# PERFORMANCE INVESTIGATION OF VEHICLE AIR CONDITIONING SYSTEM UNDER DIFFERENT COMPRESSOR SPEED

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**PERFORMANCE INVESTIGATION OF VEHICLE AIR CONDITIONING UNDER DIFFERENT COMPRESSOR SPEED**

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**A report is submitted**
**In fulfilment of the requirements for the degree of**
**Bachelor of Mechanical Engineering**

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## DECLARATION

I declared that this project report entitled "Performance Investigation of Vehicle Air Conditioning Under Different Compressor Speed" is the result of my own work except as cited in the references.

Signature : ………………………………………

Name : ………………………………………

Date : ………………………………………

## APPROVAL

I hereby declare that I have read this project report and in my opinion this report is sufficient in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering (Thermal-Fluids).

| | | |
|---|---|---|
| Signature | : | ..................................................... |
| Supervisor's Name | : | ..................................................... |
| Date | : | ..................................................... |

## DEDICATION

This thesis are dedicated to whoever shows me their love even during the hard times, gave the pure kindness when things fall apart and endless support when things seems impossible to get through. Dedication is sincerely made for the beloved family, supporting friends and dedicated lecturers

## ABSTRACT

This performance investigation of vehicle air conditioning system under different compressor speed aims to justify the best coefficient of performance (COP) of the air conditioning system while operate in vary compressor speed. Hence, the temperature, pressure, mass flow rate, enthalpy and cooling capacity are determined. By referring to the objectives, this project basically shows the interaction between the compressor speed, energy consumption and coefficient of performance (COP). The experimental work was carried out on air conditioning test rig and R134a was used as refrigerant. Based on compressor capacity, the compressor speed was controlled between 1500 rpm until 1900 rpm. While others parameter such as ambient temperature, heat load and amount of condensate water are maintained. The analysis of data was done by used thermodynamic software (REFPROP). From this project, it is found that when the compressor speed at 1700 rpm, it is work at optimum because it has highest COP which is 4.40.

# ABSTRAK

Penyiasatan prestasi sistem penyaman udara kenderaan pada kelajuan pemampat yang berbeza ini bertujuan untuk mewajarkan pekali prestasi (COP) sistem penyaman udara yang terbaik ketika beroperasi dalam berbagai kelajuan pemampat. Oleh itu, suhu, tekanan, kadar aliran jisim, entalpi dan kapasiti penyejukan bagi sistem tditentukan. Merujuk kepada objektif, projek ini secara dasarnya menunjukkan interaksi antara kelajuan pemampat, penggunaan tenaga dan pekali prestasi (COP). Kerja eksperimen dijalankan pada rig ujian penyaman udara dan R134a digunakan sebagai bahan pendingin. Kelajuan pemampat dikawal antara 1500 rpm sehingga 1900 rpm. Manakala, parameter lain seperti suhu ambient dan beban haba dimalarkan. Analisis data dilakukan dengan menggunakan perisian termodinamik (REFPROP). Akhir sekali, didapati bahawa apabila kelajuan pemampat adalah 1700 rpm, ia bekerja pada keadaan yang optimum, di mana COP pada kelajuan tersebut adalah paling tinggi iaitu sebanyak 4.40.

## ACKNOWLEDGEMENTS

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Not to be forgotten, to express my sincere gratitude to my whole family member as my pillar strength who always showed me their eternal love and support especially throughout this degree journey in order to ensure that I be able to pursue my dreams and completed my studies successfully. Without them I would never make it this far.

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## LIST OF ABBREVIATIONS

| | |
|---|---|
| **COP** | Coefficient of Performance |
| **CTC** | Compact Tube and Centre Condenser |
| **FOT** | Fixed Orifice Tube |
| **GHG** | Greenhouse Gases |
| **HTC** | Headered Tube and Centre Condenser |
| **HVAC** | Heating Ventilation Air Conditioning |
| **ICE** | Internal Combustion Engine |
| **RON** | Research Octane Number |
| **RPM** | Rotation per Minute |
| **SOP** | Standard of Procedure |
| **TFC** | Tube and Fin Condenser |
| **TXV** | Thermostatic Expansion Valve |
| **VCRS** | Vapour Compression Refrigerant System |

# LIST OF SYMBOLS

| | | |
|---|---|---|
| **h** | - | enthalpy |
| $\dot{m}$ | - | mass flow rate |
| **N** | - | Number of revolutions per minute |
| **P** | - | Pressure |
| $\dot{Q}$ | - | Heat |
| **T** | - | Temperature |
| $\dot{W}$ | - | Work done |

# CHAPTER 1

# INTRODUCTION

## 1.0 Background

Air conditioning was first proposed to automobile industry by the Packard Motor Car Company during 1940 and followed by Cadillac in 1941. The acceptance of the air conditioner to the automobile industry as a car accessory does not get a high demand when first it was introduced. However, the popularity of air conditioning keeps increasing by the year until the early 1960s where it becomes a famous option (Schnubel, 2009).

Air conditioning is the adjustment or regulation of air in a closed space by the process cooled, heated, cleaned or filtered, humidified or dehumidified and circulated or recirculated either by heating or refrigerating (cooling by removal of heat). It is important to control the quantity (volume) and quality (temperature and humidity) of conditioned air at any time in varies condition (Schnubel, 2009).

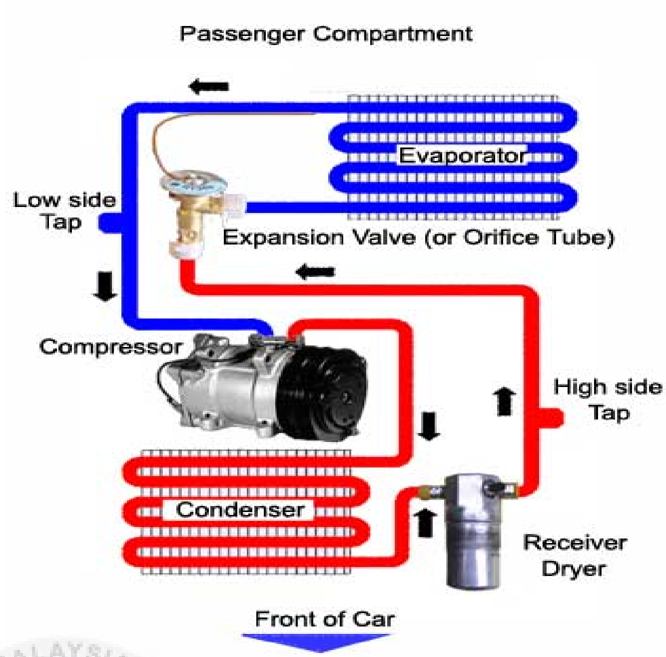



Figure1.1: Process flow diagram of the vehicle air conditioning system (Parker, 2016)

Air conditioning of vehicle is equipped with the compressor, condenser, metering devices and evaporator. Figure 1.1 shows the operation of air conditioning system of the vehicle which divided into two sides: high-pressure side, which has higher pressure and temperature, and the low-pressure side, which has low pressure and temperature (Birch, 2010). The evaporator and the expansion device are the components of the low sides while for the high sides, it major components are the compressor and the condenser.

In overall, this study is to investigate the performance of the vehicle air conditioning system that focuses more on the compressor components at various speeds. The relationship between the compressor speed, the coefficient of performance (COP) and the energy consumption is concluded at the end of the study.

## 1.2 Problem Statement

In the automotive industry, air conditioning is one of the important systems that widely used as the accessory for the vehicle such as the car that its main function is to provide thermal comfort for the users either by heating or cooling depends on the environmental condition. Geographically Malaysia is located in equatorial region which made it a hot and humid country. In this cases, air conditioning system has to work efficiently according to the climate changing which to be specified it is affected by factors; temperature, humidity and air velocity. The challenge is to ensure the air conditioning system achieve the optimum thermal comfort by control the parameter of compressor speed.

Fluctuation of fuel prices over the times in oil and gas industry especially in Malaysia made the saving in energy consumption very important. Energy consumption significantly in automobile industry tends to affect by the air conditioning system. Differ from four season country which used the heating system, in Malaysia cooling system is used in air conditioning system due to hot weather. Thus, due to low-temperature settings in the cooling system it required the compressor to work in the higher capacity which increases the fuel consumption. The compressor is known as the heart of air conditioning system is controlled by adjusting temperature of the thermostat at the relevant level to save energy consumption.

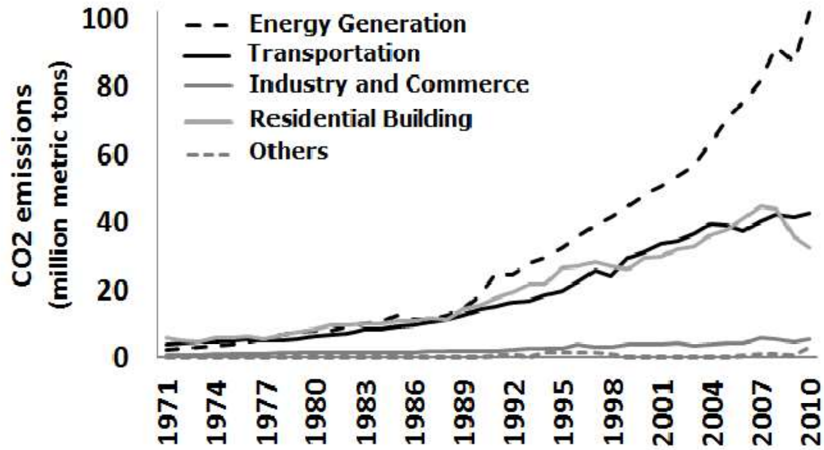



Figure1.2: Carbon emission in million metric tons by different sectors in Malaysia (1971-2010) (Tradingeconomics.com, 2013).

The annual trends in Figure1.2 showed the increment of carbon dioxide emission throughout the year (1971-2010) caused by different sectors. In 2010, the emission of $CO_2$ produced by transport sector of Malaysia is 42.43 million metric tons which equivalent to 22.9% of $CO_2$ emission in total at Malaysia (Shahid, Minhans, & Puan, 2014). The emission of $CO_2$ majorly causes by incomplete combustion of fuel. In an internal combustion engine (ICE), energy from fuel and air mixture is released by the chemical reaction of oxygen in air and hydrocarbons fuel to produce work. Type of fuels that typically used for combustion is diesel and gasoline. For example, Malaysia widely used RON95 and RON97 which contained hydrocarbons which caused the emission of greenhouse gases (GHG). GHG in the earth's atmosphere is exist in the form of water vapour which contains the small mixture of carbon dioxide ($CO_2$), methane ($CH_4$) and nitrous oxide ($N_2O$) (Abdullah, 2017). Although it seems impossible to completely prevent the emission of gases, by hope the control of compressor speed can decrease the pollutants that emit to air by lower the requirement of work with the highest efficiency.

Therefore, as major energy consumption in typical land vehicle, it is crucial to understand how to operate the air conditioning system in the most efficient way. Due to this reason, this study is proposed to investigate the effect of compressor speed on the characteristics of the air conditioning system especially its performance.

## 1.3 Objectives

To accomplish this project with huge success the objective has been set. The purposes of the project are:

I. To determine the system temperature, system pressure, cooling capacity and compressor work at different compressor speed.

II. To justify the best coefficient of performance (COP) of the air conditioning system.

## 1.4 Scope of Project

Scopes are the limitations of the study that covered from three aspects of methodology, variables covered and validity of results.

The methodology is a way that used in the project to collect the data and how to obtain the result with high accuracy. Out of analytical and simulation method, the project are conducted by the experimental method where all the work is done in a laboratory by following the standard of procedure (SOP) for ensure the safety. The data was obtained by adjusted, fixed and observed specified parameter.

For the study of air conditioning under different compressor speed, the variable that was manipulated was the speed of the compressor in the unit of rotation per minute (rpm) which the ideal value varies from 1200 rpm up to 1900 with tolerance ± 0.01. After few trials was made, the value of 1500, 1700 and 1900 rpm was chose for this experiment. Since if the speed was at 1300 rpm, the system tend to shut down by itself in the middle of operation while if the speed was less than 1200 rpm, the speed was not enough to drive the motor and compressor together. The speed supposed not exceeding 2000 rpm to prevent the motor from overheated. Meanwhile, the fixed variable of this study are maintained the ambient temperature of surrounding around 28 ºC with natural increment up to 2 ºC. The temperature of inlet evaporator was also fixed at 30 °C ± 1 ºC.

Lastly, results obtained from this project were only valid or applicable for studies air conditioning for the cooling system in Malaysia.

# CHAPTER 2

# LITERATURE REVIEW

## 2.0 Introduction

The main function of air conditioning in automobile industry are provided the thermal comfort to the occupants while operate efficiently. Hence, this research is focussed on the performance investigation of air conditioning by identify the most suitable speed of compressor which has greater COP.

HVAC is defined as technology of high indoor air quality that widely used in residential buildings, commercial building, industry and vehicular. The operation of heat removal from the passenger compartment of air conditioning system is divided into low side (low pressure and low temperature) and high side (high pressure and high temperature). The ultimate goal of the system is to properly regulate the temperature and humidity in order to provide the thermal comfort to living environment either by cooling or heating load.

Figure 2.1, it shows the overall overview study of this research which has been more specified. This chapter will discussed about the basic components of air conditioning, basic operations, how to achieve the best COP value in order the system can perform optimally and the experimental investigation by previous researcher.

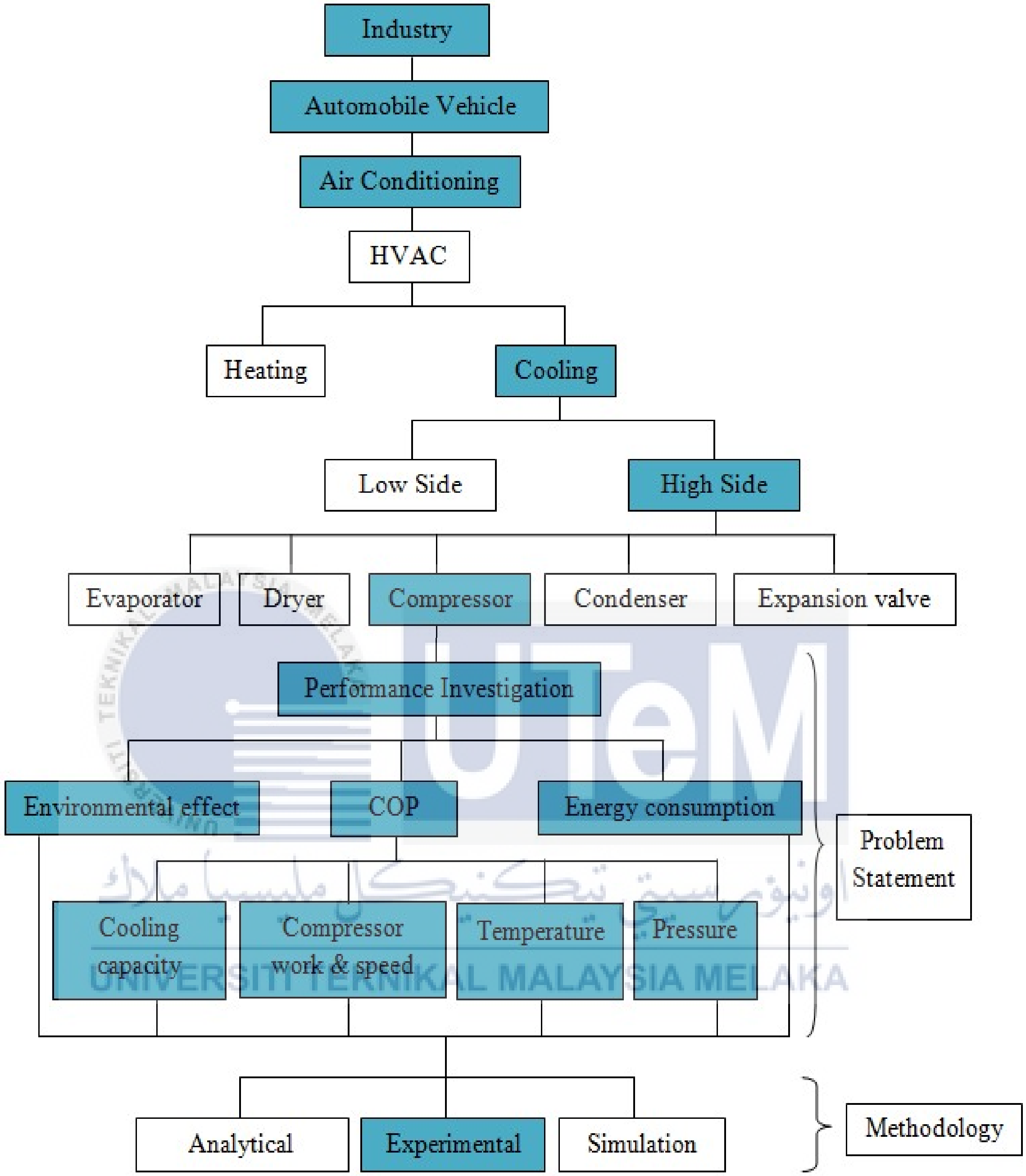



Figure 2.1: Overview study of Vapour Compression System (VCRS)

## 2.1 Vapor Compression System

In automotive industry the most widely type of cooling system used was vapour compression. Figure 2.3 shows the vapour compression system which typically designed from major components (compressor, condenser, evaporator, expansion valve and dryer) and additional components (fans and sensors) to accomplish their task (Bentrcia, Alshatewi and Omar, 2017).

The evaporator was used to absorb the cabin heat load and therefore mixture refrigerant inside the evaporator coils evaporates and become vapour. Then, the low pressure vapour was driven out from the evaporator to the compressor. The vapour refrigerant from compressor was being compressed by the low pressure vapour to high pressure vapour. Consequently, it also circulated the refrigerant throughout the cycle. The condensation of high pressure vapour to high pressure liquid takes place at the condenser. At this part, the removal of heat occurred and the heat was dissipated to the outside air. Finally, the cooled high pressure liquid leaving the condenser through the expansion valve by reduces its pressure. The low pressure liquid is send back to the evaporator as the input to ensure the continuous of cycle.



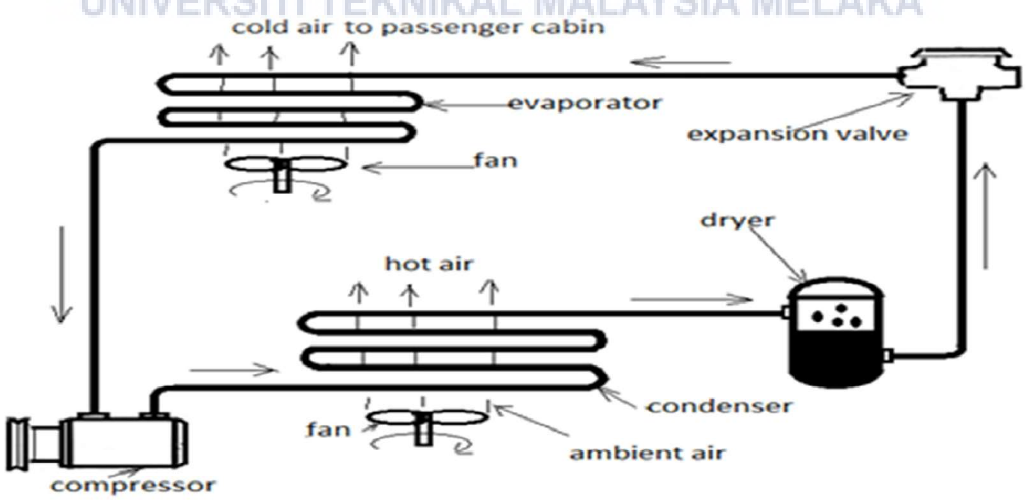



Figure 2.2: Refrigerant cycle of vapour compression A/C system for automobiles

(Bentrcia, Alshatewi and Omar, 2017)

Figure 2.4 shows the p-h diagram and t-s diagram of VCRS which consists of four processes. Process 1-2 is isentropic compression of dry saturated vapour (state 1) causes the temperature raise where the state was change to superheated vapour (state 2). Process 2-3 was isobaric condensation of saturated vapour where heat rejection occurred. The sub cooling point at the figure shown was achieved by the increase of the refrigeration effect without increased the work. Both superheating and sub cooling process affect the cooling capacity and work done by compressor. Process 3-4 was adiabatic expansion by irreversible throttling process where no heat addition and rejection during the process. Hence, the work done during throttling process is zero. Process 4-1 was isobaric and isothermal vaporization where the heat addition occurred. The heat addition (4-1) and rejection (2-3) causes the pressure drop due to friction. The drop in discharge pressure during heat rejection and suction pressure during heat addition increase the refrigeration effect, increase the work done and decrease the COP.



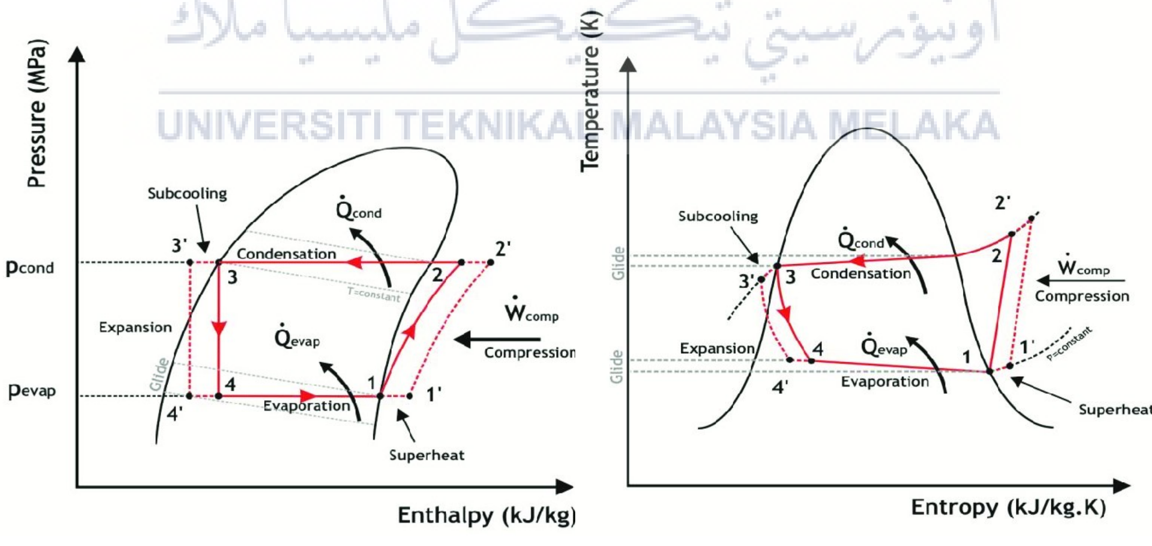



Figure 2.3: p-h diagram and t-s diagram of VCRS (Duarte, Pires, Silva & Gaspar, 2017)

Heat addition at evaporator;

$$\dot{Q}_L = \dot{m}\ (h_1\text{-}h_4) \quad \text{......} 1$$

Heat rejection at ;

$$\dot{Q}_H = \dot{m}\ (h_2\text{-}h_3)\ @\ \dot{Q}_H = \dot{Q}_L + \dot{W}_{in} \quad \text{......} 2$$

Work done by compressor;

$$\dot{W}_{in} = \dot{m}\ (h_2\text{-}h_1) \quad \text{......} 3$$

Coefficient of Performance;

$$COP = \frac{Refrigerati \quad Effect}{Work\ Done} = \frac{QL}{Win} = \frac{h1-h4}{h2-h1} \quad \text{......} 4$$

## 2.2 Basic Components

### 2.2.1 Evaporator

Similar to condenser, evaporator also acts as heat exchanger but work in opposite way from condenser which evaporator convert state of liquid to vapour. It is called evaporation process. It is located inside the passenger compartment which is hidden behind the dashboard.  Evaporator usually is made from the aluminium. Two types of evaporator is forced convection type and natural convection type.

In HVAC, evaporator provides cold air to the system by cooling and dehumidifying process. The refrigerant from expansion device (orifice tube or thermal expansion) enters the evaporator at low pressure and low temperature. The cold refrigerant passes upward through the evaporator core and flow towards the evaporator fin. Across the evaporator fins, the heat from warm air is dissipate by transfer its thermal energy into the cooler refrigerant. The low in air's temperature caused the vaporisation of the refrigerant. Hence, the enough amount of heat received by the refrigerant allows the low pressure and low temperature liquid to convert into low pressure and low temperature gas.



As mentioned earlier, the evaporator also performed dehumidification pr3ocess by the help from fan and blower motor. This operation is important especially during defroster operation because the evaporator has good efficiency of moisture removal. Thus, it helps to protect the windshield from formation of fog and frost. Some failures that can relate to the evaporator are leaking, dirty cooling fins and blocked refrigerant passage. Thus, result in bad performance of the cooling system.

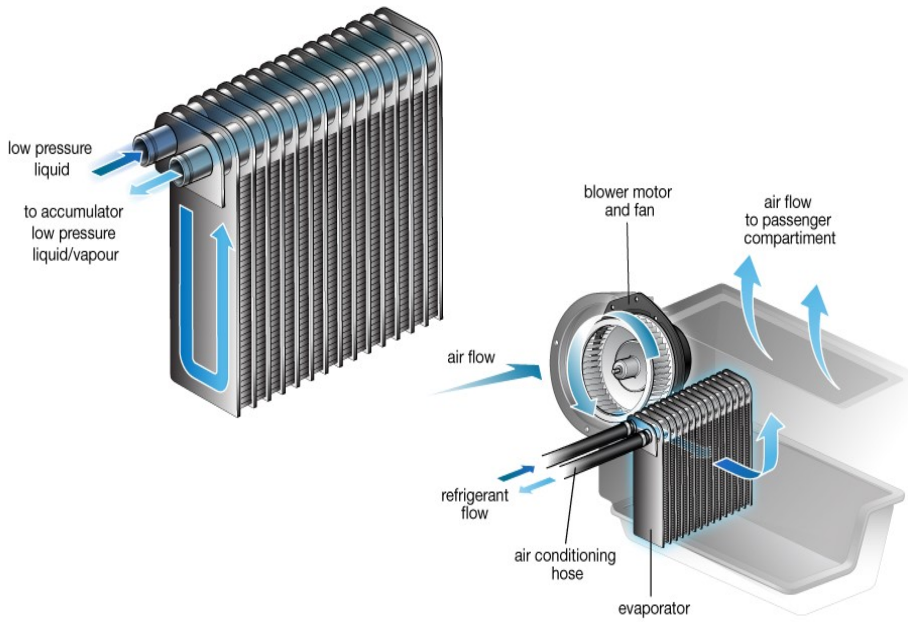



Figure 2.4: A typical evaporator in automobiles (Bevacqua, 2018)

### 2.2.2 Compressor

In air conditioning system, the compressor is known as the heart of the system. The three main tasks that performs by compressor is regulate the circulation of refrigerants through the cycle, increases the pressure and temperatures of refrigerants through the evaporation and condensation.

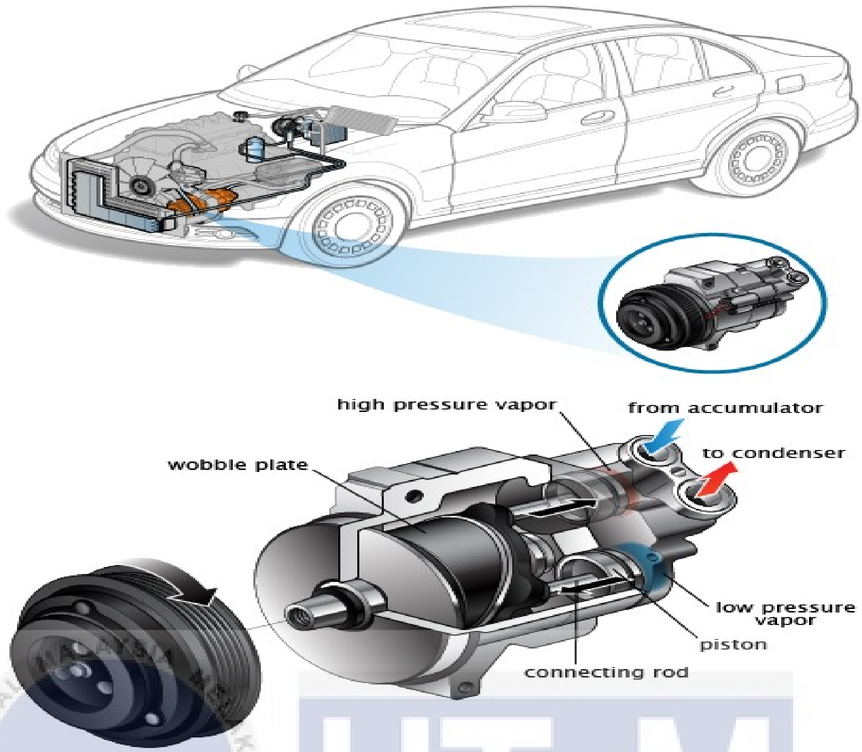



Figure 2.5: A typical compressor in automobiles (Bevacqua, 2018)



As seen in figure 2.6, the compressor is classified into two types, based on its principle of operation; dynamic and positive displacement. The dynamic in this case means the energy from the rotating impeller is transfer to the air. Meanwhile, the meaning of positive displacement is the compressor works either by reducing the volume to increase the pressure of air or carry air to discharge opening without change its volume.

The centrifugal compressor is classified under dynamic compression. The operation converts the kinetic energy (velocity) to static energy (pressure), to produced high pressure and temperature of refrigerants. The main components of centrifugal compressor is the rotating impeller

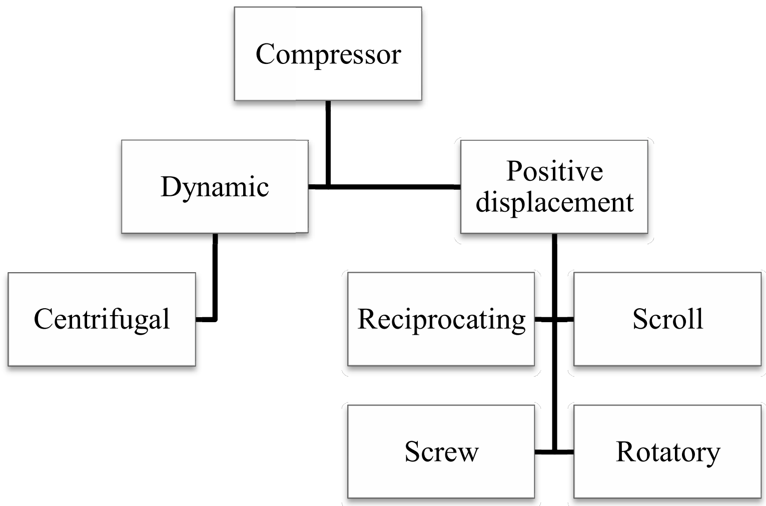



Figure 2.6: Classification of compressor types used in HVAC

### 2.2.3 Condenser

The condenser is installed in front the vehicle radiator. Main types of condenser are compact tube and centre condenser (CTC), headered tube and centre condenser (HTC) and tube and fin condenser (TFC). The condenser is simple devices that are constructed from condenser core. The core is build from series of tube which is surrounded by cooling fins. The fins provide the surface area to allow the removal of heat.

The condenser act as heat exchanger by undergoes condensation process to get rid heat from passenger compartment. The refrigerant containing heat is said to be superheated when the refrigerant is compressed by compressor. This refrigerant has high pressure and temperature flow from compressor to the condenser. The refrigerant passes the condenser coil has less heat compared to surrounding air, so the surrounding air carry the heat from the refrigerant when passing the coil and fins. The condensation begins when the heat is removes from the refrigerant caused state changing from high pressure vapour to high pressure liquid. The engine cooling fan, enhance the air flow through the condenser and radiator.

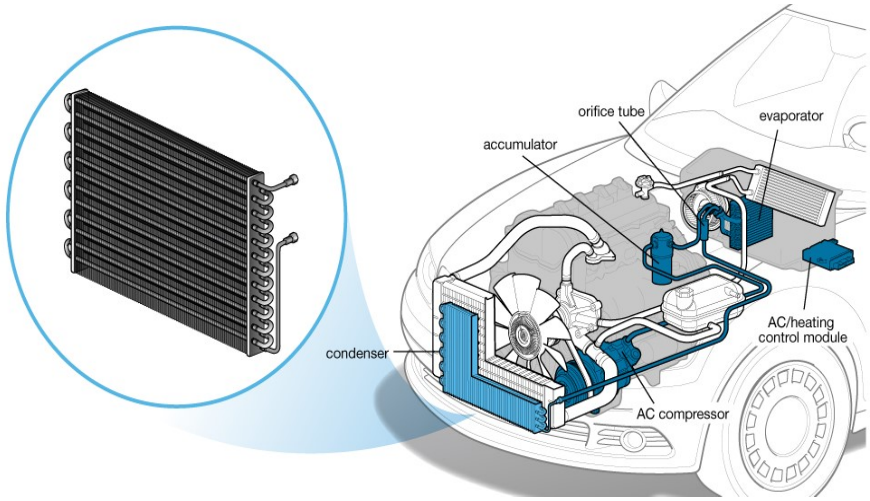



Figure 2.7: A typical condenser in automobiles (Bevacqua, 2018)

### 2.2.4 Expansion valve

The expansion valve also known as throttle valve or metering device is located in liquid line that connected the evaporator and the condenser. The expansion valve is required in VCRS due to differences in pressure between the condenser and the evaporator. Hence, the expansion valve helps to regulate the flow of refrigerant between high pressures liquid from the condenser entering the evaporator at cold low pressure liquid.

Two common type of expansion valve that used nowadays is fixed orifice tube (FOT) and thermostatic expansion valve (TXV). FOT is also known as expansion tube, it much simpler than TXV. FOT is made up from tiny brass tube enclosed in a plastic housing with a mesh filter. FOT is installed with a small mesh screen on each side of the tube. TXV function is to detect the temperature and pressure by using temperature sensing bulb and capillary tube.

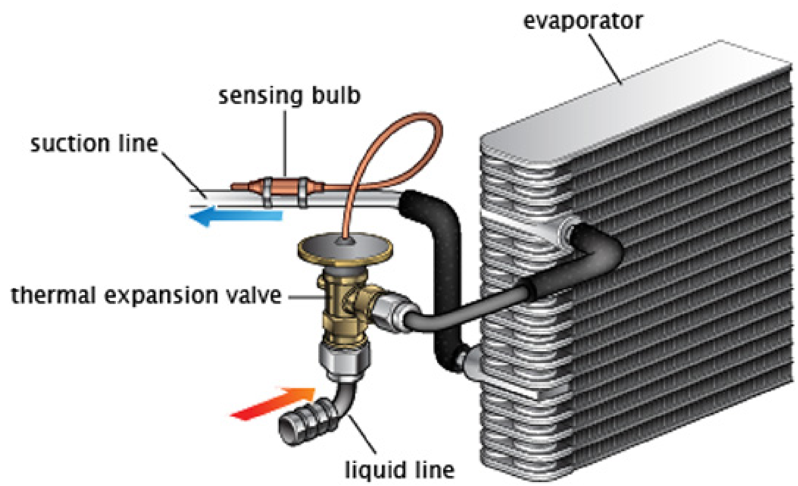



Figure 2.8: A typical expansion valve in automobiles (Bevacqua, 2018)

## 2.3 Experimental Investigation by Previous Research

Before conducting the experiment, the investigation of this experiment done by do some study based on previous research. The journal, research paper and article are studied.

Table 2.2.1: Summary of experimental studied by previous research

| Author | Title | Research Objectives | Experimental setup | Result/ Analysis |
|---|---|---|---|---|
| Afiq Aiman | Efficient and Green Vehicle Air Conditioning Using Electric Compressor | To investigated the relationship between the variable speed of compressor (1800, 2000, 2200, 2400 and 2500 rpm) with the cabin temperature and fuel consumption. 1000 W of the internal heat load is set at temperature of 21 ºC. The electrically-driven compressor (EDC) with power of 12 volt lead acid vehicle battery is proposed in this experiment. The performance of EDC and conventional belt-driven system is compared. | a) R-134a is used the refrigerant to run the system.<br>b) The vehicle at speed 50, 70, 90 and 100km/h and EDC speed at 1800, 2000, 2200, 2400 and 2500 rpm is run simultaneously.<br>c) The EDC and original automotive air conditioning (ACC) system is compared. The internal heat loads of 1000W has been set and the compressor speed is run, up to 2500 rpm. During the experiments, the set-point temperature is fixed at 21 ºC. | a) The higher the compressor speed, the higher the refrigerant flow rate, the higher the cooling capacity.<br>b) The higher the compressor speed, the higher the power consumption.<br>c) The power consumption increment ratio is higher than cooling capacity as the compressor speed increases.<br>d) The higher the compressor speed, result in thermodynamic losses, the mass flow rate of refrigerant is unstable.<br>e) The EDC has better performance compared to conventional-belt driven system. |

| Author | Title | Research Objectives | Experimental setup | Result/ Analysis |
|---|---|---|---|---|
| J.M. Saiz Jabardo, 2002 | Modelling and Experimental Evaluation of an Automotive Air Conditioning System with a Variable Capacity Compressor | Experimental evaluation on the variable capacity compressor and a thermostatic expansion valve in addition to the evaporator and micro channel parallel flow condenser. | The experimental set-up was made up of original components from the air conditioning system to emulate those in the actual vehicle. The compressor was run by an electrical motor acted upon by a frequency converter in order to cover the whole range of rotational speeds in the actual vehicle. | a) When the vehicle is operating, the refrigerating capacity does not affect by the variable parameter such as the temperature of condensing air and compressor speed. This action is due to capacity control mechanism of the compressor.<br>b) The evaporator return air temperature influenced the refrigerating capacity.<br>c) The rate of mass flow, refrigerating capacity and COP has linear effect to the condensing and return air temperatures and compressor speed.<br>d) The temperature of condensing air has slight effect to the rate of mass flow, refrigerating capacity and COP. The return air temperature of evaporator has opposite effect to the temperature of condensing air. |

| Author | Title | Research Objectives | Experimental setup | Result/ Analysis |
|---|---|---|---|---|
| Zulkifli et al., 2015 | Impact of the electric compressor for automotive air conditioning system on fuel consumption and performance analysis | To analyzed the comparison between the electric compressor and conventional compressor on the fuel consumption and air conditioning performance. | a) The car battery that charged by the alternator, powered the electric compressor.<br>b) The thermocouples which connected to the temperature data logger is used to measured the temperature.<br>c) The pressure gauge and flow meter is used to take the manual reading of pressure and mass flow rate respectively.<br>d) The flat road condition is emulated by setup the vehicle on roller dynamometer.<br>e) HFC-R134a is used the refrigerant to run the system.<br>f) The electric compressor at speed of 2500 rpm is run.<br>g) In the evaporator, the on / off controller is placed.<br>h) At temperature of 30 ºC, the tests are began.<br>i) The usage of fuel within the one hour is measured by using the fuel flow meter. | a) The fuel consumption of electric compressor is better than the conventional compressor.<br>b) COP of electric compressor is better, since the energy required to compress the refrigerant is less.<br>c) The higher the speed of vehicle, the higher the compressor speed, the higher the energy consumption, the lower the COP for conventional compressor.<br>d) The electrically driven compressor has higher COP compared to the belt-driven compressor. The higher the COP results in reduction of fuel consumption due to load reduction. |

## CHAPTER 3

## METHODOLOGY

### 3.0 Introduction

The actions that need to be carried out to achieve the objectives in this project are listed below.

I. Introduction

Thread of an idea which develops from the background of the study, problem statement, objectives and scopes.

II. Literature review

Finding information based on the previous study by refers to journal, articles and research paper that obtained via Mendeley. Additional sources also obtained through the search engine such as Google and trusted website.



III. Conduct the experiment.

The experiment is conducted by following the SOP where the details of each step, variables covered, equipment and the set up of apparatus are provided.

IV. Data collection

Data of study is collected based on the experiment of the air conditioning system at different compressor speed.

V. Data accuracy check

The data obtained is rechecking to ensure the result obtained is relevant as the previous study. If the accuracy of data is high the next step is taken but if there is any error detected the experiment is conducted again by taking the precaution steps and the new data is collected.

VI. Result analysis

The data is analysed by calculation of COP and the relationship between various compressor speeds, COP and energy consumption is presented graphically.

VII. Conclusion

Summary of the experiment is concluding based on evidence through the experimental result. The factor and effect that has relation to the result are explained thoroughly.

VIII. Report writing

Information that obtained from the beginning until the end of studies is compiling as a report and follows the writing format.

Flowchart in Figure 1.3 showed the overall process from the early studies is started until the end where the results are summarized.

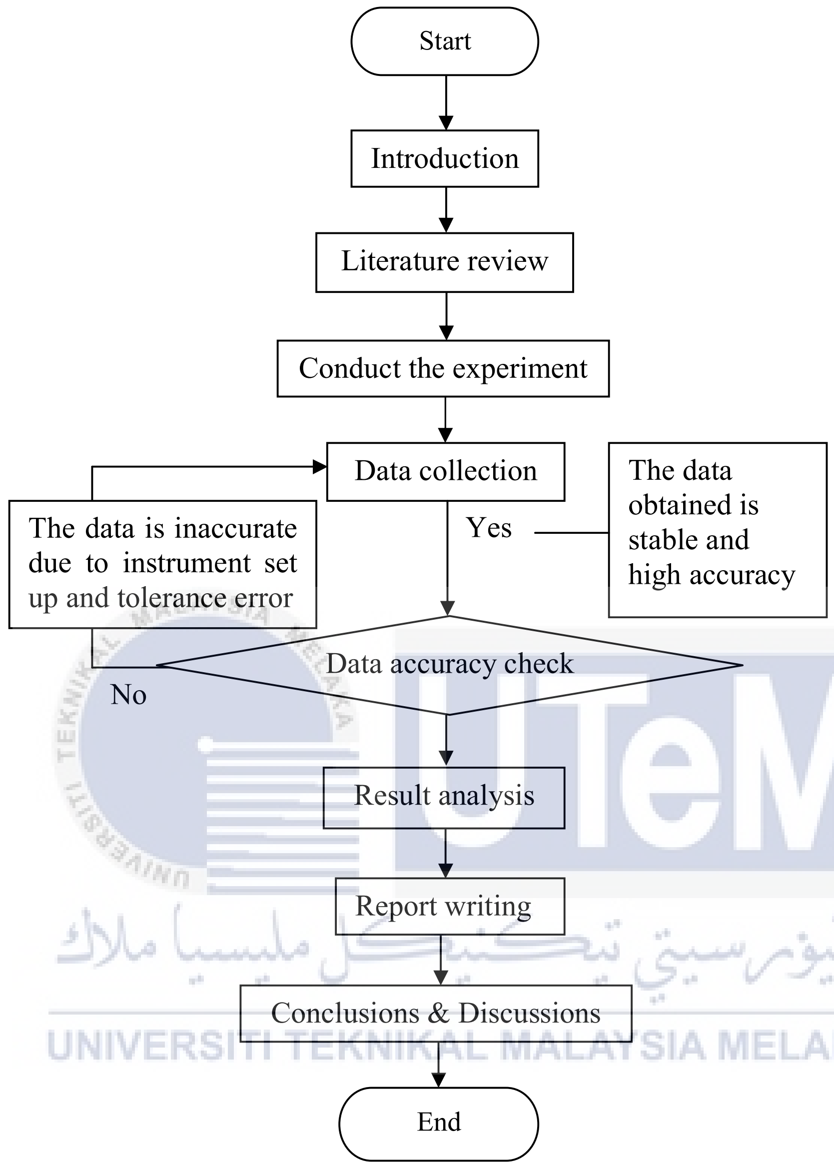



Figure 3.1: Flow chart of overall study

## 3.1 Gantt Chart

Gantt chart is the timeline of the study which consist the task need to be done in specified time so the project can be accomplished as per scheduled. The timeline for this study is shows in Table 3.1.1

Table 3.1: Gantt chart of study.

| Num | Task | Duration | week 1 | week 2 | week 3 | week 4 | week 5 | week 6 | week 7 | week 8 | week 9 | week 10 | week 11 | week 12 | week 13 | week 14 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | PSM I | | | | | | | | | | | | | |
| 1 | Find the sources of literature | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |
| 2 | Identify the problem | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |
| 3 | Construct the objectives | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |
| 4 | Understands the background of | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |
| 5 | Limit the area of study | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |
| 6 | Build the general methodology | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |
| 7 | Review the research (journal) of | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |
| 8 | Development of literature review | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |
| 9 | Build the flowchart | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |
| 10 | Selection of experimental method | Plan | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | |

| Num | Task | Duration | week 1 | week 2 | week 3 | week 4 | week 5 | week 6 | week 7 | week 8 | week 9 | week 10 | week 11 | week 12 | week 13 | week 14 | week 15 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | PSM II | | | | | | | | | | | | | | |
| 1 | Charge determination | Plan | | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | | |
| 2 | Set the baseline of the experiment | Plan | | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | | |
| 3 | Conduct the experiment | Plan | | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | | |
| 4 | Analyse data | Plan | | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | | |
| 5 | Conclude overall studies | Plan | | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | | |
| 6 | Suggest on recommendation | Plan | | | | | | | | | | | | | | | |
| | | Actual | | | | | | | | | | | | | | | |

## 3.2 Experiment Parameter

The experiment study the compressor at variable speed affect to the COP of the air conditioning system. Table 3.1 shows the parameters that influence compressor performance such as the system pressure, temperature and mass flow rate is measured at vary compressor speed. Then, the measured parameters such as the heat absorbed, heat rejected, cooling capacity, compressor work and COP value is evaluated by using the derived equation from 1st Thermodynamics Law. The derived quantity of the study is summarized in Table 3.2. The experiment is conducted at fixed surrounding temperature, heat load and blower fan speed. The test is repeated for 30 number of samples by change the setting but conduct in the same manner.

Table 3.2.1: Measured parameter

| Parameters | Variable Measured | Measuring Instrument | Tolerance |
|---|---|---|---|
| Temperature | $T_1$, $T_2$, $T_3$, $T_4$, | Thermocouple Type T (constantan & copper wire) | $\pm 0.1\,^{\circ}C$ |
| Pressure | $P_1$, $P_2$, $P_3$, $P_4$ | Pressure dial gauges | $\pm$ 0.1 bar |
| Mass flow rate | $\dot{m}$ | Mass flow meter | $\pm 1.0\ kgs^{-1}$ |
| Compressor speed | N | Tachometer | $\pm$0.01 % |

Table 3.2.2 Evaluated Parameters

| Quantity | Variable Measured |
|---|---|
| Heat rejected | $\dot{Q}_{out}$ |
| Cooling capacity | $\dot{Q}_L$ |
| Compressor work | $W_{comp}$ |
| Coefficient of performance | COP |

### 3.3 Experimental Set Up

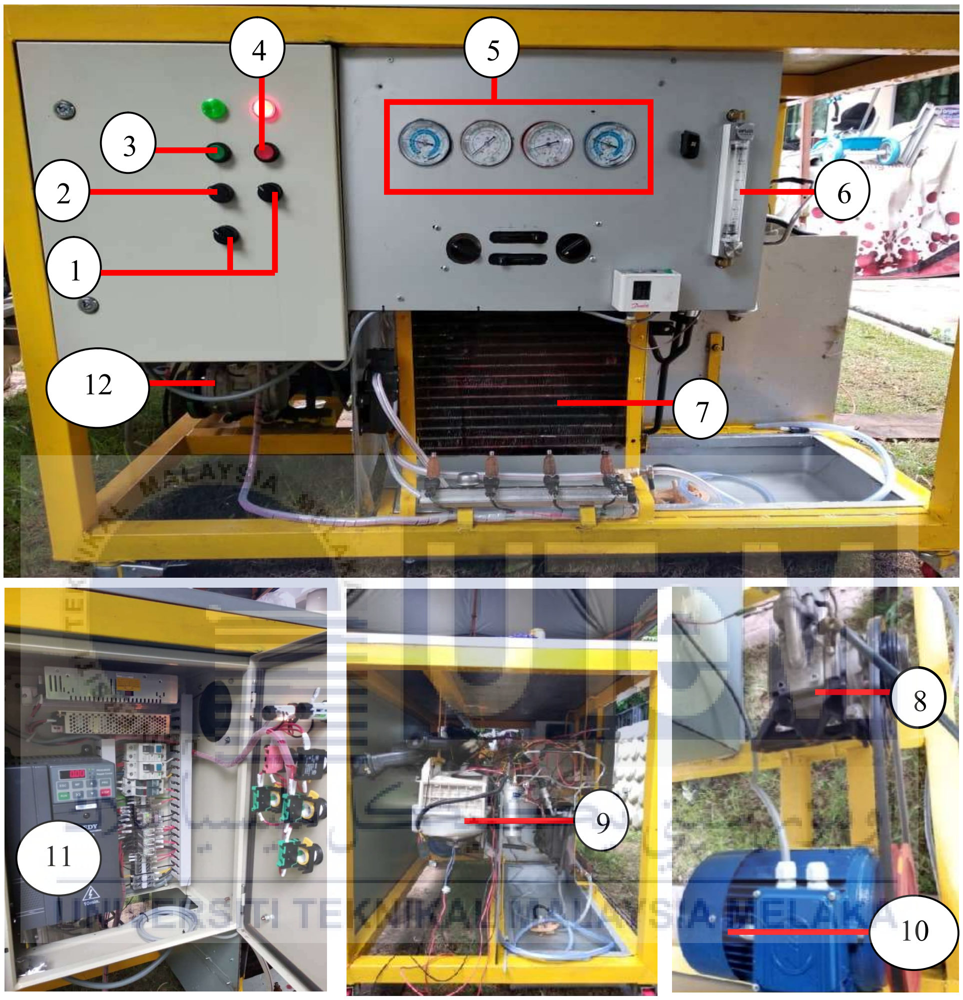



Figure 3.3.1: Experimental set up of vapour compression refrigeration test

| | | |
|---|---|---|
| 1. Control water button | 5. Pressure dial gauges | 9. Blower fan |
| 2. Control pump button | 6. Flow meter | 10. Electric motor |
| 3. ON button | 7. Condenser | 11. Frequency control setting |
| 4. OFF button | 8. Compressor | 12. High pressure pump |

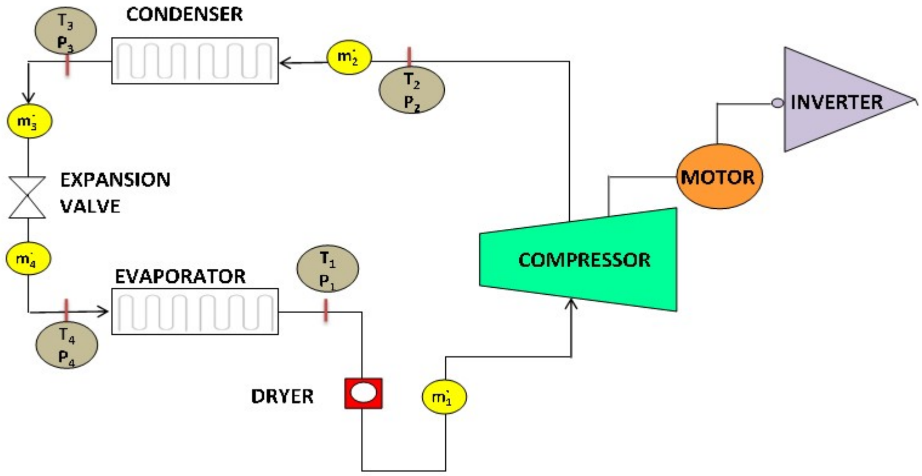


Figure 3.3.2: Schematic diagram of VCRs

The actual experiment set up of vapour compression refrigerant test rig is shown in Figure 3.2.2 while the schematic diagram of the experiment is presented in Figure 3.3.2

### 3.4 Experimental Procedures

The test rig which consists of basic components of vapour compression system such as compressor, condenser, evaporator and expansion device is setup as Figure 3.3.1 R-134a is used as refrigerant. The AC power supply with 450 W was turn ON.

For charge determination part, a set of manifold gauges which consists of pull vacuum was connected to air conditioning system. The low pressure hose and high pressure hose was connected to the low pressure port and high pressure port respectively, while the pull vacuum channel is connected to refrigerant tank. Each side was open, the refrigerant was injected to the air conditioning system for 10 psi.

By referring to Figure 3.4.1 and Table 3.4.1 the thermocouples and PICO TC08 Data Logger which used to measure the temperature of evaporator, compressor, condenser and expansion valve was connected between each part as shown in table. Another set of thermocouple was used to measure the temperature of evaporator and blower fan was set. The thermocouples was connected by used the cable tie with greater thickness in order to prevent them from melted during the operations.



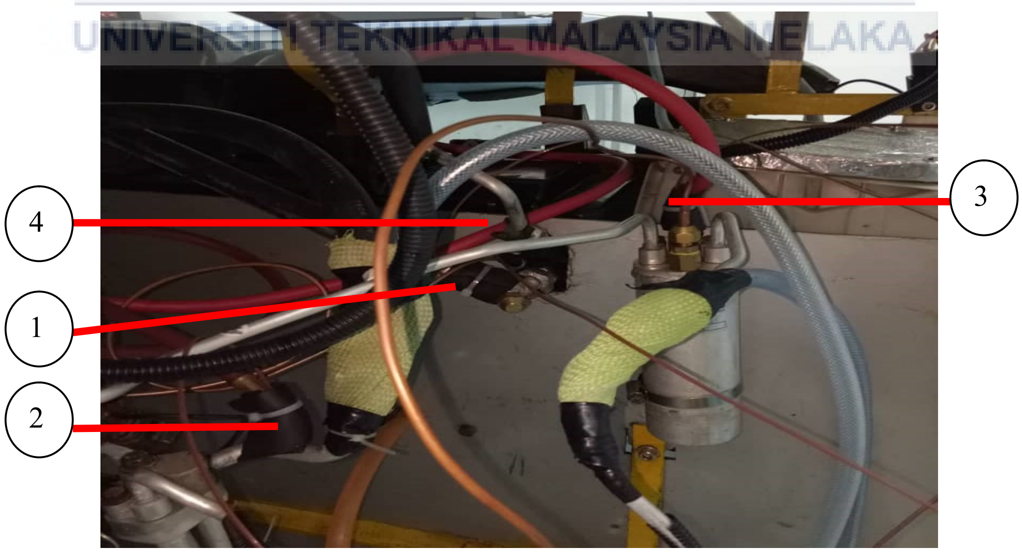



Figure 3.4.1: Placement of thermocouple at the test rig

Table 3.4.1: Thermocouple positioning

| Thermocouple | Positions |
|---|---|
| 1 | Evaporator outlet, compressor inlet |
| 2 | Compressor outlet, condenser inlet |
| 3 | Condenser outlet, expansion valve inlet |
| 4 | Expansion valve outlet, evaporator inlet |

In the beginning, the compressor was speed up to 1200 rpm (25 Hz). While, the blower fan speed and the temperature of evaporator inlet, $T_5$ are fixed at level 3 and 28 ºC (300 W) respectively. The test rig was stabilised for 20 minutes. After the system has reached the steady state, the reading of temperature and pressure were recorded. The experiment was run for 10 minutes per each compressor speed and 31 numbers of samples were obtained by used PICO Data Logger. Once completed, the system was turn OFF. The system was cooled down for five to ten minutes before proceed to another value of variables. The experiment was repeated by adjusted the speed of compressor to 1700 rpm and 1900 rpm.



For the experimental procedure, to ensure high data accuracy and consistency a few step is taken by followed the SAE International Standard (issued 2008-10). First, the test rig system need to be stabilised before the data is recorded. Means, the test rig need to be run for at least 10 minutes until the steady state condition is achieved. The system is said to be stable when the reading of data shown very minimum fluctuation and almost high consistency.

### 3.5 Data Collection

The most crucial part of the experiment is data collection. To ensure the consistency and accuracy of result, the data is gathered in the data collection form. The measured parameter is recorded in Table 3.3 while the calculated data is recorded in Table 3.4. Hence, the data is well-organised, avoid the mistake during collection of data and ease the analysis of data.

Table 3.5.1 Measured Data Collection Form

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Ambient Temperature (ºC) | | 3°C ± 0.1 | | | | | | |
| Heat Load (kW) | | 300 W | | | | | | |
| Blower Fan Speed | | Level 3 | | | | | | |
| Compressor Speed, N (rpm) | | | | | | | | |
| Position | | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
| Num. | Time (s) \ Temp (°C) | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| Mean, $T_{mean}$ | | | | | | | | |
| Median | | | | | | | | |
| Standard deviation, σ | | | | | | | | |



Table 3.5.1 Calculated Data Collection Form

| Variable \ Compressor Speed,N (rpm) | 1500 | 1700 | 1900 |
|---|---|---|---|
| Heat Rejected, $\dot{Q}_{out}$ (kW) | | | |
| Cooling Capacity, $\dot{Q}$ (kW) | | | |
| Compressor Work, $W_{comp}$ | | | |
| Coefficient of Performance, COP (%) | | | |

### 3.6 Data Analysis

The temperature ($T_1$, $T_2$, $T_3$, $T_4$) from the experiment is recorded by using the USB TC-08 Thermocouple Data Logger that used along with PicoLog Data Logging Software.

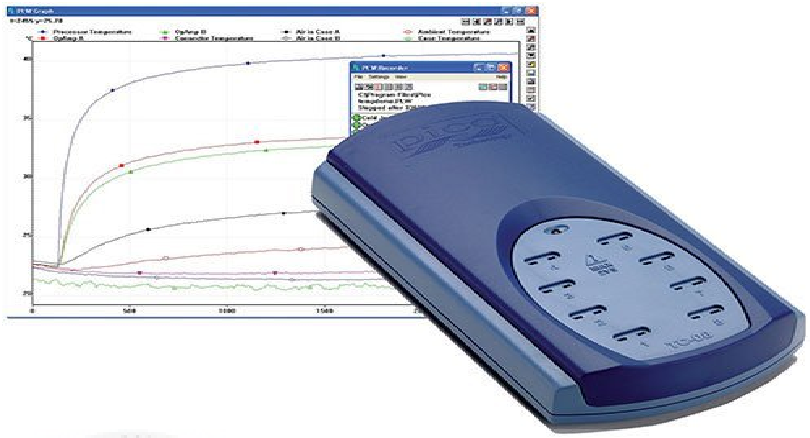


Figure 3.6.1: Thermocouple with PICO TC08 Data Logger

REFPROP is powerful programming that developed by National Institute of Standards and Technology (NIST). This program represents thermal physical properties of refrigerants. This method was used since it has high accuracy and efficient way to find the saturated temperature and enthalpy at specified pressure in a shorter time.

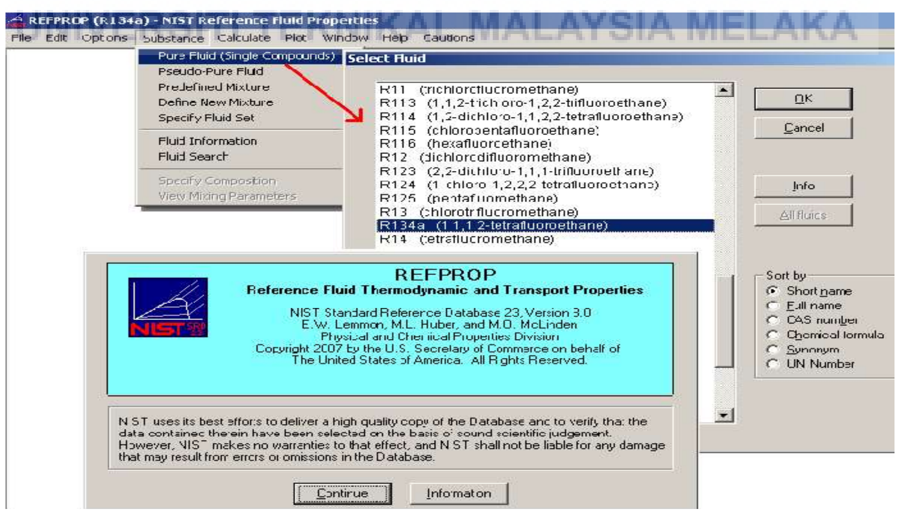



Figure 3.6.2: The dialog of choosing the refrigerant in REFPROP program (V. F. Ochkov, 2009).

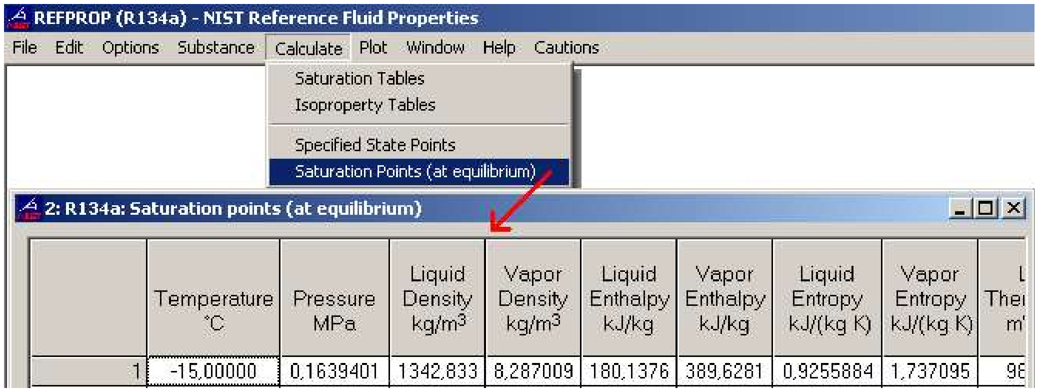
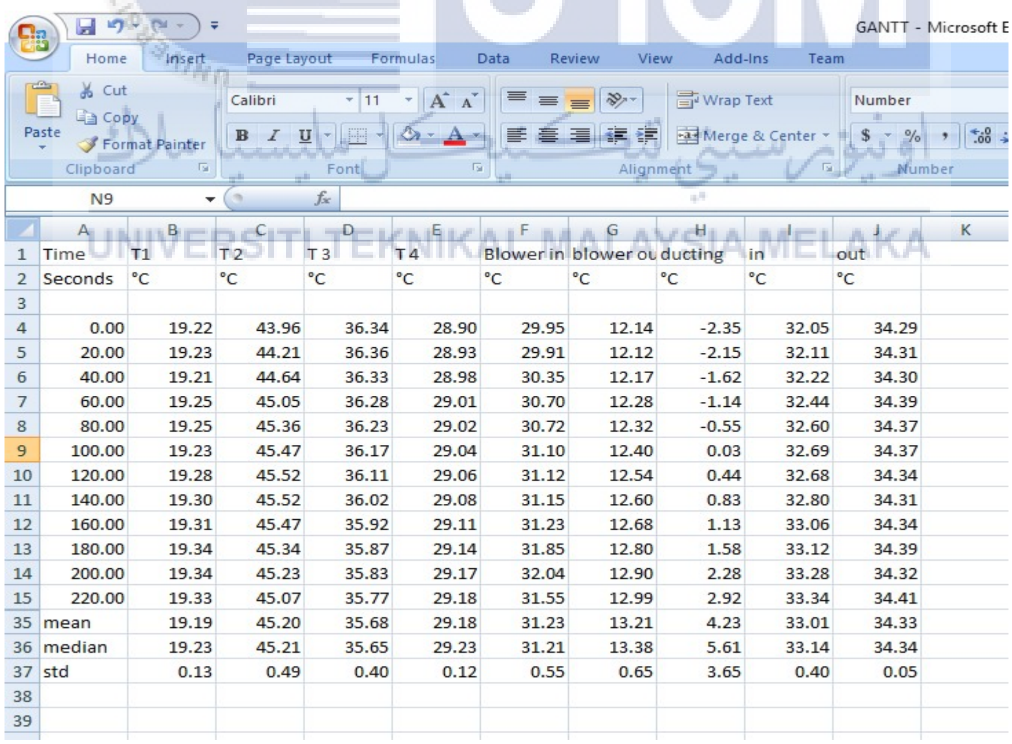



Figure 3.7: The dialog of calculation of the refrigerant thermal physical properties in REFPROP program (V. F. Ochkov, 2009).

The quantitative data is analysed and interpret by present in graphical method. The graph is generated by using the Microsoft Office Excel 2007 and the value of mean, median, standard deviation and variance is auto evaluated by inserting the formula and the value of measured variables into the worksheets.




| | A | B | C | D | E | F | G | H | I | J | K |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Time | T1 | T 2 | T 3 | T 4 | Blower in | blower ou | ducting | in | out | |
| 2 | Seconds | °C | °C | °C | °C | °C | °C | °C | °C | °C | |
| 3 | | | | | | | | | | | |
| 4 | 0.00 | 19.22 | 43.96 | 36.34 | 28.90 | 29.95 | 12.14 | -2.35 | 32.05 | 34.29 | |
| 5 | 20.00 | 19.23 | 44.21 | 36.36 | 28.93 | 29.91 | 12.12 | -2.15 | 32.11 | 34.31 | |
| 6 | 40.00 | 19.21 | 44.64 | 36.33 | 28.98 | 30.35 | 12.17 | -1.62 | 32.22 | 34.30 | |
| 7 | 60.00 | 19.25 | 45.05 | 36.28 | 29.01 | 30.70 | 12.28 | -1.14 | 32.44 | 34.39 | |
| 8 | 80.00 | 19.25 | 45.36 | 36.23 | 29.02 | 30.72 | 12.32 | -0.55 | 32.60 | 34.37 | |
| 9 | 100.00 | 19.23 | 45.47 | 36.17 | 29.04 | 31.10 | 12.40 | 0.03 | 32.69 | 34.37 | |
| 10 | 120.00 | 19.28 | 45.52 | 36.11 | 29.06 | 31.12 | 12.54 | 0.44 | 32.68 | 34.34 | |
| 11 | 140.00 | 19.30 | 45.52 | 36.02 | 29.08 | 31.15 | 12.60 | 0.83 | 32.80 | 34.31 | |
| 12 | 160.00 | 19.31 | 45.47 | 35.92 | 29.11 | 31.23 | 12.68 | 1.13 | 33.06 | 34.34 | |
| 13 | 180.00 | 19.34 | 45.34 | 35.87 | 29.14 | 31.85 | 12.80 | 1.58 | 33.12 | 34.39 | |
| 14 | 200.00 | 19.34 | 45.23 | 35.83 | 29.17 | 32.04 | 12.90 | 2.28 | 33.28 | 34.32 | |
| 15 | 220.00 | 19.33 | 45.07 | 35.77 | 29.18 | 31.55 | 12.99 | 2.92 | 33.34 | 34.41 | |
| 35 | mean | 19.19 | 45.20 | 35.68 | 29.18 | 31.23 | 13.21 | 4.23 | 33.01 | 34.33 | |
| 36 | median | 19.23 | 45.21 | 35.65 | 29.23 | 31.21 | 13.38 | 5.61 | 33.14 | 34.34 | |
| 37 | std | 0.13 | 0.49 | 0.40 | 0.12 | 0.55 | 0.65 | 3.65 | 0.40 | 0.05 | |
| 38 | | | | | | | | | | | |
| 39 | | | | | | | | | | | |

Figure 3.8: The layout of software Microsoft Office Excel 2007

# CHAPTER 4

# RESULT AND DISCUSSION

## 4.1 Introduction

After the experiment is conducted, the results that were acquired is shown and discussed in this chapter. In addition, the data is analysed by using thermodynamic software (REFPROP) and calculation from thermodynamics' law. Basically, the data analysed consists of interconnection between the compressor speeds, energy consumption and COP which will present graphically for clearer view and understanding.

## 4.2 Raw Data Analysis

On this section, the system temperature, the system pressure and mass flow rate were obtained directly from the experiment was known as raw data. Then, it was discussed in detail including the patterns, the factor influenced and the relationship between the parameter and the compressor speed.

### 4.2.1 The Effect of Compressor Speed on High Pressure and Low Pressure

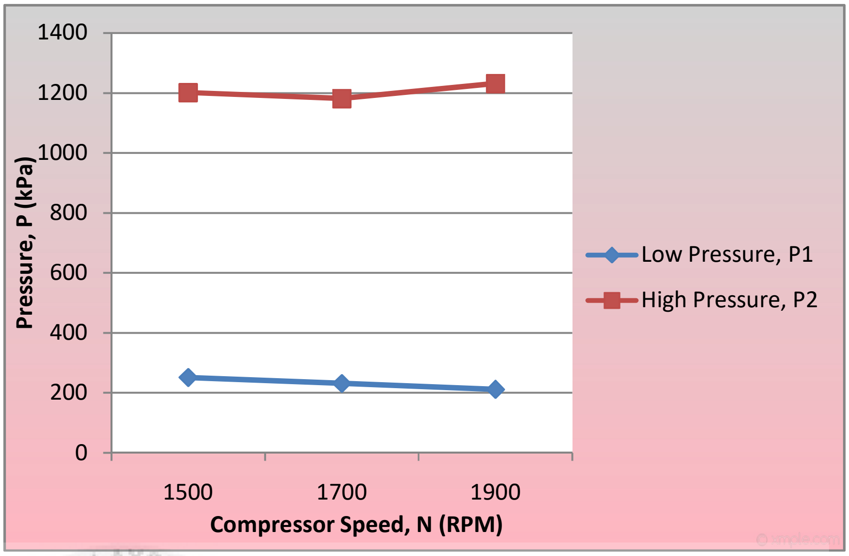



Figure 4.2.1: Graph of the system temperature of vary compressor speed

Figure 4.2.1 shows the effect of compressor speed to the low system pressure, $P_1$ and high system pressure, $P_2$. Based on the graph, $P_1$ is decreases as the compressor speed increases. At 1500 rpm to 1700 rpm, the pressure decreases by 7.95% and continue decreases by 8.64% at 1900 rpm. Hence, it shows that at speed 1900 rpm the pressure is the lowest which is 211.33 kPa while the pressure of $P_1$ is the highest at speed 1500 rpm which is 251.33 kPa. Based on the graph, $P_2$ is decreases by 1.66% from 1500 rpm to 1700 rpm but increases by 4.23% at speed 1900 rpm. Hence, the lowest pressure of $P_2$ is at 1700 rpm while the highest pressure of $P_2$ is at 1900 rpm with value 1181.33 kPa and 1231.33 kPa respectively.

### 4.2.2 The Effect of Compressor Speed on System Temperature

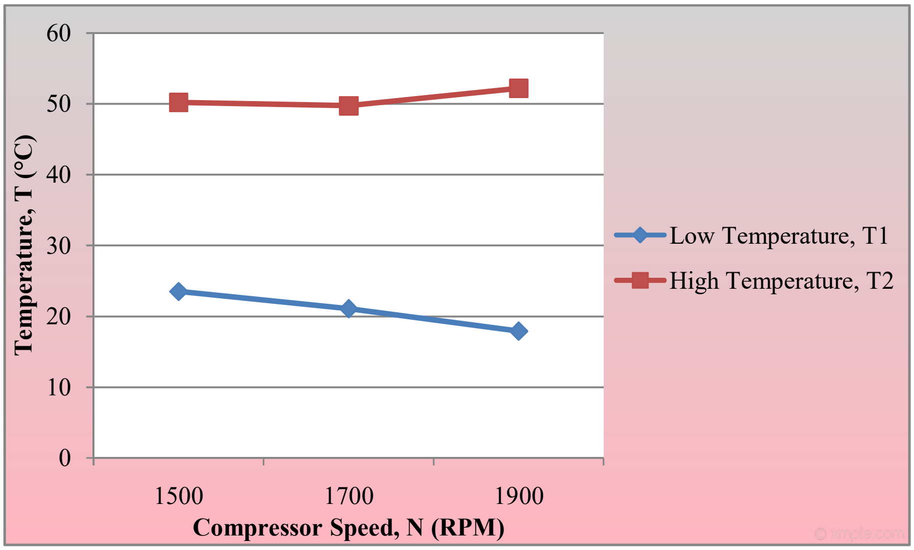



Figure 4.2.2.1: Graph of the system pressure of vary compressor speed



Figure 4.2.2.1 shows the effect of compressor speed to the low system temperature, $T_1$ and high system temperature, $T_2$.Based on the graph, $T_1$ is decreases as the compressor speed increases. At speed 1500 rpm to 1700 rpm , $T_1$ is decreases by 10.29% and continues decreases by 15.03% at 1900 rpm. Thus, at speed 1500 rpm, $T_1$ has the highest pressure which is 23.51 °C while at speed 1900 rpm, $T_1$ is the lowest which is 17.92°C.Based on the graph, $T_2$ is slightly decreases by 0.94% at speed 1500 rpm to 1700 rpm while increases by 4.74% at 1900 rpm. Thus, at speed 1700 rpm, $T_2$ has the lowest pressure which is 49.75°C while at speed 1900 rpm, $T_2$ is the highest which is 52.23°C.According to Dahlan, et al., 2014, the temperature distribution of the system was always inversely proportional to the compressor speed.

### 4.2.3 The Effect of Compressor Speed on Mass Flow Rate

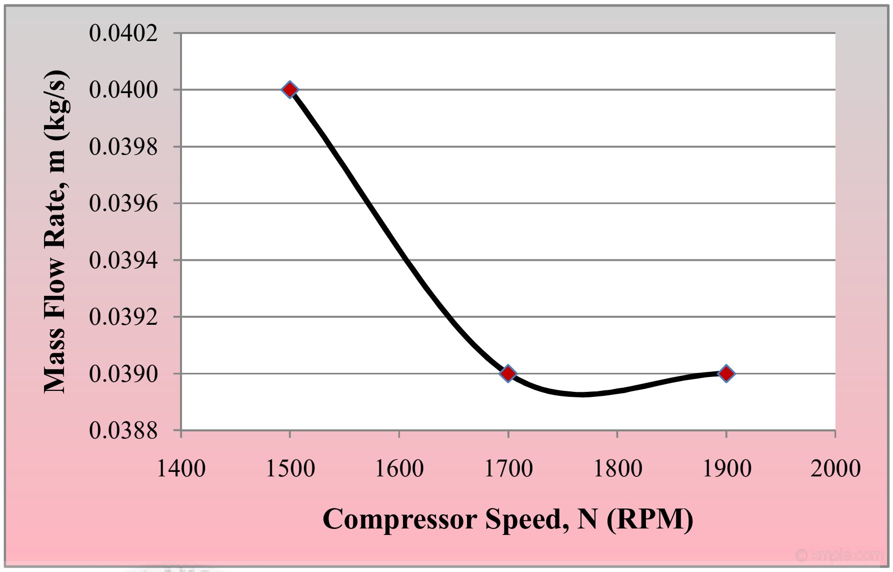



Figure 4.2.3.1: Graph of the mass flow rate of the system of vary compressor speed

Figure 4.2.3.1 shows the effect of compressor speed to the refrigerant flow rate. Based on the graph, the refrigerant mass flow rates decreased by 2.5% as the compressor speed goes from 1500 rpm to 1700 rpm while the mass flow rate at 1900 rpm remain the same as 1700 rpm. According to Dahlan, et al., 2014, the mass flow rate was always directly proportional to the compressor speed. In conclusion, the mass flow rate of the refrigerant is fluctuated which means it was not stable.

### 4.3 Calculated Data Analysis

Compressor speed at 1500 rpm

Table 4.3.1: The average of physical properties of R-134a at 1500 rpm

| Average | Temperature, T (°C) | Saturated temperature, Tsat (°C) | Pressure, P (kPa) | Mass flow rate, m (kg/s) | Enthalpy, h (kJ/kg) |
|---|---|---|---|---|---|
| State 1 | 23.51 | -4.14 | 251.33 | 0.04 | 396.17 |
| State 2 | 50.22 | 46.36 | 1201.33 | 0.04 | 426.64 |
| State 3 | 40.33 | 46.36 | 1201.33 | 0.04 | 266.01 |
| State 4 | 32.40 | 3.46 | 331.33 | 0.04 | 266.01 |

Heat addition;

$$\dot{Q}_L = \dot{m}\ (h_1\text{-}h_4)$$
$$= 0.04\ (396.17\text{-}266.01)$$
$$= 5.21\ \text{kJ /s}$$

Heat rejection;

$$\dot{Q}_H = \dot{m}\ (h_2\text{-}h_3)$$
$$= 0.04\ (426.64\text{-}266.01)$$
$$= 6.43\ \text{kJ /s}$$

Work done by compressor;

$$\dot{W}_{in} = \dot{m}\ (h_2\text{-}h_1)$$
$$= 0.04\ (426.64\text{-}396.17)$$
$$= 1.22\ \text{kJ /s}$$

Coefficient of performance; $COP = \frac{Refrigerating\ Effect}{Work\ Done}$

$$= \frac{\dot{Q}L}{\dot{W}in}$$
$$= \frac{5.21}{1.22}$$
$$= 3.99$$

Compressor speed at 1700 rpm

Table 4.3.2: The average of physical properties of R-134a at 1700 rpm

| Average | Temperature, T (°C) | Saturated temperature, Tsat (°C) | Pressure, P (kPa) | Mass flow rate, m (kg/ms) | Enthalpy, h (kJ/kg) |
|---|---|---|---|---|---|
| State 1 | 21.09 | -6.33 | 231.33 | 0.039 | 394.87 |
| State 2 | 49.75 | 45.71 | 1181.33 | 0.039 | 426.57 |
| State 3 | 35.89 | 45.71 | 1181.33 | 0.039 | 256.02 |
| State 4 | 29.42 | -26.07 | 101.33 | 0.039 | 256.02 |

Heat addition;

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

$$= 0.039\ (394.87\text{-}256.02)$$

$$= 5.42 \text{ kJ /s}$$

Heat rejection;

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

$$= 0.039\ (426.57\text{-}256.02)$$

$$= 6.65 \text{ kJ /s}$$

Work done by compressor;

$$\dot{W}_{in} = \dot{m}\ (h_2\text{-}h_1)$$

$$= 0.039\ (426.57\text{-}394.87)$$

$$= 1.24 \text{ kJ /s}$$

Coefficient of performance;

$$COP = \frac{Refrigeratin\ \ Effect}{Work\ Done}$$

$$= \frac{QL}{\dot{W}in}$$

$$= \frac{5.42}{1.24}$$

$$= 4.4$$

Compressor speed at 1900 rpm

Table 4.3.3: The average of physical properties of R-134a at 1900 rpm

| Average | Temperature, T (°C) | Saturated temperature, Tsat (°C) | Pressure, P (kPa) | Mass flow rate, m (kg/ms) | Enthalpy, h (kJ/kg) |
|---|---|---|---|---|---|
| State 1 | 17.92 | -8.67 | 211.33 | 0.039 | 393.46 |
| State 2 | 52.23 | 47.32 | 1231.33 | 0.039 | 428.28 |
| State 3 | 40.60 | 47.32 | 1231.33 | 0.039 | 267.49 |
| State 4 | 32.55 | 0.79 | 301.33 | 0.039 | 267.49 |

Heat addition;

$$\dot{Q}_L = \dot{m}\,(h_1\text{-}h_4)$$

$$= 0.039\ (393.46\text{-}267.49)$$

$$= 4.91 \text{ kJ /s}$$

Heat rejection;

$$\dot{Q}_H = \dot{m}\,(h_2\text{-}h_3)$$

$$= 0.039\ (428.28\text{-}267.49)$$

$$= 6.27 \text{ kJ /s}$$

Work done by compressor;



$$\dot{W}_{in} = \dot{m}\,(h_2\text{-}h_1)$$

$$= 0.039\ (428.28\text{-}393.46)$$

$$= 1.36 \text{ kJ /s}$$

Coefficient of performance; $COP = \frac{Refrigerating\ Effect}{Work\ Done}$

$$= \frac{QL}{\dot{W}in}$$

$$= \frac{4.91}{1.36}$$

$$= 3.61$$

### 4.3.1 The Effect of Compressor Speed on Heat Rejection

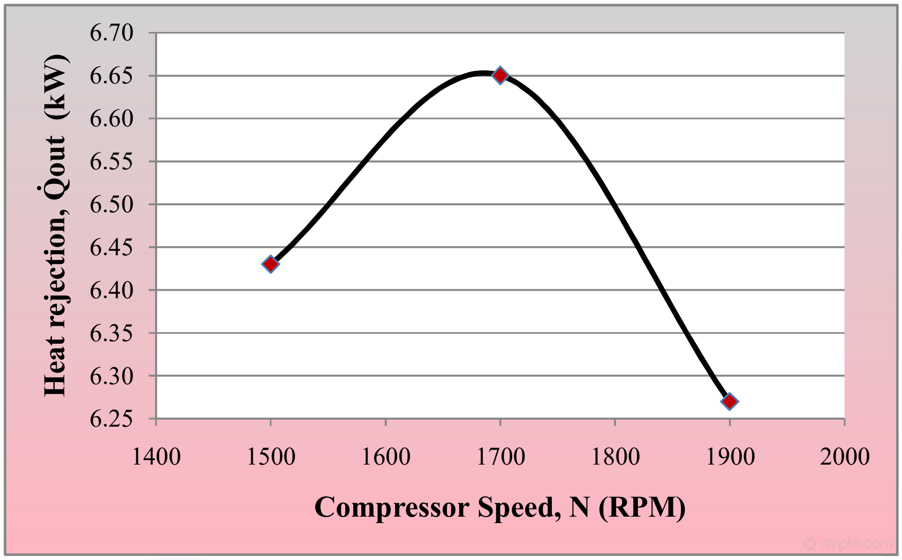



Figure 4.3.1.1: Graph of heat rejection against compressor speed

Figure 4.3.1.1 shows the effect of compressor speed to the heat rejected by the system. Based on the graph, the heat being rejected from 1500 rpm to 1700 rpm is increases by 3.3% while it decreases by 5.7% at speed 1900 rpm. In brief, the heat being rejected is the highest at speed 1700 rpm which 6.65 kW while the lowest is at 1900 rpm.

### 4.3.2 The Effect of Compressor Speed on Compressor Work

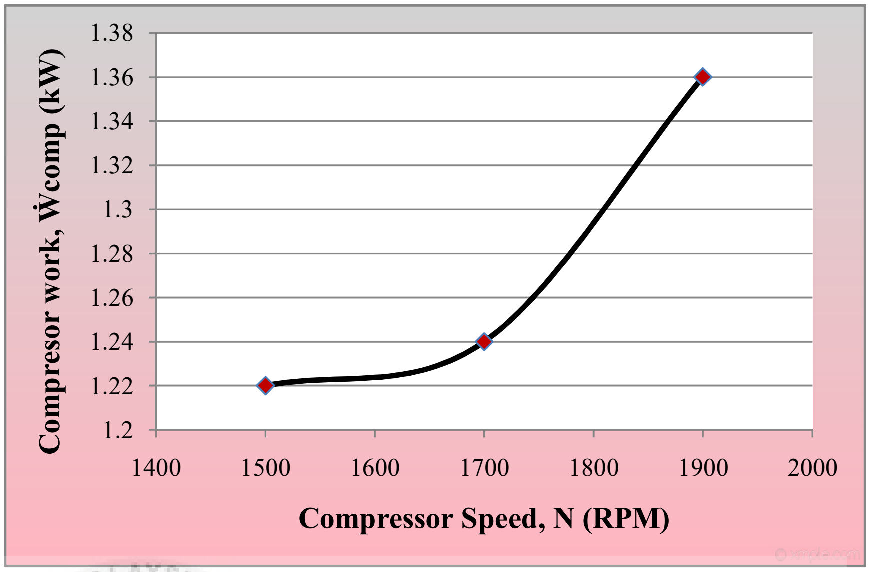



Figure 4.3.2.1: Graph of compressor work against compressor speed

Figure 4.3.2.1 shows the effect of compressor speed to the compressor work. Based on the graph, the compressor work is directly proportional to the compressor speed. The compressor work is increases by 1.6% at speed 1500 rpm to 1700 rpm and continues increases by 9.7% at speed 1900 rpm. Hence, the work done by the compressor is the lowest at speed 1500 rpm while highest at 1900 rpm. According to Dahlan, et al., 2014, the compressor work is always directly proportional to the compressor speed.

### 4.3.3 The Effect of Compressor Speed on Cooling Capacity

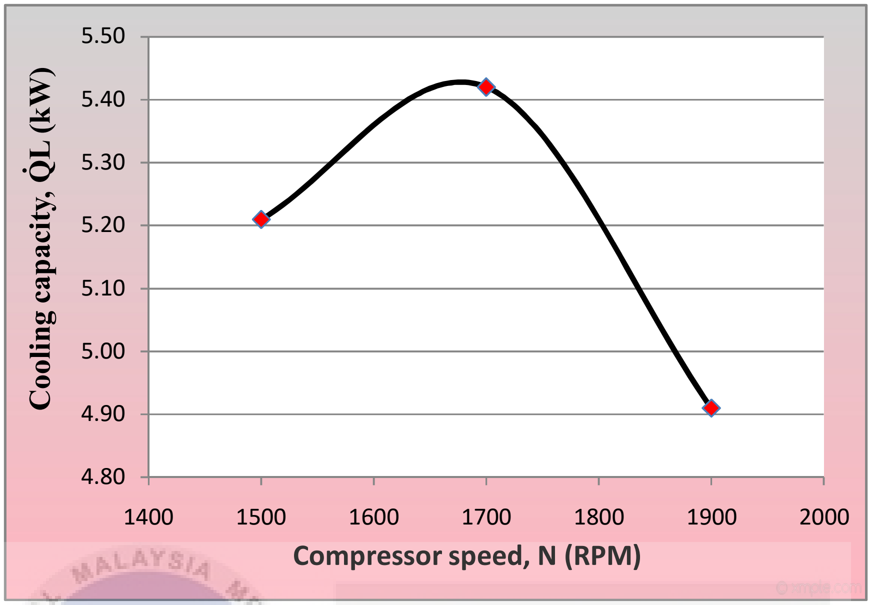



Figure 4.3.3.1: Graph of cooling capacity against compressor speed

Figure 4.3.3.1 shows the effect of compressor speed to the cooling capacity. Based on the graph, the cooling capacity is increases by 3.9% from 1500 rpm to 1700 rpm while decreases by 10.4% at speed 1900 rpm. Thus, the highest cooling capacity is at 1700 rpm and the lowest cooling capacity is at 1900 rpm which has the value 5.42 kW and 4.91 kW respectively. According to Dahlan, et al., 2014, the cooling capacity is always directly proportional to the compressor speed.

### 4.3.4 The Effect of Compressor Speed on COP

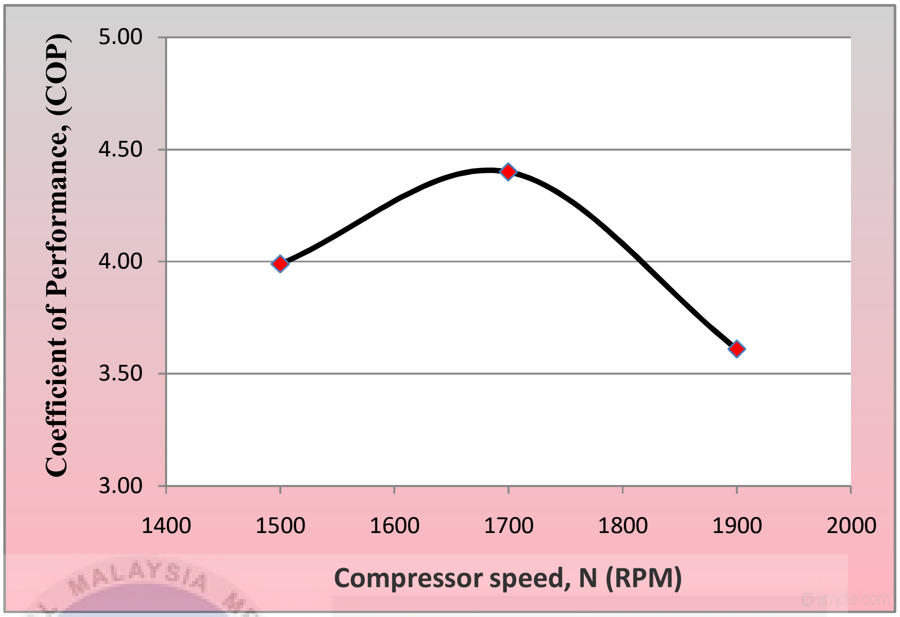



Figure 4.3.4.1: Graph of COP against compressor speed

Figure 4.3.4.1 shows the effect of compressor speed to the system coefficient of performance (COP). Based on the graph, the COP is increases by 10.3% from 1500 rpm to 1700 rpm while the COP is decreases by 18% at 1900 rpm. Thus, the highest COP is at 1700 rpm and the lowest COP is at 1900 rpm with value 4.4 and 3.61 respectively. According to Dahlan, et al., 2014, the COP is always inversely proportional to the compressor speed.

## 4.4 System P-h Diagram

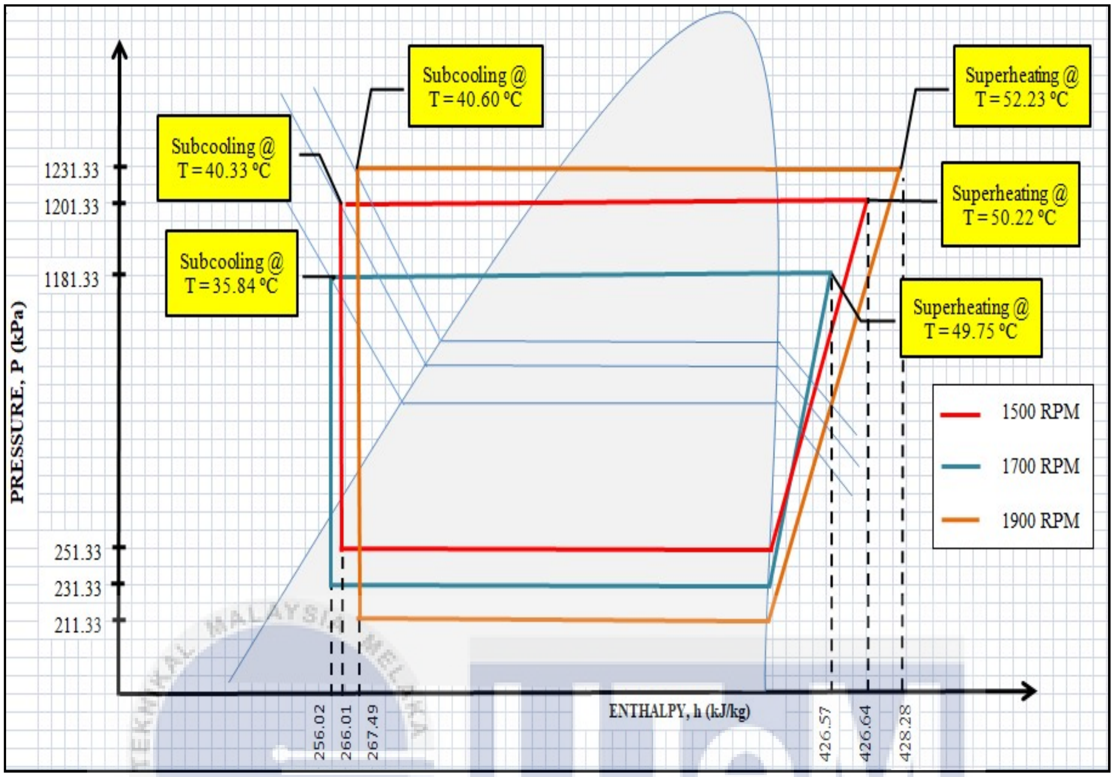



Figure 4.4.1: P-h diagram of the system for different compressor speed

The enthalpy differences based on experimental pressure is illustrates in Figure 4.4.1, where the differences between three cycle at vary speed of compressor is summarized. R-134a is used as the working fluid and operates on ideal vapour compression cycle.

For cycle with compressor speed 1500 rpm, R-134a enters the compressor refrigerator as superheated vapour at 251.33 kPa and 23.51 °C at a rate 0.04 kg/s and leaves at 1201.33 kPa and 50.22 °C. The refrigerant is sub cooled in the condenser to 40.22 °C and is throttled with the enthalpy 266.01 kJ/kg.

For cycle with compressor speed 1700 rpm, R-134a enters the compressor refrigerator as superheated vapour at 231.33 kPa and 21.09 °C at a rate 0.039 kg/s and leaves at 1181.33 kPa and 49.75 °C. The refrigerant is sub cooled in the condenser to 35.84 °C and is throttled with the enthalpy 256.02 kJ/kg.

For cycle with compressor speed 1900 rpm, R-134a enters the compressor refrigerator as superheated vapour at 211.33 kPa and 17.92 °C at a rate 0.039 kg/s and leaves at 1231.33 kPa and 49.75 °C. The refrigerant is sub cooled in the condenser to 40.60 °C and is throttled with the enthalpy 267.49 kJ/kg.

### 4.5 Discussion

Based on the data acquired, the analysis of the heat transfer, work and performance were calculated by assumed the system experienced perfect throttling process in state 3 and 4. The system is assumed to be throttled when steam is flow through an obstruction from higher pressure to lower pressure. The condition for throttling process is defined as the kinetic energy, KE, potential energy, PE and heat transfer, Q is equal to zero. The equation for steady state energy equation said to be;

$$h_3 + KE_3 + PE_3 + Q_{3\text{-}4} = h_4 + KE_4 + PE_4 + W_{3\text{-}4}$$

$$h_3 + \cancelto{0}{\frac{(c_3)^2}{2}} + \cancelto{0}{gz_3} + \cancelto{0}{Q} = h_4 + \cancelto{0}{\frac{(c_4)^2}{2}} + \cancelto{0}{gz_4} + \cancelto{0}{W}$$

$$h_3 = h_4 \rightarrow \text{isentropic process}$$

$\therefore$Hence, the value of enthalpy in state 3 and state 4 is equivalent

The leakage in refrigeration system caused the huge drop of system pressure. Thus, wasted refrigerant led to more energy losses in the system. Since, the leakage decreased the efficiencies of the system. The leakage of refrigerant was detected by used the anemometer

Referring to table 4.3.1, 4.3.2 and 4.3.3, it showed that the $T_1$ obtained has thehuge difference between the experimental data and saturation value was probably caused by the effect of ambient temperature towards the thermocouple because the thermocouple is not properly insulated on the suction side line. The heat dissipated from surroundings causes the compressor to operate at greater compressor work which led to overheated.




Figure 4.5.1: Overheated electric motor of compressor

The electric motor of compressor was overheated at 2000 rpm, approximately after 45 minute the system was operated. The poor efficiencies of compressor work were caused by oversized of air compressor which limited the compressor frequencies runs up to 50 Hz only. The increment in compressor speed led to the increment in energy per unit. Thus, when the compressor has reached their maximum capacity, the heat dissipated caused the melted on cooling fan resulting the fan to slowly stopped rotated and caused tripped to the power supply.

# CHAPTER 5

# CONCLUSIONS

## 5.1 Conclusions

In this chapter, conclusion is made in order to deliver the main message and clarify the objectives of the project.

Briefly, the increment ratio of the power consumed by the compressor is slightly higher than cooling capacity as the speed increases. For example, at 1500 rpm to 17000 rpm, the power consumed from 5.21 up to 5.42 kW compared to the cooling capacity increases from 1.22 to 1.24 kW. Thus the increment ratio is 1.04:1 to a 1.01:1. Thus, COP also increases.

Thus, from the experimental data analysed it can concluded that the compressor is work optimum at speed 1700 rpm since it has highest COP which is 4.4. Means, at this speed less compressor work needed to compress the refrigerant into the compressor compared to speed at 1500 rpm and 1900 rpm.

Table 5.1.1: Summarized of calculated data at vary compressor speed

| Variable \ Compressor Speed, N (rpm) | 1500 | 1700 | 1900 |
|---|---|---|---|
| Heat Addition, $\dot{Q}_L$ (kW) | 5.21 | 5.42 | 4.91 |
| Heat Rejected, $\dot{Q}_{out}$ (kW) | 6.43 | 6.65 | 6.27 |
| Compressor Work, $W_{comp}$ | 1.22 | 1.24 | 1.36 |
| Coefficient of Performance, COP | 3.99 | 4.4 | 3.61 |

## 5.2 Recommendation

Since the experimental conducted experienced some difficulties and error, hence the optimization is proposed in order to increase the data accuracy. First, the test rig system is leaked since the amount of refrigerant is not fully inserted, Leakage of the test rig causes by the. Second, the compressor speed cannot run in long duration and not exceed 50 Hz.

In order to get the accurate system temperature, the thermocouple must properly insulate to the suction line. Thus, protect the thermocouple from expose to ambient temperature and minimize the amount of heat added to refrigerant. The proper insulation can be done by use the foam rubber insulation as the external piping which is placed at the suction line as Figure


Figure 5.2.1: The insulation of thermocouple to the suction line

During conduct the experimental work, it was noticed that the motor compressor tend to overheated if it was run for long duration without stop. The system also overheated if it was run at 2000 rpm and above. The simplest way to minimize the compressor motor from overheated was to decrease the warm heat dissipated around the motor. Hence, the main lid used to cover motor was opened to prevent the heat from accumulated. The table fan was also placed closed to the motor in order to blown the warm heat dissipated. Thus, the compressor motor can run in longer time without overheated.

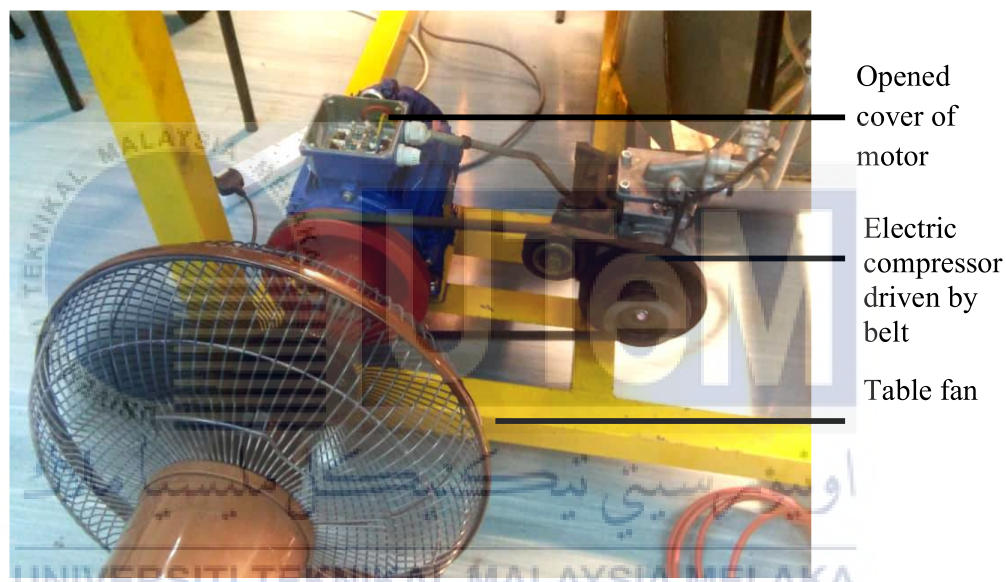



Figure 5.2.2: The table fan used as cooling medium for system compressor

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**APPENDIX 1**

| Ambient Temperature (ºC) | | 30°C ± 0.1 | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Heat Load (kW) | | 300 W | | | | | | |
| Blower Fan Speed | | Level 2 | | | | | | |
| Engine Speed, N (rpm) | | 1500 | | | | | | |
| Num. | Position / Time (s) | Temperature (°C) | | | | | | |
| | | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
| 1 | 0 | 22.55 | 50.15 | 40.59 | 32.05 | 28.96 | 10.33 | 0.05 |
| 2 | 20 | 22.61 | 50.30 | 40.53 | 32.04 | 29.62 | 10.29 | 0.01 |
| 3 | 40 | 22.70 | 50.58 | 40.46 | 32.04 | 29.03 | 10.30 | -0.25 |
| 4 | 60 | 22.80 | 50.82 | 40.45 | 32.05 | 30.39 | 10.36 | -0.34 |
| 5 | 80 | 22.89 | 51.00 | 40.37 | 32.06 | 30.86 | 10.34 | -0.48 |
| 6 | 100 | 22.98 | 51.18 | 40.37 | 32.11 | 30.38 | 10.35 | -0.71 |
| 7 | 120 | 23.07 | 51.33 | 40.36 | 32.14 | 30.45 | 10.43 | -0.76 |
| 8 | 140 | 23.18 | 51.42 | 40.36 | 32.22 | 30.37 | 10.42 | -0.85 |
| 9 | 160 | 23.26 | 51.48 | 40.39 | 32.28 | 30.97 | 10.54 | -1.02 |

| Num | Position / Time (s) | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
|---|---|---|---|---|---|---|---|---|
| 10 | 180 | 23.35 | 51.43 | 40.39 | 32.33 | 30.96 | 10.63 | -1.16 |
| 11 | 200 | 23.43 | 51.36 | 40.37 | 32.37 | 30.82 | 10.64 | -1.44 |
| 12 | 220 | 23.49 | 51.36 | 40.42 | 32.41 | 29.90 | 10.75 | -1.81 |
| 13 | 240 | 23.55 | 51.31 | 40.41 | 32.40 | 30.30 | 10.88 | -2.12 |
| 14 | 260 | 23.59 | 51.13 | 40.37 | 32.39 | 30.86 | 10.95 | -2.51 |
| 15 | 280 | 23.64 | 50.79 | 40.37 | 32.40 | 30.38 | 11.10 | -2.81 |
| 16 | 300 | 23.70 | 50.38 | 40.37 | 32.44 | 31.23 | 11.24 | -3.11 |
| 17 | 320 | 23.77 | 50.00 | 40.37 | 32.48 | 31.37 | 11.38 | -3.32 |
| 18 | 340 | 23.82 | 49.92 | 40.36 | 32.49 | 31.77 | 11.51 | -3.46 |
| 19 | 360 | 23.86 | 49.88 | 40.30 | 32.47 | 30.77 | 11.61 | -3.58 |
| 20 | 380 | 23.90 | 49.89 | 40.29 | 32.47 | 31.27 | 11.78 | -3.67 |
| 21 | 400 | 23.93 | 49.96 | 40.29 | 32.49 | 31.26 | 11.85 | -3.70 |
| 22 | 420 | 23.95 | 50.08 | 40.27 | 32.51 | 31.20 | 11.93 | -3.73 |
| 23 | 440 | 24.00 | 50.22 | 40.26 | 32.56 | 31.54 | 12.03 | -3.78 |

| Num | Position / Time (s) | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
|---|---|---|---|---|---|---|---|---|
| 24 | 460 | 24.06 | 50.27 | 40.25 | 32.63 | 32.08 | 12.12 | -3.75 |
| 25 | 480 | 24.11 | 50.02 | 40.27 | 32.68 | 31.45 | 12.19 | -3.62 |
| 26 | 500 | 24.12 | 49.44 | 40.23 | 32.67 | 31.67 | 12.27 | -3.52 |
| 27 | 520 | 24.13 | 48.78 | 40.15 | 32.64 | 31.57 | 12.34 | -3.34 |
| 28 | 540 | 24.15 | 48.34 | 40.05 | 32.62 | 30.01 | 12.48 | -3.01 |
| 29 | 560 | 23.84 | 47.86 | 39.76 | 32.51 | 29.71 | 11.93 | -1.71 |
| 30 | 580 | 23.24 | 47.88 | 40.16 | 32.65 | 30.16 | 11.42 | -1.06 |
| 31 | 600 | 23.17 | 48.17 | 40.55 | 32.74 | 30.21 | 11.70 | -1.13 |
| Average, ∑ | | 23.51 | 50.22 | 40.33 | 32.40 | 30.69 | 11.23 | -2.12 |
| Median | | 23.59 | 50.27 | 40.37 | 32.44 | 30.82 | 11.24 | -2.12 |
| Standard Deviation, σ | | 0.49 | 1.08 | 0.15 | 0.22 | 0.78 | 0.74 | 1.36 |

## APPENDIX 2

| Ambient Temperature (ºC) | | 30°C ± 0.1 | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Heat Load (kW) | | 300 W | | | | | | |
| Blower Fan Speed | | Level 2 | | | | | | |
| Engine Speed, N (rpm) | | 1700 | | | | | | |
| Num. | Position / Time (s) | Temperature (°C) | | | | | | |
| | | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
| 1 | 0 | 19.06 | 43.76 | 34.19 | 28.36 | 29.22 | 11.16 | 1.33 |
| 2 | 20 | 19.34 | 44.50 | 34.73 | 28.55 | 29.58 | 11.12 | 0.43 |
| 3 | 40 | 19.97 | 45.67 | 35.49 | 28.84 | 29.19 | 11.11 | -0.61 |
| 4 | 60 | 20.53 | 46.59 | 35.87 | 29.04 | 29.46 | 10.96 | -1.16 |
| 5 | 80 | 21.02 | 47.41 | 36.17 | 29.20 | 29.94 | 10.93 | -1.81 |
| 6 | 100 | 21.39 | 48.05 | 36.30 | 29.31 | 29.63 | 10.87 | -2.25 |
| 7 | 120 | 21.64 | 48.70 | 36.40 | 29.42 | 30.19 | 10.84 | -2.80 |
| 8 | 140 | 21.87 | 49.29 | 36.49 | 29.53 | 29.71 | 10.92 | -3.19 |
| 9 | 160 | 22.02 | 49.77 | 36.50 | 29.59 | 30.05 | 10.95 | -3.37 |

| Num | Position / Time (s) | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
|---|---|---|---|---|---|---|---|---|
| 10 | 180 | 22.15 | 50.24 | 36.53 | 29.65 | 30.70 | 11.05 | -3.85 |
| 11 | 200 | 22.25 | 50.56 | 36.49 | 29.67 | 30.68 | 11.12 | -4.06 |
| 12 | 220 | 22.30 | 50.79 | 36.42 | 29.68 | 29.96 | 11.23 | -4.25 |
| 13 | 240 | 22.36 | 51.03 | 36.41 | 29.69 | 30.95 | 11.40 | -4.64 |
| 14 | 260 | 22.41 | 51.05 | 36.32 | 29.70 | 31.05 | 11.50 | -4.76 |
| 15 | 280 | 22.44 | 51.11 | 36.24 | 29.69 | 30.38 | 11.62 | -5.08 |
| 16 | 300 | 22.49 | 51.13 | 36.18 | 29.68 | 30.43 | 11.77 | -5.44 |
| 17 | 320 | 22.49 | 50.97 | 36.06 | 29.64 | 31.06 | 11.82 | -5.46 |
| 18 | 340 | 22.49 | 50.87 | 36.01 | 29.61 | 31.08 | 11.98 | -5.64 |
| 19 | 360 | 22.51 | 50.74 | 35.94 | 29.59 | 30.43 | 12.08 | -5.72 |
| 20 | 380 | 22.48 | 50.66 | 35.85 | 29.55 | 30.47 | 12.13 | -5.56 |
| 21 | 400 | 22.45 | 50.65 | 35.81 | 29.51 | 30.77 | 12.23 | -5.60 |
| 22 | 420 | 22.44 | 50.61 | 35.77 | 29.51 | 30.03 | 12.29 | -5.46 |
| 23 | 440 | 22.40 | 50.52 | 35.73 | 29.50 | 29.70 | 12.33 | -5.26 |

| Num | Position / Time (s) | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
|---|---|---|---|---|---|---|---|---|
| 24 | 460 | 22.40 | 50.46 | 35.79 | 29.52 | 30.37 | 12.46 | -5.18 |
| 25 | 480 | 22.37 | 50.50 | 35.76 | 29.49 | 30.10 | 12.51 | -4.98 |
| 26 | 500 | 22.32 | 50.66 | 35.68 | 29.46 | 30.79 | 12.62 | -4.82 |
| 27 | 520 | 22.33 | 50.86 | 35.64 | 29.47 | 30.58 | 12.72 | -4.79 |
| 28 | 540 | 22.30 | 51.06 | 35.52 | 29.43 | 30.58 | 12.74 | -4.58 |
| 29 | 560 | 22.24 | 51.23 | 35.45 | 29.41 | 30.50 | 12.78 | -4.27 |
| 30 | 580 | 22.20 | 51.32 | 35.41 | 29.37 | 30.46 | 12.86 | -3.99 |
| 31 | 600 | 22.15 | 51.42 | 35.29 | 29.32 | 30.56 | 12.85 | -3.41 |
| | Average, ∑ | 21.90 | 49.75 | 35.89 | 29.42 | 30.28 | 11.77 | -3.88 |
| | Median | 22.30 | 50.65 | 35.87 | 29.51 | 30.43 | 11.77 | -4.58 |
| | Standard Deviation, σ | 0.94 | 2.06 | 0.53 | 0.32 | 0.53 | 0.71 | 1.85 |

**APPENDIX 3**

| Ambient Temperature (ºC) | | 30°C ± 0.1 | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Heat Load (kW) | | 300 W | | | | | | |
| Blower Fan Speed | | Level 2 | | | | | | |
| Engine Speed, N (rpm) | | 1900 | | | | | | |
| Num. | Position / Time (s) | Temperature (°C) | | | | | | |
| | | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
| 1 | 0 | 18.22 | 47.87 | 40.53 | 32.08 | 29.54 | 16.64 | 10.10 |
| 2 | 20 | 18.27 | 48.38 | 40.55 | 32.16 | 29.57 | 16.57 | 10.08 |
| 3 | 40 | 18.30 | 49.34 | 40.64 | 32.29 | 29.73 | 16.45 | 10.17 |
| 4 | 60 | 18.17 | 50.14 | 40.62 | 32.38 | 29.54 | 16.32 | 10.20 |
| 5 | 80 | 18.18 | 50.97 | 40.67 | 32.52 | 29.78 | 16.28 | 10.33 |
| 6 | 100 | 18.06 | 51.71 | 40.69 | 32.61 | 29.97 | 16.24 | 10.51 |
| 7 | 120 | 17.98 | 52.24 | 40.68 | 32.67 | 29.77 | 16.22 | 10.65 |
| 8 | 140 | 17.93 | 52.71 | 40.73 | 32.70 | 30.26 | 16.24 | 10.80 |
| 9 | 160 | 17.80 | 53.00 | 40.76 | 32.73 | 30.51 | 16.22 | 10.99 |

| Num | Position / Time (s) | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
|---|---|---|---|---|---|---|---|---|
| 10 | 180 | 17.81 | 53.22 | 40.82 | 32.75 | 30.39 | 16.27 | 10.80 |
| 11 | 200 | 17.76 | 53.44 | 40.86 | 32.79 | 30.52 | 16.34 | 10.83 |
| 12 | 220 | 17.76 | 53.57 | 40.85 | 32.81 | 30.54 | 16.38 | 11.16 |
| 13 | 240 | 17.81 | 53.66 | 40.85 | 32.82 | 31.09 | 16.43 | 11.37 |
| 14 | 260 | 17.77 | 53.63 | 40.81 | 32.80 | 30.21 | 16.50 | 11.43 |
| 15 | 280 | 17.83 | 53.54 | 40.82 | 32.81 | 31.18 | 16.54 | 11.26 |
| 16 | 300 | 17.83 | 53.53 | 40.86 | 32.83 | 31.64 | 16.56 | 11.03 |
| 17 | 320 | 17.87 | 53.41 | 40.86 | 32.85 | 30.70 | 16.60 | 10.94 |
| 18 | 340 | 17.94 | 53.23 | 40.90 | 32.91 | 30.34 | 16.69 | 10.92 |
| 19 | 360 | 17.97 | 52.87 | 40.88 | 32.90 | 30.64 | 16.76 | 10.94 |
| 20 | 380 | 18.07 | 52.57 | 40.91 | 32.95 | 31.28 | 16.83 | 10.96 |
| 21 | 400 | 18.10 | 52.62 | 40.93 | 32.99 | 31.31 | 16.87 | 11.04 |
| 22 | 420 | 18.11 | 52.68 | 40.95 | 32.99 | 31.97 | 16.92 | 11.16 |
| 23 | 440 | 18.10 | 52.79 | 40.93 | 32.95 | 32.24 | 16.90 | 11.22 |

| Num | Time (s) \ Position | Compressor in | Condenser in | Expansion valve in | Evaporator in | Blower in | Blower out | Ducting |
|---|---|---|---|---|---|---|---|---|
| 24 | 460 | 18.09 | 52.92 | 40.88 | 32.86 | 31.11 | 16.90 | 11.33 |
| 25 | 480 | 18.08 | 53.08 | 40.88 | 32.81 | 30.69 | 16.88 | 11.35 |
| 26 | 500 | 18.01 | 52.97 | 40.72 | 32.65 | 30.84 | 17.03 | 11.47 |
| 27 | 520 | 17.89 | 52.32 | 40.33 | 32.33 | 30.88 | 17.14 | 11.57 |
| 28 | 540 | 17.68 | 51.85 | 39.89 | 31.97 | 31.57 | 16.87 | 11.41 |
| 29 | 560 | 17.49 | 51.69 | 39.52 | 31.65 | 32.16 | 16.71 | 11.76 |
| 30 | 580 | 17.34 | 51.70 | 39.29 | 31.44 | 32.50 | 16.61 | 11.77 |
| 31 | 600 | 17.17 | 51.60 | 39.11 | 31.18 | 32.89 | 16.83 | 11.81 |
| Average, ∑ | | 17.92 | 52.23 | 40.60 | 32.55 | 30.82 | 16.60 | 11.01 |
| Median | | 17.94 | 52.71 | 40.81 | 32.75 | 30.69 | 16.60 | 11.03 |
| Standard Deviation, σ | | 0.26 | 1.50 | 0.48 | 0.47 | 0.91 | 0.27 | 0.49 |