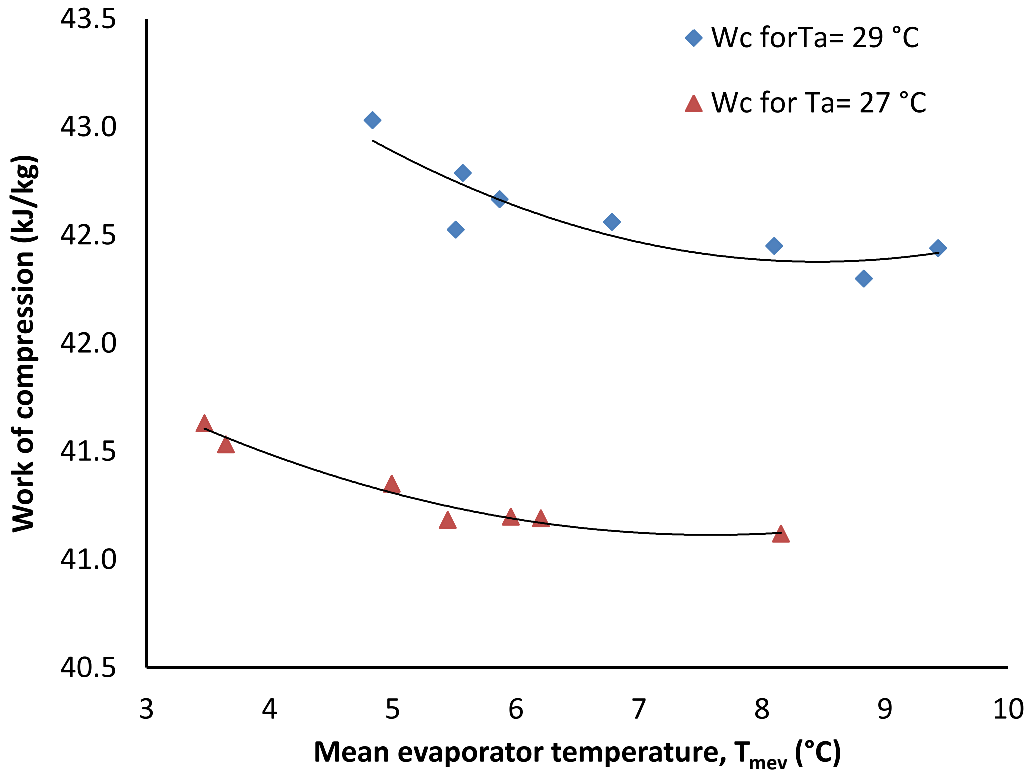


Figure 4.19:Variation of work of compression with mean evaporating temperature, T_{mev} (°C) at ambient temperature, T_a = 27 and 29 °C.

Work of compression is found higher at higher ambient temperature. Work of compression at ambient temperature 29°C is 4-4.5 % higher than that of at 27 °C. Figure 4.19 shows the variation of work of compression with different evaporator temperature at ambient temperatures 27°C and 29°C, respectively. Here, also work of compression decreases with the increase of mean evaporator temperature. When the evaporator temperature increases, the difference between the room temperature and ambient temperature also decreases. This indicates less cooling is necessary. Hence less work of compression is required.

4.4.5 Variation of power consumption with pressure ratio of the compressor

Figure 4.20 shows the variation of power consumption with pressure ratio changes. It is shown in Figure 4.20 that power consumption by the compressor varies with the variation in pressure ratio in the compressor. Pressure ratio is the ratio of outlet pressure of the compressor to inlet pressure of the compressor. The higher value of pressure ratio (Pr) indicates that the refrigerant is compressed to a higher pressure compared to the inlet pressure.

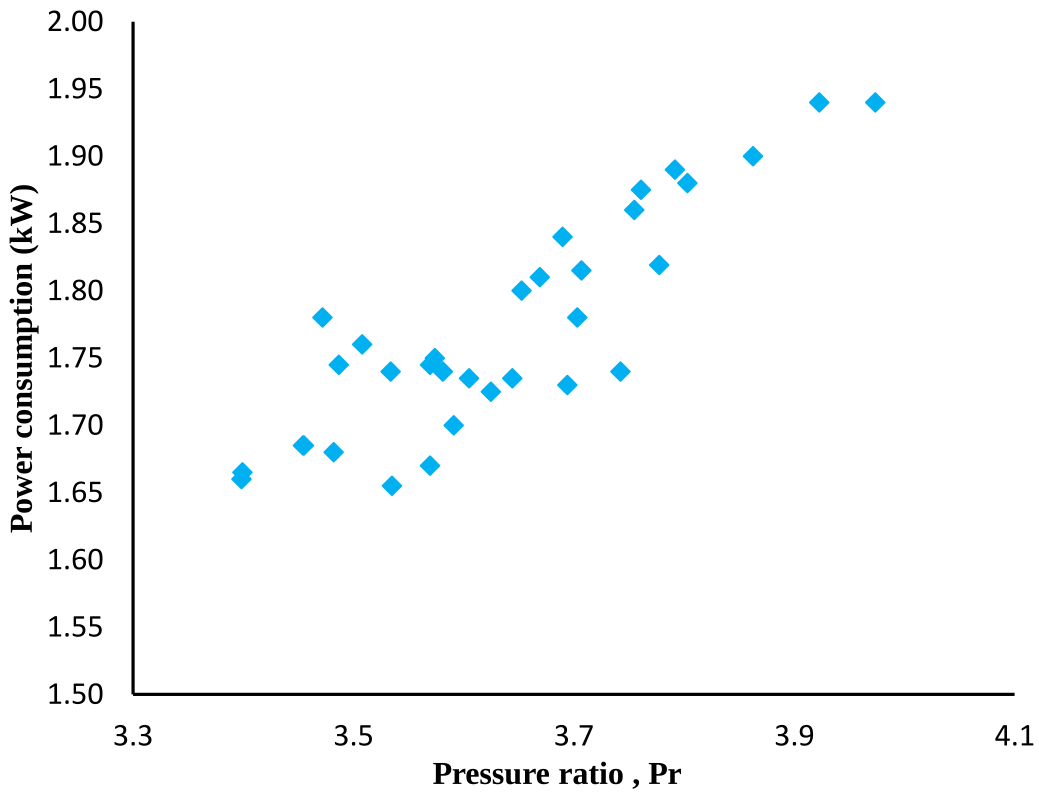


Figure 4.20:Variation of power consumption with pressure ratio of the compressor.

Thus the required/energy consumption is higher. It also varies with the environmental temperature. Thus the trend of the power consumption is not following the straight line. There is a small variation in power consumption with ambient temperature also. Performance of the system is degraded with ambient temperature. Hence, the power requirement is increased.

4.5 Correlations developed for coefficient of performance

In this section, a correlation relating to coefficient of performance with the operating temperatures is developed using the statistical software SPSS. From Figures 4.2 to 4.5, it is observed that coefficient of performances of the vapor compression system are changed with the change of mean evaporator temperature, condenser temperature and ambient temperature for given refrigerants. From the Figures 4.2-4.5, it is found a relation to the variation of COP with one variable while other two variables are kept constant. But in actual situation during operation, ambient temperature, condenser temperature and mean evaporator temperatures could not be fixed. These variables will be changed at the same time. So, it is applicable to correlate the COP with three variables instead of single variable. Consider condenser temperature, evaporator temperature and ambient temperature as independent variables and coefficient of performance as dependent variables. Using SPSS software, from the regression analysis, Table 4.2 has been obtained.

Table 4. 2: Regression results considering three variables for R22

(1)Model Summary[b]

Model	R	R^2	Adjusted R^2	Std. Error of the Estimate	Change Statistics				
					R^2 Change	F Change	Degrees of freedom, df1	Degrees of freedom, df2	Significance Change
1	.933[a]	0.87	0.859	.0856	0.87	72.03	3	32	0.00

a. Predictors: (Constant), T_{cond}, T_{mev}, T_a

b. Dependent Variable: COP

(2)ANOVA[b]

Model	Sum of Squares	df	Mean Square	F	Significance
Regression	1.596	3	0.532	72.03	0.00[a]
Residual	0.236	32	0.007		
Total	1.833	35			

a. Predictors: (Constant), T_{cond}, T_{mev}, T_a

b. Dependent variable: COP

(3) Coefficients

Model	Unstandardized Coefficients		Standardized Coefficients	t	Significance
	B	Std. Error	Beta		
Constant	6.929	0.303		22.885	0.00
T_a	-0.014	0.016	-0.110	-.873	0.39
T_{mev}	0.052	0.013	0.437	3.943	0.00
T_{cond}	-0.077	0.010	-1.161	-7.625	0.00

a. Dependent Variable: COP

From Tables 4.2 (1)-(3), independent variables are T_a, T_{cond} and T_{mev}. The dependent variable is COP. R-square value is 0.871 and it is significant for range of degrees of freedom from 3 to 32. ANOVA result shows that this result is significant. Table 4.2 (3) shows that the terms constant, T_{mev} and T_{cond} are very significant (indicates zero significance). But the variable T_a is not much significant. This means that changes in T_{mev} and T_{cond} will cause a great change in COP but changes in T_a will not effect on COP. This is also proved from the Figures 4.2 to 4.5. Variation of ambient temperature cause less effect in COP changes. But changes in evaporator temperature and condenser temperature have large effects. From this table, a general correlation of COP with other variables is found as follows:

$$COP = 6.929 - 0.014T_a + 0.052T_{mev} - 0.077T_{cond} \qquad (4.4)$$

Here this correlation is valid for mean evaporator temperature ranges from 2.5 to 10.5 °C, ambient temperature ranges from 25.2 to 31.8 °C. Condensing temperatures ranges from 40.0 °C to 50.0 °C. The unit of temperatures is considered as °C.

Table 4. 3: Regression results considering three variables for M1.

(1) Model Summary

Model	R	R^2	Adjusted R^2	Std. Error of the estimate
1	0.878[a]	0.77	0.71	0.034

a. Predictors: (Constant), Tcond, Tmev, Ta

(2) ANOVA[b]

Model	Sum of squares	Degrees of freedom, df	Mean Square	F	Significant
Regression	0.046	3	0.015	13.447	.00[a]
Residual	0.014	12	0.001		
Total	0.060	15			

a. Predictors: (Constant), T_{cond}, T_{mev}, T_a

b. Dependent variable: COP

(3) Coefficients[a]

Model	Unstandardized coefficients		Standardized coefficients	t	Significance
	B	Std. Error	Beta		
Constant	4.542	0.200		22.71	0.000
T_a	-0.007	0.009	-0.155	-.75	0.466
T_{mev}	0.035	0.006	1.111	6.22	0.000
T_{cond}	-0.020	0.006	-0.748	-3.40	0.005

a. Dependent variable: COP

From Table 4.3 (1)-(3), independent variables are T_a, T_{cond} and T_{mev}. The dependent variable is COP. R-square value is 0.771 and it is significant for range of degrees of freedom from 3 to 12. ANOVA result shows that this result is significant (0).Table 4.3 (3) shows that the terms constant, T_{mev} and T_{cond} are very significant (indicates 0 or less significance). But the variable T_a is not much significant. This means that changes in T_{mev} and T_{cond} will cause a great change in COP but changes in T_a will not effect on COP. This is also proved from the Figures 4.2 to 4.5. Variations in ambient temperatures cause less effect in the changes of COP. But changes in evaporator temperatures and condenser temperatures have higher effects. From these Tables, the general correlation for COP for M1 with other variables will be as follows:

$$COP = 4.542 - 0.007T_a + 0.035T_{mev} - 0.02T_{cond} \qquad (4.5)$$

Here, this correlation is valid for mean evaporator temperature ranges from 3.5 to 9.60°C, ambient temperature ranges from 25.5 to 30.30°C. Condensing temperatures ranges from 37.80°C to 46.25°C.

Payne and Domanski (2011) also developed a correlation for COP with operating temperatures like outdoor temperature. They also found that the COP deceased with the increase of outdoor temperature.

4.6 Parametric influence on exergy analysis

4.6.1 Effect of evaporator temperature on exergy losses at different ambient temperatures

Figure 4.21 shows the effect of mean evaporator temperature on total exergy losses in the system at two distinct ambient temperatures. Exergy losses or destruction depends on evaporator temperature. In this figure, it is observed that exergy losses per kg of refrigerant flow are increased with the increase of evaporator temperature. Here, the evaporator temperature is taken as the mean evaporator temperatures.

Entrance temperature of the evaporator is higher than the outlet temperature of the evaporator because the pressure is lost during the flow through the tubes in the evaporator. When the evaporator temperature is increased, the outlet pressure required for the compressor is increased and hence, the outlet temperature of the compressor is also increased. Thus, the exergy losses in the compressor are increased. So, total irreversibility or exergy losses are increased with the increase of evaporator temperature.

Exergy losses at ambient temperatures 31 ˚C and 27 ˚C for different evaporative temperatures are investigated in the same figure. It is observed that at both ambient temperatures, the exergy loses are decreased with the increase of mean evaporator temperature. It is also observed that exergy losses are lower at T_a =31°C than that of at T_a= 27°C for all the evaporator temperatures. At higher evaporator temperatures, the deviation of the room temperature from the atmospheric/ambient temperature is reduced. Hence exergy destructions are reduced. Irreversibility of system decreases as the deviation of any system decreases from the atmospheric condition. Due to decrease of irreversibility exergy losses are also decreased. Hence the performance of the system is increased.

In Figure 4.21, the variations in exergy losses with different evaporator temperature for R22 and M1 at ambient temperature 27 $^{\circ}$C is observed. Exergy losses are reduced 4-6% for the M1 compared to R22. Figure 4.21 shows the variation in exergy of the two refrigerants at different evaporator temperatures. For both the refrigerants, the exergy losses are reduced with the increase of evaporator temperature. But the decreases of exergy losses are not similar for both ambient temperatures. Figure 4.22 shows that decreases of exergy losses are higher in higher atmospheric temperature. This means that at T_a= 31 ˚C , the vapor compression system shows better performance than T_a= 27 ˚C .

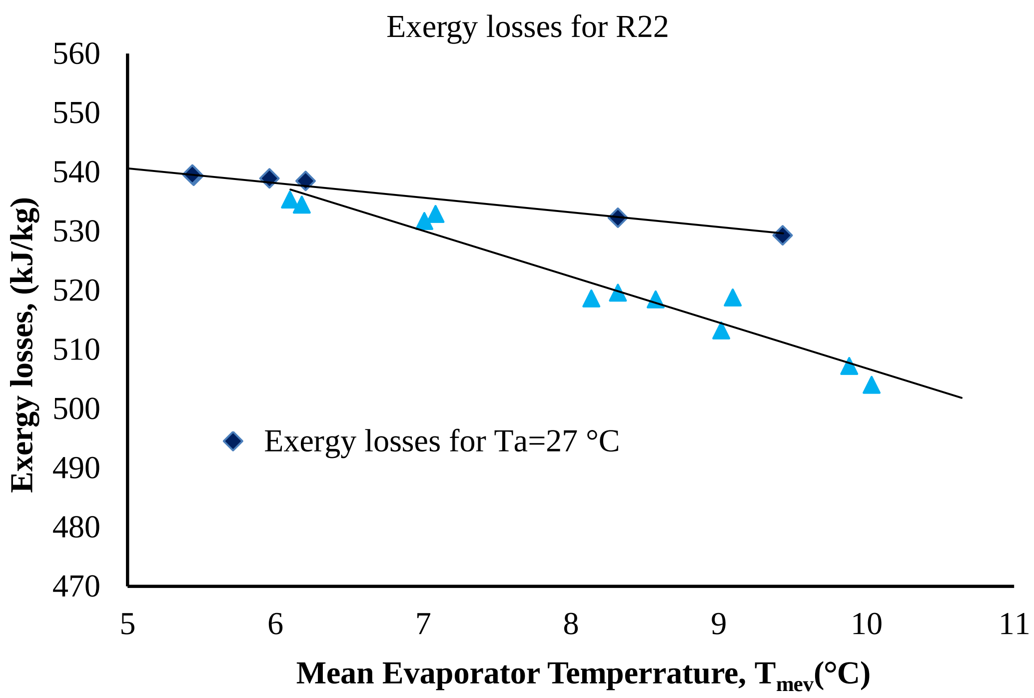


Figure 4.21: Variation of exergy losses of the refrigerants with evaporator temperatures at ambient temperature, T_a= 27 $^{\circ}$C and T_a = 31°C.

The nature of the exergy loss in the system is decreasing with the evaporating temperature and it can be explained by Yumrutas et al. (2002). The main cause is that the average temperature difference between the evaporator chamber and the control room decreases with increasing evaporating temperature. The higher the temperature difference the higher the exergy loss. Hence, the total exergy losses with the increase of evaporator temperature are decreased.

4.6.2. Effect of refrigerants on exergy losses with different evaporator temperature

Figure 4.22 describes the variations in exergy losses with different mean evaporating temperature for different refrigerants. Exergy losses for different evaporating temperature for both the refrigerants at ambient temepature T_a = 29°C are described in the Figure 4.22. For the refrigerant R22, the exergy losses are higher at every evaporating temperature than that of the refrigerant M1. The trend also shows that the exergy losses are decreased with the increase of evapoarator temeprature. This figure shows that M1 is more efficient than that of R22 based on second law analysis. Thermodynamic performance of M1 is also higher than that of R22. It also indicates that the evaporator temperature has a great effect on exergy losses for both refrigerants. Exergy losses have an effect on ambient temperature also.

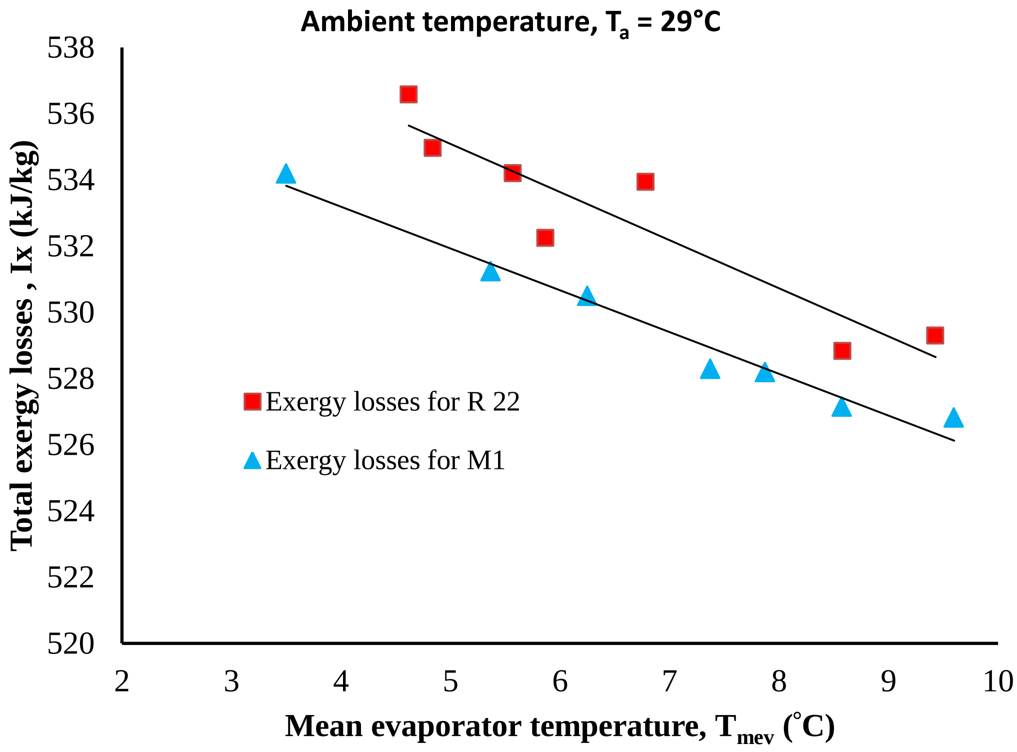


Figure 4.22: Variation of exergy losses of the refrigerants with evaporator temperatures.

Figure 4.23 shows the variation in total exergy losses at ambient temperature T_a= 27°C. Exergy loss decreases with the increase of mean evaporator temperature. The decreasing trend of exergy loss is similar to that of Figure 4.22. But it is also clear from the Figures 4.22 and 4.23 that exergy losses at ambient temperature 27°C are higher than that of at 29°C.

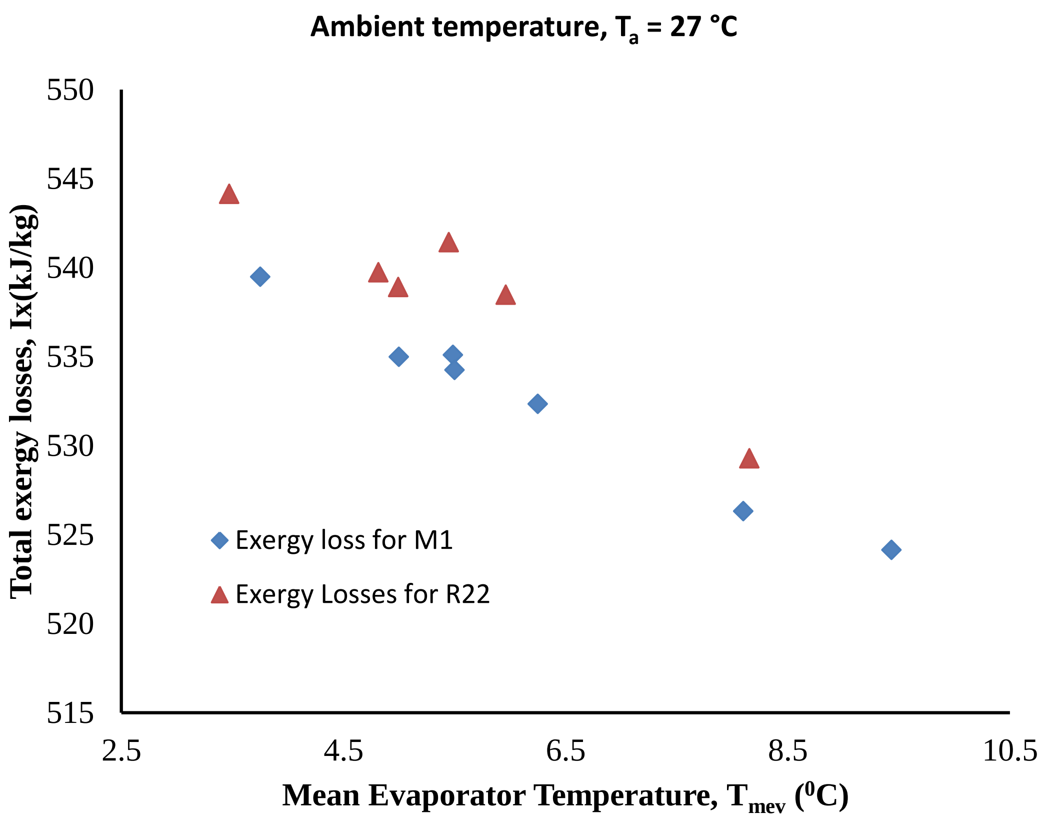


Figure 4. 23: Variation of total exergy losses of the refrigerants with evaporator temperatures.

4.6.3 Effect of condenser temperature on exergy losses

Figure 4.24 shows that exergy losses increase with the increase of condensing temperature for a given evaporator and ambient temperature. Total irreversibility rate is increased with the increase of condensing temperature for any kind of refrigerant. When the temperature difference between the ambient and respective component become higher, the exergy losses is higher i.e. availability of work is reduced. Chances or possibility of irreversibility increases. Figure 4.26 shows that the total exergy losses for refrigerant R22 and M1 at different condenser temperature. But the ambient temperature is remained within the range of 27.0 °C to 28.5°C and the evaporator temperature is remained within the range of 4.5 °C to 8.5°C. The variation of the evaporator and ambient temperatures is very small. So, the variations in exergy losses shown in Figure

4.26 are only depended on condensing temperatures. Bayrakci and Ozgur (2009) found also similar results for R134a, R600a, R600, R22, R290 and R1270. Kalaiselvam and Saravanan (2009) also found that with the increase of condensing temperature, exergy losses increases and exergy efficiency decreased for all the refrigerants.

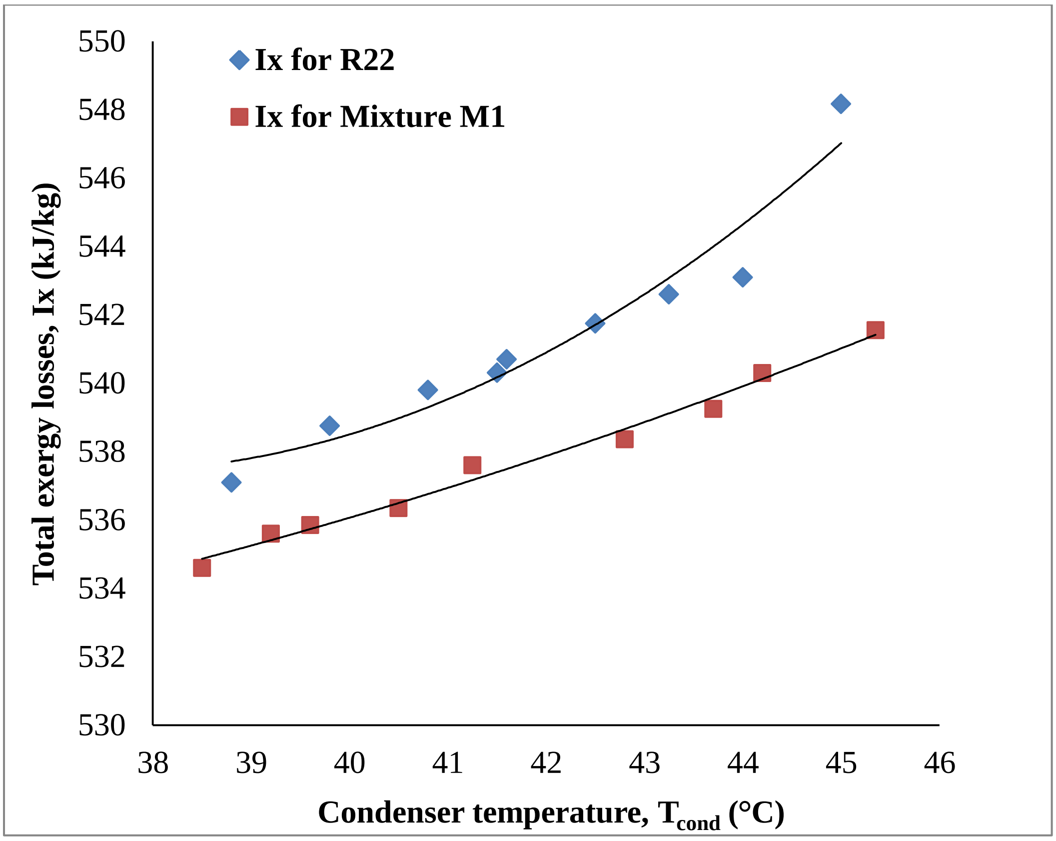


Figure 4.24: Variation of exergy losses at different condensing temperatures using both refrigerants.

4.6.4 Effect of evaporator and dead state temperatures on exergy efficiency

Figure 4.25 shows the effect of mean evaporator and ambient (dead state) temperatures on exergy efficiency of the system. From Figures 4.21-4.24, it is clear that exergy losses depend on mean evaporator and ambient temperatures. So, exergy efficiency also varies with respect to mean evaporator and ambient temperatures. It is shown in the Figure 4.25 that exergy efficiency increases with the increase of mean evaporator temperature for both ambient temperatures (29°C and 31°C). Figure 4.25 also shows that exergy efficiency at lower ambient temperature (29°C) is higher than that of at higher ambient temperature (31°C). It is because the exergy means workability of any system decreases with the increase of ambient temperature. The difference in temperatures between ambient and evaporator are decreased when the ambient temperature is increased.

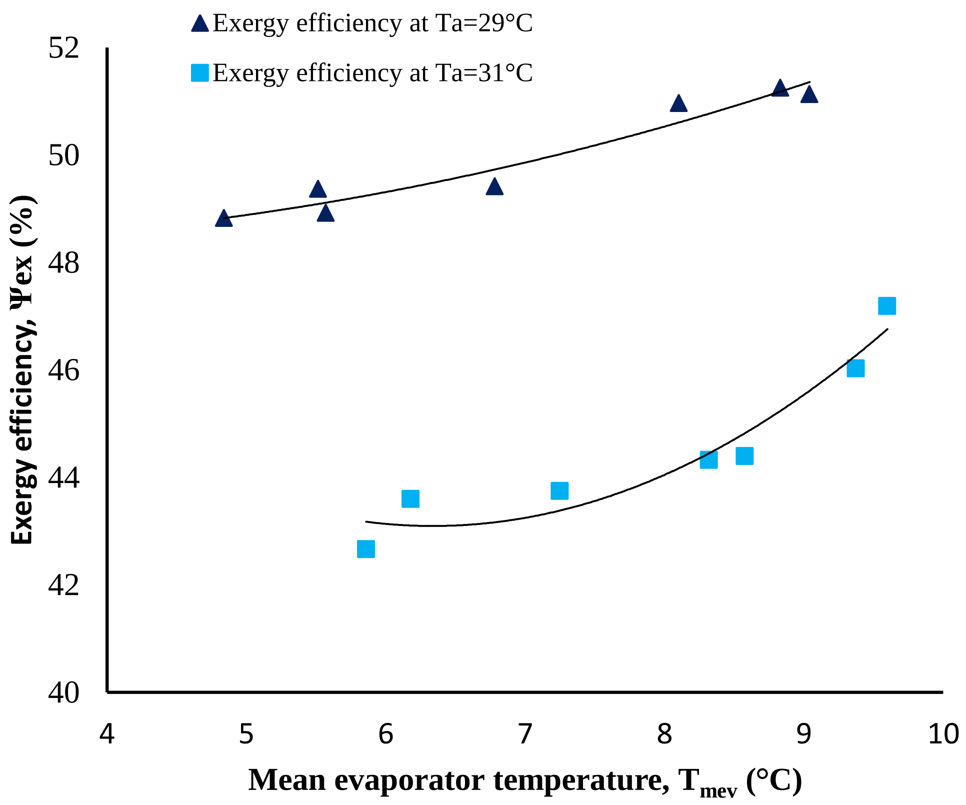


Figure 4.25: Variation of exergy efficiency (Ψ_{ex}) with mean evaporator temperature (T_{mev}) at T_a=29°Cand T_a=°C for R22.

Exergy efficiency of air conditioning system as well as vapor compression air conditioning system increases with the increase of evaporator temperature. It indicates that the second law performance of the system increases with the evaporator temperature. The efficiency varies from 49 % to 51.5 % at T_a=29°C and from 42.25% to 46.5% at T_a=31°C with the given mean evaporator temperature shown in Figure 4.25. Taufiq *et al*. (2007) also found that exergy efficiency varied from 36-42% for evaporative cooling with relative humidity 70% and exergy efficiency decreased with the increased of ambient temperature. Stegou-Sagia and Paignigiannis (2003) found that exergy efficiency for R404, R410a, R401, etc. varied from 50-55% and this range decreased with the increase of ambient temperature. With the increase in evaporator temperature, COP also increases, exergy losses decreases. Hence, the exergy performance increases.

In Figure 4.26, variation of exergy efficiency with mean evaporator temperature at ambient temperature $T_a = 29^{\circ}C$ for refrigerant R22 and M1 is shown. It is observed that exergy efficiency increases with the increase of mean evaporator temperature for both the refrigerants. As exergy losses are decreased at higher evaporator temperatures, thus the exergy efficiency also increases. But for refrigerant M1, the exergy efficiency shows higher values at all of the mean evaporator temperatures. This is due to lower exergy losses and higher refrigerating effect of M1 than those of R22.

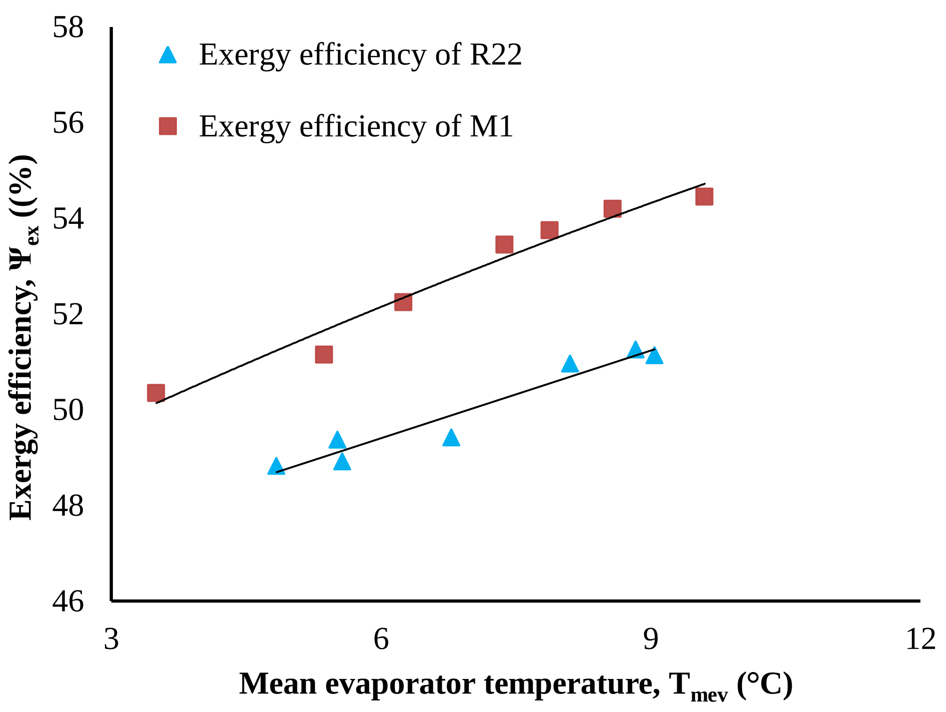


Figure 4. 26: Exergy efficiency of refrigerant R22 and M1 for different mean evaporating temperature at $T_a = 29°C$.

The increasing trend of exergy efficiency is similar to that of studied by Yumrutas et al. (2002). Bejan (1997) found that exergy efficiency depends on refrigeration temperature. Exergy efficiency deceases with the increase of evaporator temperature.

4.6.5 Variation of exergy losses in the different components of the system with evaporator temperature

Exergy losses depend on evaporator temperature of the air conditioner. It also depends on the temperature and process of the components. There are four components in the vapor compression system. In the condenser, the inlet temperature is higher compared to the other components. So, exergy losses in the condenser are higher than those in the other components. Due to the irreversibility in the condenser and friction, exergy losses are high compared to that in other components. It is observed that due to friction pressure there are pressure losses in the condenser and evaporator. So, higher exergy losses are occurred in these two parts compared to that of the others (Figure 4.27).

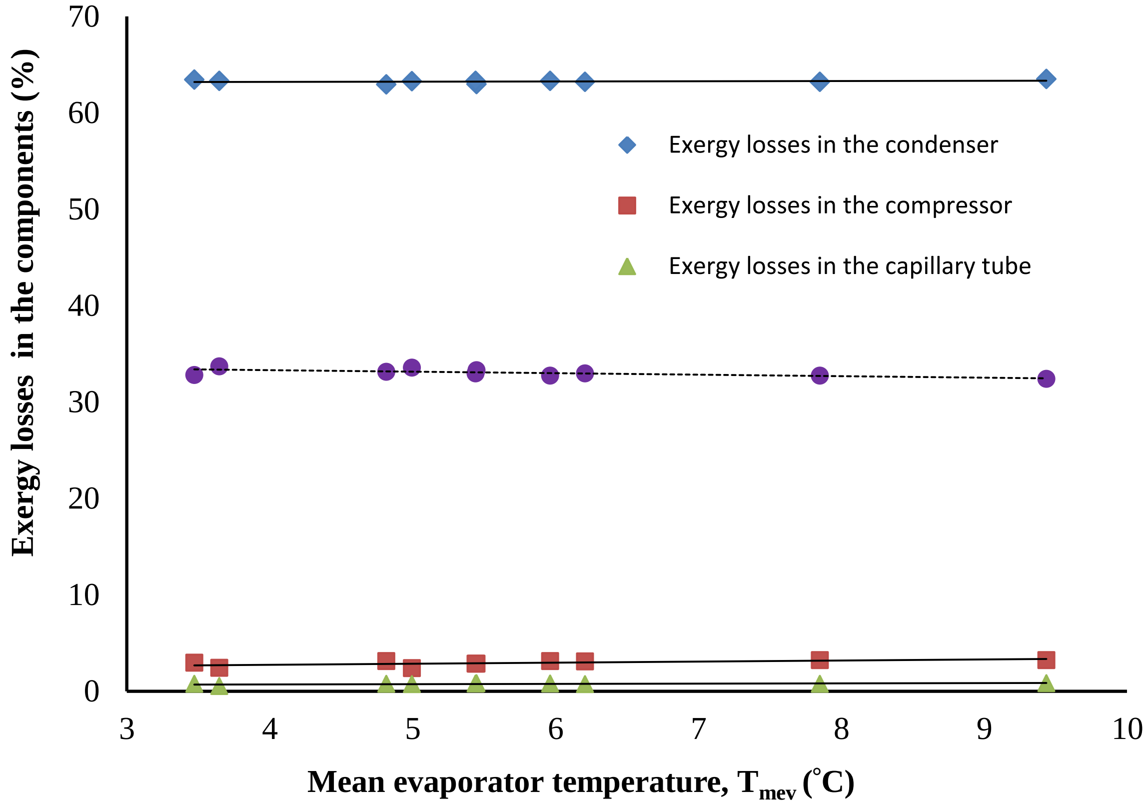


Figure 4. 27: Variation of exergy losses in the different components at different evaporator temperature (T_a= 27 °C).

Exergy losses in the capillary tube are lowest among the components. Exergy losses decrease with the increase of evaporator pressure or temperature. In Figure 4.27, the temperatures are taken as the mean evaporative temperatures in the evaporator at ambient temperature 27°C. Figures 4.27-4.29 show the exergy losses in the different components at ambient temperatures 27 °C, 29 °C and 31 °C.

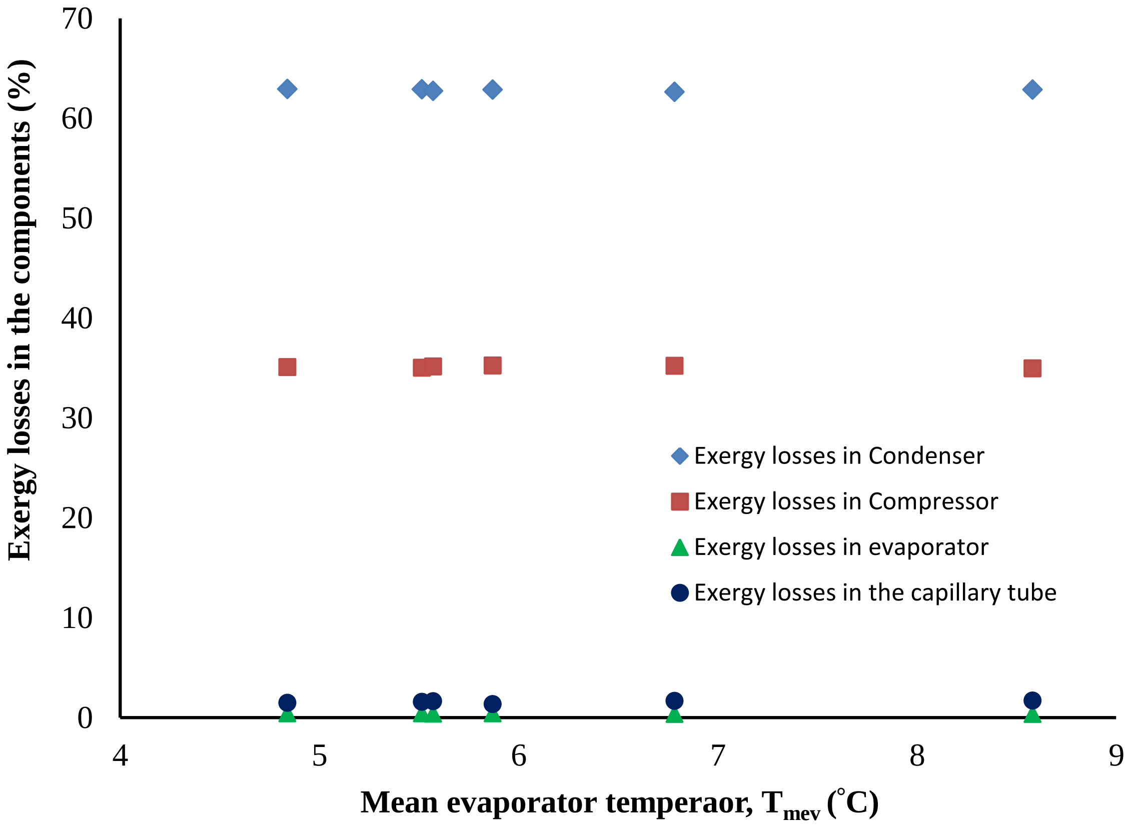


Figure 4. 28: Variation of exergy losses in the different components at different evaporator temperature (T_a= 29 °C).

Actually, the pressures in the evaporator are decreased along the path due to friction. The outlet temperatures in the evaporator are lower than the inlet temperatures. The increase in the evaporator temperature reduces the temperature difference between the evaporator and the atmospheric temperature. The more the deviation of temperature of any system from the atmosphere, the more the exergy destruction of that system occurs.

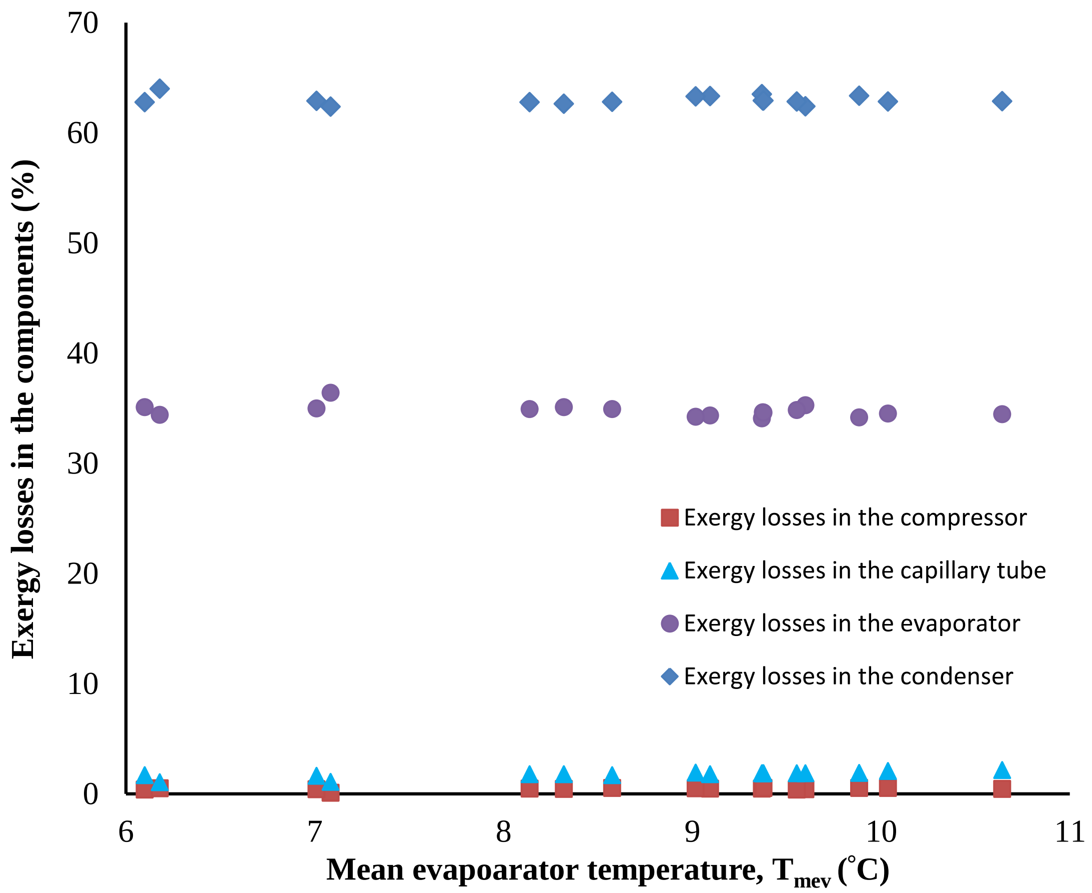


Figure 4.29:Variation of exergy losses in the different components at different evaporator temperature (T_a= 31°C).

Most of the exergy losses occur in the condenser (63-64%). Second highest losses occur in the compressor (about 34-36%). At capillary tube and evaporator, the exergy losses seem to be lowest. Similar results are found from the study of Yumrutas et al.(2002). Greater percentage of losses is found in the condenser. But their studies were based on computational.

Exergy losses can be stated also by using Grassmann diagram shown in Figure 4.30 for a particular operating temperature. This Figure shows that maximum exergy loss occurs in the condenser (62-65%) and in the evaporator (35%). The lowest losses have been occurred in the capillary tube.

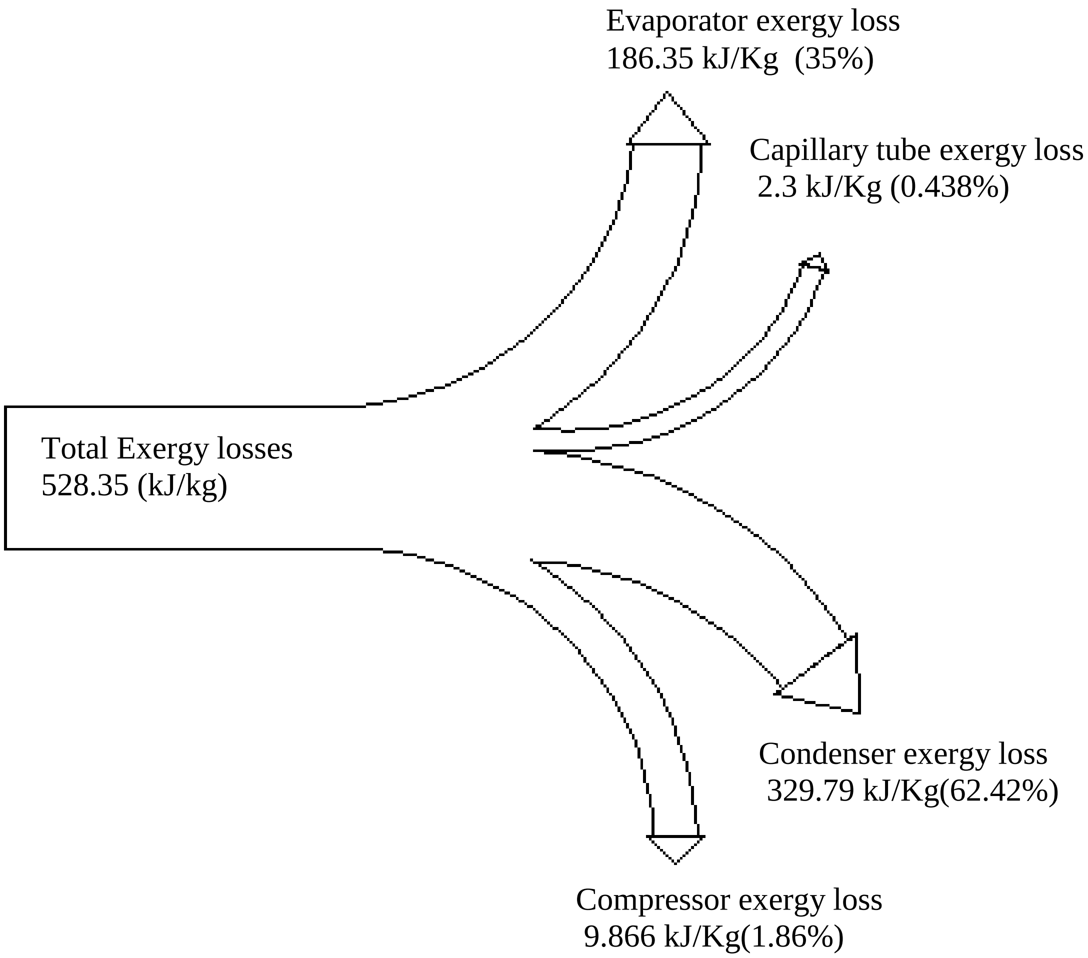


Figure 4. 30: Grassmann diagram for exergy losses flow through the components of the system.

4.6.6 Effect of dead state/reference temperature on exergy losses

It is shown in Figure 4.31 that exergy loses are reduced with the increase of ambient temperature. It is found that with the increase of reference or dead state temperature, the term $(1\text{-}T_0/T_r)$ is reduced. Hence, it caused to reduce the exergy losses. Arora and Kaushik (2008) also studied a theoretical investigation on some refrigerants like R502, R404A and R507A. They found that exergy efficiency and exergy destruction (losses) reduced with the increase of ambient temperature.

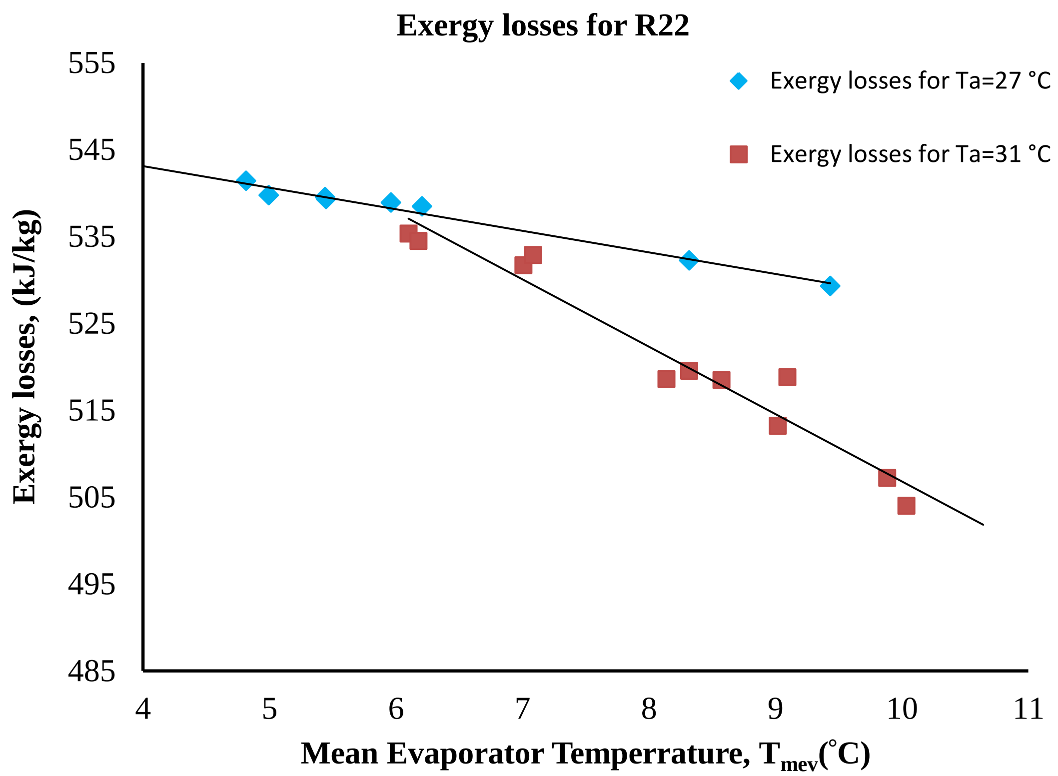


Figure 4.31: Variation of exergy losses of the different ambient temperatures and at the different evaporator temperature.

4.7 Heat transfer analysis

Heat transfer occurs in the condenser and evaporator of the vapor compressor system. Refrigerating capacity of the evaporator, condenser duty and heat rejection ratio of the systems are the parameters to measure the heat transfer performance of vapor compression system. Evaporator temperature, ambient temperature and condenser temperature have some effects on the heat transfer parameters. The number of turn of the coils of evaporator is 10. The length of each turn is 900 mm. The inside diameter of the coil is 7 mm and outside diameter is 9 mm. Hence the total surface area of the evaporator is 0.17325 m^2. The length of the capillary tube is 560 mm and its diameter is 1.9 mm. The metal of the evaporator and condenser is copper and attached with fins.

4.7.1 Effect of evaporator temperatures on condenser duty

Figure 4.32 shows the effect of mean evaporator temperature on condenser duty at ambient temperatures 31°C and 27 °C. Condenser releases the heat to the atmosphere. Condenser duty means heat rejection by the condenser which depends on mean evaporator temperature as well as ambient temperature. Figure 4.32 shows that heat rejection in the condenser is decreased with the increase of evaporator temperature at ambient temperature T_a= 31°C.

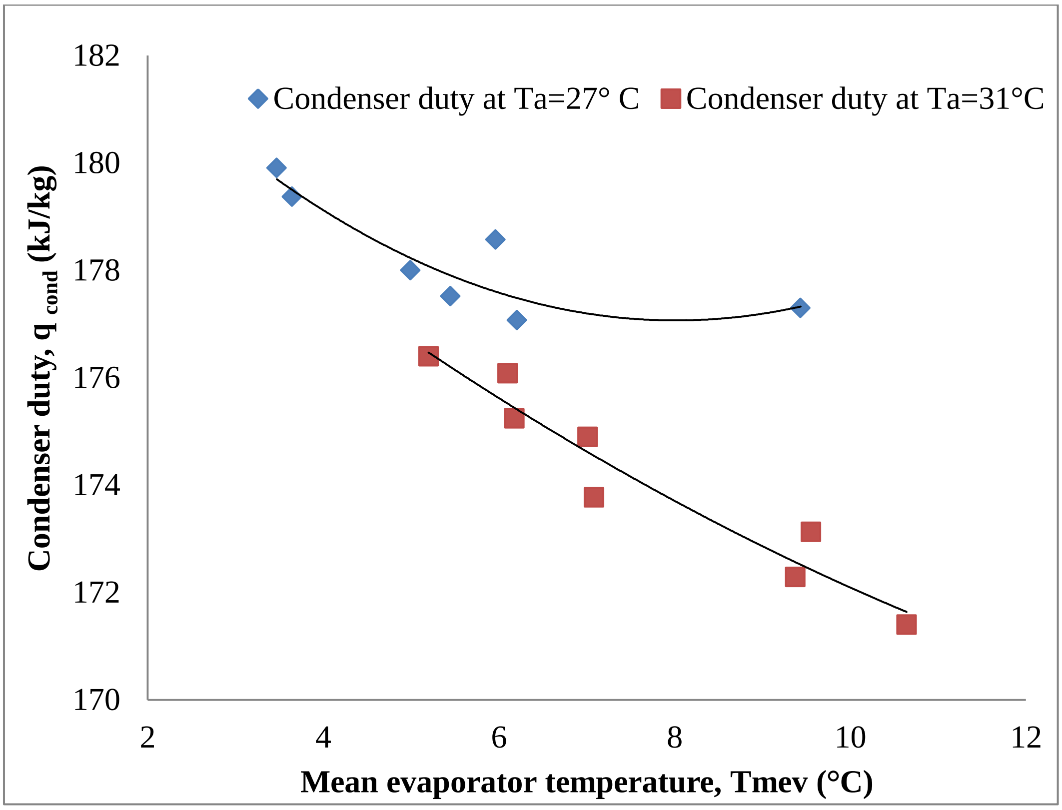


Figure 4.32: Variation in condenser duty or heat rejection capacity at different evaporator temperature (T_a= 31°C).

Figure 4.32 shows the variation of condenser duty at T_a= 27 °C and T_a= = 31 °C with different evaporator temperatures. This figure shows that condenser duty is decreasing with the increase of evaporator temperature. But from Figure 4.32, it is clear that condenser duty at T_a= 27 °C is greater than that at T_a= 31 °C for the evaporator temperatures. At lower evaporator temperatures, the system perform better than at evaporator temperature. At lower ambient temperature, the system undergoes more heat rejection from the condenser. Thus, the condenser duty is increased.

4.7.2 Effect of ambient temperature/evaporator temperature on cooling capacity

Figure 4.33 shows that the cooling or refrigeration capacity of the evaporator is increased with the increase of evaporator temperature. Figure 4.33 shows the variation in cooling capacity with evaporation temperature for R22. At higher evaporation temperature, the rate of cooling is gradually increased. Figure 4.33 also shows the variations in cooling capacity at different evaporation temperatures at different ambient temperatures. From the Figure 4.33, it is clear that cooling capacity is higher at T_a= 27° C than that at T_a= 31°C for the evaporator temperatures. At lower ambient temperature, the cooling capacity is higher because the difference of the temperatures between the atmospheric conditions and conditioned space is decreased.

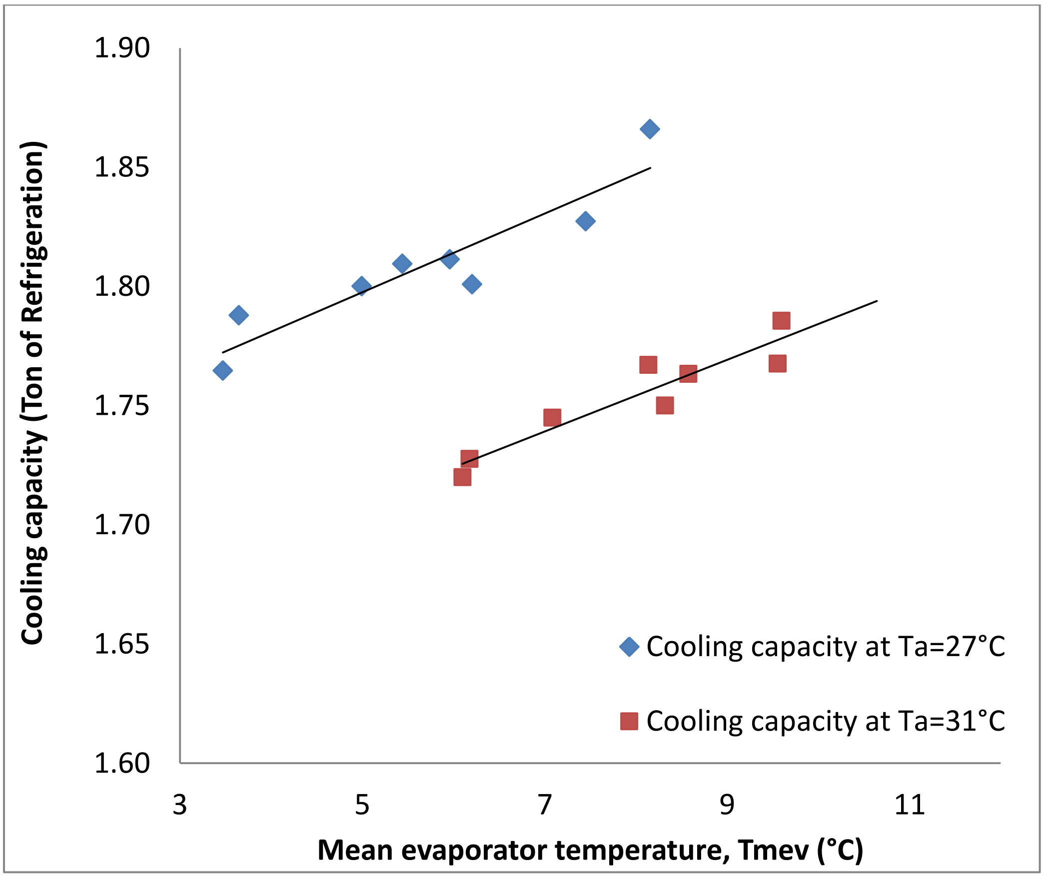


Figure 4.33: Variation in cooling capacity at different evaporator temperatures for refrigerant R22 at ambient temperature, T_a= 31 ^{0}C and 27°C.

That means the deviation of the system from the atmospheric condition is decreased. In that case, works of compression are also reduced. Generally, thermal performance is increased at lower ambient temperature. Increase of cooling capacity at T_a= 27°C for mean evaporator temperature range from 3.47-8.15 °C is 5.74% but at T_a= 31°C for mean evaporator temperature range from 6.17 to 8.57°C is only 2.06%.

4.7.3 Effect of refrigerants on cooling capacity at different evaporator temperature

Figure 4.34 shows the cooling capacity of R22 and M1 (mixture of R290 and R22 with ratio 1:3) at various evaporator temperatures. Figure 4.34 shows the variation in cooling capacity at ambient temperature T_a= 29° C. It shows that cooling capacity of M1 is higher than that of R22 at every evaporation temperature. M1 has higher latent heat of vaporization than that of R22. It indicates that the room will be cooled faster. The performance of M1 is better than that of R22.

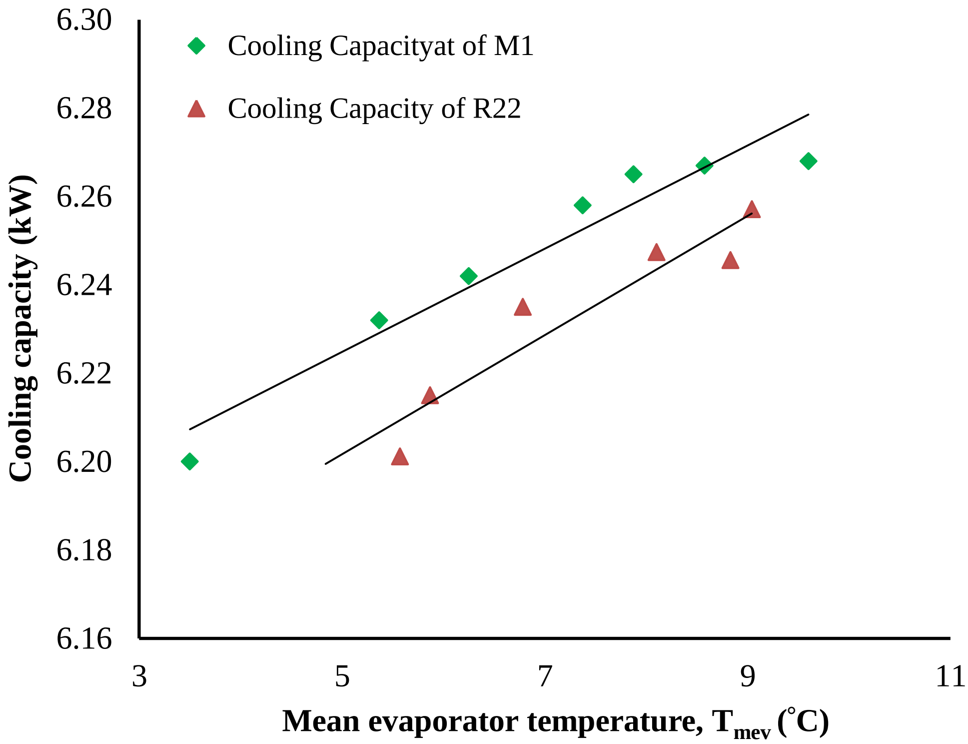


Figure 4.34: Variation in cooling capacity at different evaporator temperatures for different refrigerants at ambient temperature, T_a= 29°C.

4.7.4 Effect of evaporator temperatures on heat transfer coefficient

Figure 4.35 shows the effect of evaporator temperatures on heat transfer coefficient as well as heat transfer rate when the ambient temperature is 29 ^{0}C. It also varies with refrigerants. Two types of refrigerants have been taken for comparison. M1 shows higher heat transfer than that of R22 at all the evaporator temperatures. Similar results for R22 are found in the study of Park and Jung (2007).

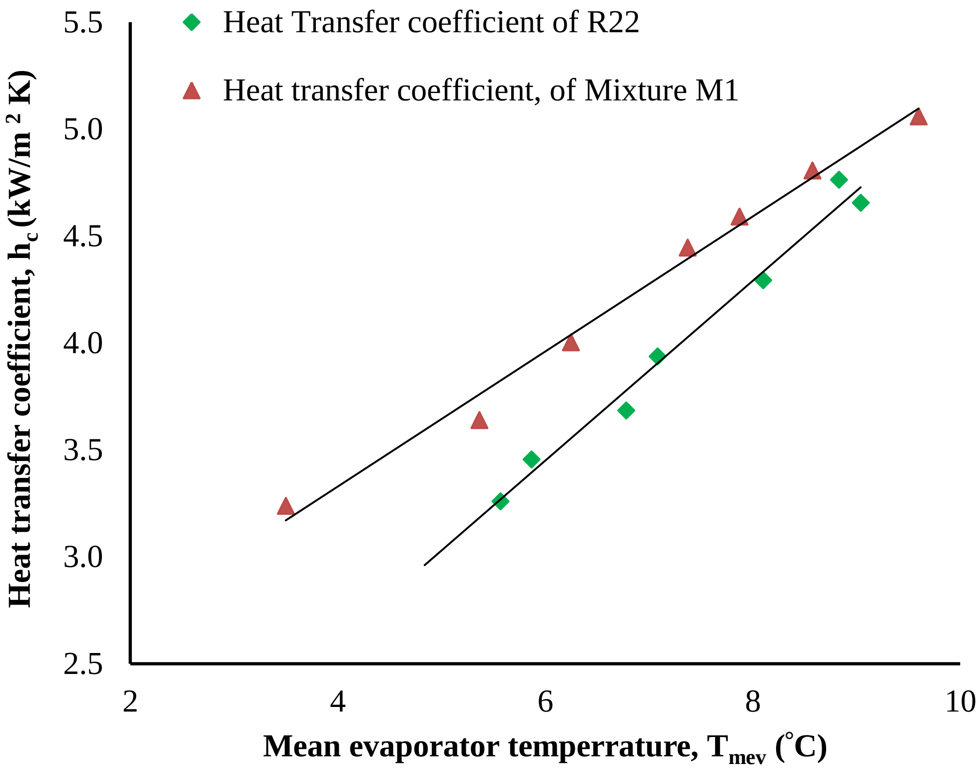


Figure 4.35 Variation of heat transfer coefficient on mean evaporator temperature at T_a= 29°C.

From Figure 4.36, it is observed that heat transfer coefficient varies with evaporator temperature as well as refrigerants for ambient temperature 27°C. At higher evaporator temperature, it increases and for M1 it is always higher than that of R22. Hence, this refrigerant is beneficial over the existing refrigerant. M1 has higher thermal capacity to transfer heat. Improvement based on heat transfer helps to reduce cost and energy uses also.

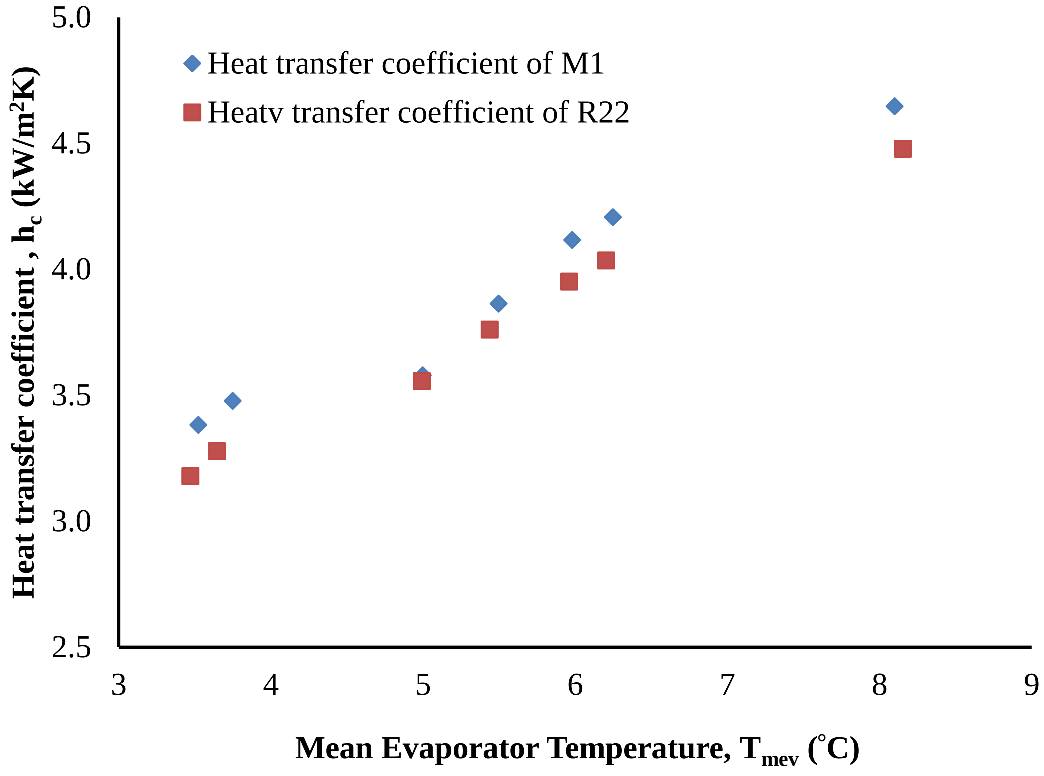


Figure 4.36: Variation of heat transfer coefficient on mean evaporator temperature on at T_a= 27°C.

4.7.5 Effect of operating temperatures on heat rejection ratio

Figures 4.37 and 4.38 show the effects of mean evaporator temperatures and condenser temperatures on heat rejection ratio for the air conditioner. Figure 4.38 shows the variation of heat rejection ratio for different evaporator temperatures for both the refrigerants. From Figure 4.38, it is observed that HRR decreases with the increase of mean evaporator temperature. M1 has lower HRR values than that of R22 at corresponding evaporating temperature. But this should be greater than one.

Heat rejection ratio increases with the increase of condenser temperature shown in Figure 4.38. For a given range of mean evaporator temperature, heat rejection in the condenser increases with the increase of condenser temperature. Similar trend was found from the study conducted by Sattar et al.(2007) .

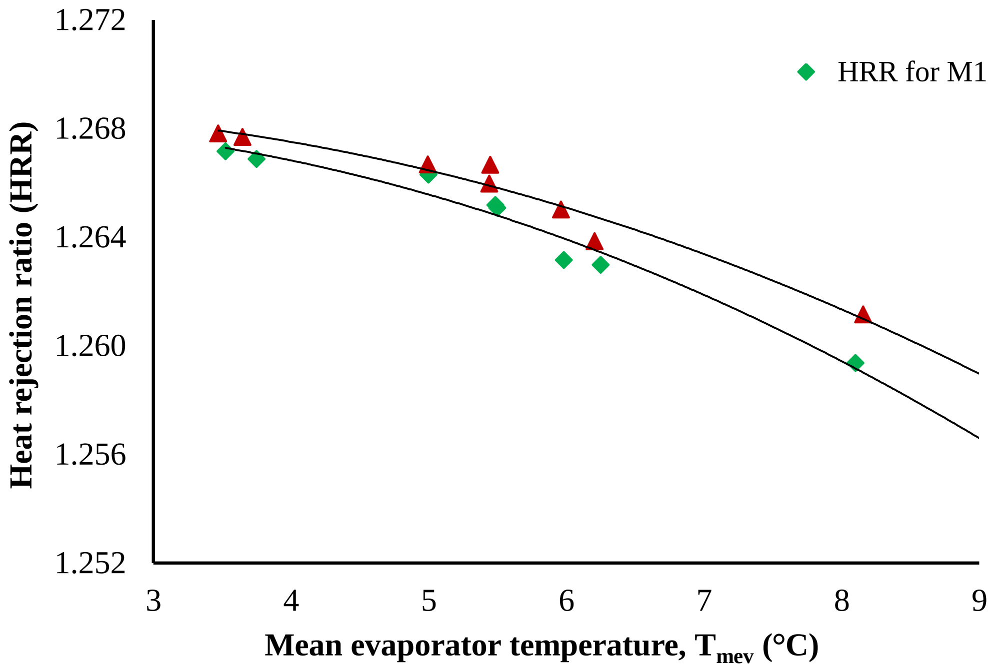


Figure 4. 37: Variation of heat rejection ratio with mean evaporator temperature.

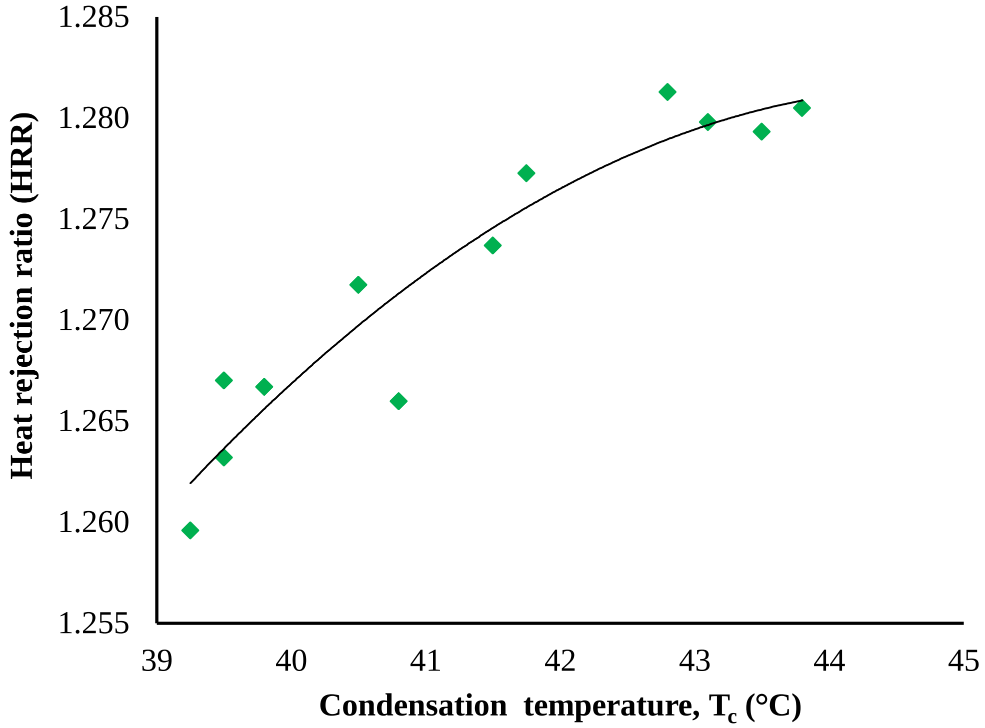


Figure 4. 38: Variation of heat rejection ratio with mean condenser temperature for evaporator temperature ranges from 4.8 to 5.9 °C.

4.8 Analytical study of nano-fluids on heat transfer performance

Nano particles are nowadays used in many sectors to enhance heat transfer rate. If nano-fluid is used as small percentage (2% to 4%) by volume to lubricant then lubricant is mixed with maximum 10% with the refrigerant. It is called nano lubricant –refrigerant mixture. It is also able to increase the refrigerating effect and heat transfer performance. Thus the heat rejection ratio and COP will be increased. Many literatures are available for using the nano fluid with refrigerant–lubricant. But the nano fluid should be miscible with lubricant. It has higher thermal conductivity. The energy consumption of R22 using mineral oil and nano particles (0.1% TiO_2) mixtures as lubricant can be saved 26.1% (Sheng-Shan et al., 2008). The same test with Al_2O_3 showed that the different nano particles have little effect on refrigerant effect. Nano fluid enhanced the solubility of the R22 and mineral oil. Another investigation showed that 4% CuO gives better enhancement in boiling heat transfer rather than 2% CuO for R22 (Murshed et al., 2008). It is because of more interactions of nano particles with the bubbles. But in this study, due to some unavoidable reasons, the study of the effect of nano-refrigerant for enhancement cannot be done.

Analytically, it is easy to find a relation between mass fractions of nanoparticles and thermal conductivities of heat transfer medium which are directly related to increase the heat transfer rate of the system. At first, percentage of nano fluid with the refrigerant by mass (ω) can be measured as follows (Peng et al., 2009):

$$\omega = \frac{M_n}{M_n + M_r} \tag{4.6}$$

Where, M_n represents the mass of nano particles, kg; M_r represents the mass of the refrigerant, kg with which the nano particles are mixed. This mass of nano particles occupies a volume in the refrigerant. The effect of the nano particles on heat transfer depends on their density and volume in the medium. Volume fractions (ϕ) of the particles are measured as follows:

$$\phi = \frac{\omega\rho_{r,L}}{\omega\rho_{r,L} + (1-\omega)\rho_n} \tag{4.7}$$

Where, ω is the mass fraction of the nano particles, $\rho_{r,L}$ and ρ_n are the densities of the liquid pure refrigerant and nanoparticles. The thermal conductivity of the mixture of refrigerant and nanofluid can be calculated by Hamilton-Crosser (Hamilton & Crosser, 1962) equation written as:

$$\lambda_{n,r,L} = \lambda_{r,L} \frac{\lambda_n + 2\lambda_{r,L} - 2\phi(\lambda_{r,L} - \lambda_n)}{\lambda_n + 2\lambda_{r,L} + \phi(\lambda_{r,L} - \lambda_n)} \tag{4.8}$$

Where, $\lambda_{n,r,L}$ represents the thermal conductivity of the refrigerant in liquid state containing nanoparticles with it, $\lambda_{r,L}$ represents the thermal conductivity of the pure refrigerant, λ_n represents the thermal conductivity of nano particles alone, ϕ is the volume fraction of the nano particles in the refrigerant-nano particle mixture.

From Equations (4.6) and (4.8), it is clear that high volume fraction of nano particles with high conductivity will results higher thermal conductivity of the mixture. But higher concentration of the nano fluid will cause higher deposition of nanoparticles and make nanofluid unstable. Figure 4.39 shows the increase of thermal conductivity of the nano-refrigerant with the increase of volumetric nano particle Al_2O_3 by percentage in the refrigerant.

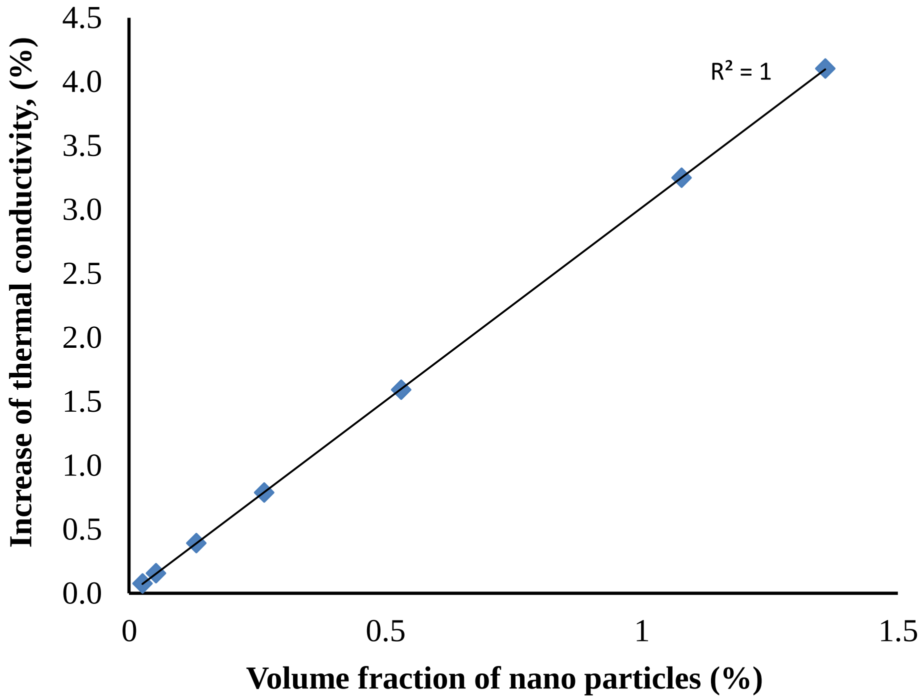


Figure 4.39: Increase of thermal conductivity of nano refrigerant with nano fluid of different volume.

Figure 4.39 shows that thermal conductivity increases with the increase of nano particles. By adding 1.5% of nanoparticles, the thermal conductivity increases up to 4.5% which is directly related to the heat transfer phenomena.

Another parameter, heat transfer coefficient is changed due to the addition of nano particles/fluid in the refrigerant. Peng et al. (2009) studied that heat transfer increased in the case of refrigerant R113 while using CuO as a nanofluid.

$$h_{c,n,r} = h_{c,r} \times \exp\left\{\phi\left[0.8\frac{\lambda_n}{\lambda_{n,L}} - 39.94\frac{(\rho C_p)_n}{(\rho C_p)_{r,L}} - 0.028G - 733.26x(1-x)\right)\right]\right\} \qquad (4.5)$$

Where, $h_{c,r}$ is the heat transfer coefficient of pure refrigerant $h_{c,r,\ n}$ is the heat transfer coefficient of refrigerant based nano fluid, x is the dryness fraction of vapor or vapor quality, C_p is the specific heat of the refrigerant and refrigerant based nano fluid, G is the mass flux.

4.9 Energy savings

Energy savings is the main objective of the most researches in the energy sectors. Usages of hydrocarbon mixture are capable to reduce energy expenses in the vapor compression air conditioning sector. Besides, some techniques are very much useful to reduce the energy usage also. Most of the techniques were utilized for analytical study. But in practice, some techniques are applied in some cases. Following methods are proposed to be used for the enhancement of the energy savings for this experimental study:

a) Changing thermostat temperature

b) Using hydrocarbon mixture as a refrigerant

c) Changing operating temperatures

4.9.1 Changing thermostat temperature

Generally, the thermostat temperature of the indoor room is a set point at 20 °C. But in this experimental investigation, it was changed in order to save the energy usage. The air conditioner was running from 10.00 am up to 6.00 pm. Total running time daily was 8 hrs. Total consumption of energy at different set point temperatures is shown in Table 4.4. Using Equations 3.30-3.32 with the data found from the experimental work are tabulated and shown in Table 4.4. From the literature (Saidur, 2009), it is found that total energy consumption in Malaysia was 127,752 MWh. Author found that energy consumption in the residential sector was 19% of the total energy uses. The air conditioner consumed 20% of the energy consumed by the residential sector in Malaysia. Hence, it is found that a total energy used by air conditioner in the residential sector is as follows:

$$TEC_{ac} = 127752 \times \frac{19}{100} \times \frac{20}{100} \Rightarrow 4854.576 \text{ MWh}$$

Total energy usage by domestic air conditioner = 4854.576 MWh

Table 4. 4: Annual energy and bill savings in the air conditioning sectors in Malaysia

<table>
<tr><th>Set point (T)</th><th>Average daily Energy Usage, AEU</th><th>%Energy Savings (%EST)</th><th>Average Energy Savings per ° C (%)</th><th>Total annual Energy usage (in air conditioner) TEC_{ac}</th><th>Total Energy savings (TES)/year</th><th>Total bill savings yearly (Thousands)</th></tr>
<tr><td>°C</td><td>kWh</td><td></td><td rowspan="6">6.09</td><td>MWh/year</td><td>MWh/year</td><td>RM (k)</td></tr>
<tr><td>20</td><td>7.738</td><td>-</td><td rowspan="5">4754.576</td><td>-</td><td>-</td></tr>
<tr><td>21</td><td>7.250</td><td>6.31</td><td>30001.37</td><td>6540.30</td></tr>
<tr><td>22</td><td>6.810</td><td>11.99</td><td>57007.37</td><td>12427.61</td></tr>
<tr><td>23</td><td>6.280</td><td>18.84</td><td>89576.21</td><td>19527.61</td></tr>
<tr><td>24</td><td>5.850</td><td>24.40</td><td>116011.7</td><td>25290.54</td></tr>
</table>

4.9.2 Using hydrocarbon mixture as a refrigerant

Normally, R22 is used as a refrigerant in the air conditioner but in this experimental work, M1 is considered as an alternative refrigerant to R22. Using data from Table 4.1 and incorporating these data into the Equation 3.29, average energy savings are obtained as follows:

$$AES\% = \frac{AEU_{R22} - AEU_{M1}}{AEU_{R22}} \times 100\%$$

$$\Rightarrow \frac{7.65_{R22} - 7.25}{7.65} \times 100\% \Rightarrow 5.22\%$$

The above result indicates that using only 25% propane with R22, energy consumption will be reduced. Another thing is that this mixture caused to increase the coefficient of performance.

4.9.3 Changing the operating temperature

Operating temperatures have a great influence on the coefficient of performance. Hence, it has an effect on energy consumption also. In the case of vapor compression system, operating temperatures are condensing temperature and ambient temperature. But changes of operating temperatures are not easy. Evaporator temperature and condenser temperatures were changed due to the changes of refrigerants. It is observed in the Figure 4.14 that in the morning, average power consumption was 1.58 kW and at noon, power consumption was 1.68kW for a particular range of evaporator temperatures. But during experimental work, in the morning, the ambient temperature was around 26°C and at noon it was 31.8°C. So, for 5.8 °C increase in ambient temperature (though other operating temperatures are not constant) caused 6.33% increase of power consumption. If it is possible to maintain ambient temperature for whole day at 26°C, the energy consumption will be reduced 6.33%. But this cannot be possible exactly. By limiting the use of substances having more global warming potential in the energy sectors, it is possible to maintain the atmospheric temperature at minimum level. Planting more trees in the country will also help to reduce emission, warming, etc. Thus, environment will become cool. Indirectly using high thermal capacity refrigerant, enhancement devices in the evaporator and condenser, pressure ratio as well as power consumption will be reduced.

4.10 Reduction of exergy destruction or exergy savings

4.10.1 Replacement of refrigerant R22 with M1

Total exergy losses of the refrigerant R22 and the M1 at different mean evaporator temperatures for two different ambient temperatures are calculated and presented in Table 4.5. By using Equation 3.33 and Figures 4.22-4.29, exergy losses reductions are calculated. It is observed that exergy loss can be reduced using M1 instead of R22. Exergy savings are found higher at lower ambient temperatures. Table 4.5 shows the differences in exergy losses for both refrigerants used in the experimental work.

Table 4. 5: Average exergy savings at different mean evaporator temperatures and different ambient temperatures for both the refrigerants.

At T_a= 27 °C				At T_a= 29 °C			
Mean evaporator temperature, T_{mev}°C	Exergy losses, Ix (kJ/kg) For R22	Exergy losses, Ix (kJ/kg) for M1	Exergy Savings %	Mean evaporator temperature, T_{mev}°C	Exergy losses, Ix (kJ/kg) For R22	Exergy losses, Ix (kJ/kg) for M1	Exergy Savings %
4	542	537	0.92	4.5	535.0	532	0.56
5	538	534	0.74	5	532.5	531	0.28
7	532	528	0.75	6	531.0	529	0.38
8	528.5	525.5	0.57	8	528.5	527	0.28

4.10.2 Increasing mean evaporator temperature

Total exergy losses are calculated for the refrigerants at different mean evaporator temperatures using the Figure 4.24 and Equation 3.22. Ambient different are also varied. But for clear understanding and comparison, data for two distinct ambient temperatures are described and presented in Table 4.6.

Table 4.6: Exergy savings at different mean evaporator temperatures for different ambient temperatures.

Mean evaporator Temperature, T_{mev} (°C)	Exergy loss at T_a= 27 °C	Exergy savings % at T_a=27 °C	Mean evaporator Temperature, T_{mev} (°C)	Exergy loss at T_a= 31°C	Exergy savings % at T_a=31°C
3.47	543.96	-	6.10	540.31	-
4.82	541.43	0.47	6.17	534.5	1.08
5.45	539.34	0.85	7.01	531.69	1.60
5.96	538.92	0.93	8.32	519.57	3.84
6.21	538.48	1.01	9.00	513.00	5.05
8.15	533.25	1.97	10.00	504.00	6.72
9.44	529.30	2.70			

Total exergy losses and exergy savings at T_a= 27°C and 31°C for different mean evaporator temperatures are calculated using Equation 3.34. It is clear that exergy savings increased with the increase of evaporator temperature for both ambient temperatures. It is also concluded that exergy savings are higher at T_a= 31°C than that of at T_a= 27°C. It is because at higher ambient temperature, the deviation of condenser temperature from the atmosphere is reduced. Hence, the exergy losses are reduced.

4.10.3 Exergy savings by decreasing condenser temperature

Table 4.7 shows a variation of exergy savings with different condenser temperatures. From Figure 4.22, it is observed that exergy losses depend on condenser temperature also. By reducing condensation temperature, exergy losses can be reduced for the refrigerants. Using data from Figure 4.26 and Equation 3.35, exergy savings for the refrigerant R22 are calculated. These results are summarized and shown in Table 4.7. Table 4.7 shows that exergy savings increased with the decrease of condensation temperature. In this experiment, the base condensation temperature for comparison was taken to be 45°C. The lowest condensation temperature was 38.8 °C.

Table 4.7: Exergy savings with decreasing condenser temperature.

Condenser temperature, T_c (°C)	Exergy losses, Ix (kJ/kg)	Temperature decrease, ΔT_c (°C)	Exergy savings (%)
45.0	548.17	-	-
44.0	543.10	1.0	0.93
43.3	542.60	1.75	1.02
42.5	541.75	2.5	1.17
41.5	540.31	3.5	1.43
40.8	539.80	4.2	1.536
39.8	538.75	5.2	1.72
38.8	537.10	6.2	2.019

4.11 Comparative study with other works

Some of the researchers worked with R22 for air-conditioning system. Most of the researches were performed based on computational models. Some of the experimental works are performed related to the vapor compression system using refrigerator. Park and Jung (2008) found that compressor outlet temperature was higher for refrigerant R22 than that of R1270 and R290 but the lowest discharge temperature was found for R290. The coefficient of performance for R1270 also found better than that of the existing refrigerants. Karkri et *al.*(2009) also stated that using R410A as refrigerant, COP varied from 3.0 to 4.20 for evaporator outlet temperature ranges from 0°C to 8°C. In that study, authors also found that COP increased with the increase of evaporator temperature. Sachdev (2008) and Arora et al. (2007) described about the performance of the refrigerants R22, R407C, R410A with evaporator temperatures.

Figure 4.40 shows a comparative study of work of Sachdev (2008) and the present work based on coefficient of performance with different mean evaporator temperature. Sachdev (2008) used R22 and R407C as the working fluid. Sachdev (2008) also studied with other refrigerants. A comparative study of the present work and work of Sachdev (2008) are shown in Figure 4.40. Both the researchers showed that COP increased with the increase of evaporator temperature. Sachdev (2008) studied through analytical approach but present work based on experimental work. At the higher evaporator temperatures, the results tend to be closer.

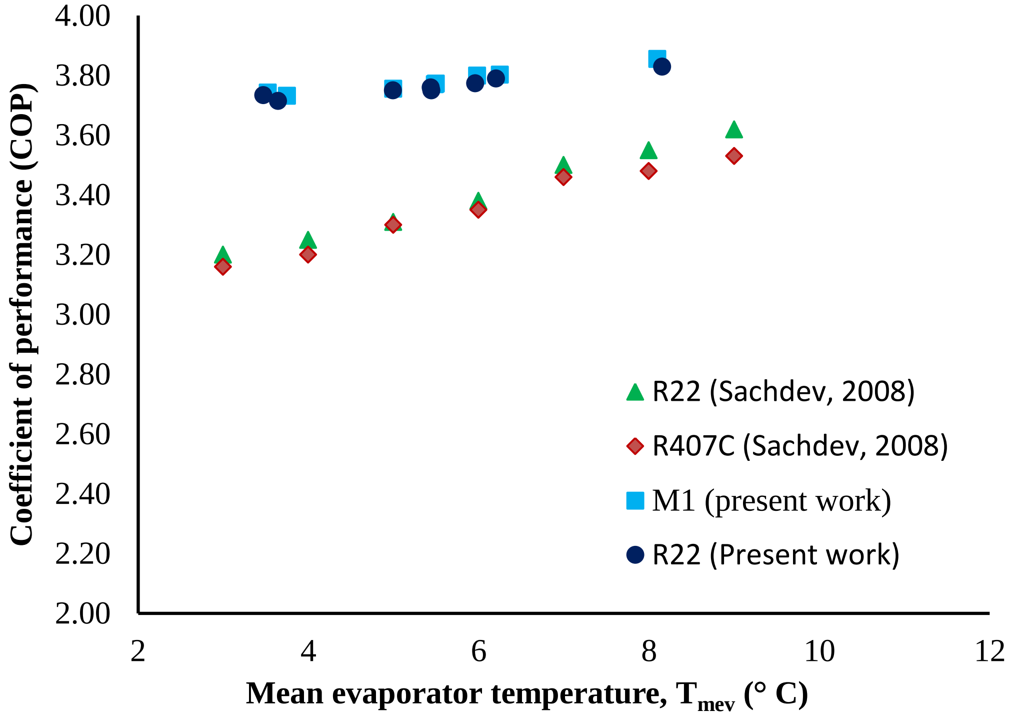


Figure 4. 40: Comparative study of the present work to the work of Sachdev (2008) with different mean evaporator temperature.

CHAPTER 5: CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

Hydrocarbon is found as the most suitable refrigerant for the air conditioners due to its zero ODP and low GWP. Although hydrocarbons and their mixtures have some flammability problems, but they have better energy performance (COP) and high cooling capacity as well as exergy efficiency compared to the CFC/HCFC refrigerants. In this experimental investigation, the amount of hydrocarbons (R290) is 25% of the total mixture by mass whereas, R22 is about 75%. For running a room air conditioner, a small amount of hydrocarbon is essential. This amount should not create any harmful or dangerous effect for the people though it is flammable. Therefore, many researchers and manufacturers suggest that it can be used with care i.e. without any leakage. Results are obtained based on thermal performances at various operating conditions for pure R22 and the M1 are summarized here and has been taken for comparison. The findings according to the objectives are described as follows:

(a) Based on thermal performance of the refrigerants

- ❖ It is found that the coefficient of performance decreases with the increase of the pressure ratio for all the refrigerants. The COP decreases by 35.22% for unit increase of the pressure ratio.
- ❖ M1 shows higher refrigerating effect than that of R22 at all evaporating temperatures.
- ❖ Work of compression increases up to 17% for the ranges of condenser temperatures from 35 to 51°C without considering the effect of ambient temperatures for R22. But at constant ambient temperature, the variation in work of compression with condenser temperature at given range is not significant. When the ambient temperature is 29 °C, the work of

compression increases up to 3% for condenser temperature changes from 42.5 to 44.5 ˚C.

- ❖ The power consumption decreases by 4.5 to 5.5 % while using M1 instead of R22.
- ❖ Seasonal energy efficiency ratio (SEER) increases with the increase of evaporator temperature, and it is higher at lower ambient temperature. At T_a= 27°C, seasonal energy efficiency ratio is higher compared to those at T_a=29° and 31°C. SEER for the M1 is found about 2% higher than that of R22.
- ❖ SEER also depends on condenser temperature. At higher condenser temperature, the values of SEER are decreased. With the given condition, SEER increases upto 8% with the increase of condenser temperature.

(b) Based on operating temperatures

- ❖ The COP is increased with the increase of evaporator temperatures for both the refrigerants. It is obserbed that a small fluctuation is occured in COP for the different ambient temepratures.
- ❖ At T_a= 27˚C and for mean evaporator temperature ranges of 3.47-8.15 ˚C, the cooling capacity increased up to 5.74% but at T_a= 31˚C and for mean evaporator temperature ranges of 6.17 to 8.57˚C, cooling capacity increased only up to 2.06%.
- ❖ COP at T_a =27 ˚C is found to be10-15% higher than that at T_a= 31 ˚C for the given mean evaporator temperature. The COP is decreased with the increase of condenser temperature. However, when other parameters such as evaporator and condenser temperatures, humidity and controlled room temperature remain constant, the effect of ambient temperature on the performance is not significant. It is also found that evaporator temperature

changes with the changes of the ambient temperature. Hence, the COP changes significantly. At lower ambient temperatures, COP increases very slowly with the increase of evaporator temperature.

- ❖ Exergy losses or destruction depends on evaporator temperature. From Figure 4.23, it can be concluded that exergy losses per kg of refrigerant flow are increased with the increase of mean evaporator temperature. It is also observed that for both ambient temperatures, the exergy losses are decreased with the increase of mean evaporator temperature. It is also observed that exergy losses are lower at T_a =31 $^{\circ}$C than that at T_a= 27 $^{\circ}$C for every mean evaporator temperature.
- ❖ Exergy efficiency of the vapor compression air conditioning system is increased with the increase of evaporator temperature. It indicates that the second law performance of the system increases with the increase of evaporator temperature. Exergy efficiency at T_a=27°C is found higher than that at T_a=31°C.

(c) Based on exergy performance

- ❖ Exergy losses in the condenser are found to be higher than that in other components. Due to the irreversibilities and frictions in the condenser, exergy losses are higher than that in other components and the condenser is worked at a higher temperature compared to the evaporator.
- ❖ Exergy efficiency of the system of M1 is 10-15% higher than that of R22. Exergy efficiency of the system can be increased up to 10% by decreasing the atmospheric temperature.

(d) Based on heat transfer performance

- ❖ For a given ambient temperature, cooling capacity of M1 is higher than that of R22 at all evaporator temperatures.

- Heat transfer coefficient (h_c) of M1 is higher than that of R22 at all operating conditions. Heat transfer coefficient increases with the increase of mean evaporating temperature.
- M1 shows lower heat transfer coefficient than that of R22 at all evaporating temperatures. Heat rejection ratio (HRR) depends on the operating temperatures. Heat rejection ratio decreases with the increase of evaporator temperature. On the other hand, it increases with the increase of condenser temperature.

(e) Based on correlation developed

- Two correlations for COP are developed for the two refrigerants used in this experimental investigation for different operating temperatures. It is observed that COP varies highly with the changes of condenser and evaporator temperatures. But the ambient temperature has less effect on COP for both refrigerants.

(f) Based on energy and exergy savings

- With increasing the set point indoor temperature by 4°C, 24.40% of the energy consumption can be saved annually. Using M1 as a refrigerant instead of R22, up to 5.22% energy can be saved.
- More exergy savings can be possible by increasing mean evaporating temperature. At T_a= 27°C, by increasing evaporator temperature from 3.5 to 9.5°C, 2.7% exergy loss can be reduced whereas, at T_a= 31°C, by increasing evaporator temperature from 6.1 to 10°C, 6.72% exergy loss can be reduced.

5.2 Recommendations

In this study, the mixture of R22 and R290 with the mass ratio of 3:1 respectively has been used. However, different kind of mixture of R22 and R290 can be used for the analysis of energy, exergy and environmental performance. From the present experiment, it can be recommended the following issues:

- For the new mixtures, new properties will be achieved and new relationship for the performance ratio of the air conditioner can be derived.
- Besides, tribo-logical factors can be considered for selecting refrigerant. Some additives may improve the surface quality or reduce the wear formation in the compressor parts.
- Exergy analysis can be taken into consideration for the new mixtures. It is also necessary to modify the compressor for checking all the combination of the refrigerants.
- Some techniques like uses of porous twisted tape inserts can improve heat transfer in the condenser as well as in the evaporator. This technique is already found better performance for the heat transfer in the heat exchanger for turbulent flow.
- An exergoeconomic (which is a combination of exergy and economics) analysis can be considered for the improvement of this study.

APPENDICES

APPENDIX A: SPECIFICATIONS OF THE EQUIPMENT

Table A1: Specifications of the Evaporator, Condenser and Capillary tube

Specifications	Ranges/values
Number of tubes	10
Length of each turn	900 mm
Inside diameter of the tube	7.0 mm
Outside diameter of the tube	9.0 mm
Total area	0.17325 m^2
Capillary tube length	560 mm
Capillary tube diameter	1.9 mm
Metal of the evaporator tube	copper
Condenser materials	Similar to evaporator

Table A2: Specifications of Power Meter

Specifications	Ranges
Voltage	15-600 V
Current	0.5- 50 Amp
Frequency	10 Hz-50Hz
Accuracy	± 0.2% of rdg

Table A3: Specifications of Data Logger

Specifications	Ranges
No. of channels	20 Multiplexer
Voltage	100 V-120 V/220 V-240 V
Accuracy	±0.1^0 C and 0.2%

Table A4: Specifications of Pressure Transducer

Specifications	Unit	Results
Pressure Range and Type	barG	0~ 50
Output range	mA	4.00~ 20.0
Temperature effect on zero and span	%FS/10°C	0.5
Supply voltage	V dc	12~36
Compensated temp. Range	°C	-10~+60
Operating Temperature Range	°C	-40 ~125
Model	132F	

Table A 5: Specifications of Flow meter

Specification	Unit
Type	ABB Flow meter
Model	FAM541F1YOF1
Voltage	12-46v
Output	4~20 mA
Ambient temperature	20~70^0 C
Diameter, DN	25

Table A6: Specifications of the Yellow Jacket Electrical Scale

Specifications	Ranges/values	Specifications	Ranges/values
Capacity	100 kg	Storage temperature	-20 to 70°C
Resolution	.01 kg	Operating temperature	0 to 50 °C
Accuracy	0.1%	Operational method	Oversized strain gauge bridge
Unit weight	5.4 kg	Components	Industrial grade
Case size	457x330x108	Battery	9V alkaline
Plat form size	287x187		

APPENDIX B : UNCERTAINTY ANALYSIS

In this experiment there may be some inaccuracies in measuring the primary data. It needs a description of such inaccuracies. It is known that an appropriate idea for expressing inaccuracies is a “uncertainty” and the actual value should be provided by an “uncertainty analysis”. An uncertainty is not the same as an error. The error in the measurement is the difference between the true value and the measured value. Since uncertainty can take various values over a range, it is inherently a statistical variable.

1. Uncertainties in Measurands

To eliminate the uncertainties in the experiment, experimenters are advised to report the uncertainties in every measurands. For measuring the uncertainties following information should be considered:

a. Precision limit, P: This is an estimate of the lack of repeatability caused by random errors and process unsteadiness. This element can be sampled with the available procedures and apparatus. It should be based on statistical estimations from samples whenever possible.

b. Bias limit, B: The bias limit is an estimate of the magnitude of the fixed constant error. This element cannot be sampled within available procedure and its existence is what mandates the need of cross-checks.

c. Total uncertainty, W: There is 5 intervals about the nominal results of the band within which the experiment is 95% confident that the true value of the results lies. And it is calculated from the following way:

$$W = \sqrt{[P^2 + B^2]} \qquad \text{(B-1)}$$

2. Propagation of uncertainties into results

In nearly all other experiments, it is necessary to compute the uncertainty in the results from the estimations of uncertainty in the measurands. This computation process is called "propagation of uncertainty".

According to Kline and McClintock (1953), the propagation equation of a result, R computed from m measurands x_1, x_2, x_3, -----------xm having absolute uncertainty W_R is given by the following equation:

$$W_R = \left[\left(\frac{\delta R}{\delta X_1} w_{x1} \right)^2 + \left(\frac{\delta R}{\delta X_2} w_{x2} \right)^2 + ------ \left(\frac{\delta R}{\delta X_m} w_{xm} \right)^2 \right]^{\frac{1}{2}} \quad \text{[B-2]}$$

3. Calculation of uncertainties in the present experiment:

Primary measurands are T_a, T_{mev}, T_c, P_i, P_{out}, W_c and m_r . Results of these uncertainties are presented in the Tables 3.2 and 3.3, respectively. From these results, uncertainty in Coefficient of performance, exergy losses, exergy efficiency, refrigeration capacity and condenser duty are calculated.

Determination of uncertainty in condenser duty:

Room temperature, $T\infty=27^0$, error 0.05 ^{0}C of 27 ^{0}C

The inlet and outlet temperature of the compressor are 4.8 and 17.19 bar respectively. Enthalpies are found using REFPROF7 software with corresponding pressure and temperature. At this pressure and temperatures the enthalpies are as follows:

h_3= 431.62 kJ/kg, h_4=254.32 kJ/kg, mass flow rate mr= 0.041 kg/s

Heat rejection in the condenser, Q_r=7.2693 kJ/sec

The uncertainties in these values are calculated as follows:

w_{h3}=0.2158 kJ/kg $^{\circ}$C

w_{h4} = 0.12716 kJ/kg

w_{mr} =2.2 x 10^{-5} kg/s

Now, uncertainty in the heat rejection capacity of the system is

$$W_{Qr} = \left[\left(\frac{\partial Q_r}{\partial h_3} \times dh_3 \right)^2 + \left(\frac{\partial Q_r}{\partial h_4} \times dh_4 \right)^2 + \left(\frac{\partial Q_r}{\partial m_r} \times dm_r \right)^2 \right]^{\frac{1}{2}}$$

$$= \left[(m_r \times dh_3)^2 + (m_r \times dh_4)^2 + \{(h_3 - h_4) \times dm\}^2 \right]^{1/2}$$

$$= \left[(0.041 \times .2158)^2 + (.041 \times .1271)^2 + \{(431.62 \ \ - 254.32) \times 2.195 \times 10^{-5}\}^2 \right]^{1/2}$$

$$=0.010989 \text{ kJ}$$

% of error/uncertainty in heat rejection $= \frac{.0.010989}{7.2693}$

$$=0.1511\%$$

Determination of uncertainty of refrigeration effect

Room temperature, $T\infty=27^0$, error 0.05^0 C of 27^0 C

The inlet and outlet temperature of the compressor are 4.8 and 17.19 bar respectively. Pressure in the evaporator inlet is 7.72 bar and outlet pressure of the evaporator is 4.9 bar. Enthalpies are found using REFPROF7 software with corresponding pressure and temperature. At this pressure and temperatures the enthalpies are as follows:

H_1= 406.55 kJ/kg, h_4=253.5 kJ/kg, mass flow rate mr= 0.041 kg/s

Heat Absorption in the evaporator, Q_e=6.275 kJ/sec

The uncertainties in these values are calculated as follows:

w_{h1}=0.2032 kJ/kg $^{\circ}$C

w_{h4} = 0.12675 kJ/kg

w_{mr} =2.2058 x 10^{-5} kg/s

Now, uncertainty in the heat rejection capacity of the system is

$$W_{Qr} = \left[\left(\frac{\partial Q_r}{\partial h_1} \times dh_1 \right)^2 + \left(\frac{\partial Q_r}{\partial h_4} \times dh_4 \right)^2 + \left(\frac{\partial Q_r}{\partial m_r} \times dm_r \right)^2 \right]^{\frac{1}{2}}$$

$$= \left[(m_r \times dh_1)^2 + (m_r \times dh_4)^2 + \{(h_1 - h_4) \times dm\}^2 \right]^{1/2}$$

$$= \left[(0.041 \times .2032)^2 + (.041 \times .12675)^2 + \{(406.55 - 253.5) \times 2.2058 \times 10^{-5}\}^2 \right]^{1/2}$$

=0.010386 kJ

% of error/uncertainty in heat rejection $= \frac{.0.010386}{6.275}$

=0.1655%

Determination of uncertainty of coefficient of performance (COP):

Room temperature, T∞=27^0, error 0.05^0 C of 27^0 C

The inlet and outlet temperature of the compressor are 4.8 and 17.19 bar respectively. Pressure in the evaporator inlet is 7.72 bar and outlet pressure of the evaporator is 4.9 bar. Enthalpies are found using REFPROF7 software with corresponding pressure and temperature. At this pressure and temperatures the enthalpies are as follows:

H_1= 406.55 kJ/kg, h_4=253.5 kJ/kg, mass flow rate mr= 0.041 kg/s

Heat Absorption in the evaporator, Q_e=6.275 kJ/sec

Enthalpy of the Compressor outlet, h_2= 450.699 kJ/kg

The uncertainties in these values are calculated as follows:

w_{h1}=0.2032 kJ/kg °C

w_{h4} = 0.12675 kJ/kg

w_{h2} =0.2253 kg/s

It is known that $COP = \frac{h_1 - h_4}{h_2 - h_1}$

Now, uncertainty in the heat rejection capacity of the system is

$$W_{Qr} = \left[\left(\frac{\partial Q_r}{\partial h_1} \times dh_1 \right)^2 + \left(\frac{\partial Q_r}{\partial h_4} \times dh_4 \right)^2 + \left(\frac{\partial Q_r}{\partial h_2} \times dh2 \right)^2 \right]^{\frac{1}{2}}$$

$$=\left[\left(\frac{h_2-h_4}{(h_2-h_1)^2}\times dh_1\right)^2+\left(\frac{h_1-h_4}{(h_2-h_1)^2}\quad\times dh_2\right)^2+\left\{(\frac{1}{h2-h1})\times dh_4\right\}^2\right]^{1/2}$$

$$=\left[\begin{array}{l}\left(\frac{450.55_2-253.5_4}{(450.699-406.55)^2}\times 0.2032\right)^2+\left(\frac{406.55-253.5}{(450.699-406.55)^2}\quad\times 0.2253\right)^2\\+\left\{(\frac{1}{450.699-406.55})\times 0.12675\right\}^2\end{array}\right]^{1/2}$$

=0.02873 kJ

% of error/uncertainty in COP = $\frac{0.02873}{3.6063}$

=0.7574%

APPENDIX C: RELEVANT PUBLICATIONS

Published and Accepted Journal Papers:

1. **J. U. Ahamed,** R. Saidur, H. H. Masjuki, 2011.A review on exergy analysis of vapor compression refrigeration system. Journal of Renewable and Sustainable Energy Reviews. Vol. **15**(3) pp. 1593-1600. (ISI cited, Q1).

2. **J. U. Ahamed**, Saidur R.,   H. H. Masjuki**,** Energy and exergy analysis of vapor compression refrigeration system using hydrocarbon as refrigerant. International Journal of Green Energy. **DOI:** 10.1080/15435075.2011.621491 Accepted (ISI-Q3).

3. **J.U. Ahamed**, R. Saidur, H.H. Masjuki and S. Mehzabeen Prospect of hydrocarbon uses based on exergy analysis in the vapor compression refrigeration system, International Journal of Renewable Energy Research (IJRER), Vol.1 (2011), No.2, 68-71

Journal Papers Under Review

1. **J. U. Ahamed**, Saidur R, Masjuki, H. H.Review on hydrocarbons as refrigerant: Based on thermodynamic and environmental analysis. Journal of Renewable and Sustainable Energy Reviews. Manuscript Number: RSER-D-11-00827. (ISI cited, Q1).

2. **J U Ahamed**, H. H. Masjuki, R. Saidur, Effect of operating temperatures and refrigerants on environmental and heat transfer performance of air conditioner for hydrocarbon mixture with R22. International Journal of Refrigeration. Paper ID number: JIJR-D-11-00368. (ISI cited, Q1).

3. **J. U. Ahamed,** Saidur R, Masjuki H. H. M. A**.,** Energy, Thermodynamic and environmental performance of hydrocarbon as refrigerant. Journal of Energy Conversion and Management. (ISI cited. Q1).

4. **J. U. Ahamed**, H. H. Masjuki, R. Saidur, Effect of operating temperatures on Energy savings of domestic air conditioner using R22 as refrigerant: An experimental investigation, Energy Education Science and Technology Part A: Energy Science and Research. (ISI cited, Q1).

Conference/Seminars:

1. **J. U. Ahamed**, M. M. K. Bhuiya, R Saidur, H H Masjuki, Prospect of Hydrocarbon Uses Based on Exergy Analysis in the Vapor Compression Refrigeration System, Prec. Of IEEE, Clean Energy and Technology, UMPEDAC Research Centre, University Malaya, 27-29 June, 2011, pp. 300-304.
2. **J U Ahamed,** R Saidur, H. H. Masjuki, M.M. K. Bhuiya, Alternative refrigerant of domestic air conditioner system based on exergy analysis. International Conference on Mechanical Engineering and Renewable Energy (ICMERE - 2011), CUET, 22-24 December, 2011, Paper Number: PI-01.

REFERENCES

ACRIB. (2001). Guidelines for the use of Hydrocarbon Refrigerants in Static Refrigeration and Air Conditioning Systems, Air Conditioning and Refrigeration Industry Board, Kelvin House, 76 Mill Lane, Carshaltan, Surrey. Web: www.ACRIB.org.uk.

Adnot, J., Lopes, C., & Cauret. (2000). *Expanding the electricity chain: the provision of active energy services in the peculiar French system", - Electricity for a sustainable urban development November 2000. In Proceedings of the UIE International Conference, Lisbon. .*

Ahamed, J. U., Saidur, R., & Masjuki, H. H. (2011). A Review on Exergy analysis of vapor compression refrigeration system. *Renewable and Sustainable Energy Reviews, 15*(3), 1593-1600.

Akash, B. A., & Said, S. A. (2003). Assessment of LPG as a possible alternative to R-12 in domestic refrigerators *Energy Conversion and Management 44*, 381-388.

Akintunde, M. A. (2011). Validation of a vapor compression refrigeration system design model. *American Journal of Scientific and Industrial Research, 2*(4), 504-510.

Alsaad, M., & Hammad, M. (1998). The application of propane/butane mixture for domestic refrigerators. *Applied Thermal Engineering 18,* 911-918.

Aprea, C., & Greco, A. (2002). An exergetic Analytic of R-22 substitution. *Applied Thermal Enineering, 22*, 1455-1469.

Arcaklioglu, E. (2004). Performance comparison of CFCs with their substitutes using artificial neural networks. *International Journal of Energy Research, 28*(12), 1113-1125.

Arcaklioglu, E., Cavosuglu, A., & Erisen, A. (2005). An algorithmic approach towards finding better refrigerant substitutes of CFCs in terms of the second law of thermodynamics. Energy Conversion and Management, 46; 1595-1611.

Arora, A., Arora, B. B., Pathak , B. D., & Sachdev, H. L. (2007). Exergy analysis of a Vapour Compression Refrigeration system with R-22, R-407C and R-410A. *International Journal of Exergy 4,* 441 - 454

Arora, A., & Kaushik, S. C. (2008). Theoritical Analysis of vapor compreession refrigeration system with R502, R404A and R507A. *Inernational Journal of Refrigeration, 31*, 998-1005.

ASHRAE 34 (Ed.). (1997). *Refrigerant Data and Safety Classifications*. Atlanta: American Society of Heating, Refrigerating and Air - Conditioning Engineering. Incorporation.

AZR. (2011). Practical efficieny or COP. Retrieved on 13 Aprill, 2011. Website:http://www.alephzero.co.uk/ref/practcop.htm.

Bayrakci, H. C., & Ozgur, A. E. (2009). Energy and exergy analysis of vapor compression refrigeration system using pure hydrocarbon refrigerants. *International Journal of Energy Research, 33*(12), 1070-1075.

Bejan, A. (1989). Theory of heat transfer-irreversible refrigeration plants *International Journal of Heat Mass Transfer, 32*(9), 1631-1639.

Bejan, A. (1997). Advanced Engineering Thermodynamics. 2nd Edition, Wiley: New York, p. 888.

Beyerlein, A., DesMarteau, D., HWang,, Smith, N., & Joyner, P. (1993). Physical properties of fluorinated propane and butane derivatives as alternative refrigerants. *ASHRAE Transaction, 99*, 368-379.

Bi, S.-s., Shi, L., & Zhang, L.-l. (2008). Application of nanoparticles in domestic refrigerators. *Applied Thermal Engineering*, *28*(14-15), 1834-1843.

Brown, S. (1996). Single-site and industrial-scale schemes. *Applied Energy 53*, 149–155.

Carl, B. A. (2008). *Variable Condition Energy Efficiency Rating of a Room Air Conditioner.* Unpublished Masters Of Engineering Thesis, The University of Texas, San Antonio.

Cavallini, A. (1996). Working fluids for mechanical refrigeration. *International Journal of Refrigeration, 19,* 485-496.

Cengel, Y. A. (2007). Green thermodynamics. *International Journal of Energy Research*, *31*(12), 1088-1104.

Choi, D. K., Domanski, P. A., & Didion, D. A. (1996). *Evaluation of flammable refrigrants for use in water -to-water residential heat pump. In. Proceedings of IIR applications for natural refrigerants, Aarhus, Denmark. p. 467-76*

Corberan, J. M., Segurado, J., Colbourne, D., & Gonzalvez, J. (2008). Review of standards for the use of hydrocarbon refrigerants in A/C, heat pumps and refrigeration equipment. *International Journal of Refrigeration*, *31*(4), 748-756.

Cornelissen, R. L. (1997). Thermodynamics and sustainable development. PhD Thesis, University of Twenty, The Netherlands.

Devotta, S., Padalkar, A. V., & Sane, N. K. (2005). Performance assessment of HC-290 as a drop-in substitute to HCFC-22 in a window air conditioner. *International Journal of Refrigeration, 28*(4), 594-604.

Devotta, S., Padalkarb, A. S., & Sane, N. K. (2005b). Performance assessment of HCFC-22 window air cnditioner retrofitted with R-407C *Applied Thermal Engineering, 25*, 2937-2949.

Devotta, S., Waghmare, A. V., Sawant, N. N., & Domkundwar, B. M. (2001). Alternatives to HCFC-22 for air conditioners. *Applied Thermal Engineering, 21*, 703-715.

DOE. (2007). U. S. Department of Energy Report: Anual Energy Outlook 2007, Document Number: DOE/EIA-0383 (2007). Available at: Available at: www.EIA.DOE.govt.

Dosat, R. J., & Haron, T. J. (2002). *Principle of Refrigeration* (Fifth ed.): Prentice Hall, New Jersey, USA

DuPont, P., & Vauvert, J. (2004). *Benchmarking Air Conditioner Performance: Results of a 5- Country Comparison APEC Conference Proceedings- Air Conditioning and Energy Performance, Sydney, Australia*.

Edgerton, R. H. (1992). Available Energy and Environmental Economics. D. C. Heath, Toronto.

EIA. (2011). World Energy Demand and Economic Outlook: U. S. Energy Administration: Independant Statistics and Analysis. Retrieved on June, 2011. Available at:http://www.EIA.gov/oiaf/archive/aeo10/index.html.

Fatouh, M., & EI Kafafy, M. (2006). Essessment of propane/ commercial butane mixtures as possible alternatives to R134a in domestic refrigerators. Energy Conversion and Management 47, 2644-2658.

Gong, G., Zeng, W., Wang, L., & Wu, C. (2008). A new heat recovery technique for air-conditioning/heat-pump system. *Applied Thermal Engineering 28*, 2360-2370.

Good, D. A., Fransisco, J. S., Jain, A. K., & Wuebles, D. J. (1998). Lifetimes and global warming potentials for dimethyl ether and for flourinated ethers:CH3OCF3 (E143a), CHF2OCHF2(E134), CHF2OCF3(E125). *Journal of Geophysical Research, 103 (D21)*, 9230-9240.

Granryd, E. (2001). Hydrocarbon as refrigerants -an overview. *International Journal of Refrigeration, 24*, 15-24.

Grant, M., Wicks, F., & Wilk, R. (2001). *Identification and Analysis of Pshychometric means for enhancing Air Conditioner Efficiency and Capacity* Paper presented at the IECEC-36th Intersociety Energy Conversion Engineering Confeence; Proceedings.

Hamilton, R. L., & Crosser, O. K. (1962). Termal conductivity of heterogeneous two-component systems. *Industrial and Engineering Chemistry Fundamentals, 1*(3), 187-191.

Hammad, M. A., & Alsaad, M. A. (1999). The use of hydrocarbon mixtures as refrigerants in domestic refrigerators. *Applied Thermal Engineering, 19*, 1181-1189.

Hasanuzzaman, M., Saidur, R., & Masjuki, H. H. (2008). Investigation of energy consumption and energy savings of refrigerator-freezer during open and closed door condition. J. Appl. Sci., 8(10): 1822-1831.

Hepbasli, A. (2007). Exergetic Modeling and Assessment of Solar Assisted Domestic Tank Integrated Ground Source Heat Pump Systems for Residences. *Energy and Buildings 39*(12), 1211-1217.

Hepbasli, A. (2008). A key review on Exergetic Analysis and Assessment of Renewable Energy Resources for sustainable Future. *Renewable and Sustainable Energy Reviews 12*(3), 593-561.

Highwood, E. J., & Shine, K. P. (2000). Radiative forcing and global warming potentials of 11 halogenated compounds. *Journal of Quantitative Spectroscopy and Radiative Transfer, 66*, 169-183.

Honaka, T., Ishitani, H., Matsuhashi, R., & Yoshida, Y. (2002). Recovery of flourocarbons in Japan as a measure for abating global warming *Applied Energy, 72*, 705-721.

Hychill. (2010). The Facts on Hydrocarbon Refrigerants. Retrived on 10 th November, 2010. Retrieved on June, 2010. Available at: website:www.hychill.com.

IPCC. (2006). Intergovernmental Panel on Climate Change: Special Report on Safegaurding the Ozone Layer and the Global Climate System; Issues related to Hydroflourocarbons and Perflourocarbons. Cambidge University Press, available at: www.wmo.ch.

Jafar, M. F. Z. B. (2010). The Case Of Air Conditioning Use In Malaysian Homes,Proceedings of Conference:Adapting to Change: New Thinking on Comfort Cumberland Lodge, Windsor, UK, 9-11 April, London.

Jahanson, A. (2003). *Phase out of Refrigrrant R22*. Unpublished PhD thesis, Royal Institute of Technology, Stockholm, Sweden.

Jiangtao, W., Yingjie, C., Jing, H., & Zhigang, L. (2009). Performance of mixture refrigerant R152a/R125/R32 in domestic air conditioner. *International Journal of Refrigeration, 32*(5), 1049-1057.

Jose, M., Corberan, Seguradoa, J., ColbourNEB, D., & lvez, J. G. (2008). Review of standards for the use of hydrocarbon refrigerants in A/C, heat pump and refrigeration equipment. *International Journal of Refrigeration, 31*, 748-756.

Jung, D., Kim, C., Lim, B., & Lee, H. (1996). Testing of a hydrocarbon mixture in domestic refrigerators *ASHRAE Trans, 102*, 1077-1084.

Jung, D., Kim, C. B., Song, K., & Park, B. (2000). Testing of propane iso-butane mixture in domestic refrigerants *International Journal of Refrigeration, 23*, 517-527.

Kabul, A., Kizilkan, O., & Yakut, A. K. (2008). Performance and exergetic analysis of vapor compression refrigeration system with an internal heat exchanger using a hydrocarbon, isobutane (R600a). *Inernational Journal of Energy Research, 32*, 824-836.

Kalaiselvam, S., & Saravanan, R. (2009). Exergy analysis of Scroll Compressors working with R22, R407 and R717 as Refrigerant for HVAC System. *Thermal Science, 13*(1), 175-184.

Kanoglu, M. (2002). Exergy analysis of the multistage cascade refrigeration cycle used for natural gas liquefaction *Internattional Journal of Energy Research, 26*(8), 763-774.

Karkri, M., Boussehain, R., & Feidt, M. L. (2009). Frederic Sicard: Exergy analysis of a vapour-compression refrigerating system using 410A as refrigerant. *International Journal of Exergy, 6*(3), 295-322.

Kedzierski, M. A., & Gong, M. (2007). Effect of CuO Nanolubricant on R134a Pool Boiling Heat Transfer with Extensive measurement and Analysis Details. NISTIR 7454, U.S. Department of Commerce, Washington, DC.

Khan, S. H. (1992). Second law based thermodynamics analysis of vapor compression system. A thesis of Master of Science in Engineering, Dept of Mechanical Engg. King Fahad University of Petrolium and Minerals, Saudi Arabia.

Kilicarslan, A., & Hosoz, M. (2010). Energy and irreversibility analysis of a cascade refrigeration system for various refrigerant couples. *Energy Conversion and Management, 51*(12), 2947-2954.

Kim, S. G., Kim, M. S., & Ro, S. T. (2002). Experimental investigation of the performance of R22, R407C and R410A in several capillary tubes for air-conditioners. *International Journal of Refrigeration 25*, 521-531.

Kline, S. J., & McClintock, F. A. (1953). Describing Uncertainties in Single- Sample Experiments. *Mechanical Engineering, 75*, 3-8.

Kongkiatumpai, P. (1999). Study of impact of indoor setpoint temperature on energy consumptin of air-conditioner and green-house gases emission. A special study report for the Degree of Master of Engineering, School of Energy Technology Program, King Mongkut's University of Technology Thonburi, Thailand, p.28.

Kotas, T. J. (1985). THe Exergy Method of Thermal Plant AnalysisThe Exergy Method of Thermal Plant Analysis (Reprint Edition)Krieger, Malabar, FL, pp.72-73 (19).

Lee, Y. S., & Su, C. C. (2002). Experimental studies of isobutene (R600a) as refrigerant in domestic refrigeration system. Applied Thermal Engineering 22, 507-519.

Leidenfrost, W., Lee, K. H., & Korenic, K. H. (1980). Conservation of energy estimated by second law analysis of power-consuming process, Energy 5; 47–61.

Lesson 26. (2008). Lesson 26 :Refrigerants [Electronic Version]. *R and AC lecture 26, Version 1 ME, IIT Kharagpur. Available at:http://nptel.iitm.ac.in/courses/Webcourse-contents/IIT%20Kharagpur/Ref%20and%20Air%20Cond/pdf/R&AC%20Lecture%2026.pdf*, 1-14,

Liu, B. Y., Tomasek, M. L., & Radermacher, R. (1995). Experimental results with hydrocarbon mixtures in domestic refrigerator/freezers. *ASHRAE Transaction, 101*, 1415-1421.

Lorentzen, G. (1995). The use of natural refrigerants: a complete solution to the CFC/HFC predicament. *International Journal of Refrigeration, 18*, 190-197.

Maclaine-cross, I. L. (2004). Usage and Risk of hydrocarbon refrigerants in motor cars for Australia and the United States. *Int. J. of Refrig, 27*(4), 339-345.

Maclaine-Cross,& Leornadi, E. (1996). Comparative performances of hydrocarbon refrigerants, IIF-IIR-Commissions E2, E1, B1, B2-Melbourne (Australia), 11-14 February.

Mahlia, T. M. I., Masjuki, H. H., & Choudhury, I. A. (2002). Development of energy labels for room air conditioner in Malaysia: methodology and results *Energy Conversion and Management, 43*(15), 1985-1997.

Mansouri, I. (1996). Energy Consumption and Environmental Impacts of Electrical Appliances in UK households . PhD Thesis, Cranfield University, Bedford, UK.

Mehrpooya, M., Jarrahian, A., & Pishvaie, M. R. (2006). Simulation and exergy-method analysis of an industrial refrigeration cycle used in NGL recovery units. *International Journal of Energy Research, 30*, 1336-1351.

Mizuhara, K., Akei, M., & Matsuzaki, T. (1994). The friction and wear behavior in controlled alternative refrigerant atmospher *STLE TribologyTransaction, 37*(1), 120-128.

Mohanraj, M., Jayaraj, S., & Muraleedharan, C. (2007). Improved energy efficiency of a domestic refrigerator retrofitted with hydrocarbonrefrigerant mixture (HC290/HC600a) as drop in substitute. *Energy for SustainableDevelopment, 11*(29-33).

Molina, M. J., & Rowland, F. S. (1974). Stratospheric sink for chlorofluommethanes—chlorine atomic catalysed destruction of ozone. *Nature*, *249*, 810-812.

Moran, M. J. (1989). Availability Analysis — A Guide to Efficient Energy Use. *ASME*.

Moran, M. J., & Sciubba, E. (1994). Exergy Analysis: Principles and Practice. Journal of Engineering for Gas Turbines and Power 116, 285–290.

MP. (1987). Montreal Protocolon the substances that deplete the ozone layer. United Nations Environmental Programme, United Nations.

Murshed, S., Leong, K., & Yang, C. (2008). *Simultaneous measurement of the effective thermal conductivity and effective thermal diffusivity of nanofluids. , v PART A 2008.* Paper presented at the In: Proceedings of the ASME Micro/Nanoscale Heat Transfer International Conference, MNHT 2008.

NEB. (2007). *National Energy Balance Malaysia*: Ministry of Energy, Water and Communications, Malaysiao. Document Number)

Oh, T. H., & Chua, S. C. (2010). Energy efficiency and carbon trading potential in Malaysia. *Renewable and Sustainable Energy Reviews*, *14*(7), 2095-2103.

Padilla, M., Revellin, R., & Bonjour J. (2010). Energy Conversion and Management,. *Exergy analysis of R413a as replacement of R12in a domestic refrigeration system, 51*, 2195-2201.

Palm, B. (2008). Hydrocarbons as refrigerants in small heat pump and refrigeration systems - A review. *International Journal of Refrigeration, 31*, 552-563.

Park, K. J., &Jung, (2008). Performance of R290 and R1270 for R22 applications with evaporator and condenser temperature variation. *Journal of Mechanical Science and Technology 22,* 532~537.

Park, K. J., &Jung, S. (2006). Thermodynamic performance of HCFC 22 alternative refrigerants for residential air conditioning applications. *Energy and Buildings, 39*, 675-680.

Park, K. J., &Jung, S. (2007). Thermodynamic performance of R502 alternative refrigerant mixtures for low temperature and ransport applications. *Energy Conversion amd Management*, *48*(12), 3084-3089.

Payne, W. V., & Domanski, P. A. (2011). A compaison of an R22 and an R410A air conditioner operating at high ambient temperatures. Available at: web.mit.edu/nnf/publications/GHM114.pdf.

Pelletier, O., & Palm, B. (1996). Propane for heat pump applications using brazed plate heat exchangers, Report number 3. (8).

Peng, H., Ding, G., Jiang, W., Hu, H., & Gao, Y. (2009). Heat transfercharacteristicsof refrigerant-basednanofluid flowboilinginsude a horizontal smooth tybe. *International Journal of Refrigeration*, *32*, 1259-1270.

Philip, C. H. Y. (2001). A Study of Energy Use for Ventilaion and Air Conditioning System in Hong Kong. PhD thesis, Department of Mechanical Engineering, The Hong Kong Polytechnic University, p. 294.

Purkayastha, B., & Bansal, P. K. (1998). An experimental study on HC290 and a commercial liquefied petroleum gas (LPG) mix as suitable replacements for HCFC22. *International Journal Refrigeration*, *21*(1), 3-17.

Radermacher, R., &Jung, (1993). Theoretical analysis of replacement refrigerants for R22 for residential uses. *ASHRAE Transaction*, *99*(1), 333-343.

Radermacher, R., & Kim, K. (1996). Domestic refrigeration: Recent development. *Internaional Journal of Refrigeration*, *19*, 61-69.

Rogstam, J. (1996). Application for natural refrigerants, Proceedinngs of IIF/IIR Conference, Arhus, Denmark, p. 457-466

Rosen, M. A., Dincer, I., & Kanoglu, M. (2008). Role of exergy in increasing efficiency and sustainability and reducing environmental impact, Energy Policy 36,128-137.

Rosenquist, G. J. (1998). Energy Conservation Standards for Room Air Conditioners *ASHRAE Transaction*, *104*, 817-832.

Sachdev, H. L. (2008). Performance analysis of R22 and its alternative refrigerants, Masters Thesis, Department of Mechanical Engineering, University of Delhi, Delhi, India, p.145.

Sahin, A., Dincer, I., & Rosen, M. A. (2005). Thermodynamic analysis of wind energy. International Journal of Energy Research, vol.30 No:8, pp.553-566..

Saidur, R. (2008). Energy, exergy and emission analysis in Malaysian residential sector. PhD Thesis, Mechanical Engineering Department, University of malaya, Kuala Lumpur, p. 189.

Saidur, R. (2009). Energy consumption, energy savings, and emission analysis in Malaysian office buildings. *Energy Policy, 37*, 4104-4113.

Saidur, R. (2010). A Review on Electrical Motors Enedrgy Use and Energy Savings. *Renewable and Sustainable Energy Reviews, 14*, 877-898.

Saidur, R., Ahamed, J. U., & Masjuki, H. H. (2010). Energy, exergy and economic analysis of industrial boilers. *Energy Policy, 38*(5), 1188-1197.

Saidur, R., Masjuki, H. H., & Jamaluddin, M. Y. (2007). An application of energy and exergy analysis in residential sector in Malaysia. *Energy Policy, 35*, 1050-1063.

Saidur, R., Rahim, N. A., Masjuki, H. H., Mekhilef, S., Ping, H. W., & Jamaluddin, M. F. (2009). End-use energy analysis in the Malaysian industrial sector. *Energy, 34* (2), 153-158.

Sattar, M. A. (2008). Performance investigation of domestic refrigerator using hydrocarbons and mixture of hydrocarbons as refrigerant. A Thesis of Masters in Science and Engineering, Dept. of Mech. Engg. University of Malaya, Malaysia.

Sattar, M. A., Saidur, R., & Masjuki, H. H. (2007). Performance investigation of domestic refrigerator using pure hydrocarbons and blends of hydrocarbons as refrigerants. *World Academy of Science, Engineering and Technology, 29*, 223-228.

Sattar, M. A., Saidur, R., & Masjuki, H. H. (2008). Pure Butane as Refrigerant in Domestic Refrigerator-Freezer, GMSARN International Journal. 2(1) 29-34. Available at: www.waset.org.

Schwarz, W. (2005). Emission, Aktivitatsraten and Emissions faktoren von flourierten Treibhausgasen (F-gasen) in Deutshland fur die Jahre 1995-2002, Umweltbundesamt, UBA-Texte 4/05.

Sekhar, S. J., & Lal, D. M. (2005). R134a/R600a/R290 a retrofit mixture for CFC12 systems *International Journal of Refrigeration, 28*, 735-743.

Sheng-shan, B., Lin, S., & Zhang, L. L. (2008). Application of nanoparticles in domestic refrigerators. *Applied Thermal Engineering, 28*(14-15), 1834-1843.

Shilliday, J. A., Tassou, S. A., & Shilliday, N. (2009). Comparative energy and exergy analysis of R744, R404A and R290 refrigeration cycles. *International Journal of Low-Carbon Technologies, Advance Access published June,16*, 1-8.

Somasundaram, P., Dinakaran, R., Iniyan, S., & Samuel, A. A. (2004). Exergy based refrigerant selection and simulation of auto refrigeration cascade (ARC) system. *International Journal of Exergy, 1*(1), 60-81.

Somchai, W., & Nares, C. (2005). Experimental study of hydrocarbon mixtures to replace HFC-134a in a domestic refrigerator. *Energy Conversion and Management, 46*, 85-100.

Stegou-Sagia, A., & Paignigiannis, N. (2003). Exergy losses in refrigerating systems: A study for performance comparison in compressor and condenser. *International Journal of Energy Research 27*(12), 1067-1078.

Szargut, J. (1980). International progress in second law analysis. Energy 5, 709-718.

Szargut, J., Morris, D. R., & Steward, F. R. (1988). Exergy Analysis of Thermal, Chemical and Metallurgical Processes. Hemisphere, New York.

Tashtoush, B., Tahat, M., & Shudeifat, M. A. (2002). Experimental study of new refrigerant mixtures to replace R12 in domestic refrigerator *Applied Thermal Engineering, 22*, 495-506.

Taufiq, B. N., Masjuki, H. H., Mahlia, T. M. I., Amalina, M. A., Faizul, M. S., & Saidur, R. (2007). Exergy analysis of evaporative cooling for reducing energy use in a Malaysian building. *Desalination 209*, 238-243.

Teng, T. P., Mo, H. E., Lin, H., Tseng, Y. H., Liu, R. H., & Long, Y. F. (2012). Retrofit assessment of window air conditioner. *Applied Thermal Engineering, 32*, 100-107.

UNDP. (1991). United Nations Development Programme, Study on the Potential for Hydrocarbon Replacements in Existing Domestic and Small Commercial Refrigeration Appliances. United Nations Publication ISBN 92-807-1765-0(1991).

UNEP. (1987). United Nations Environmental Programme: Montreal Protocol on Substances that Depletes the Ozone Layer, Final Act, New York, USA.

United Nations Development Programme. (1999). Study on the Potential for Hydrocarbon Replacements in Existing Domestic and Small Commercial Refrigeration Appliances. United Nations Publication ISBN 92-807-1765-0.

USDE. (2011). U. S. Deparment of Energy: Energy efficiency and renewable energy, Room air Conditioner. Available at:http://www.energysavers.gov/your_home/space_heating_cooling/index.cfm/mytopic=12420.

Venkataramanamurthy, V. P., & Kumar, P. S. (2010). Experimental Comparative energy, exergy flow and second law efficiency analysis of R22, R436b vapor compression refrigeration cycles, . *International Journal of Science and Technology. , 2* (5), 1399-1412.

Wang, R. X., & Xie, H. B. (2003). *A refrigerating system using HFC134a and mineral lubricant appended with n-TiO2(R) as working fluids.* Paper presented at the Proceedings of the 4th International Symposium on HAVC, Beijing, China.

Wang,, Dicskson, T., Dexter, J., & Mcgragor, I. (2005). Vapor quality and performance of an automotive air conditioning system. *Experimental Thermal and Fluid Science, 30*(1), 59-66.

Wen, D. S., & Ding, Y. L. (2005). Experimental investigation into the pool boiling heat transfer of aqueous based c-alumina nanofluids *Journal of Nano particle Research 7*, 265-274.

Wing, Y. F. (2004). Optimum condensing temperature at part-load operation of air-cooled chillers for energy efficiency. PhD Thesis, Department of Building Service Engineering,The Hong Kong Polytechnic University, Hong Kong, p.281.

Wongwises, S., & Chimres, N. (2005). Experimental study of hydrocarbon mixtures to replace HFC 134a in a domestic refrigerator *Energy Conversion and Management, 46*, 85-100.

Yamtraipat, N., Khedari, J., & Hirunlabh, J. (2003). Thailand Thermal confort standards of air-conditioning buildings, Proceedings of the 1st International Conference on Sustainable Energy and Green Architecture, Bangkok, Thailand, GA, pp. 36-41.

Yamtraipat, N., Khedari, J., Hirunlabh, J., & Kunhhornrat, J. (2004). Assessment of Thailand indoor set point impact on energy consumption and environment. Available at: http://www.bsrc.net

Yilmaz, N., Sara, O. N., & Karsli, S. (2001). Performance evaluation criteria for heat exchanger based on second law analysis. *Journal of Exergy: An International Journal, 4, 278-294.*

Yumrutas, R., Kunduz, M., & Kanoglu, M. (2002). Exergy analysis of vapor compression refrigeration systems, Exergy, an International Journal 2, 266-272.

Zhentao, W. (2006). *Analysis of Energy, Exergy and Green House Gas emissions for Variable Air Volume Systems in an Office Building, Masters of Applied Science, Building, Departement of Civil and Envionmental Engineering, Concordia University, Montreal, Quebec, p.158.* Unpublished MastersThesis.

Zhu, Y. X., & Lin, B. (2004). Sustainable housing and urban construction in China. *Energy and Buildings, 36*, 1287-1297.