

PERFORMANCE OPTIMIZATION OF DOMESTIC REFRIGERATOR USING
EXPERIMENTAL METHOD

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SUPERVISOR'S DECLARATION

I hereby declare that I have checked this project and in my opinion, this project is adequate in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering.

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STUDENT'S DECLARATION

I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

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Dedicated to my beloved parents

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ABSTRACT

This thesis deals with optimization performance testing for a domestic refrigerator using experimental method. As the refrigeration system has become one of the basic needs to us, the actual working principle is essential to be known by us so that we can always optimize the system performance and minimize the resources used. The objectives of this thesis is to improve a previous developed mini domestic refrigerator test rig by modifying the configuration of its tubing system, investigate the effect of refrigerant quantity on refrigeration system, and optimization performance of the mini domestic refrigerator test rig. The thesis describes the step by step fabrication of the tubing system of the mentioned test rig to achieve a simplest tubing system configuration for this refrigerator test rig. Vapor compression refrigeration system was studied as it is the most widely used refrigeration system especially in small scale of refrigeration application such as in domestic application. The design of the simpler tubing system will be depend on the location of other components in refrigeration system are located. The previous developed lengthy tubing system was greatly reduced in length by removing the thermocouple wires from the tubing system and moved it to the outer surface of the tubing. The improved refrigerator test rig was run and observed by using different quantity of refrigerant. The pressures and temperatures at the points of interest were observed and collected for all quantity of refrigerant. Finally, obtained the enthalpies for each of the points of interest and the optimum refrigeration coefficient of performance, COP_R , by using equations given. From the results, it is observed that the quantity of refrigerant does give significant effect on the performance of the refrigeration system. The obtained results indicate that there is a optimum point of refrigerant quantity for a particular refrigeration system depends on the capacity of the components in the system. The results concluded that the optimum performance achieved at the 35 psi initial charged refrigerant quantity which is at the lower refrigerant quantity out of the testing range of 35 to 45 psi initial refrigerant charged quantity. The results can contribute to understanding of the actual working principle of refrigeration system.

ABSTRAK

Tesis ini membentangkan kecekapan optimum bagi suatu peti sejuk domestik dengan menjalankan eksperimen. Oleh kerana sistem penyejukbekuan telah menjadi salah satu keperluan asas bagi kita, oleh itu prinsip berfungsi sebenar bagi sistem penyejukbekuan adalah penting untuk pengetahuan kita supaya kita dapat selalu mengoptimumkan kecekapan bagi suatu sistem penyejukbekuan dan meminimumkan kegunaan bahan mental. Objektif bagi tesis ini adalah untuk mempertingkatkan peti sejuk domestik kecil yang dibina sebelum ini dengan mengubah susunan sistem tiub-nya, mengkaji kesan kuantiti bahan penyejukan atas sistem penyejukbekuan, dan mengoptimumkan kecekapan peti sejuk kecil uji tersebut. Tesis ini menerangkan proses penghasilan sistem tiub yang baru dengan langkah demi langkah untuk mencapai susunan sistem tiub yang paling mudah bagi peti sejuk uji ini. Kemampatan wap sistem penyejukan dipelajari oleh sebab ianya adalah sistem yang paling banyak digunakan terutamanya di dalam sistem kecil seperti sistem bagi kegunaan domestik. Reka bentuk bagi susunan sistem tiub yang mudah adalah bergantung pada kedudukan komponen-komponen lain dalam sistem penyejukan tersebut. Sistem tiub yang panjang dibina sebelum ini telah banyak dikurangkan panjangnya dengan memindah termoganding yang sebelum dipasang di dalam sistem tiub ke permukaan tiub-tiub. Peti sejuk uji yang telah dipertingkatkan kualitinya dihidupkan dan diperhatikan dengan menggunakan kuantiti bahan penyejukan yang berbeza. Tekanan dan suhu di titik-titik yang berminat diperhatikan dan dicatat untuk semua kuantiti bahan penyejukan yang diuji. Akhirnya dapatkan enthalpi untuk setiap titik yang berminat dan koefisien kecekapan penyejukan optimum, COP_R , dengan menggunakan persamaan-persamaan dibagi. Dari keputusan, ia menunjukkan bahawa kuantiti bahan penyejukan dapat memberi kesan yang nyata atas kecekapan bagi suatu sistem penyejukbekuan. Keputusan yang diperolehi menunjukkan terdapat satu titik optimum kuantiti bahan penyejukan bagi sesuatu sistem penyejukbekuan yang tertentu dan ia bergantung pada kebolehan komponen-komponen dalam sistem tersebut. Keputusan diperolehi memutuskan kecekapan optimum tercapai pada 35 psi kuantiti awal isi bahan penyejukan yang mana ia ialah yang paling kurang di antara julat 35 psi dan 45 psi kuantiti awal isi bahan penyejukan yang diujikajikan dalam eksperimen ini.. Keputusan ini dapat menyumbang ke atas kefahaman dalam prinsip berfungsi sebenar dalam sistem penyejukbekuan.

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LIST OF SYMBOLS

COP	Coefficient of performance
COP_R	Refrigeration coefficient of performance
h	Enthalpy
\dot{m}	Mass flow rate
p	Pressure
Q_H	Heat rejection per unit mass
\dot{Q}_H	Heating capacity
Q_L	Refrigerating effect
\dot{Q}_L	Refrigeration capacity
s	Entropy
T	Temperature
T_H	High temperature
T_L	Low temperature
V	Voltage
W_c	Compressor work per unit mass
\dot{W}_c	Compressor work rate

LIST OF ABBREVIATIONS

CFCs	Chlorofluorocarbons
CFM	Coriolis flow meter
EMF	Electromotive force
EPA	Environmental Protection Agency
GWP	Global warming potential
HCFCs	Hydrochlorofluorocarbons
<i>J</i>	Thermocouple junction
MAPP	Methylacetylene-propadiene
R	Refrigerant
UMP	Universiti Malaysia Pahang

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

Refrigeration may be defined as the process to achieve and keep an enclosed space at a temperature lower than its surrounding temperature. This is done by continuous extraction of heat from the enclosed space whereas the temperature is below than that of the surrounding temperature.

Nowadays refrigeration is something that is indispensable in our daily life. One of the most important applications is the preservation of perishable foods and keeps the food in fresh condition. There is no doubt that food, is just like air and water are necessities for livings. People often utilize refrigeration to chill their drinks, making it more scrumptious. In additional, refrigeration also being used in providing thermal comfort to people by means of air conditioning process.

Historically, it is generally agreed that the first refrigeration machine was introduced in 1755 which was made by Scottish professor William Cullen. However, he did not use his discovery for any practical purpose. In the following 50 years, an American inventor, Oliver Evans, designed the first refrigeration machine. An American physician, John Gorrie, built a refrigerator based on Oliver Evans' design in 1844 to make ice to cool the air for his yellow fever patients. A German engineer named Carl von Linden patented not a refrigerator but the process of liquefying gas in 1876 that is part of basic refrigeration technology.

Generally refrigeration systems can be classified in 3 main cycle systems which are vapor compression refrigeration system, vapor absorption refrigeration system, and gas cycle refrigeration system. However the vapor compression refrigeration system is the most widely used in the refrigeration process. It is adequate for most refrigeration applications. The ordinary vapor compression refrigeration systems are simple, inexpensive, reliable and practically maintenance free.

However for large industrial applications, other refrigeration systems will be used to fulfill the effectiveness need. Jacob Perkins, an American living in London actually had designed the vapor compression refrigeration system and was built by Jacob Perkins in 1835 and had received his patent in 1834. The first practical vapor compression refrigeration system was made by James Harrison who took a patent in 1856 for vapor compression refrigeration system using ether, alcohol or ammonia as refrigerant.

Most of the domestic refrigerators today are running based on the vapor compression refrigeration system. It is somewhat analogous to a reverse Rankine cycle. The vapor compression refrigeration system contains four main components which are compressor, condenser, expansion device, and evaporator (Schmidt, 2006).

Compressor is used to compress the low pressure and low temperature of refrigerant from the evaporator to high pressure and high temperature. After the compression process the refrigerant is then discharge into condenser. In the condenser, the condensation process requires heat rejection to the surroundings. The refrigerant can be condensed at atmospheric temperature by increasing the refrigerant's pressure and temperature above the atmospheric temperature.

After the condensation process, the condensed refrigerant will flow into the expansion device, where the temperature of refrigerant will be dropped lower than the surrounding temperature caused by the reducing pressure inside the expansion device. When the pressure drops, the refrigerant vapor will expand. As the vapor expands, it draws the energy from its surroundings or the medium in contact with it and thus produces refrigeration effect to its surroundings. After this process, the refrigerant is

ready to absorb heat from the space to be refrigerated. The heat absorption process is to be done in the evaporator. The heat absorption process is normally being called as evaporation process. The cycle is completed when the refrigerant returns to the suction line of the compressor after the evaporation process.

The performance of the domestic refrigerator is to be analyzed by using experimental method and attempt to improve and achieve the maximum performance for a unit of domestic refrigerator. In order to have more accurate results for analyzing the performance of the domestic refrigerator, the suitable locations of parameters to be recorded down to determine the performance of the domestic refrigerator is crucial to be identified. The experiment is carried out by using the previous developed test rig (mini size domestic refrigerator). The test rig is improved and modified if necessary.

Several different charge quantities of refrigerant will also be tested in the system and its effect on the performance of the refrigeration system is observed. This report is to give the basic understanding on vapor compression refrigeration system to the readers.

1.2 PROBLEM STATEMENT

As refrigeration has become one of the basic needs to modern people, it is important to know the actual working principle of the domestic refrigerator so that the users can always maintain the refrigeration system at its maximum performance. Therefore by understanding the working principle of the refrigeration system, it will allow the users to make the best use of the domestic refrigerator without wasting any electricity and materials used. There are 2 problem statements in this project, first is that study is needed to analyze the actual performance of refrigerator and second, is to determine the optimum COP_R by using different quantities of refrigerant charges on the improved previous developed refrigerator test rig.

1.3 OBJECTIVES

The main objective of this report is to improve the configuration of the refrigerator test rig to a simpler configuration. It is to avoid any significant pressure

drop and heat transfer to and from the system throughout the refrigeration cycle and the effect of the charge quantity of the refrigerant on the refrigeration system. Finally, obtain the optimum COP_R by using the data collected from the experiment. To be able to do this, the exact locations of the points of interest at where the data (temperature and pressure) should be collected must be identified correctly.

1.4 SCOPES

1.4.1 Literature study

The literature study is mainly focused on the fundamental of working principle of vapor compression refrigeration cycle. The working principle of each of the 4 main components, compressor, condenser, expansion device and evaporator are also in the region of concerned. In this section, however, there is also a review of some techniques on how to fabricate the refrigeration tubing system.

1.4.2 Thermodynamics analysis

Clear understanding of vapor compression refrigeration cycle is needed so that the conditions of each critical state in the vapor compression refrigeration cycle can be easily determine and analyze. The temperature and pressure at the predetermined states or points in the cycle are the parameters of concern in the experiment of this project.

1.4.3 Improvement of experiment test rig

Set of experiments are conducted using previous developed refrigeration system test rig (mini size domestic refrigerator) in order to identify the best location where the parameters of concern will be recorded for the analysis of performance of the domestic refrigerator. Then the optimum COP_R of the test rig is obtained. Improvement and modification of the test rig should be done when it is necessary to get better results.

1.4.4 Testing and analysis

Analyze the data collected from the experiment by using pressure-enthalpy ($p-h$) diagram and determine the COP_R by using the second law of thermodynamics. The results are then discussed.

CHAPTER 2

LITERATURE REVIEW

2.1 THE SECOND LAW OF THERMODYNAMICS

There are 2 classical statements of second law of thermodynamics which are the Kelvin-Planck statement and the Clausius statement. Both of Kelvin-Planck and Clausius statements are 2 equivalent expressions of the second law of thermodynamics. For refrigerators or heat pump, Clausius statement is being related to which is expressed as “It is impossible to construct a device that operates in a cycle and produces no effect other than the transfer of heat from a lower-temperature body to a higher temperature-body” (Cengel, 2008).

There is commonsense that heat does not naturally transfer on its own from a colder medium to a warmer medium. The Clausius statement simply means that if a cyclic device that transfers heat from a colder medium to a warmer medium will be impossible to be achieved or construct, unless this cyclic device produce a net effect on other (Cengel, 2008).

For an example, a cyclic device that transfers heat from a cold medium to a warmer one has long been constructed which is the domestic refrigerator. A domestic refrigerator is in complete compliance with the Clausius statement of the second law of thermodynamics. The Clausius statement simply states that a refrigerator can not operate unless its compressor is driven by an external power source, such as an electric motor. In this case, the compressor leaves a trace in the surroundings by consuming some energy in the form of work by the electric motor so that to transfer heat from the colder body to a warmer one (Cengel, 2008).

2.2 REFRIGERANTS

Over the last decade, the choice of refrigerant used in a refrigeration system has been becoming a worldwide issue as mainly in response to the environmental issues of “holes in the ozone layer” and “global warming or greenhouse effect”.

Previously people had no much discussion on the selection of refrigerant. The refrigerants chosen were all based on the capability of heat absorption and releasing of the fluids, which depends on the latent heat of vaporization of the fluids. As the majority of applications could be met by the well known and well tested fluids, R-11, R-12, R-22, R-502 and ammonia (R-717). However only ammonia can be considered environmental friendly today, but still it is not readily suited to commercial or air-conditioning refrigeration applications because of its toxicity, flammability and attack by copper.

The ozone layer beyond the atmosphere provides a filter for ultraviolet radiation, which is harmful to us. The ozone depletion potential of the refrigerants such as R-11, R-12, R-114, and R502 is due to the emissions into the atmosphere of chlorofluorocarbons (CFCs). The Montreal Protocol in 1987 agreed that the production of hydrochlorofluorocarbons (HCFCs) would be phased out by 1995 with a consumption cap, followed by a 35 % reduction in consumption beginning in 2004 and alternative fluids developed (Trott, 2000). The phaseout of HCFCs is earlier in some European countries, with for example Germany having a phaseout of R-22 in new equipment starting in 2000, and Sweden banning HCFC use for new equipment after 1997, and service after 2001 (Murphy, 1998).

Global warming is the increasing of the world's temperatures, which results in melting of the polar ice caps and rising the sea levels. It is because the so-called “greenhouse” gases release into the atmosphere, which form a blanket and reflect heat back to the earth's surface, or hold heat in the atmosphere. The most infamous greenhouse gas is carbon dioxide (CO_2) (Trott, 2000).

A newly developed refrigerant gas, R-134a is harmless to ozone layer. However, R-134a is not completely environmental friendly as it has a global warming potential (GWP) if released into the atmosphere. R-134a has a GWP of 1300, which means that the emission of 1 *kg* of R-134a is equivalent to 1300 *kg* of CO₂ (Trott, 2000).

In this experiment, R-134a will be used for the refrigerant. Therefore because of its GWP of 1300, additional care must take during the charging process and zero leakage must be ensured in the refrigeration system not only for the accuracy of the experimental results but also for the good for the environment.

2.3 VAPOR COMPRESSION REFRIGERATION SYSTEM

2.3.1 Ideal vapor compression refrigeration cycle

The Temperature-Entropy ($T-s$) and Pressure-Enthalpy ($p-h$) diagram for the ideal vapor compression refrigeration cycle are shown in Figure 2.1.

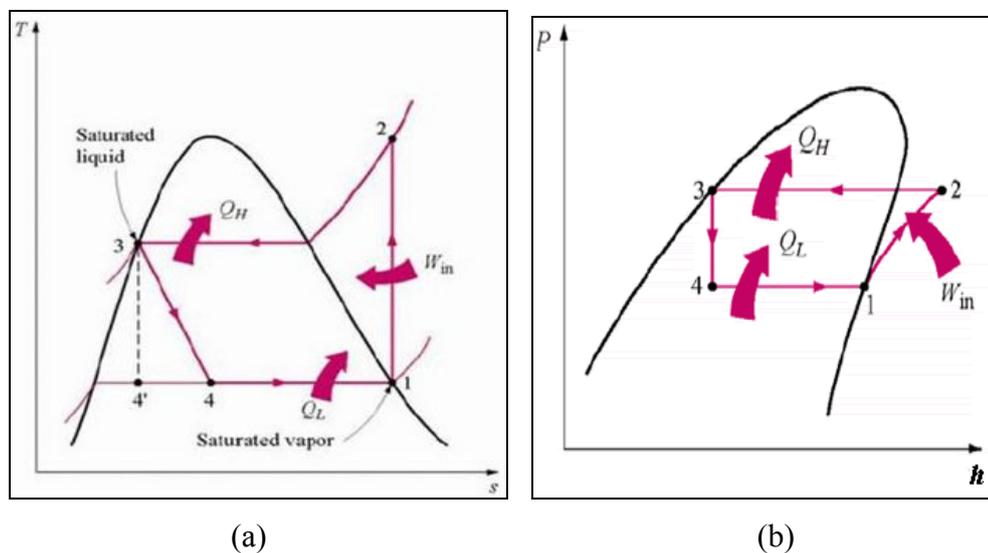


Figure 2.1: (a) $T-s$ and (b) $p-h$ diagrams for the idea vapor compression refrigeration cycle

Source: Cengel and Boles (1998)

The vapor compression refrigeration system is the most common refrigeration system that is being used nowadays. It is widely used for all purpose refrigeration. It is commonly used for all industrial purposes from a small domestic refrigerator to big air conditioning plant. The vapor compression refrigeration system is an improved type of system of air refrigeration system. In this system, a particularly suitable working fluid is used to run the whole system and we called this working fluid as refrigerant. The refrigerant used is circulating throughout the system alternately condensing and evaporating without leaving the system (Khurmi, 2006).

It named vapor compression refrigeration system because the low pressure vapor refrigerant from the evaporator is being compressed into high pressure vapor refrigerant by the compressor in this system. There are 4 main components in vapor compression refrigeration system which is compressor, condenser, expansion device or capillary tube for the simple and small capacity units of refrigerators, and evaporator. The vapor compression refrigeration cycle consists of 4 processes.

For an ideal case, these 4 processes are isentropic compression by compressor, constant pressure heat rejection by condenser, throttling by expansion device and constant pressure heat absorption by evaporator (Khurmi, 2006).

In vapor compression refrigeration cycle, the cycle is said to be started when the refrigerant enters the compressor. In an ideal case, the refrigerant is in saturated vapor form when comes out from the evaporator. The saturated vapor refrigerant is then compressed isentropically by the compressor to high pressure and high temperature state where it reached state 2 as shown in Figure 2.1. At this state, the refrigerant is superheated and its temperature is well above the surrounding temperature which is usually the ambient temperature.

The refrigerant then enters the condenser at state 2 as superheated vapor. The superheated refrigerant vapor will be cooling down by the surrounding air that flows through and leaves the condenser coils as saturated liquid at state 3 as shown in Figure 2.1 (Cengel, 1998). As the superheated refrigerant changes phase from vapor to liquid phase, a large amount of heat will be released by the refrigerant at constant temperature

which refers to as at saturated temperature with respect to a certain pressure (condensation pressure). This heat released normally refers to as latent heat of vaporization.

Latent heat of vaporization is the heat needed to overcome the molecular forces between the molecules of a substance. The latent heat of vaporization is measured by the change of enthalpy of the substance. If sufficient heat is added to the molecules of a solid or liquid, the molecular forces will break away and turning the substance into a gas. The same amount of latent heat of vaporization will be release when the phase-changes process is reversed.

In vapor compression refrigeration system, for example in the condensation process, same amount of latent heat of vaporization will be released rather than absorbed by the refrigerant when it changes phase from saturated vapor back to saturated liquid.

Therefore during the condensation process, a large amount of heat need to be removed from the refrigerant so that it can change form from vapor to liquid phase The heat removed from this process is sensible as we stand near to the condenser.

However the temperature of the refrigerant at state 3 is still above the temperature of the ambient condition even though a large amount of heat has been released from the refrigerant by the ambient air. After the condenser, the saturated liquid refrigerant will enter the capillary tube which is the expansion device for this test rig at where the refrigerant is being throttled adiabatically back to low pressure and low temperature. The enthalpy of the refrigerant remains unchanged throughout the throttling process.

As the pressure drops, the refrigerant will expand, and while it expands it will absorb heat from surrounding and causes the surroundings temperature to drop to the saturated temperature. At this state the refrigerant temperature will be below the temperature of the refrigerated space. At the end of the throttling process, the saturated

liquid refrigerant at condensation pressure will become saturated mixture refrigerant at evaporator pressure inside the evaporator.

The saturated mixture refrigerant is then enters the evaporator at state 4. During the evaporation process, the saturated mixture refrigerant will continue boil inside the evaporator by absorbing the required latent heat of vaporization from the refrigerated space at constant temperature thereby cooling the space until it is fully vaporized (Ananthanarayanan, 2005). The saturated vapor refrigerant then once again enters the compressor to complete the cycle. The schematic diagram for the ideal vapor compression refrigeration cycle is shown in Figure 2.2.

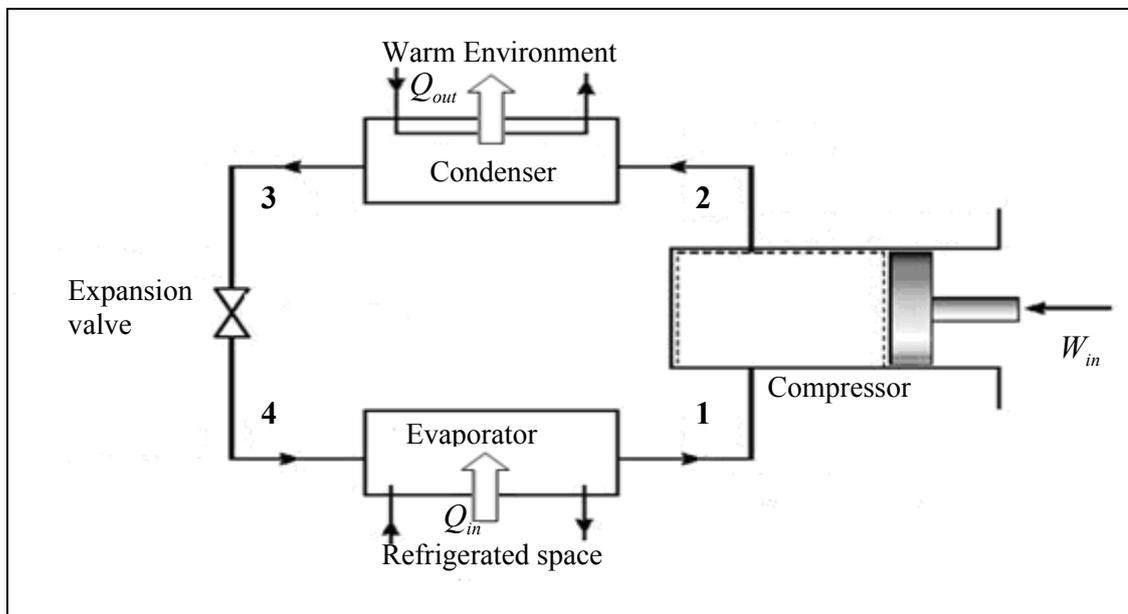


Figure 2.2: Schematic diagram for the idea vapor compression refrigeration cycle

2.3.2 Actual vapor compression refrigeration cycle

The Temperature-Entropy ($T-s$) diagram of actual vapor compression refrigeration cycle is shown in Figure 2.3.

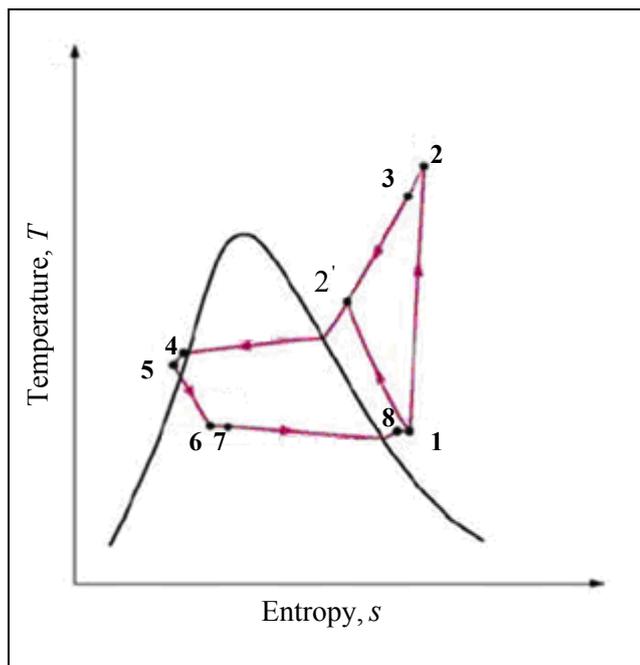


Figure 2.3: $T - s$ diagram for an actual vapor compression refrigeration cycle

Source: Cengel and Boles (1998)

There are some differences between the actual and the ideal case of vapor compression refrigeration cycle primarily due to pressure and temperature drops throughout the system. Some unavoidable losses such as the pressure drops associated with refrigerant flow inside the connecting tube lines and also the components and the temperature raises or drops caused by the heat transfer from or to the system.

The refrigerant vapor leaving the evaporator is normally superheated due to the pressure drop in the evaporator and connecting line and the heat transfer into the system. This is undesirable because the superheated refrigerant vapor will increase the work done by the compressor since the refrigerant entering will have an increased specific volume.

In actual case, there is heat transfer to the vapor refrigerant during the compression process in the compressor. This is due to the heat transfer from the compressor cylinder walls to the vapor refrigerant because the temperature of the cylinder walls is somewhat in between the temperatures of cold suction vapor refrigerant and hot discharge vapor refrigerant. Thus the compression process of vapor

refrigerant is neither isentropic nor polytropic as in the ideal case. The entropy may increase (process 1-2) or decrease (process 1-2') as shown in the Figure 2.3 during the actual compression process depending on which effect is dominating. However the compression process 1-2' is even more desirable than the isentropic compression process since the smaller specific volume of the refrigerant and thus smaller input power needed by the compressor (Khurmi, 2006; Dincer, 2003).

The pressure of the liquid refrigerant will somehow slightly drops across the condenser. The liquid will be normally sub-cooled after leaving the condenser. This represents a gain. It is because as a result of this heat transfer, the refrigerant enters the evaporator with a lower enthalpy which permits more heat to be transferred to the refrigerant in the evaporator. Some pressure will drop when the refrigerant travels through the evaporator too (Dincer, 2003). Finally the refrigerant leaves the evaporator slightly superheated, complete the cycle.

2.4 REFRIGERATION SYSTEM COMPONENTS

In vapor compression refrigeration system, there are 4 major components and some auxiliary equipment associated with these major components. The major components are condensers, evaporators, compressor and expansion device whereas the auxiliary equipment will be components such as receivers, refrigerant capacity controls and accumulators (Dincer, 2003).

2.4.1 Compressors

There are variable types of compressor used in refrigeration system such as hermetic compressors, semi hermetic compressors, open compressors, displacement compressor and dynamic compressor. However the hermetic type of compressor is in most used in domestic refrigerators, freezers and air conditioners because of its low cost and easy in installation. The hermetic compressors are available for small capacity only. Its motor and drive are sealed in compact welded housing. Hermetic compressors can work for a long time in small capacity refrigeration systems without any maintenance requirement and any gas leakage. However they are sensitive to electric voltage

fluctuations, which may make the copper coils of the motor burn. A hermetic compressor is shown in Figure 2.4 (Dincer, 2003).



Figure 2.4: Hermetic compressor

2.4.2 Condensers

The action in the condenser is just opposite to the action in the evaporator where condenser gives off heat to change the vapor back to liquid, whereas the evaporator absorbs heat to change the liquid to vapor.

There are several types of condensers to be considered when choosing for the particular application. They are air-cooled, water-cooled, shell and coil, tube within tube and evaporative condensers. Air-cooled condenser is mostly used in domestic application. An air-cooled condenser functions in removing heat by the air. It consists of steel or copper tubing through which the refrigerant flows. The size of tube usually ranges from 6 mm to 18 mm outside diameter, depending upon the size of the condenser. Fins will be attached on the surface of tubes to increase the efficiency of heat transfer. The fins are usually made from aluminum because of its light weight. Spacing between the fins is quite wide to prevent the dust clogging.

The single row tubing condenser provides the most efficient heat transfer. However there is condenser with up to 6 rows of tubing also in used, anyway the condensers up to 8 rows of tubing are usually not efficient already. This is because the air temperature will be too close to the condenser temperature to absorb any more heat after passing through 8 rows of tubing (Khurmi, 2006; Dincer, 2003; Langley, 2008).

There are 2 types of air cooled condenser which are the natural convection and forced convection. In natural convection air-cooled condenser, the heat transfer from the condenser coils to the air is being done by the natural air flow that created when the air comes in contact with the warm condenser tubes, it absorbs heat from the refrigerant and thus become warm air therefore lighter and rises up letting the cold air from below rises to take away the heat from the condenser, and therefore produce a natural circulation across the condenser tubing.

However the rate of heat transfer in natural convection condenser is slower, therefore larger surface area is require compared to the force convection condenser. The natural convection air-cooled condensers are used only in small capacity applications such as domestic refrigerators, freezers, water coolers and room air conditioners. An air-cooled condenser is shown in Figure 2.5 (Khurmi, 2006).

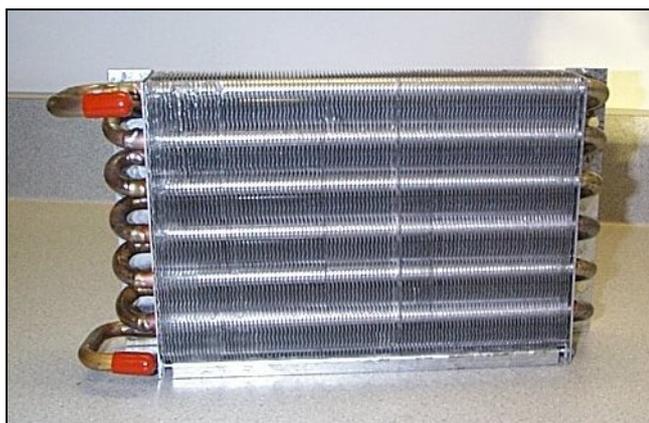


Figure 2.5: Air-cooled condenser

2.4.3 Evaporators

Evaporator is the part in the refrigeration system where the heat is captured from the refrigerated space and provides the cooling effect required for any particular application. Evaporators are generally consists of a series of tubes through which refrigerant flows are finned to increase the heat transfer rate from the medium to be cooled, let says air inside the refrigerated space (Dincer, 2003).

Evaporators are divided into 2 categories which are direct cooler evaporators and indirect cooler evaporators. As it is named, the direct cooler evaporators cool the medium in the refrigerated space directly, such as cooling the air in the room, so that to cool the room. Where as, indirect cooler evaporators that cool a liquid such as brine solution so that cool the product. However, the direct cooler evaporators are usually being used in domestic applications.

Direct cooler evaporators can be divided into 2 general types which are natural-draft evaporators and force-draft evaporators. To cool the refrigerated space, air in the refrigerated space must be made to flow from the evaporator to all parts of the refrigerated space (Dincer, 2003).

If natural- draft evaporators are used, the air circulation is driven by the natural behavior of the air that the tendency of cold air goes down and warm air rises up. This phenomenon enables creating continuous movement and circulation of air through the evaporator and thus eventually cools the refrigerated space. But anyhow natural convection is not effective enough for most of the domestic applications and thus the forced-draft evaporators have been introduced.

The forced-draft evaporators which utilize the forced convection are the most effective evaporators of obtaining rapid, positive air circulation. In forced-draft evaporators, air is being forced flow through the evaporator coil by the fan mounted at the end of the evaporators and thus increases the rate of heat transfer process. Figure 2.6 shows a roll-bond refrigerator evaporator (Langley, 2008).



Figure 2.6: Roll-bond refrigeration evaporator

2.4.4 Expansion devices

Expansion devices are also called throttling devices. They are the component used in the refrigeration system to reduce the refrigerant pressure from condensing pressure to evaporating pressure by throttling operation and regulate the liquid-refrigerant flow to the evaporator to match the equipment and load characteristics. These devices are designed so that to meet the requirement of refrigerant flow rate need to be fed into evaporator. The amount of refrigerant to be fed, of course depends on the amount of heat need to be removed from the refrigerated space (Dincer, 2003).

Among the most common expansion devices are thermostatic expansion valves, constant pressure expansion valves, float valves and capillary tubes. However for small capacity refrigeration and domestic systems, capillary tube is being chosen because of its simplicity, low cost and maintenance free. The capillary tube is not a valve at all and it is the simplest type of flow control device. The capillary tube is nothing more than a coil of several feet of very fine tubing, usually having an orifice of about 0.4 to 3 mm internal diameter and 1.5 to 5 m in length. The pressure of the high side of the system will force the liquid refrigerant passes through the small diameter and long capillary tube. The length of the capillary tube is designed to allow the proper amount of liquid refrigerant to trickle through to the evaporator. The only adjustment of capillary tube is

its length and when it is installed, it is no more changes in the system. The right size of the capillary tube and the right refrigerant charge is somehow self-regulating.

The capillary tube however not as efficient in operation as an expansion device that adjusts itself to the refrigerant flow required. However, for domestic and small refrigerators the savings possible with a better device are insignificant. There is no receiver required when capillary tube is used as it passes condensed refrigerant to the evaporator. Thus, the liquid storage is in the evaporator. Cooper type capillary tube and its drier filters are shown in Figure 2.7 (Dincer, 2003; Langley, 2008).



Figure 2.7: Copper capillary tube and drier filter

2.5 THEORY AND CALCULATION OF VAPOR COMPRESSION REFRIGERATION CYCLE

Figure 2.8 shows a Temperature-Entropy ($T-s$) diagram for a vapor compression refrigeration cycle. A vapor compression refrigeration cycle consists of 4 processes as mentioned above and can be analyzed by applying steady-state flow according to the 1st law of thermodynamics in each of the component in the refrigeration system separately (Dincer, 2003).

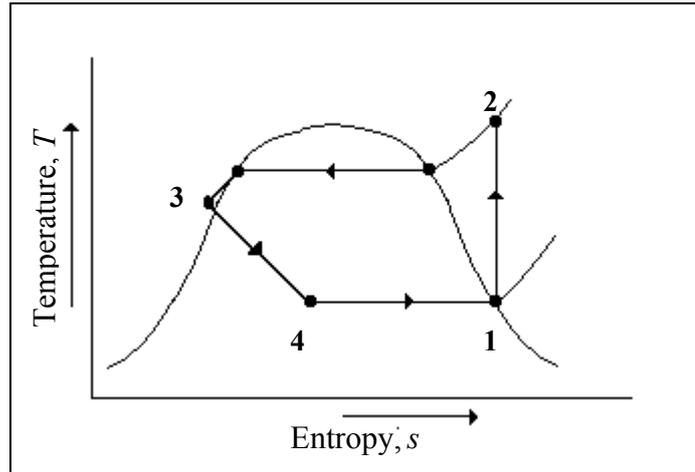


Figure 2.8: T-s diagram for a vapor compression refrigeration cycle

2.5.1 Compression process

Work is consumed by the compressor to compress the low temperature refrigerant vapor into a small volume high temperature and high pressure superheated vapor refrigerant. The compressor work per unit mass and compressor work rate are expressed as in Eq. (2.1) and (2.2)

$$W_c = (h_2 - h_1) \text{ kJ/kg} \quad (2.1)$$

$$\dot{W}_c = \dot{m}(h_2 - h_1) \text{ kW} \quad (2.2)$$

2.5.2 Condensation process

During condensation process, heat is rejected from the superheated vapor refrigerant to the warm space at high temperature, T_H . The condensed vapor is cooled at constant pressure to a temperature below that of the saturation temperature corresponding to the same condensation pressure (Eastop, 1993). The heat rejection per unit mass and heating capacity are expressed as in Eq. (2.3) and (2.4)

$$Q_H = (h_2 - h_3) \text{ kJ/kg} \quad (2.3)$$

$$\dot{Q}_H = \dot{m}(h_2 - h_3) \text{ kW} \quad (2.4)$$

2.5.3 Throttling process

The expansion process in the capillary tube is adiabatic, therefore the enthalpy throughout the process remain unchanged is expressed as in Eq. (2.5) (Nag, 1995).

$$h_3 = h_4 \text{ kJ/kg} \quad (2.5)$$

2.5.4 Evaporation process

This is the device where there is heat exchange for providing refrigeration. Thus, heat will be transferred into the liquid-vapor refrigerant and boils the mixture refrigerant at a low temperature, T_L . The heat absorbed is known as the refrigerating effect, the amount of heat removed from the refrigerated space per unit mass flow of refrigerant. It is desirable to extend the evaporation process to give the vapor a definite amount of superheated so that to make complete use of its latent heat of vaporization and also prevent the compressor from damage by the carry over liquid refrigerant. The refrigerating effect and refrigeration capacity are expressed as in Eq. (2.6) and (2.7) (Dincer, 2003; Nag, 1995; Iynkaran, 2004).

$$Q_L = (h_1 - h_4) \text{ kJ/kg} \quad (2.6)$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) \text{ kW} \quad (2.7)$$

2.5.5 Refrigeration coefficient of performance, COP_R

The measure unit of refrigeration system and heat pump is called coefficient of performance, COP . In refrigeration system COP_R is used. The small notation of R in the term COP_R is represent refrigeration rather heat pump. COP_R is the ratio of

refrigerating effect, Q_L to the compressor work per unit mass, W_c . The refrigeration coefficient of performance is expressed as in Eq. (2.8)

$$COP_R = \frac{Q_L}{W_c} \quad (2.8)$$

2.6 PREVIOUS RESEARCHES AND DEVELOPMENTS RELATED TO THE TITLE OF THIS THESIS

2.6.1 A test rig to record mass flow rate, pressure and temperature of a household refrigerator during on/off cycling mode

An electronic test rig was developed to record mass flow rate, pressures, temperatures, and power consumption of the compressor of a household refrigerator during on/off cycling by J. Philipp and his colleagues. The electronic test rig consists of 2 main parts. First part is a test rig for getting the needed data to forecast the energy consumptions and cabinet temperature of a household refrigerator. Second part is the simulation software which enables the data storage and presents them on screen for instant process analysis.

According to J. Philipp, simulation software is worthwhile to be used on testing of a household refrigerator. There is still common procedure people build the refrigerator and measure and test and modify it in order to improve the performance. In sum, the savings could be remarkable when the simulation software is used instead of the common procedure used.

The electronic test rig was set up with 4 thermocouple glued into the refrigerant lines, 1 at the discharge line of compressor, 1 at the suction line, 1 at the filter dryer and finally another 1 in the middle of the condenser. Other thermocouples were glued on the heat exchanging surfaces of condenser, evaporator and compressor can. 3 thermocouples were set up in different locations of the cabinet to record the mean temperature there. 2 pressure transducers were connected 1 to the suction line of the

compressor and the other 1 to the discharge line. Both at a distance 300mm from the condenser can.

The coriolis flow meter (CFM) was chosen to measure the mass flow rate of the refrigerant. The location chosen to put CFM into the refrigerator component was in between the compressor and the condenser. This was because CFM needs higher densities of refrigerant in order to get a reasonable accuracy data.

The data points obtained were verified by performing energy balance. The data was taken after a near steady state condition was reached. J. Philipp and his colleagues obtained a cabinet constant of 1.1 W/K and this value was compared with 1.03 W/K by Environmental Protection Agency (EPA) refrigerator analysis. It was found that the results obtained using the test fixture compared to the conventional method differs in refrigeration capacity by 12%. Given the accuracy of the sensors and the assumptions taken this difference is acceptable (Philipp, 1996).

2.6.2 Steady state characteristics of failures of a household refrigerator

A conference working paper on steady state characteristics of failures of a household refrigerator by M. G. McKellar and D. R. Tree was a response to the large percentage of all parts returned under warranty were still in working order. This problem arose because of the misjudged of some failures as the components failures by the repairman. This is a tremendous cost to both the manufacturer and the consumer to solve every failure by replacing with a new component. The main thrust of this work was to help develop a diagnostic system for a household refrigerator to eliminate this problem.

Several transducers such as thermocouples and pressure transducers were put into the system at some crucial points where the data collected from that point could help in detection of the actual failures. However transducers used must keep in the least number. Therefore the major objective of instrumenting the refrigerator was to determine the minimum number of transducers needed to locate and identified the most failures.

The failures characteristics were tested and analyzed and recorded down in the ratio form with normal condition test. This paper presented the characteristics from 3 failures of overall failures which are a blocked capillary tube, a failed evaporator fan, and a leaking compressor.

In the case of partially blocked capillary tube can cause the compressor to over heat because the refrigerant flow is slower or stopped, and therefore can not cool the compressor effectively. A refrigerator with a partially blocked capillary tube will have a lower than normal mass flow rate characteristics. A partially blocked capillary tube will increase the resistance to refrigerant flow, thus reducing the mass flow rate. A second characteristic of blocked capillary tube is liquid refrigerant accumulating in the condenser. The refrigerant condensed along the first third of the condenser length, dropped almost to the temperature of the surrounding air in very short distance, and continued to subcool gradually for the remainder of the length. The low mass flow rate allowed more refrigerant to condense along a shorter than normal distance. The constriction caused this excess refrigerant to build up in the condenser. The evaporator had a lower than normal pressure. The evaporator inlet temperature was above the saturation temperature. Therefore the refrigerant entering the evaporator was superheated. The low mass flow rate combined with the low pressure caused the refrigerant to completely evaporate in the capillary tube. The most effective way to identify a blocked capillary tube, once can locate 2 pressure transducers 1 at high side and the other at low side, locate thermocouple at the inlet of evaporator, and the thermocouples along the condenser.

Evaporator fan failure caused the lower rate of heat transfer therefore increase the temperature difference between the refrigerant and the surroundings. The increased temperature difference was reflected by a decrease in the average temperature of the evaporator. This decrease was also reflected by the decrease in the suction pressure. The reduction in suction pressure was propagated to the discharge pressure resulting in its decrease. The air temperature at the top of the evaporator was lower than the temperature at the bottom. This characteristic was due to the fact that the evaporator was vertical. When the fan was on, the air flowed upward across the evaporator. This caused the temperature at the top to be lower than the temperature at the bottom. When the fan

was off, however, the cooler air sank to the bottom of the evaporator, while the warm air rose, thus causing the temperature at the top of the evaporator to be warmer. The most effective transducers used to determine this type of failure were the suction and discharge pressure transducers and the thermocouples used to measure the temperatures of the air around the evaporator.

Another researcher, Jankov has found that a leak in the discharge valve will cause the compressor to work harder. This is due to high pressure gas leaking back into the cylinder during suction process. Jankov also shown that it takes less time from start up to reach the discharge pressure due to the fact the higher pressure gas is in the cylinder at the beginning of the compression process. However, Jankov found that a leak in the suction valve reduces compressor work due to the reduction of the resistance to compression. During the compression cycle, some of the gas leaks through the suction valve into the compressor return line. This causes the return line to have a higher than normal pressure. Jankov also indicated that it takes more time from start up to reach the discharge pressure. The most effective transducers used to detect a leaking compressor were the discharge and suction pressure transducers (Tree, 1988).

2.6.3 Dynamic behavior of a vapor compression refrigerator: A theoretical and experimental analysis

C. Melo and his colleagues had done an experiment to validate the dynamic modeling of the refrigerator. The particular model considered was limited to the reciprocating compressor, forced-draft condenser, accumulator, capillary tube, and forced-draft evaporator system. The space being cooled, for example, is a top-mount domestic refrigerator, which was a new feature that made both theoretical and experimental analysis, more complex.

Each of the refrigeration system components was modeled by 1 or more control volumes. Thus 3 for the condenser, in order to accommodate the region of superheated vapor, mixture of saturated vapor and saturated liquid, and subcooled liquid. 2 control volumes for the evaporator, 1 for liquid region and the other for the vapor region. The compressor and compressor shell were modeled as 2 separate control volumes. 2 control

volumes also for accumulator, 1 for saturated condition and the other for the superheated condition. 3 control volumes for capillary tube in order to take into account for the case containing only superheated vapor, containing only saturated vapor and containing only subcooled liquid.

The conservation equations for mass and energy was utilized to do the transient analysis of the refrigeration system. However, only condenser (superheated flow region), capillary tube, and space being cooled were presented in this paper.

In condenser, as soon as the condensation process started the model neglects the superheated flow region and employed only the saturated vapor control volume. The remaining control volume was employed when some subcooled liquid was formed in the condenser. The assumption of disregarding the influence of the superheated flow region, is due to the fact that this region occupies only 5-10% of the total coil volume which is not valid for natural-draft condenser. The control volume was treated as a stirred tank, in which the conditions existing at the outlet of the tank were the same as the bulk conditions within the tank. The capillary tube was modeled in a straight, horizontal, constant inner diameter tube. Flow in the capillary tube was assumed one-dimensional, homogenous, and adiabatic. The choked and metastable flow phenomena were neglected. The space being cooled is a $0.42m^3$ domestic top mount refrigerator. The required results for all 3 cases were calculated by using the numerical integration of equations.

In order to validate the model, an experimental test was performed. It was necessary, before the beginning of the test, to insulate the refrigerator capillary tube since one of its assumptions was adiabatic flow. The refrigerator was placed in an environmental test chamber, in which the air temperature was maintained in $43^{\circ}C$. The temperature and pressure in several locations of the refrigerator were measured respectively by copper constantan thermocouples and strain gage pressure transducers. Both the thermocouples and the pressure transducers were connected to a data acquisition system. The results of such test were then compared with the computational results. The results achieved good agreement between laboratory and model data even with the simplifying assumptions used in the model (Melo, 1988).

2.7 MEASUREMENT DEVICES

There will be 2 important parameters of concern in determining the coefficient of performance of the refrigerator, COP_R in this experiment. They are temperatures and pressures. Temperatures and pressures throughout the system are being observed and recorded down at chosen points of interest and then use the data collected to obtain enthalpy, h for each of the points of interest and finally compute the COP_R of the refrigeration system using the equations shown above.

2.7.1 Temperature measurement devices

Thermocouple is used to measure the temperatures at the chosen points in the refrigeration system. The thermocouple is a device that made of two wires of different materials which are twisted together at their ends to form a closed loop. The combination of 2 dissimilar materials allows EMF generated in the thermocouple circuit when its ends are at different temperatures. The different of EMF generated in different materials gives the differential EMF which is called the Seebeck EMF and this is the EMF on which thermoelectric temperature measurements are based (Leigh, 1988).

Physically, thermoelectric EMF arises because of the temperature gradient in the material when its ends are at different temperature. Electrons will experience a force that is directed down the temperature gradient and if the material is a conductor, electrons are free to move inside the material thus a thermally generated potential difference will appear between the ends of the material.

A thermocouple measures the temperature difference between the hot junction and a cold junction. Cold junction is usually referred to as reference junction and the reference junction is usually the ambient temperature. Of course the ambient temperature must be known. The hot junction is attached at the chosen location in the system. Where as the cold junction is an open end and is connected to the head of the thermocouple meter. Below are general construction procedures for a thermocouple. Figure 2.9 shows a schematic diagram for a type K thermocouple.

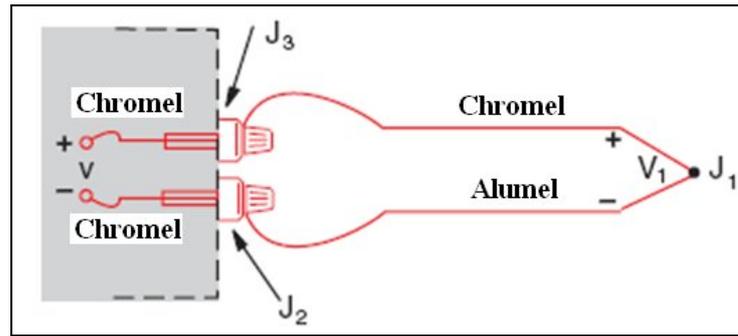


Figure 2.9: Schematic diagram of type K thermocouple

Source: Omega Engineering, Inc.

The thermocouple shown in Figure 2.9, J_1 is refer to the measuring junction where it is going to put at the chosen location on the tubing system. J_2 and J_3 are refer to the reference junction where the temperature is known. In our experiment, the temperature of the reference junction is the ambient temperature. Temperature of J_2 and J_3 are the same. The equivalent circuits are shown in Figure 2.10.

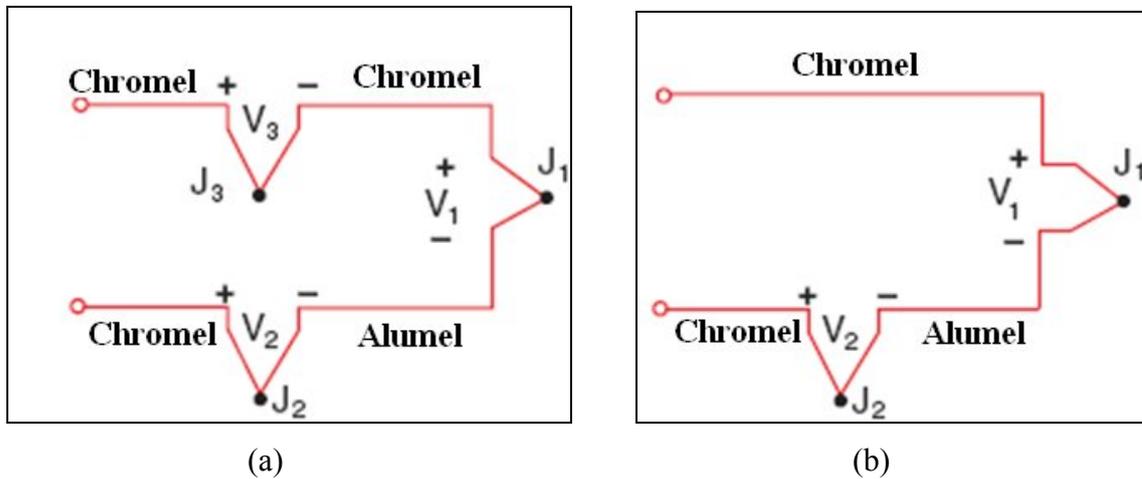


Figure 2.10: Equivalent circuits of the type K thermocouple shown in Figure 2.10 (a) can be reduced to (b)

Source: Omega Engineering, Inc.

From the circuit, J_3 is a copper-to-copper junction, it creates no thermal EMF ($V_3 = 0$). But J_2 is copper-to-constantan junction which will add an EMF (V_2) in opposite to V_1 . The resultant voltmeter reading V will be proportional to the temperature difference between J_1 and J_2 . A K type thermocouple with its meter that is used in this experiment is shown in Figure 2.11.



Figure 2.11: K type thermocouple that is used in this experiment

2.7.2 Pressure measurement devices

The bourdon tube pressure instrument will be used to measure the pressures at the chosen location in the refrigeration system. It consists of a thin walled tube with a cross sectional area elliptical in shape is bent into an arc of a circle of about 270° to 300° . When pressure is applied on the bourdon tube, the pressure causes distention of the flat section of the tube and it will then tend to restore its original round cross-section. This change in cross section causes the tube to straighten slightly. Since the tube is permanently fastened at one end, the tip of the tube traces a curve that is the result of the change in angular position with respect to the center. Inside the limit gauge, this angular displacement of the tube will move the position of the pointer then gives the exact pressure reading at that point. However, bourdon tube pressure instrument only measures the pressure above or below the atmospheric pressure and the pressures measured by it are said to be gauge pressure. A bourdon tube pressure gauge is shown in Figure 2.12.

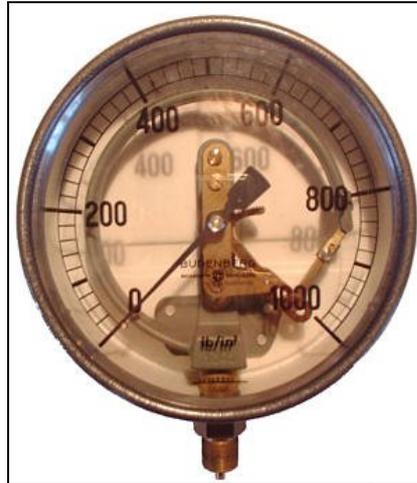


Figure 2.12: Bourdon tube pressure gauge and its inner construction

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

This chapter is mainly discussed about the assembly process of the thermocouples and bourdon tube pressure gauges and the procedures for the experiment or testing processes. A previous developed experimental test rig which is small capacity domestic refrigerator is used in this experiment. Some improvements are going to be done on this test rig so that more accurate data could be obtained.

Both of temperatures and pressures measurement devices are being rearranged to the better locations that as nearer as possible to the exact locations of points of interest in the refrigeration system for the results improvement purpose. Furthermore the previous developed tubing system will be improved by modifying to a more simple tubing system. This is to prevent any significant pressure drop of refrigerant travelling throughout the tubing lines in the refrigeration system. As mentioned in chapter 2, the irreversible pressure drops will be caused by the frictional effect between the tubes wall and the refrigerant. Therefore it is desirable to reduce the length of the tubing lines to the shortest and simplest in the refrigeration system whenever it is possible to avoid from any significant pressure drop of refrigerant that is travelling throughout the system.

The amount of refrigerant to be charged into the system depends on the operating temperature and pressure of the system. Before refrigerant can be charged into the system, evacuation has to be done first to ensure that there is no others substances or contaminants inside the refrigeration system other than refrigerant. Evacuation is done

to prevent the compressor from damage and also ensure the maximum coefficient of performance can be achieved.

3.2 FLOW CHART

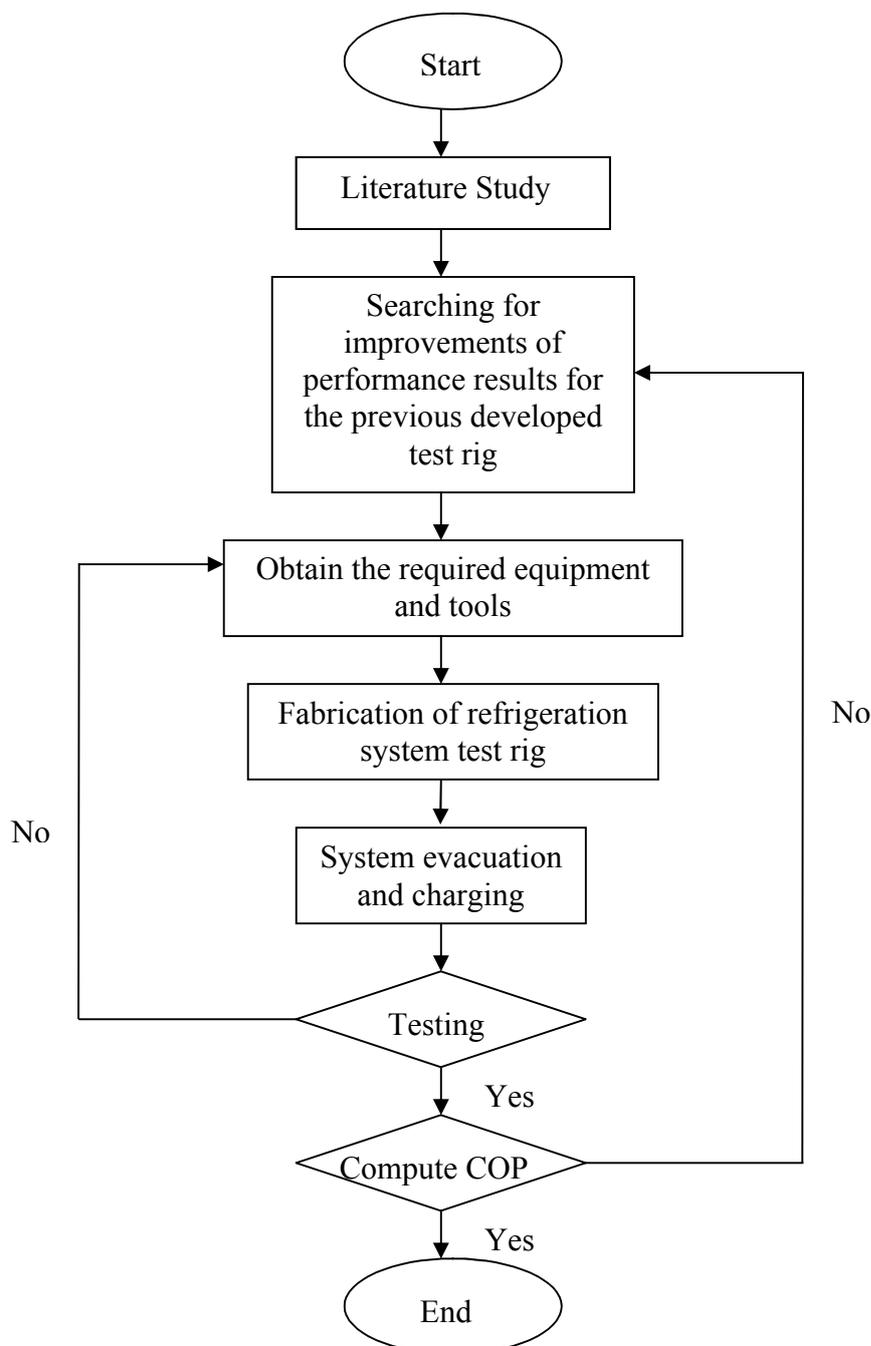


Figure 3.1: Flow chart shows the overall flow of the process of this project

3.3 REFRIGERATION SYSTEM TEST RIG

3.3.1 SPECIFICATION

Table 3.1: Technical specification of the refrigerator model URF-M50A

Particulars	Specifications
Model	UPSON
Model no	URF-M5)A
Rated voltage	240V
Electricity consumption	0.5kWH/24h
Power	90W
Weight	19kg
Total effective volume	47L

Source: Wan Ramli (2008)



Figure 3.2: Experimental test rig

3.4 EQUIPMENTS

Figure 3.3 shows the equipments needed in this experiment. From picture (a) to (d) shows a K type thermocouples with its indicator meter, a R-134a cylinder tank, a bourdon tube pressure gauge, and a vacuum pump respectively.



(a)



(b)



(c)



(d)

Figure 3.3: Equipments that are used in this experiment: (a) K-type thermocouple with its meter, (b) Refrigerant cylinder tank, (c) Pressure gauge, (d) Vacuum pump

3.5 TOOLS

Figure 3.4 shows the tools needed in this experiment. From picture (a) to (h) shows a set of screw type flaring tools, spanners in size 17, wires and a test pen, a stick of silver rod, a cooper tube cutter, a roll of copper tube, a soldering cylinder, and a charging manifold respectively.

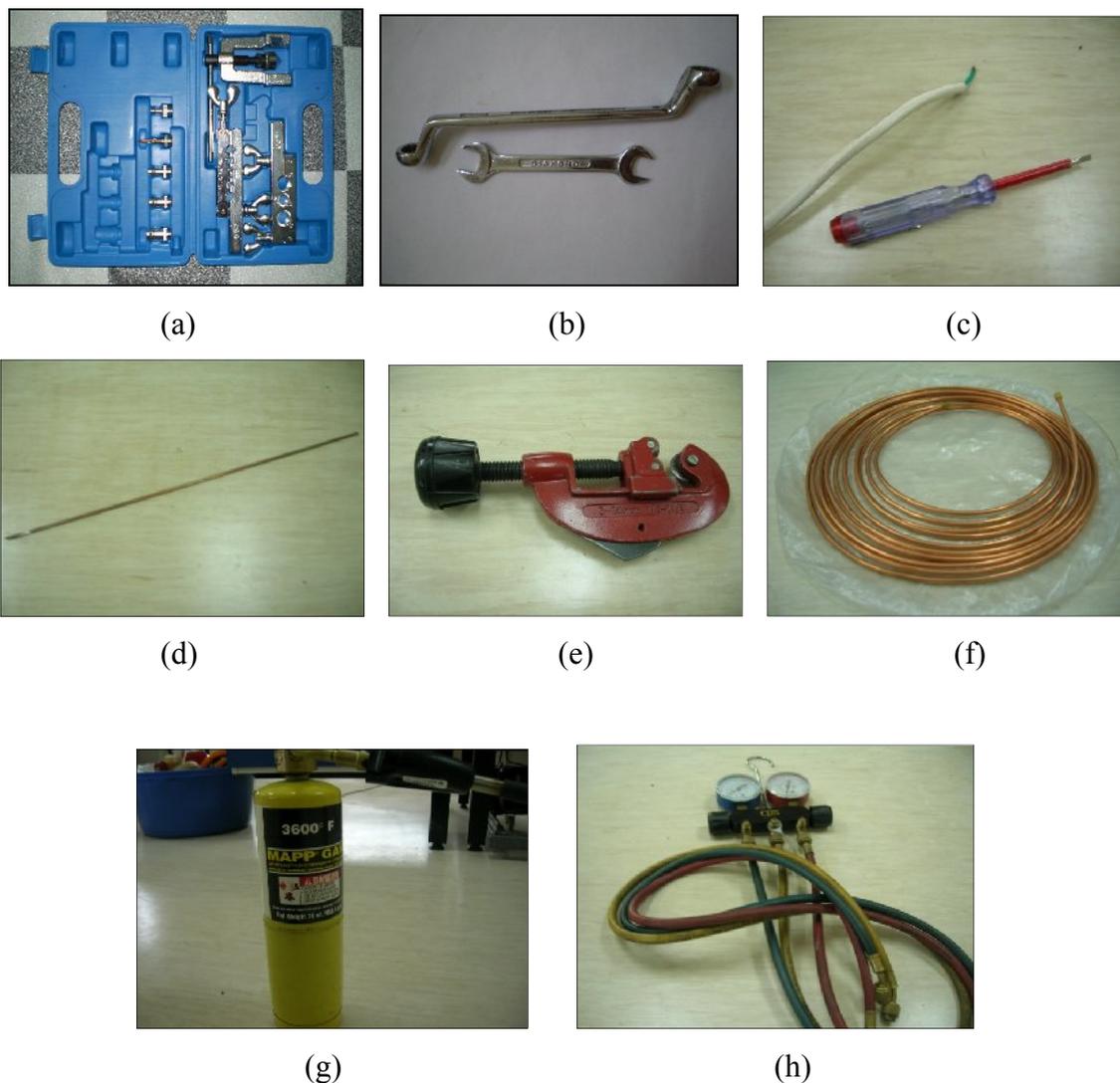


Figure 3.4: Tools needed in this experiment: (a) Screw type flaring tools, (b) Spanner size 17, (c) Wires and test pen, (d) Silver rod, (e) Cooper tube cutter, (f) Copper tube, (g) Soldering cylinder, (h) Charging manifold

3.6 FABRICATIONS

Figure 3.5 shows the schematic diagram of where the measuring devices are located in the refrigeration system test rig. Where T_s indicates the thermocouple located at compressor suction, T_d indicates the thermocouple located at compressor discharged, T_{ci} indicates the thermocouple located at capillary tube inlet, P_s indicated the pressure transducer located at compressor suction, and P_d indicates the pressure transducer located at compressor discharged.

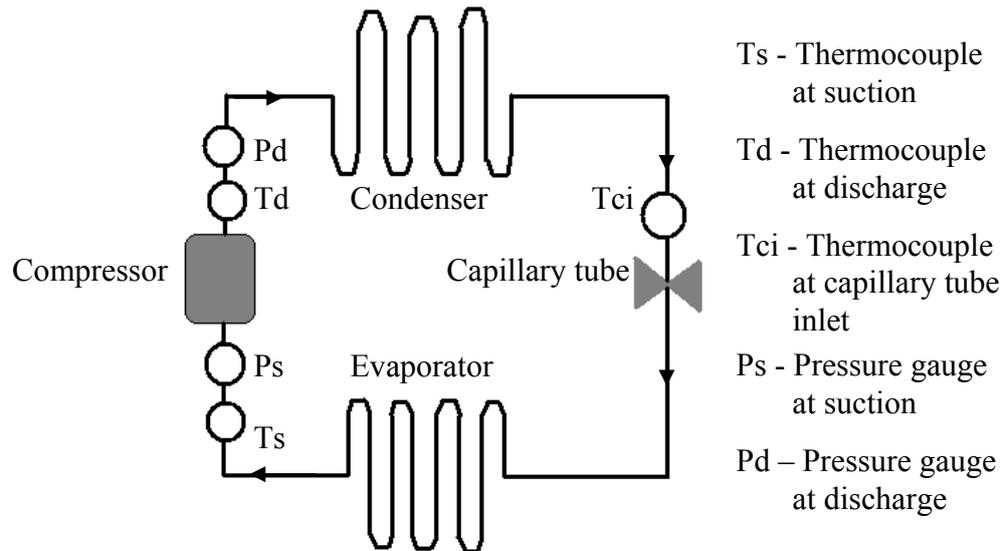


Figure 3.5: Locations of the measuring devices in the refrigeration system test rig

3.6.1 Thermocouples fabrication

To determine the performance of the experimental test rig, the measuring junctions of the thermocouples must be attached at the locations as nearer as possible to the exact locations of points of interest on the outside surface of the tubes. Attaching the junctions on the outside surface of the tube can easily move the thermocouple junctions to the exact locations. This enables the movement of the thermocouple junctions to the location of points of interest with no restraint.

Comes to compare with the previous developed test rig, the thermocouples measuring junctions were installed inside the tubing system by using cooper T-junctions and fixed with steel epoxy adhesive so that to have direct contact with the flowing refrigerant inside the tubes. Therefore in the previous developed test rig, the accuracy of the measuring data can be maintained high.

However, if cooper T-junctions are to be used, extra space will be required for the T-junctions to be connected into the tubing system. This is undesirable since extra length of tubing will lead to the significant pressure drop of refrigerant when it is

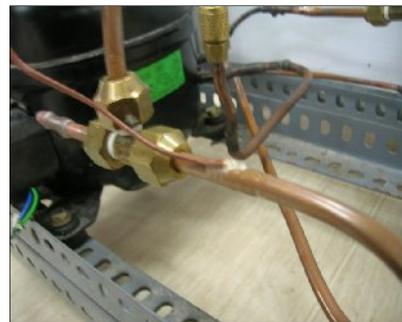
travelling throughout the system especially for the mini size of refrigerant system of this experiment.

On the other hand if the thermocouple junctions are to be attached only on the outside surface of the cooper tubes, the thermocouples junctions will not have the direct contact with the refrigerant and this might cause the thermocouples could not sense the temperature of refrigerant in the fastest rate and might effect the accuracy too. To deal with this problem, insulation must be done perfectly at least at the point where thermocouples measuring junctions are attached.

There are at least 3 points where the temperature of refrigerant need to be collected. Thus, 3 thermocouples are needed in this experiment. Figure 3.6 shows the thermocouples attached to the refrigeration system test rig before and after simplification.



(a)



(b)

Figure 3.6: K-type thermocouple attached to the refrigeration system test rig: (a) Thermocouple wires mounted into cooper tubing system by using cooper T-junction, (b) Thermocouple wires attached on the outer surface of cooper tube by using tape

The installation of the thermocouples is simplified from using cooper T-junction in the previous developed test rig (Figure 3.6 (a)) to just attached on the outer surface of the cooper tube by using tape (Figure 3.6 (b)). The insulation work is then to be done at the locations where the thermocouples wires are attached for the new developed refrigeration system test rig.

3.6.2 Bourdon tube pressure gauge fabrication

There are 2 points of system pressures to be determined by connecting the bourdon tube pressure gauges at the points where specifically chosen. 2 points of pressure on the system are needed to combine with 3 temperatures obtained at the points of interest in the system to get the enthalpies of that mentioned 4 points in chapter 2 which are h_1 , h_2 , h_3 and h_4 where h_3 equals to h_4 so that the COP_R of the refrigeration system test rig can be determined.

The 2 pressure points are suction and discharge pressure of the compressor. However the outlet pressure of the condenser will be assumed to be same as the discharge pressure and outlet pressure of the capillary tube will be assume to be same as the suction pressure. These assumptions are correct when the tubing system of the testing rig is not too long and complicated. Figure 3.7 shows the detail construction of the bourdon tube pressure gauge into the tubing system by flared joint.

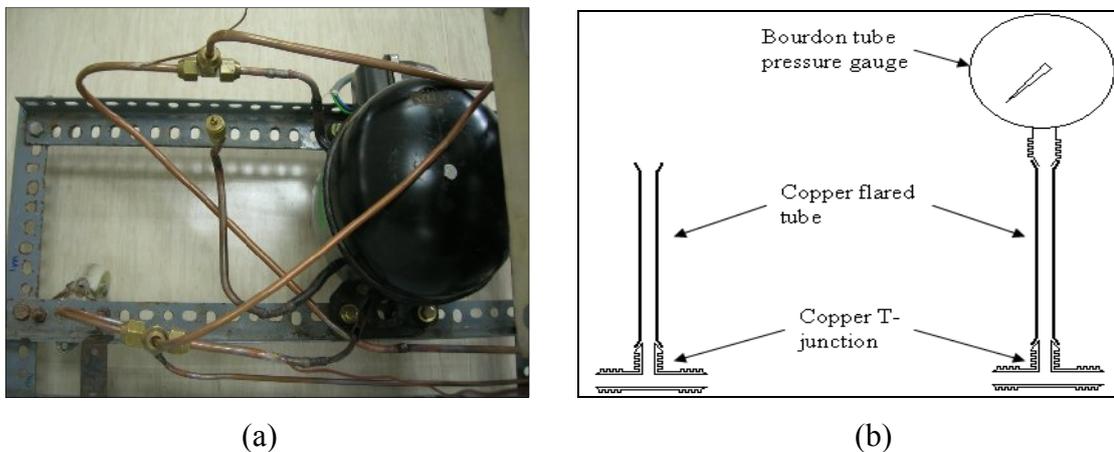


Figure 3.7: Assembly diagrams of bourdon tube pressure gauge: (a) Bourdon tube pressure gauge is installed into the tubing system, (b) Visualization of the flaring connection of bourdon tube pressure gauge

The bourdon tube pressure gauges are connected at the points mentioned by simply flared joints with copper T-junction. Firstly prepare a copper flared tube by using screw-type flaring tool like what have done in thermocouple fabrication. Then

connect the copper flared tube with copper T-junction as shown on the left hand side in Figure 3.7 (b). After the connection have been done, the bourdon tube pressure gauge is ready to connect to the other end of the connected cooper flared tube as shown on the right hand side in the Figure 3.7 (b) to complete the bourdon tube pressure gauge installation to the refrigeration system test rig.

3.6.3 Soldering

Soldering is a process used to join piping or tubing to fittings. It also being called soft soldering and its operating temperature is under $450\text{ }^{\circ}\text{C}$. The tools need to be prepared for soldering are the soldering cylinder which is the heat supply source and filler metal that used to adhere to the fittings.

In this experiment the MAPP type of soldering cylinder is used. To join 2 lengths of cooper tubing together, a fitting or coupling need to be placed between them into which each tube slides. Hence a clearance will be created between the surfaces of 2 lengths of the cooper tubing which allows the tubing to slide into the coupling or fitting. Therefore a fillet metal must be used to fill this clearance. The filler metal actually acts as the adherent for the joint. In this experiment, a silver rod is chosen to use as the filler metal.

Soldering must be done with extra care to prevent any destructive accident from occurring and the other one is to prevent any leaky tube from producing. Any leakage not only will affect the performance of the refrigeration system but also eventually causes the refrigerant charged into system to be finish leaking out.

As with most craft skills, however, the keys to success are practice and planning. Even for small projects, draw up a plan that minimizes the number of fittings. Avoid any unnecessary soldering joint. Preassemble and solder as many sections as possible on a workbench and make sure that every fitting that has to be soldered in place will be easily accessible (Monroe, 2004). The actual condition when the fillet metal is filling into the clearance between surfaces of the fittings is shown in Figure 3.8.

3.6.3.1 Soldering techniques

- 1) Make a swaged joint at the end of 1 length of cooper tubing that to be joined together by using swaging tool shown in Figure 3.4 (a).
- 2) Clean both the fittings with a brush and sand cloth.
- 3) Join the fittings together by sliding the end of a length of cooper tubing into the end of the other length of cooper tubing which has been swaged before.
- 4) Make the swaged joint upright and apply heat surround the top of the fitting.
- 5) Keep moving the flame surround the entire joint, spread the heat evenly and do not over heat any area. (It is very important that the flame be in motion and not remain on any one point long enough to damage the tube)
- 6) Touch the filler metal to the joint, when the joint is hot enough, the filler metal will melt and flow into the clearance by capillary attraction.
- 7) Let the filler metal continue to melt and flow into the clearance until the joint is seem to be fully filled, take away the filler metal and heat.
- 8) Let the joint to cool down.

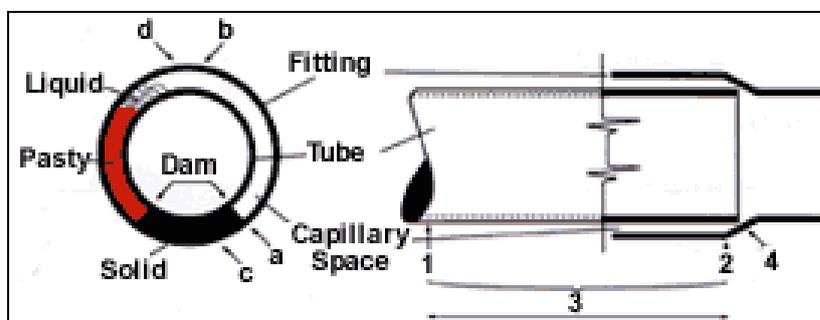


Figure 3.8: The phase distribution condition of the fillet metal while filling the clearance soldered joint by capillary attraction

Source: Copper Tube Handbook: VII. Brazed Joints (2009)

3.6.4 Comparison of the present and the past developed test rig

Figure 3.9 shows the process changes of the previous developed refrigeration system test rig to the present improved refrigeration system test rig.

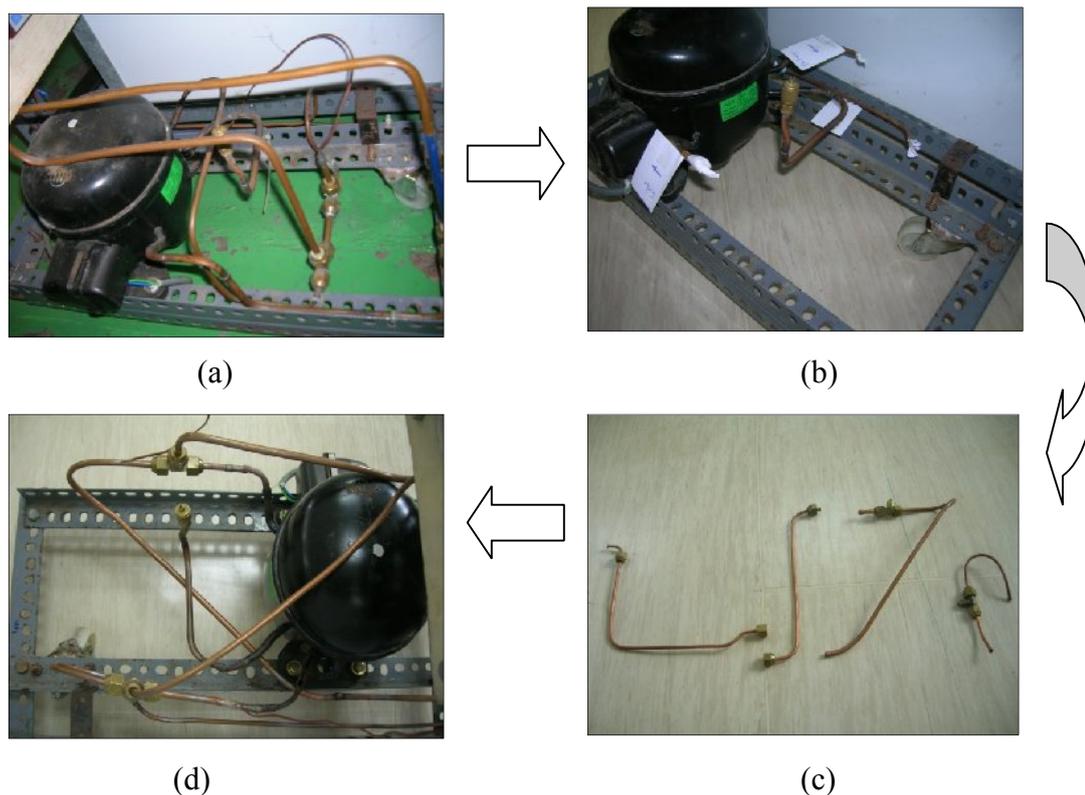


Figure 3.9: Overall process flow of the improved test rig fabrication: (a) Configuration before modification, (b) After disassembled, (c) New improved connecting cooper tubing, (d) Completed new reassembled test rig

3.7 SYSTEM EVACUATION

The term evacuation means to remove the undesirable or non-condensable gases, air and water vapor from the refrigeration system at the same time pulling a vacuum in the refrigeration system with a vacuum pump. Those of these substances mentioned are detrimental to the system. They would come into the system when the system is assembled or serviced. The non-condensable gases such as nitrogen will have occupied the condenser space when it moves through the condenser like liquid refrigerant. This will cause the discharge temperature and compression ratio to increase and cause unwanted inefficiency. The oxygen in the air will react with the refrigeration oil to form organic solids. Where as the water vapor will react with R-134a which is contain of either chlorine or fluorine to form acids and more water. Acids that formed will cause metal corrosion and sludge can produce inside the system (Whitman, 2005). That is why

evacuating the refrigeration system is essential to ensure the system can operate last longer and efficiently.

However the perfect vacuum has never been achieved. But there are instrument that enables the approach to the perfect vacuum.

Leak should be checked before system charging can be done. This is important because leakage allows air to enter the system during system evacuation is being done and it also allows refrigerant to leak out from the system during and after the system charging. There is strictly undesirable letting the refrigerant leak out from the system for the sake of the God's works since R-134a will contribute green house effect to the environment. Figure 3.10 shows the refrigeration system test rig is being evacuated.

3.7.1 Evacuation procedure

- 1) There are 3 ports at the charging manifold. Right port is connected to the charging port at the compressor, middle port is connected to the refrigerant cylinder and the left port is connected to the vacuum pump.
- 2) Open both the valves of the low and high pressure gauges on the left and right charging manifold while the valve of the refrigerant cylinder must be ensured closed at this state.
- 3) All connections are checked properly.
- 4) Switch on the vacuum pump.
- 5) Observe both the low and high pressure gauges at the charging manifold until vacuum is indicated.
- 6) The vacuum pump is switched off and the system is allowed to stand for same time period to see whether the pressure rises.
- 7) If pressure does rise and stop at some point, a material such as water is boiling in the system. If this occurs, continue the evacuation by switch on the vacuum pump again.
- 8) Stops the vacuum pump when both the low and high pressure gauges at the charging manifold show vacuum once again.

- 9) Observe both of the low and high pressure gauges. If the pressure continues to rise, there is a leak. The system should be leak checked and fixed. To check out for leakage, use soap bubbles, put the soap bubbles throughout on the tubing system, if there is a leak, it will be easily detected by observing the soap bubbles.
- 10) Switch on the vacuum pump and allow it run for 30 minutes.
- 11) Switch off the vacuum pump, the system evacuation is completed. The system is ready for refrigerant charging.



Figure 3.10: The refrigeration system test rig is being evacuated by vacuum pump

3.8 SYSTEM CHARGING

System charging is a process to add the refrigerant into the refrigeration system. The correct charge should be well defined and added into the system so that the system can operate well as it was designed.

However, the correct amount of charge depends on the operation temperatures and pressures of the system which had been designed. The refrigerant may be added to the system in the vapor state or liquid state by weighting, measuring, or using operation pressure chart.

In this refrigeration system test rig, R-134a will be used by the mean of vapor refrigerant charging. Vapor refrigerant will be charged into the refrigeration system from the high pressure side charging port of compressor. Liquid refrigerant charging normally accomplished in the liquid line such as charge into the king valve on the receiver (Whitman, 2005). The refrigeration system test rig is being charged with R-134a is shown in Figure 3.11.

3.8.1 Charging procedure

- 1) Disconnect the vacuum pump from the system by disconnecting the left port of the charging manifold from the vacuum pump. The left port of charging manifold is free of connection now.
- 2) Connect the refrigerant cylinder to the middle port of the charging manifold as mentioned early.

Note: With any blended refrigerant the refrigerant cylinder should be upside down when charging the system.

- 3) Connect the right port of the charging manifold to the charging port at the compressor as mentioned early.
- 4) The left port of the charging manifold remains free of connection.
- 5) Open the middle manifold valve and valve on the refrigerant cylinder while both the right charging manifold valves remain closed. Leave the cylinder for at least 10 seconds and then close the middle manifold valve again to give the system a low pressure static charge before starting the system up.
- 6) Slowly open the right side manifold valve until the pressure read about 45 psi, and then leaves it for few minutes to wait the refrigerant travels through the capillary tube into the low pressure side. The pressure of high pressure side will gradually decreases while the pressure of the low side will gradually increases, and eventually equalizes each other.
- 7) Repeat the process when the high and low side pressures equal about 0 to 10 psig and allow the system to equalize until the static pressure of the system is about 45 psi.
- 8) Turn on the compressor. You can observe the high side pressure gauge rises to the region of 160 to 190 psi depending on the component of the system

and the ambient temperature. The low side pressure can be observed decreases to slight vacuum and again the value depends on the component of the system.



Figure 3.11: The refrigeration system test rig is being charged with refrigerant 134a

3.9 EXPERIMENT FLOW CHART

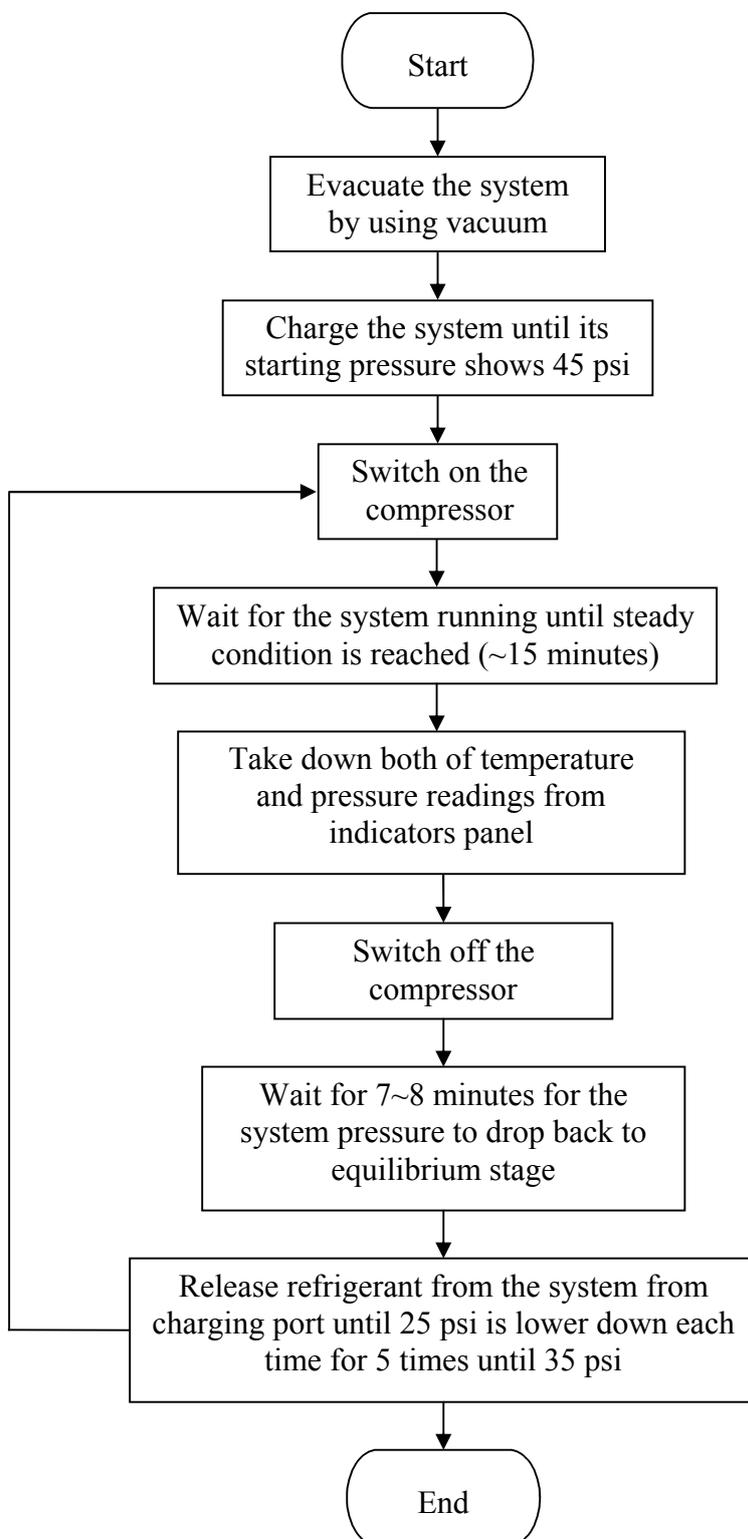


Figure 3.12: Flow chart shows the flow process of this experiment

CHAPTER 4

RESULTS AND DISCUSSION

4.1 INTRODUCTION

This chapter will discuss about the results obtained from the experiment done before. The terms to be discussed including compressor work per unit mass, W_c , heat rejection per unit mass, Q_H , refrigerating effect, Q_L , and refrigeration coefficient of performance, COP_R of the refrigeration system test rig.

The temperatures and pressures collected at the locations for each of the points of interest in the refrigeration system from this experiment, are used to determine the COP_R of the tested refrigeration system by using Eq. (2.8). In this experiment 5 sets of data were collected for 5 different quantities of refrigerant initial charges.

The most conventional method is used which is by using $p-h$ diagram for refrigerant 134a to obtain the enthalpy for each of the point of interests in this refrigeration system. Once the temperature and pressure for a particular point in the system is known, the enthalpy of that particular point can be easily obtained from the $p-h$ diagram.

The raw data for steady state condition are shown in Figure 4.1 and 4.2.

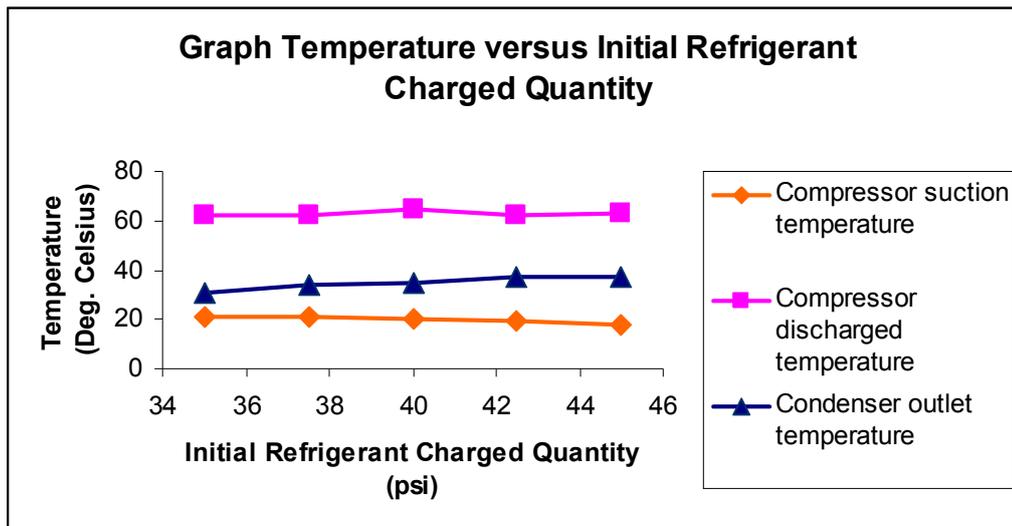


Figure 4.1: Temperature versus initial refrigerant charged quantity

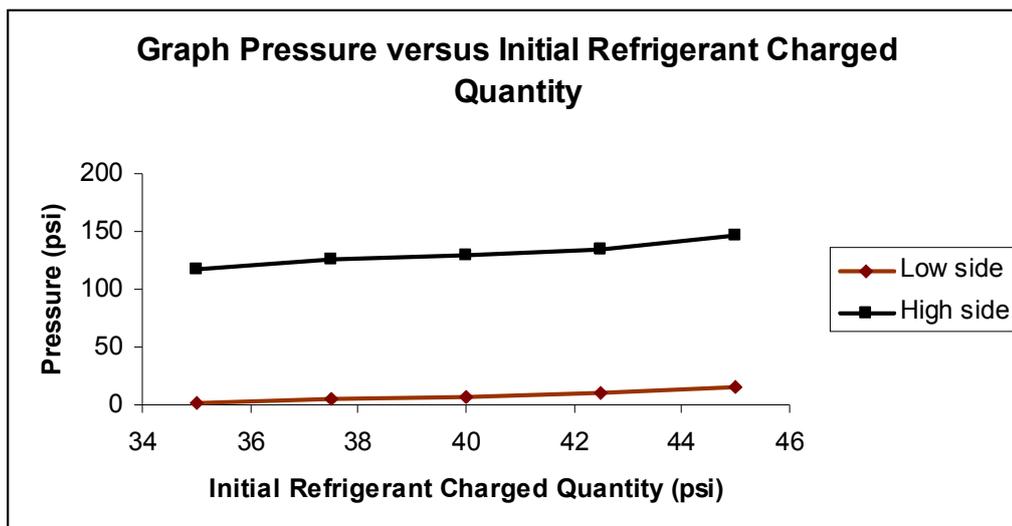


Figure 4.2: Pressure versus initial refrigerant charged quantity

4.2 RESULTS

4.2.1 Enthalpy obtained from the experiment

Enthalpy is a new extensive thermodynamic property which is the combination of other thermodynamic properties of internal energy and work done on its surroundings during the process. The values of enthalpy for each of the point of interest are

determined by using $p-h$ diagram shown in Appendix A and Eq. (2.5) is used to calculate for h_4 (Sonntag, 2003). The results are shown in Table 4.1.

Table 4.1: Enthalpies obtained for different initial refrigerant charged quantity

Initial Refrigerant Charged Quantity (psi)	Enthalpy (kJ/kg)			
	h_1	h_2	h_3	h_4
45.0	405	449	243	243
42.5	409	448	243	243
40.0	410	450	240	240
37.5	411	450	238	238
35.0	411	450	236	236

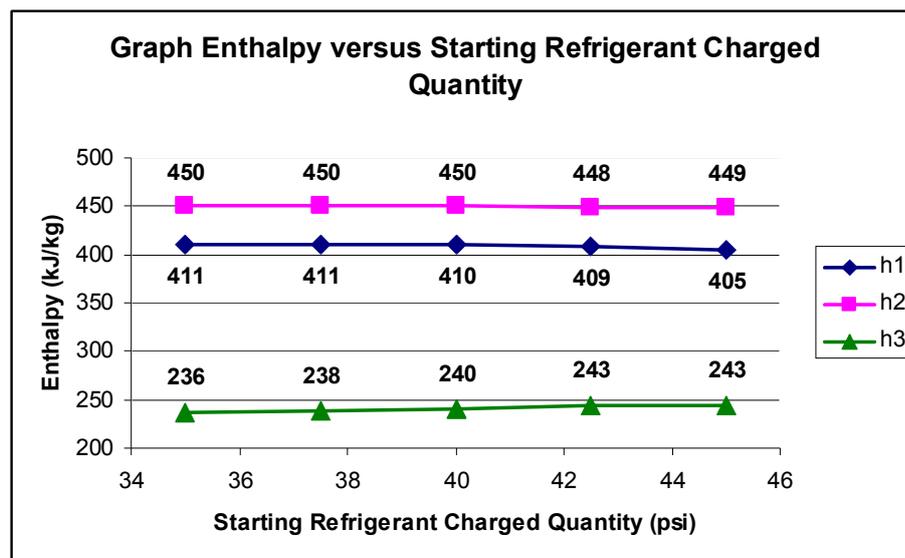


Figure 4.3: Graph enthalpy versus initial refrigerant charged quantity

The results obtained are tabulated and plotted. From Figure 4.3, it is observed that there is only a slightly change in enthalpy throughout the experiment. The variation of the value of enthalpy h_1 , h_2 , and h_3 show almost straight line up from 35 psi to 45 psi of initial refrigerant charged quantity. Even though it is only a slight change in enthalpy but it can lead to a big change in COP_r as will be shown later.

4.2.2 Compressor work per unit mass, W_c

Compressor work per unit mass is the work done needed to compress the refrigerant per unit mass drawn from evaporator from lower pressure to higher pressure and at the same time increase refrigerant temperature to at least 7°C higher than the ambience for effective heat transfer. The value of W_c is calculated by using Eq. (2.1) (Potter, 2004). The results are shown in Table 4.2.

4.2.2.1 Sample calculation for W_c at 45.0 psi

$$W_c = h_2 - h_1 = (449 - 405) \text{kJ} / \text{kg} = 44 \text{kJ} / \text{kg}$$

Table 4.2: Compressor work per unit mass calculated for different initial refrigerant charged quantity

Initial Refrigerant Charged Quantity (psi)	W_c (kJ/kg)
45.0	44
42.5	39
40.0	40
37.5	39
35.0	39

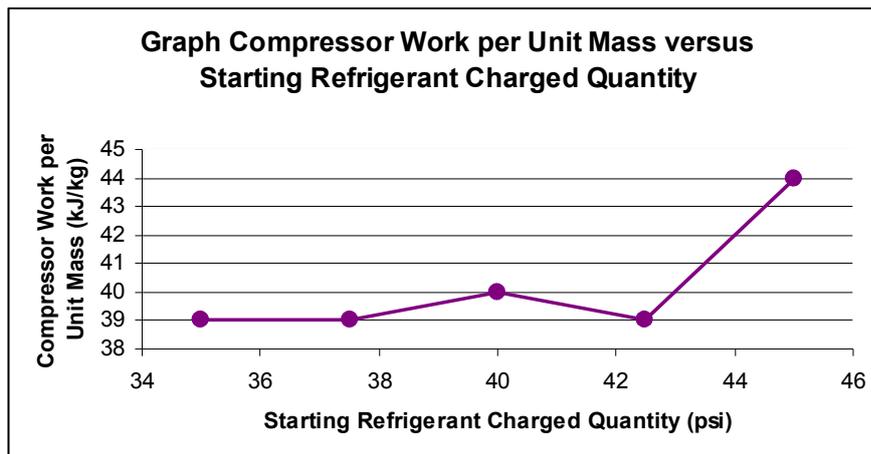


Figure 4.4: Graph compressor work per unit mass versus initial refrigerant charged quantity

From Figure 4.4 it can be observed that the work done by the compressor increases while the initial refrigerant charged quantity increases. It makes sense as the starting pressure is higher, then the compressor needs to do more work to start off the compression process. If the refrigerant charged quantity continues to increase until the limit of the compressor, it will finally cause the compressor to burn.

4.2.3 Heat rejection per unit mass, Q_H

Heat rejection per unit mass is the heat which transfers from the refrigerant to the ambience by condensation process in condenser. This heat transfer out the refrigeration system is including the heat absorbed from the refrigerated space by the refrigerant in evaporator and the heat produced by the compressor during compression process. The value of Q_H is calculated by using Eq. (2.3). The results are shown in Table 4.3.

4.2.3.1 Sample calculation for Q_H at 45.0 psi

$$Q_H = h_2 - h_3 = (449 - 243) \text{kJ/kg} = 206 \text{kJ/kg}$$

Table 4.3: Heat rejection per unit mass calculated for different initial refrigerant charged quantity

Initial Refrigerant Charged Quantity (psi)	Q_H (kJ/kg)
45.0	206
42.5	205
40.0	210
37.5	212
35.0	214

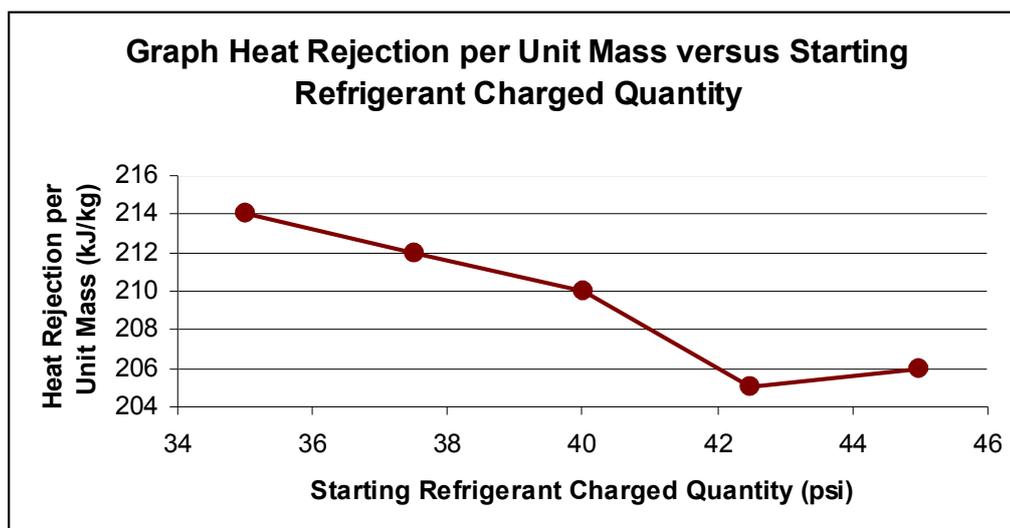


Figure 4.5: Graph heat rejection per unit mass versus initial refrigerant charged quantity

From Figure 4.5 it is observed that the highest heat rejection per unit mass from the system is at 35 psi initial refrigerant charged quantity. It is supposed that, the more refrigerant in the system, the more heat will be adsorbed and transferred out the system.

However too much increase in refrigerant charged quantity into a refrigeration system would somehow lead to a negative effect which is too much increase in working pressure in the system and due to the increase of the working pressure, the refrigerant

circulates faster inside the system, therefore it will lead to insufficient time for the refrigerant to remove the heat to ambience. Because the same size of condenser is used throughout the experiment. Thus, the refrigerant charged into the system will also lower down the heat rejection efficiency.

4.2.4 Refrigerating effect, Q_L

The refrigerating effect is the heat per unit mass absorbed by the refrigerant from the refrigerated space during evaporation process. The value of Q_L is calculated by using Eq. (2.6). The results are shown in Table 4.4.

4.2.4.1 Sample calculation for Q_L at 45.0 psi

$$Q_L = h_1 - h_3 = (405 - 243) \text{kJ} / \text{kg} = 162 \text{kJ} / \text{kg}$$

Table 4.4: Refrigerating effect calculated for different initial refrigerant charged quantity

Initial Refrigerant Charged Quantity (psi)	Q_L (kJ/kg)
45.0	162
42.5	166
40.0	170
37.5	173
35.0	175

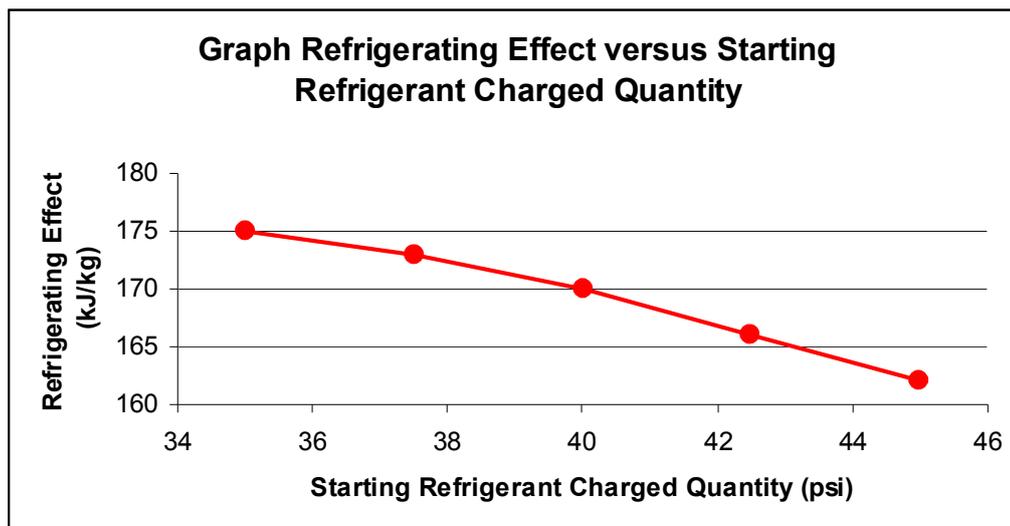


Figure 4.6: Graph refrigerating effect versus initial refrigerant charged quantity

Figure 4.6 shows an inversely proportional relationship between refrigerating effect and initial refrigerant charged quantity in this experiment. As Q_H , the refrigerating effect had an optimum value at 35 psi initial refrigerant charged quantity and decreases as the initial refrigerant charged quantity increases until the lowest value of 162 kJ/kg at 45 psi initial refrigerant charged quantity. This occurred due to the same reason as in Q_H . From Figure 4.4 and 4.5 the initial refrigerant charged for optimum performance is at 35 psi in this refrigeration system test rig. Hence, the exceed refrigerant charged into the system that beyond 35 psi initial pressure will cause too high working pressure in the system when the system running and lead to the insufficient time for the refrigerant to absorbed heat from the refrigerated space.

4.2.5 Refrigeration coefficient of performance, COP_R

The refrigeration coefficient of performance can be described as the ratio of the amount of energy received by the refrigeration system undergoing the cycle from the cold body, Q_L , to the net work into the refrigeration system to accomplish this effect, W_c . Thus the value of COP_R is calculated by using Eq. (2.8) (Moran, 2008). The results are shown in Table 4.5.

4.2.5.1 Sample calculation for COP_R at 45.0 psi

$$COP_R = \frac{Q_L}{W_c} = \frac{162kJ/kg}{44kJ/kg} = 3.68$$

Table 4.5: Refrigeration coefficient of performance calculated for different initial refrigerant charged quantity

Initial Refrigerant Charged Quantity (psi)	COP_R
45.0	3.68
42.5	4.26
40.0	4.25
37.5	4.44
35.0	4.49

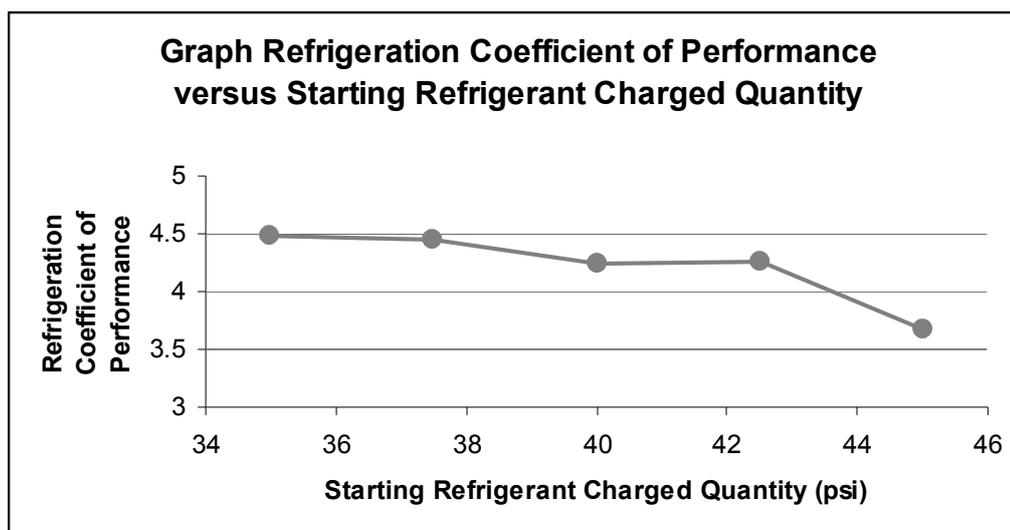


Figure 4.7: Graph refrigeration coefficient of performance versus initial refrigerant charged quantity

From Figure 4.7 the maximum COP_R is at 35 psi initial refrigerant charged quantity. The more the refrigerant charged into the refrigeration system, the higher the starting pressure hence the more work done needed by the compressor to compress the refrigerant, of course the discharge temperature and pressure of the refrigerant will be higher too. In fact, within the compressor limit, more refrigerant charged into a

refrigeration system, more work input by the compressor is needed but in return, the higher discharge temperature and pressure will be produced, if the system is large enough this is desirable because higher discharge temperature and pressure will increase the rate of heat rejection from the refrigeration system, thus in this case the COP_R will not be affected.

However in this experiment the size of evaporator is remain the same all the time as the refrigerant charged quantity increasing, it will turn out to be more compressor work done on the refrigerant but the limited heat, Q_L which depends on the size of evaporator can be absorbed by the refrigerant from the refrigerated space causes the value of COP_R which is the ratio of Q_L to W_c to drop.

Furthermore another effect that has been discussed above for Q_H and Q_L will be taking place and causes the value of COP_R drop furthermore. In this experiment, it is observed that the optimum initial refrigerant charge quantity that leads to the optimum performance of this refrigeration system test rig is at 35 psi which gives the value of COP_R 4.49.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 CONCLUSION

A previous developed refrigeration system test rig (mini size domestic refrigerator) was improved by modifying the tubing system from the previous more complicated design to the present more simple design. It was done by removing 2 of the cooper T-junctions from the tubing system. The 2 cooper T-junctions that had been removed, previously was used for the installation of thermocouple wires inside the tubing system. After the study, the author decided that letting the thermocouple wires attached on the outside surface of cooper tubing at the point of interest instead of putting the thermocouple wires inside the tubing system. With this simplification, 2 cooper T-junctions could be removed and hence the total length of the tubing system could be greatly reduced. By reducing the total length of tubing system, the performance of the refrigeration system test rig can be improved. The improvement had successfully been done by the author.

Besides, the locations at where the temperatures and pressures were collected for the determination of COP_R for the system had also been moved to the locations more nearer to the exact location of point of interest. Sometimes it is hard to locate the temperature and pressure measuring devices to the exact location of point of interest due to the structural configuration of the tubing system and the components of the system. However those temperature and pressure measuring locations could still be moved to the locations more nearer to the exact location of point of interest after the simplification of thermocouple installation had been done. The author had decided there were in total 3 temperature measuring points and 2 pressure measuring points in this

experiment. Those 5 measuring points had been moved to more suitable locations after the simplification of tubing system had been done.

For another concern, 5 different refrigerant quantities had been charged and tested in this refrigeration system test rig. The refrigerant quantity charged into the system was measured by the initial charged pressure within the range of 35 psi to 45 psi. The results obtained showed that the COP_R of the tested refrigeration system reducing while the refrigerant quantity increasing. The optimum COP_R occurred at 35 psi initial refrigerant charged quantity which has the value of 4.49. From this experiment, the results showed that the quantity of refrigerant charged into system does affect the performance of the refrigeration system.

The COP_R for each of the 5 different refrigerant quantities were then computed based on the data collected from the experiment by using the equation given. The results for COP_R obtained were as follow, 4.49, 4.44, 4.25, 4.26 and 3.68 for the initial refrigerant charged quantity of 35 psi, 37.5 psi, 40 psi, 42.5 psi, and 45 psi respectively.

5.2 RECOMMENDATIONS

The objectives of this project were successfully achieved as discussed in the conclusion before. However there is still room for improvement of the improved refrigeration system test rig. If the highly accurate results are desirable, there are some recommendations for it.

Firstly, 2 more pressure measuring devices usually bourdon tube pressure gauges could be added on the test rig so that the pressure drop can be investigated while the refrigerant travelling through inside the tubing system. Even though the pressure change during the condensation and evaporation process is small enough to be ignored in small size of refrigeration system but it cannot be denied that there are changes in refrigerant pressure during those processes although it is small. However for bigger size of refrigeration system, the mentioned pressure changes would be large enough to highly affect the accuracy of the calculated results.

Finally, insulation could be done better especially at the locations where the temperatures points are to be collected. In this experiment, thermocouple wires were put outside on the surface of the cooper tubing. Therefore insulation is very important at these temperature measuring points to ensure the correct temperature could be sensed correctly by the thermocouple since now the thermocouple wires had no direct contact with the refrigerant inside the tubing system. Insulation also important at other parts of the tubing system in order to prevent any heat transfers from and to the refrigeration system. Any heat transfers from and to the refrigeration system will lower down the performance of the system.

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APPENDIX A

Pressure-Enthalpy ($p-h$) diagram for refrigerant 134a at optimum condition (35 psi initial refrigerant charged)

