

**DETERMINATION OF CO-EFFICIENT OF PERFORMANCE (COP) OF A
VAPOR COMPRESSION REFRIGERATION CYCLE USING LIQUEFIED
PETROLEUM GAS (R-290) AS REFRIGERANT AND COMPARE IT WITH
R-134a**

A THESIS SUBMITTED TO THE DEPARTMENT OF “MECHANICAL ENGINEERING” IN
PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE DEGREE OF BACHELOR OF
SCIENCE IN MECHANICAL ENGINEERING

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A thesis submitted to the Department of Mechanical Engineering, Military Institute of Science and Technology, Dhaka, on December 2017 in partial fulfillment of the requirements for the degree of B.Sc in Mechanical Engineering.

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SUPERVISOR'S DECLARATION

I hereby declare to the Department of Mechanical Engineering, Military Institute of Science & Technology, Dhaka, Bangladesh that I have checked this thesis, **“DETERMINATION OF CO-EFFICIENT OF PERFORMANCE (COP) OF A VAPOR COMPRESSION REFRIGERATION CYCLE USING LIQUEFIED PETROLEUM GAS (R-290) AS REFRIGERANT AND COMPARE IT WITH R-134a”** and in my opinion this project is satisfactory in terms of scope and quality for the partial achievement of the degree of Bachelor of science in Mechanical Engineering.

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STUDENT'S DECLARATION

This is to certify that the work presented in this thesis titled “**DETERMINATION OF CO-EFFICIENT OF PERFORMANCE (COP) OF A VAPOR COMPRESSION REFRIGERATION CYCLE USING LIQUEFIED PETROLEUM GAS (R-290) AS REFRIGERANT AND COMPARE IT WITH R-134a**” is an outcome of the investigation carried out by the authors under the supervision of **Lieutenant Colonel Golam Saklayen**, Associate Professor, Department of Mechanical Engineering, MIST. This thesis or any part of it has not been submitted elsewhere for the award of any other degree or other similar title.

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ABSTRACT

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. Performance of the system becomes main issue and many researches are still ongoing to evaluate and improve efficiency of the system. In this modern age, various refrigerants are used like R-12, R-22, R-134a etc. All these refrigerants are of CFC/HCFC group. These refrigerants contain high Global Warming Potential (GWP) and Ozone Depletion Potential (ODP). It has become a crying need to use such a refrigerant that is highly efficient as well as environment friendly. We have chosen Liquefied Petroleum Gas as our experimental object. We have investigated the result of an experimental study carried out to determine the performance of domestic refrigerator when a liquefied petroleum gas (LPG) which is locally available which comprises of 80% propane, 20% butane which is varied from company to company is used as a Refrigerant. The LPG is cheaper and possesses an environmental friendly nature with no Ozone Depletion Potential (ODP) and no Global Warming Potential (GWP). It is used in world for cooking purposes. The refrigerator used in the present study is designed to work on LPG. The performance parameters investigated is the refrigeration effect in certain time. The refrigerator worked efficiently when LPG was used as a refrigerant instead of R134a. The evaporator temperature reached -2.2°C with an ambient temperature of 28°C . Also from the experiment which done in atmospheric condition, we can predict the optimum value of cooling effect with the suitable operating condition of regulating valve and capillary tube of the system.

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Nomenclature

HCFC	Hydro Chloro fluoro Carbon
CFC	Chloro Fluoro Carbon
H	Enthalpy
T	Temperature
P	Pressure
M	Mass Flow rate
COP	Co efficient of Performance
ODP	Ozone Depletion Potential
GWP	Global Warming Potential
S	Entropy
V	Specific Volume
D	Density of Refrigerant

SUBSCRIPT

	Liquid
V	Vapor
Sat	Saturated
Theo	Theoretical
Act	Actual
Inl	Inlet
Con	Condenser
Evp	Evaporator

CHAPTER -01

INTRODUCTION

1.1 BACKGROUND

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.

Now-a-days 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. The refrigerant that are used in it is somewhat analogous to a reverse Rankin cycle. The vapor compression refrigeration system contains four main components which are compressor, condenser, expansion device, and evaporator.

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 refrigerants 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. Besides it has become important to prevent any harmful effect to our environment. Like, now some types of refrigerants that are used in now a days, for example R-134a, R-12, R-22 are very useful but very much harmful for our environment. They contain high amount of ozone depletion potential and global warming potential. These types of gases cause damage to ozone layer and cause global warming. So it has become a crying need to use another type of refrigerant that can be efficient as well as environment friendly. In our view we are using liquefied petroleum gas as refrigerant. Liquefied petroleum gas is the mixture of 80% of Propane and 20% of Butane in our countries perspective. It has a very low global warming potential and zero ozone depletion potential. It can be charged to our refrigeration system and can be used as refrigerant by maintaining the properties of a common refrigerant. Now we are using Liquefied Petroleum Gas (LPG) or R-290 and going to compare it with a common refrigerant like R-134a.

1.2 PROBLEM STATEMENT

Now-a-days, refrigeration system is important because it is widely used for domestic application. However, the actual performance of the refrigerator is still unknown. So, we need some research to analyze the coefficient of performance of liquefied petroleum gas as a refrigerant. At the same time we need to analyze the coefficient of performance of another useful refrigerant like R-134a under same condition and power. For this we have to construct a test rig to find the coefficient of performance.

1.3 OBJECTIVES OF RESEARCH

The main objectives of this thesis are given below:

- To develop an experimental rig for vapor compression refrigeration system.
- To determine the coefficient of performance (COP) of the refrigeration system using liquefied petroleum gas as refrigerant.
- To learn and understand the basic vapor compression refrigeration systems.
- To identify correctly the exact locations of points where the data for temperature and pressure should be collected.
- Compare the properties of refrigerants with another useful refrigerant R-134a.
- Analyze the properties of liquefied petroleum gas and determine whether it can be replaced with usual refrigerant R-134a.

CHAPTER – 02

LITERATURE REVIEW

2.1 HISTROY OF VAPOR COMPRESSION REFRIGERATION SYSTEM

History of refrigeration is long and refrigeration changed along the way from the pretty primitive yet ingenious to modern technology which allowed people to have refrigerators in their house and not depend on nature. Find out more about refrigeration and different forms of cooling.

The basis of modern refrigeration is the ability of liquids to absorb enormous quantities of heat as they boil and evaporate. Professor William Cullen of the University of Edinburgh demonstrated this in 1755 by placing some water in thermal contact with ether under a receiver of a vacuum pump. The evaporation rate of ether increased due to the vacuum pump and water could be frozen. This processing involves two thermodynamic concepts, the vapor pressure and the latent heat. A liquid is in thermal equilibrium with its own vapor at a pressure called the saturation pressure, which depends on the temperature alone. If the pressure is increased for example in a pressure cooker, the water boils at higher temperature. The second concept is that the evaporation of liquid requires latent heat during evaporation. If latent heat is extracted from the liquid, the liquid gets cooled. The temperature of ether will remain constant as long as the vacuum pump maintains a pressure equal to saturation pressure at the desired temperature. This requires the removal of all the vapors formed due to vaporization. If a lower temperature is desired, then a lower saturation pressure will have to be maintained by the vacuum pump. The component of the modern day refrigeration system where cooling is produced by this method is called evaporator. If this process of cooling is to be made continuous the vapors have to be recycled by condensation to the liquid state. The condensation process requires heat rejection to the surroundings. It can be condensed at atmospheric temperature by increasing its pressure. The process of condensation was learned in the second half of eighteenth century. U.F. Clouet and G. Monge liquefied SO_2 in 1780 while van Marum and Van Troostwijk liquefied NH_3 in 1787. Hence, a compressor is required to maintain a high pressure so that the evaporating vapors can condense at a temperature greater than that of the surroundings. The man responsible for making a practical vapor compression refrigeration system was James Harrison who took a

patent in 1856 for a vapor compression system using ether, alcohol or ammonia. Charles Tellier of France patented in 1864, refrigeration system using diethyl ether which has a normal boiling point of -23.6°C . Carl von Linde in Munich introduced double acting ammonia compressor. It required pressures of more than 10 atmospheres in the condenser. Since the normal boiling point of ammonia is -33.3°C , vacuum was not required on the low pressure side. Since then ammonia is used widely in large refrigeration plants. David Boyle, in fact made the first NH_3 system in 1871 in San Francisco. John Enright had also developed a similar system in 1876 in Buffalo N.Y. Franz Windhausen developed carbon dioxide (CO_2) based vapor compression system in Germany in 1886. The carbon dioxide compressor requires a pressure of about 80 atmospheres and therefore a very heavy construction. Its lowest pressure was high enough to prevent the leakage of air into the system. Palmer used $\text{C}_2\text{H}_5\text{Cl}$ in 1890 in a rotary compressor. He mixed it with $\text{C}_2\text{H}_5\text{Br}$ to reduce its flammability. Edmund Copeland and Harry Edwards used iso-butane in 1920 in small refrigerators. It disappeared by 1930 when it was replaced by CH_3Cl . Di-chloro-ethylene (Dielene) was used by Carrier in centrifugal compressors in 1922-26.

2.2 TYPES OF AIR REFRIGERATION CYCLE

There are mainly two types of air refrigeration cycle. They are-

- Open Air Refrigeration Cycle
- Closed Air refrigeration Cycle

2.2.1 Open Air Refrigeration Cycle

In an open air refrigeration cycle, the air is directly led to the space to be cooled, allowed to circulate through the cooler and then returned to the compressor to start another cycle. Since the air is supplied to the refrigerator at atmospheric pressure, therefore, volume of air handled by compressor and expander is large. Thus, the size of compressor and expander should be large. Another disadvantage of open air cycle is that, the moisture is regularly carried away by the air circulated through the cooled space. This leads to formation of frost at the end of the expansion process and clog in line. In this process, a dryer is used.

In this case, we have used open refrigeration cycle, since we have used Liquefied Petroleum Gas (LPG) or R-290 as primary refrigerant and R-134a as secondary refrigerant for comparison. Both of them require open air refrigeration cycle.

2.2.2 Closed Air Refrigeration Cycle

In a closed or dense air refrigeration cycle, the air is passed through the pipes and component parts of the system at all time. The air in this system is used for absorbing heat from other fluid and this cooled brine is circulated into the space to be cooled. There are some thermodynamic advantages of closed air refrigeration cycle. They are

- Since it can work at suction pressure higher than that of atmospheric pressure, therefore the volume of air is handled by compressor or expander are smaller as compared to an open air refrigeration cycle system.
- The operating pressure ratio can be reduced, which results in higher coefficient of performance.

2.3 HISTORY OF REFRIGERANTS

In the process of refrigeration the heat is carried from the low temperature reservoir to the high temperature (against the second law of thermodynamics). The refrigerant is the fluid that is used in the vapor compression refrigeration system for carrying the heat from the substance or the fluid to be cooled and throw it to the atmosphere. The substance to be cooled is at low temperature and the atmosphere is at higher temperature. During the process of transfer of heat the refrigerant repeatedly gets converted into the vapor state when it absorbs the heat and into the liquid state when it gives up the heat to the atmosphere.

The history that we are talking about essentially refers to these substances. Since these substances have to evaporate and condense at required temperatures (which may broadly lie in the range of -100°C to 100°C) at reasonable pressures, they have to be essentially volatile. Hence, the development of refrigerants started with the search for suitable, volatile substances. Historically the development of these refrigerants can be divided into three distinct phases, namely:

- Refrigerants prior to the development of CFCs
- The synthetic fluorocarbon (FC) based refrigerants
- Refrigerants in the aftermath of stratospheric ozone layer depletion.

2.3.1 Refrigerants Prior to the Development of CFCs:

2.3.1.1 First Generation Refrigerants:

Beginnings of mechanical refrigeration, starting from early 19th century were characterized by use of natural refrigerants. Refrigerators that were built in the late 1800s to 1929 used the first generation refrigerants like methyl chloride, ammonia and sulphur dioxide. The common refrigerants for the first hundred years included

whatever worked and whatever was available. Nearly all the first generation refrigerants were flammable, toxic or both and some were also highly reactive.

Table 2.1: Properties of Refrigerants

Substance	R Number	M (kg/kmol)	NBP (°C)	GWP
Carbon dioxide	R-744	44.01	-55.6	1
Ammonia	R-717	17.03	-33.3	0
Sulphur dioxide	R-764	64.06	-10.0	0
Ethyl ether	R-610	74.12	35	0
Dimethyl ether	R-170	46.07	-25	0
Methyl chloride	R-40	50.49	-24.2	16

2.3.1.2 Second Generation Refrigerants:

The second generation refrigerants were distinguished by a shift to chlorofluoro chemicals for safety and durability. Thomas Midgley and his associates studied the property tables of elements of periodic table. They disregarded compounds that are unstable, toxic, yielding insufficient volatility and inert gases based on their low boiling point. In 1928, Midgley and his colleagues made critical observations regarding flammability and toxicity of compounds containing elements like carbon, nitrogen, oxygen, sulphur, hydrogen, fluorine, chlorine and bromine. Their first publication was on fluorochloro refrigerants and it showed how the variation of chlorination and fluorination of hydrocarbons influences boiling point, flammability and toxicity of the refrigerants. Thus CFC refrigerants made the second generation of refrigerants. CFC is a non-toxic, non-flammable gas with relatively high mass. It is a good refrigerant because it can be compressed easily to liquid and carries away lots of heat when it evaporates. It is very stable that only UV rays can break it down. In fact, it's well suited to a variety of applications because it doesn't react with anything; it works well as a solvent, a blowing agent, a fire extinguishing agent and an aerosol propellant. Because it is a single molecule, not a mixture, it doesn't separate out at different pressures or temperatures. Some of the refrigerants of this generation are presented here along with their thermodynamic properties and applications.

2.3.1.3 Third Generation Refrigerants:

They were mostly low ozone depletion potential refrigerants. Another new class of fluorocarbon refrigerants called hydrofluoro-olefin (HFO) with potential for reduced GWP have been developed. Their primary advantage, other than their low GWP, is that they can be used with existing refrigeration system designs. This is good for the industry and their customers, but it is still a fluorinated gas. There is growing political pressure to regulate it out of production and force the industry to develop an even lower-impact refrigeration technology. So the search continues.

2.3.2 THE SYNTHETIC CFCs/HCFCs

Almost all the refrigerants used in the early stages of refrigeration suffered from one problem or other. Most of these problems were linked to safety issues such as toxicity, flammability, high operating pressures etc. As a result large-scale commercialization of refrigeration systems was hampered. Hence it was felt that “refrigeration industry needs a new refrigerant if they expect to get anywhere”. The task of finding a “safe” refrigerant was taken up by the American Thomas Midgley, Jr., in 1928. Midgley was already famous for the invention of tetra ethyl lead, an important anti-knock agent for petrol engines. Midgley along with his associates Albert L. Henne and Robert R. McNary at the Frigidaire Laboratories (Dayton, Ohio, USA) began a systematic study of the periodic table. From the periodic table they quickly eliminated all those substances yielding insufficient volatility. They then eliminated those elements resulting in unstable and toxic gases as well as the inert gases, based on their very low boiling points. They were finally left with eight elements: carbon, nitrogen, oxygen, sulphur, hydrogen, fluorine, chlorine and bromine. These eight elements clustered at an intersecting row and column of the periodic table, with fluorine at the intersection.

A look at the refrigerants discussed above shows that all of them are made up of seven out of the eight elements identified by Midgley (fluorine was not used till then). Other researchers have repeated Midgley’s search with more modern search methods and databases, but arrived at the same conclusions. Based on their study, Midgley and his colleagues have developed a whole range of new refrigerants, which are obtained by partial replacement of hydrogen atoms in hydrocarbons by fluorine and chlorine. They have shown how fluorination and chlorination of hydrocarbons can be varied to obtain desired boiling points (volatility) and also how properties such as toxicity, flammability are influenced by the composition. The first commercial refrigerant to come out of Midgley’s study is Freon-12 in 1931. Freon-12 with a chemical formula CCl_2F_2 , is obtained by replacing the four atoms of hydrogen in methane (CH_4) by two atoms of chlorine and two atoms of fluorine. Freon-12 has a normal boiling point of -29.8°C , and is one of the most famous and popular synthetic refrigerants. It was exclusively used in small domestic refrigerators, air conditioners, water coolers etc. for almost sixty years. Freon-11 (CCl_3F) used in large centrifugal air conditioning systems was introduced in 1932. This is followed by Freon-22 (CHClF_2) and a whole series of synthetic refrigerants to suit a wide variety of applications. Due to the emergence of a large number of

refrigerants in addition to the existence of the older refrigerants, it has become essential to work out a numbering system for refrigerants. Thus all refrigerants were indicated with "R" followed by a unique number (thus Freon-12 is changed to R-12 etc.). The numbering of refrigerants was done based on certain guidelines. For all synthetic refrigerants the number (e.g. 11, 12, and 22) denotes the chemical composition.. Refrigerant mixtures begin with the number 4 (zeotropic) or 5 (azeotropic), e.g. R-500, R-502 etc.

The introduction of CFCs and related compounds has revolutionized the field of refrigeration and air conditioning. Most of the problems associated with early refrigerants such as toxicity, flammability, and material incompatibility were eliminated completely. Also, Freons are highly stable compounds. In addition, by cleverly manipulating the composition a whole range of refrigerants best suited for a particular application could be obtained. In addition to all this, a vigorous promotion of these refrigerants as "wonder gases" and "ideal refrigerants" saw rapid growth of Freons and equally rapid exit of conventional refrigerants such as carbon dioxide, sulphur dioxide etc. Only ammonia among the older refrigerants survived the Freon magic. The Freons enjoyed complete domination for about fifty years, until the Ozone Layer Depletion issue was raised by Rowland and Molina in 1974. Rowland and Molina in their now famous theory argued that the highly stable chlorofluorocarbons cause the depletion of stratospheric ozone layer. Subsequent studies and observations confirmed Rowland and Molina theory on stratospheric ozone depletion by chlorine containing CFCs. In view of the seriousness of the problem on global scale, several countries have agreed to ban the harmful Ozone Depleting Substances, ODS (CFCs and others) in a phase-wise manner under Montreal Protocol. Subsequently almost all countries of the world have agreed to the plan of CFC phase-out. In addition to the ozone layer depletion, the CFCs and related substances were also found to contribute significantly to the problem of "global warming". This once again brought the scientists back to the search for "safe" refrigerants. The "safety" now refers to not only the immediate personal safety issues such as flammability, toxicity etc., but also the long-term environmental issues such as ozone layer depletion and global warming.

2.3.3 REFRIGERANTS IN THE AFTER MATH OF OZONE LAYER DEPLETION

The most important requirement for refrigerants in the aftermath of ozone layer depletion is that it should be a non-Ozone Depleting Substance (non-ODS). Out of this requirement two alternatives have emerged. The first one is to look for zero ODP synthetic refrigerants and the second one is to look for "natural" substances. Introduction of hydro fluorocarbons (HFCs) and their mixtures belong to the first route, while the re-introduction of carbon dioxide (in a supercritical cycle), water and various hydrocarbons and their mixtures belong to the second route. The increased use of ammonia and use of other refrigeration cycles such as air cycle refrigeration systems and absorption systems also come under the second route. Both these routes have found their proponents and opponents. HFC-134a (synthetic substance) and hydrocarbons (natural substances) have emerged as alternatives to Freon-12. No clear

pure fluid alternative has been found as yet for the other popular refrigerant HCFC-22. However several mixtures consisting of synthetic and natural refrigerants are being used and suggested for future use. Table shows the list of refrigerants being replaced and their alternatives. Mention must be made here about the other environmental problem, global warming. In general the non-ODS synthetic refrigerants such as HFC-134a have high global warming potential (GWP), hence they face an uncertain future. Since the global warming impact of a refrigerant also depends on the energy efficiency of the system using the refrigerant (indirect effect), the efficiency issue has become important in the design of new refrigeration systems. Though the issues of ozone layer depletion and global warming has led to several problems, they have also had beneficial effects of making people realize the importance of environmental friendliness of technologies. It is expected that with the greater awareness more responsible designs will emerge which will ultimately benefit the whole mankind.

2.4 HISTORY OF LPG

Liquid petroleum gas or liquid petroleum gas (LPG or LP gas), also referred to as simply propane or butane are flammable mixtures of hydrocarbon gases used as fuel in heating appliances, cooking equipment, and vehicles.

First mention of propane and butane mixture comes from as early as 1910. It was then that Walter O. Snelling, an American chemist researching properties of petrol, separated gaseous fractions from liquid ones, thus discovering the existence of propane. Two years later, in 1912, he started his first domestic propane installation, and in 1913 he patented producing propane on an industrial scale. Later that year the patent was bought by Frank Philips, the founder of the Conoco Philips oil company. Even so, LPG consumption did not grow considerably.

LPG was gaining momentum rapidly. In 1932, it was used for cooking and water heating during the Olympic Games in Los Angeles. Propane-butane industry was growing stronger by the year and managed to produce and sell 56 million gallons of LPG in 1934. In the few following years, demand for liquefied petroleum gas was further boosted by the popularity of airships, regularly traveling between Europe and the US. The then latest-generation Zeppelin series airships were propelled by engines fueled with the so called Blau gas (invented by Herman Blau), itself very much alike butane – one of LPG's ingredients. Using gaseous fuel with roughly the same mass as air was actually very convenient for airships as it did not alter a zeppelin's overall weight the way liquid fuels did (airships would become considerably lighter once liquid fuels were burned away, thus forcing the release of hydrogen, which was extremely dangerous). Unfortunately, when the Hindenburg – the largest airship ever built – was destroyed in a disaster in 1937, killing 36 people, the zeppelin era ended abruptly.

When World War II was over and industrial production rose again, LPG sales in the USA topped 1 billion gallons. Nearly 62% of all American households had LPG installations at the time. By 1947, the first liquefied gas tanker was built and entered service. In

1950, Chicago Transit Authority, a public transport operator from Chicago, ordered 1000 LPG-powered buses, while in Milwaukee 270 taxis were converted in the same year. In 1958, LPG sales reached 7 billion gallons and in 1965 Chevrolet introduced 4 new LPG-powered engines for commercial vehicles.

Initial international export contracts were not made until the 1950's. However, the amount of exported LPG was still low in the 1960's – below 1 million tons were shipped outside of the US. Within the next 20 years, export had grown to 17 million tons and reached 48 million tons in the year 2000.

LPG first appeared in Europe when it was imported from the USA and introduced in France in mid-1930's. It was bottled and came from an East Coast refinery. In 1938, the Italian company Liquigas started filling bottles with LPG in a facility near Venice. The outbreak of World War II hampered further development for a few years on the Old Continent.

R-290 is a hydrocarbon that was introduced into the refrigeration industry in the early 1900s. Like other natural refrigerants, it was used through the 1930s until it was replaced by chlorofluorocarbons (CFCs). Since the 2000s, R-290 has been regaining popularity in a wide range of applications. With its increase in capacity and low global warming potential (GWP), R-290 has emerged as an alternative to hydro fluorocarbons (HFCs) like R-404A and HFC-134a. Global regulatory actions to ban high-GWP refrigerants have placed renewed emphasis on R-290 and other natural refrigerants. It was officially approved in a recent EPA SNAP ruling, and has been identified as a viable HFC alternative in the E.U.'s F Gas regulations. And, with R-404A and HFC-134a officially delisted by the EPA in many commercial refrigeration applications, natural refrigerants including R-290 are primed for a broader resurgence.

2.5 EXTRACTION OF LPG

2.5.1 LPG STRIPPED FROM NATURAL GAS

Separating the Gas from the Oil:

- The gas/oil mixture is piped out of the well and into a gas trap, which separates the stream into crude oil and "wet" gas, which contains LPG and natural gas.
- The heavier crude oil sinks to the bottom of the trap and is then pumped into an oil storage tank for refining.
- The "wet" gas, off the top of the gas trap, is processed to separate the gasoline (petrol) from the natural gas and LPG.

Processing the Methane:

- Impurities such as water, carbon dioxide, nitrogen and hydrogen sulfide must be removed before either the natural gas (methane) or LPG can be used. Some of these can be sold as by-products.
- The refined and purified natural gas, which is mostly methane, is fed into the pipelines that supply our cities and towns.
- Distribution of refined natural gas (methane) is typically handled by gas utility companies.

Processing the LPG:

- The LPG must be separated from an intricate mixture of hydrocarbons, which includes the methane, ethane, ethene, propene, isobutene, butadiene, pentane, and pentene.
- The plants that do the processing are frequently called “stripper plants”, as the LPG is stripped from the natural gas flow.
- Impurities must also be removed from the propane and butane or they will prevent the LPG from liquefying properly.
- The refrigeration technique is common for recovery of LPG from a natural gas stream.
- With this technique, they refrigerate the gas stream to obtain the LPG.
- Refrigeration is employed in three different processes: expander plants, low temperature separation and combined processes.
- Other separation techniques may also be employed, including lean oil absorption.
- This LPG can be used as a mixture or further separated into propane, butane and isobutane, as needed.
- The LPG enters its own distribution network, where it eventually finds its way to end users, including 45kg home LPG and commercial LPG users all around Australia and the world.

2.5.2 LPG Processed from Oil Refining:

Refining the Crude Oil:

- Crude oil undergoes a variety of refining processes including fluid catalytic cracking, delayed cokers, and crude distillation.
- One of the refined products is LPG.
- After desalting, the heated crude oil is pumped into the distillation tower.
- Fractions of the flow are extracted from the side of the distillation tower at various heights between the bottom and the top.
- Each extraction point is temperature controlled to extract a specific fraction including gasoline, naphtha, kerosene, diesel, light gas oil and heavy gas oil.
- These are then sent to unique streams for storage or possible further processing.

Refining the LPG:

- LPG, with the lowest boiling point, is extracted from the top of the distillation tower.
- This LPG component can be used as a mixture or further separated into its three primary parts: propane, butane and iso-butane.
- This further fractionation can be achieved with Depropaniser, Debutaniser and Deisobutaniser fractionators.

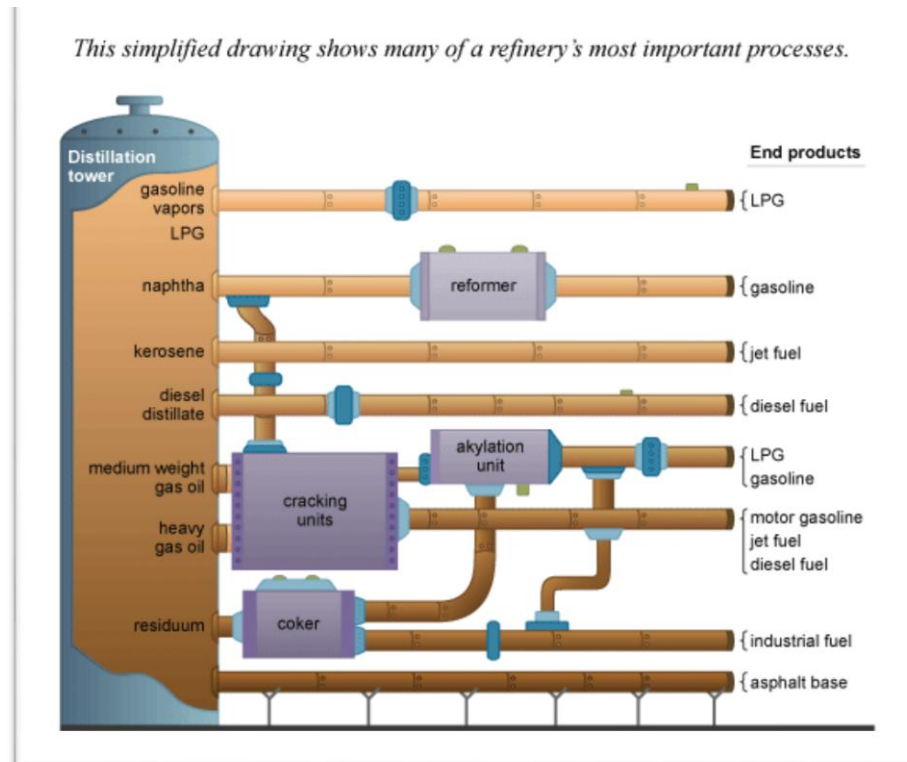


Fig 2.1: Refining crude oil

2.6 ADVANTAGES AND DISADVANTAGES OF LPG

Our prime substance, liquefied petroleum gas has some advantages as well as some disadvantages. Considering all these properties we can consider it as a good or bad refrigerant

2.6.1 ADVANTAGE OF USING LPG(R-290)

- Because of its long history of usefulness, some properties and performance efficiencies of LPG(R-290) have been well vetted. These properties include some thermodynamic properties. In case of pressure, low back pressure, discharge temperature, volumetric efficiency capacity loss and co-efficient of performance(COP) it has been found that, R-290(LPG-Propane Gas) is very much similar to R-22(Chloro-di-fluoro methane). R-22 is a colorless gas that has been used as refrigerant because of the compound's ozone depletion potential(ODP) and global warming potential(GWP).
- We have found LPG(R-290) capable of high performing efficient operation. Compared to the refrigerants it will likely be called upon to replace — like the

recently delisted R-404a and R-134a — R-290 yields more capacity with lower wattage consumption . We've developed a full line of Copeland hermetic compressors to be compatible with R-290.

- In terms of achieving regulatory compliance, R-290 is very appealing. First, its global warming potential (GWP = 3) is well below the global threshold of 150 GWP, which places it in an elite class of refrigerants from the standpoint of minimal environmental impact. And, its high-efficiency characteristics also qualify it as a candidate for meeting the Department of Energy's (DOE) energy reduction rules .
- R-290 or LPG has small charge limit that is about 150g. So it can be used for commercial application such as beverage cooler, frozen drink machine, ice machines, small ice cream freezer and small rich in units. We can install some freezing machines for food that is designed according to R-290.
- R-290 is a hydrocarbon based non synthetic refrigerant. It has very low global impact which is it's most advantageous side. It has GWP (Global Warming Potential)= 3 and (Ozone Depletion Potential)= 0 . It is a relatively affordable that has high efficiency, high performance and reliability.

2.6.2 DISADVANTAGE OF USING LPG(R-290)

- LPG or R-290 is considered as Class A3 refrigerant. Class A3 refrigerant are considered as highly flammable refrigerant. It is risky to use LPG in any kinds of activities whereas we are thinking to use it as refrigerant in a refrigeration system.
- Using R-290 or LPG as a refrigerant need special requirements as well as Certification. In many cases LPG or R-290 can not be used with the equipment and compressor that are used for other HFC refrigerants. For example, we will need different kind of Compressor with special properties if we use R-290 commercially. Otherwise accident may occur.
- It has globally mandated low charge limits that has restricted the application range.
- There is a lack of trained and certified field technicians which has created a great problems is case of charging LPG(R-290) in refrigeration systems or doing any field work related to refrigerant(R-290). It may also create problems where physical implement of work is needed to work with R-290.

2.6.3 PROPERTIES OF GOOD REFRIGERANT

- Colorless
- Odorless. (It is normal to odorise LPG by adding an odorant prior to supply to the user, to aid the detection of any leaks).

- Flammable
- Heavier than air.
- Approximately half the weight of water.
- Non-toxic but can cause asphyxiation.
- LPG expands upon release and 1 litre of liquid will form approximately 250 litres of vapor.

2.6.4 PROPERTIES OF REFRIGERANTS THAT CAN MAKE US TO CONSIDER IT AS A GOOD REFRIGERANT

A Good Mixture

LPG is mainly Propane (C₃H₈), Butane (C₄H₁₀) or a mix of Propane/Butane. Since LPG has such a simple chemical structure, it is among the cleanest of any alternative fuel.

Boiling Point

LPG's boiling point ranges from -42 °C to 0 °C depending on its mixture percentage of Butane and Propane.

Combustion

The combustion of LPG produces carbon dioxide (CO₂) and water vapor but sufficient air must be available. Inadequate appliances flueing or ventilation can result in the production of carbon monoxide which can be toxic.

Vapor Pressure

LPG is stored as a liquid under pressure. It is almost colorless and its weight is approximately half that of an equivalent volume of water. The pressure inside a closed container in which LPG is stored is equal to the vapor pressure of the liquid and corresponds to its temperature.

Ignition Temperature:

The temperature required to ignite LPG in air is around 500°C

Calorific Value

The calorific value of LPG is about 2.5 times higher than that of main gas so more heat is produced from the same volume of gas.

Toxicity

LPG is a colorless, odorless and non-toxic gas. It is supplied commercially with an added odorant to assist detection by smell. LPG is an excellent solvent of petroleum and rubber product and is generally non-corrosive to steel and copper alloys.

Safety

LPG is just as safe as any other fuel. In fact it is safer than most fuels because neither LPG itself nor the end products that are produced by burning LPG in a suitable

appliance, are poisonous to inhale. Since LPG cannot burn without air, there can never be a 'Flash-back' into the cylinder.

2.7 LPG IN AIR REFRIGERATOR CYCLE (BELL-COLEMAN CYCLE)

A Bell-Coleman air refrigeration machine was developed by Bell Coleman by reversing the Joule's air cycle. It is one of the earliest types of refrigeration system. In this process, we have used LPG gas as refrigerant. This process can be described through schematic diagram and p-v and T-s chart.

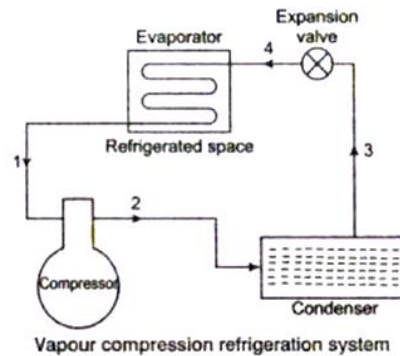


Fig 2.2 : Bell-Coleman Cycle

A Bell Coleman cycle is a modification of reversed Carnot Cycle. About four steps are considered in the cycle.

- Isentropic Compression process
- Constant pressure Cooling Process
- Isentropic Expansion Process
- Constant Pressure Expansion Process

Isentropic Compression Process

Here, refrigerant R-290 (LPG) from the refrigerator is drawn into the compressor cylinder where it is compressed isentropically in the compressor by the curve p-v (pressure-volume) and T-s (temperature-entropy). During the compression stroke, both pressure and temperature increases and specific volume of refrigerant at delivery from compressor reduces. We know that during the isentropic compression process no heat is absorbed or rejected by refrigerant.

Constant Pressure Cooling Process

The warm LPG from compressor is then passed to the cooler (condenser) that is cooled at constant pressure reducing the temperature shown in p-v and T-s diagram. The specific volume also reduces. Here refrigerant is condensed without changing pressure.

Isentropic Expansion Process

Now refrigerant from condenser is drawn into the expander cylinder where it is expanded isentropically from compressed pressure to refrigerator pressure that is equal to atmospheric pressure. The temperature of LPG then falls. The expansion process is shown by p-v and T-s diagrams. The specific volume of refrigerant at entry to the refrigerator increases. We know that during the isentropic expansion process no heat is absorbed or rejected by refrigerant.

Constant Pressure Expansion Process

Cold LPG from expander is then passed to refrigerator where it is expanded at constant pressure. Temperature of LPG refrigerant increases. This process can be shown by graph in p-v and T-s diagram. Due to heat from refrigerator, the specific volume of LPG changes.

CHAPTER – 03

DESIGN AND CONSTRUCTION

3.1 TEST RIG DESIGN

The schematic diagram of our test rig is shown in figure. The main frame and basement is made of steel. The length, breadth and height of the frame are 0.76m, 0.61m and 1.37m respectively. We have used a board at the back of the frame which is made of plywood. We have mounted four temperature gauges, two pressure gauges, flow meter and dryer on the board. The cooling chamber is placed on angle bar at a height of 0.46m above the basement. The evaporator is welded inside the cooling chamber. We have used 0.05m insulation for the cooling chamber. The dimension of the cooling chamber is 0.63m×0.51m×0.45m.

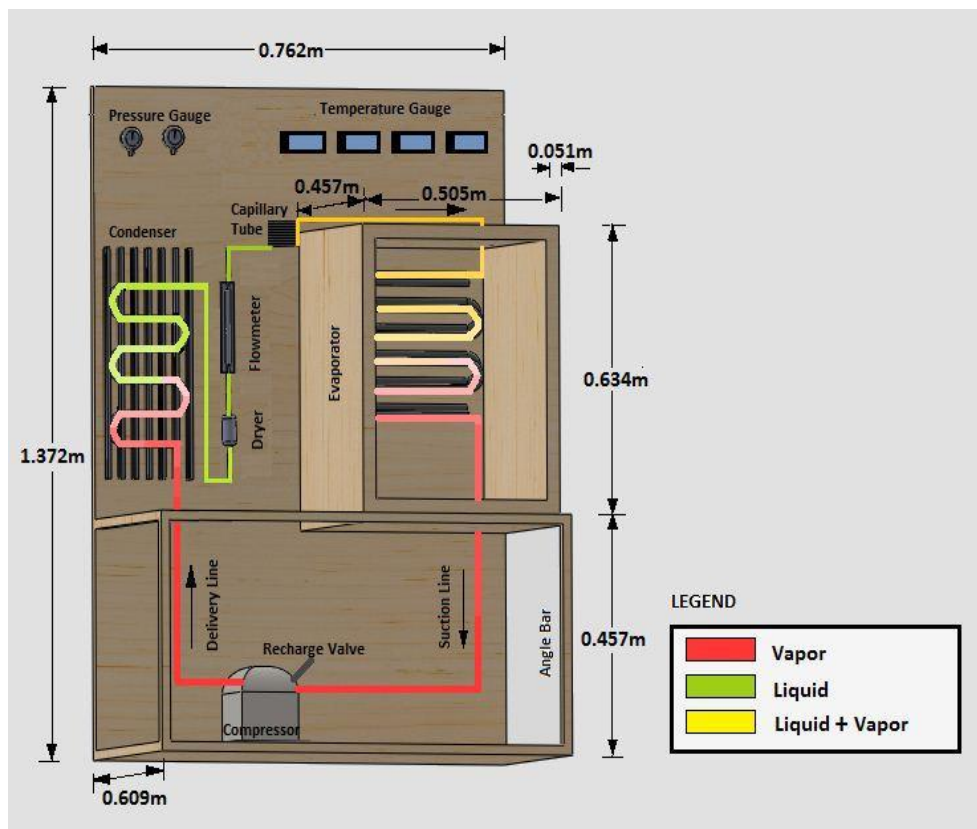


Fig 3.1 : Schematic diagram of vapor compression refrigeration test rig

3.1.1 EVAPORATOR DESIGN

Evaporator cooling chamber of the test rig is mainly a rectangular box made of steel sheet which is shown in Figure. We have attached a door at the front side of the compartment which is made of aluminum frame and glass. Outer dimension of the

cooling chamber is 0.63m×0.51m×0.45m. The insulation thickness of the cooling chamber is 0.05m. The evaporator coil is welded inside the cooling chamber. The diameter of the coil is 0.003m with a total length of 10.795m. Loop to loop distance of the evaporator coil is 0.09m.

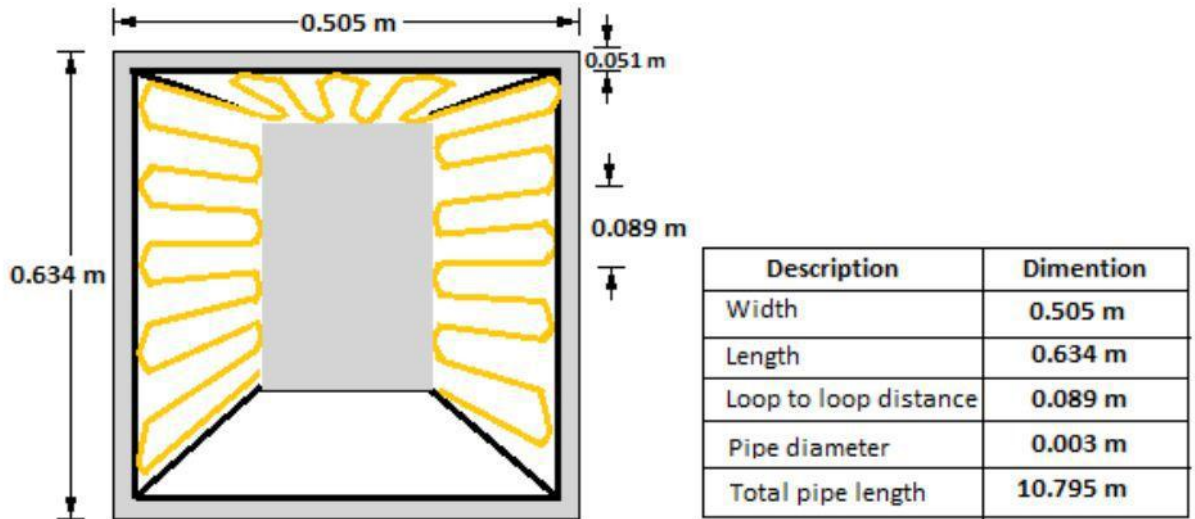


Fig 3.2 : Design of evaporator of vapor compression refrigeration system

3.1.2 CONDENSER DESIGN

The design of condenser of vapor compression refrigeration system is shown in Fig 3.4. We have used naturally air cooled condenser in the test rig. The length and width of the condenser is 0.76m and 0.46m respectively. There are 08 loops in this condenser. Loop to loop distance of the condenser is 0.10m. Total length of the pipe is 8.63m.

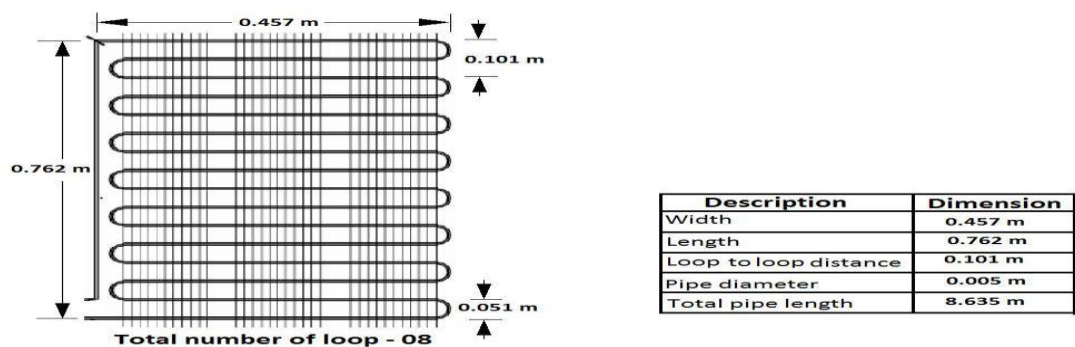


Fig 3.3 : Design of condenser of vapor compression refrigeration system

3.3 DIFFERENT COMPONENTS

Different components of the vapor compression test rig are given below:

3.3.1 COMPRESSOR

In a refrigeration cycle, the compressor has two main functions within the refrigeration cycle. The function of the compressor is to pump the refrigerant vapor from the evaporator so that the desired temperature and pressure of the system. The second function is to increase the pressure of the refrigerant vapor through the process of compression, and simultaneously increase the temperature of refrigerant vapor. The compressor is shown in Figure



Fig 3.4 : Compressor

3.3.2 CONDENSER

The heat extracted from the substance to be cooled by the refrigerant in the evaporator is rejected to the atmosphere through the condenser. The refrigerant is forced through the condenser in order to remove as much heat as possible with the tubes arranged to provide maximize surface area. In the condenser, the temperature of the superheated vapor has to be brought down to its saturation temperature before condensed into a liquid. The condenser is shown in Figure



Fig 3.5 : Condenser.

3.3.3 CAPILLARY TUBE

The capillary tube is the simplest type of refrigerant flow control device and may be used in place of an expansion valve. The capillary tube is small-diameter tubes through which the refrigerant flows into the evaporator. It reduce the condensing pressure to the evaporating pressure in a copper tube of small internal diameter, maintaining a constant evaporating pressure independently of the refrigeration load range. The capillary tube is shown in Figure



Fig 3.6 : Capillary tube.

3.3.4 EVAPORATOR

Evaporator can be considered the point of heat capture in refrigeration system and provides the cooling effect required for any particular application. During the same process, the high pressure liquid is lowered to what is called the evaporator pressure, or suction pressure. During evaporation, the refrigerant remains the same temperature throughout the coil until all droplets of liquid are vaporized, or totally saturated. The evaporator is shown in Figure.



Fig 3.7 : Evaporator

3.3.5 PRESSURE GAUGE

There are two types of pressure gauge used in refrigeration system. One is high pressure gauge and other is low pressure gauge. High pressure gauge is connected to the delivery line and low pressure gauge is connected to the suction line. The pressure gauge is shown in Figure



Fig 3.8 : Pressure gauge.

3.3.6 TEMPERATURE METER

Temperature meter gives the temperature reading of different point of the refrigeration system. Digital temperature meter gives more accurate value than analog temperature meter. It shows reading in degree Celsius. The temperature meter is shown in Figure.



Fig 3.9 : Digital temperature meter

3.3.7 AMMETER

An ammeter is a measuring instrument used to measure the electric current in a circuit. Electric currents are measured in amperes (A), hence the name. It is connected in series in the circuit. Improved instruments were designed which could be mounted in any position and allowed accurate measurements in electric power systems. There are generally two types of ammeter called analog and digital ammeter. The ammeter is shown in Figure.

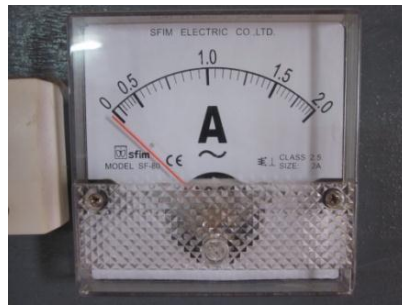


Fig 3.10 : Ammeter

3.3.8 VOLTMETER

A voltmeter is a measuring instrument used to measure the voltage in a circuit. Electric voltages are measured in volts (V). It is connected parallel in the circuit. Improved instruments were designed which could be mounted in any position and allowed accurate measurements in electric power systems. The voltmeter is shown in Figure.

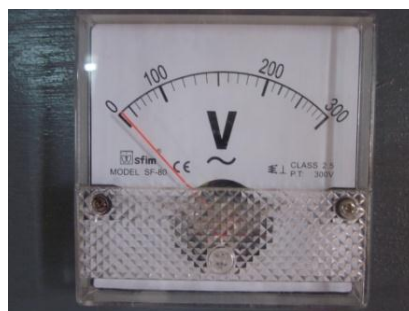


Fig 3.11 : Voltmeter

3.3.9 DRYER

Filter-dryer is a device used to trap the moisture, small metal chips, and dirt, in the refrigerant from entering the system which can restrict the free flow of refrigerant into the expansion valve. Every time a repair work is carried out in your refrigerator or freezer, it must be replaced with a new one. The dryer is shown in Figure.



Fig 3.12 : Dryer

3.3.10 COOLING FAN

The cooling fan motor for compressor is used to force the air into its surface in order to dissipate the heat into the atmosphere and to bring down its temperature rapidly; and it sucks in the cool air from the surroundings and forces its way through the compressor surface where the heat is being removed. The cooling fan is shown in Figure.



Fig 3.13 : Cooling fan.

3.3.11 THERMOSTATE

Refrigerator thermostats are usually located inside the refrigerator and have a knob that allows users to adjust them. Once a user sets the desired temperature, the thermostat maintains that temperature by sensing internal fluctuations and prompting the compressor to turn on or off in response to those changes. When the refrigerator is appropriately adjusted to the designated temperature, the thermostat stops the flow of electricity to the compressor, stopping it from cooling the appliance. The thermostat is shown in Figure



Fig 3.14 : Thermostat

3.3.12 LIQUID LINE INDICATOR

Moisture liquid indicators ensure a fast and safe inspection of the conditions of the refrigerant fluid in the circuit concerning regular flow and moisture. Liquid indicators also ensure inspection of the regular return of oil to the compressor crankcase. The liquid indicators consist of a sensitive element as a ring, which changes color passing from blue to pink according to the percentage of moisture in the system. The liquid line indicator is shown in Figure



Fig 3.15: Liquid line ind

3.3.13 PICTURE OF THE PROJECT SETUP

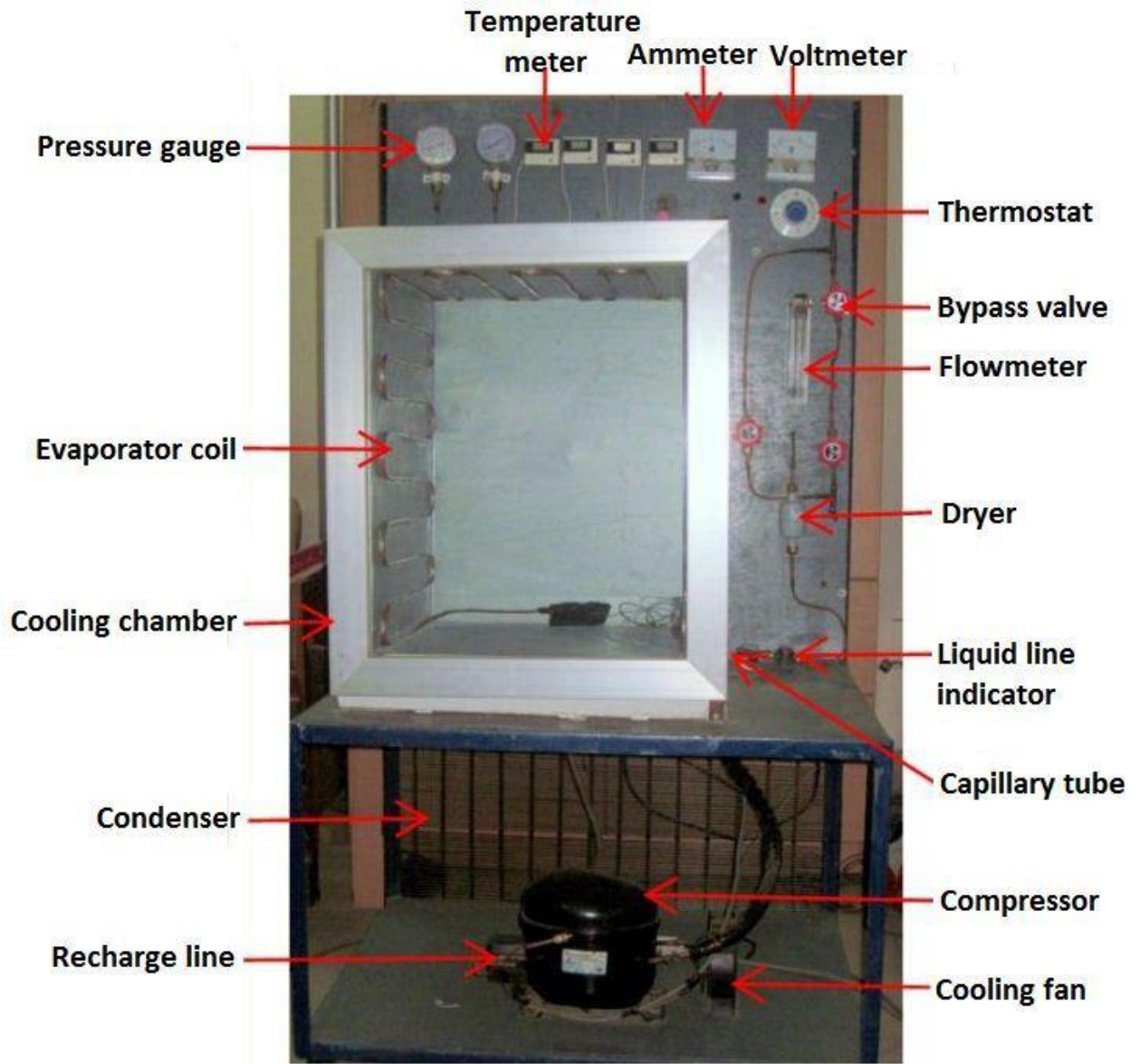


Fig 3.16: Picture of the Project Setup

3.4 CONSTRUCTION AND SET UP

We have ensured proper insulation of the evaporator compartment with polystyrene and foam to prevent heat loss from the compartment. We attached a cooling fan near the compressor to avoid the overheating of the compressor. It increases the pumping effort of the compressor and gives better performance to the overall system. The condenser is placed in the system such a way that it can dissipate heat to the environment efficiently.

We have used digital temperature meter to get more accurate temperature reading of different points of the system during the experiment. To calculate the total power consumed by the system we must know the voltage and current reading, so we have attached an ammeter and a voltmeter in the system. To measure the flow rate of the refrigerant directly a liquid flow meter is attached.

To construct the test rig at first the main steel frame is made and then we have colored it. Then we have made the cooling chamber and welded the evaporator coil inside it. We have also added a booster line to the cooling chamber. A door has been attached in front of the cooling chamber which is made of aluminum bar and glass. Then the cooling chamber is placed on the main steel frame. The condenser is placed at the back of the test rig.

We have collected all necessary equipment like pressure gauges, dryer, liquid line indicator, cooling fan etc. Then we have attached a ply wood board in the test rig. All the equipment are mounted on the board and then we have conned them sequentially. Then we have attached a light inside the cooling chamber. After that we have vacuumed the test rig to check any leakage. Finally we have recharged the required refrigerant in exact amount. The working sequence is shown in Figure.

3.4.1 DESIGN AND CONSTRUCTION

We have to build a design a vapor compression test rig which should be compact and represent the whole vapor compression refrigeration system. The system will be air cooled refrigeration system. That means the condenser is naturally air cooled. So the test rig is designed in such a way that the refrigerant and easily dissipate heat to the environment while passing through the condenser. For easy movement from one place to another, we have attached four wheels at the bottom of the frame.

3.4.2 DESIGN REQUIREMENTS

Some requirements should be fulfilled to design a vapor compression rig

- Our construction should be simple and the components that are used in this experiment should be well equipped.
- We have built this vapor compression system for experimental purposes. So, we should be aware of cost of this project. Since it is not a commercial vapor

compression refrigeration system so we should keep our costs as low as possible.

- The refrigeration system we have built is environment friendly. We are going to use a very short amount of refrigerant (both Freon and LPG) to operate the system.
- Our system should be quite enough to operate in a room. This should be user friendly to operate and very easy to maintain.
- Our system should be safe and have to ensure that there will be no harm due to this refrigeration system. We have to keep in mind that the refrigerant we are using (LPG) is flammable and it is very much risky to charge the gas. So we should be careful and ensure our safety purposes.
- It should be compact and light in weight. It is important for easy movement and further installment to the system.
- Our system should ensure proper insulation of the evaporator.

Considering above requirements, we have built our test rig that is described

3.5 EXPERIMENTAL PROCEDURE

The experimental procedure is described below:

- When the experimental setup was ready and inspected visually we have to vacuum the whole system for 30 minutes. This is done to charge refrigerant properly.
- By observing the pressure gauge reading we have ensured that the system is totally leak proof. If there is any leakage, we have used soap water at the joint of pipelines to find the leakage.
- Then we have charged right amount refrigerant in the system. In our experiment, we have charged about 200 g of R-134a and 100 g of LPG.
- The tests are conducted under room condition. We have taken the room temperature from digital temperature sensor. We have got our room temperature 26°C.
- Then we turned the power supply on and ensure that all the electrical instruments are working properly. Our main concern will be the running of compressor, ammeter and voltmeter.
- After that we have waited until the system reached the steady state condition.

- We have to check whether the flow meter is working or not. We should check the liquid refrigerant passing through the flow meter and the indicator is providing us the reading.
- Then we will have to check whether the cooling chamber is cooling our load that means whether the evaporator is working or not. If the cooling chamber is taking more time than required then we will have to consider it as a problem and check the whole system properly.
- Then in steady state condition we have taken the reading from pressure gauge and temperature meter carefully. We will have to take readings from temperature sensor and pressure gauge after 30 min and take 5 min break between next reading. Then the procedure will go on.
- We have to take readings from pressure gauge and temperature sensor for half an hour. The time and duration between the readings should be similar for both refrigerants for the ease of calculation.

CHAPTER - 04

DATA COLLECTION

We have taken data for two different refrigerants in two different days. It is impossible to take all data in a single day as it takes time to vacuum and recharge the refrigerants in the system. Moreover, we have used M-seal to prevent leakage that takes a whole day to get dry. To calculate COP, we will have to know the value of enthalpy of the system, especially of compressor, evaporator and condenser. We have used these pressure and temperature for calculating COP of both refrigerants. After taking the value of pressure and temperature, we have to take the value of enthalpy using different charts and pressure enthalpy diagram or p-h diagram. Finally, theoretical COP is calculated by putting these values in mathematical formula.

DATA TABLE 4.1 : For refrigerant R-290

No.	Calc. symbol	Description	Unit	Time(min)						
				30	35	40	45	50	55	60
1	P_1	Evaporator Pressure	Bar	1.1378	1.124	1.1171	1.103	1.110	1.083	1.076
2	T_{evp}	Evaporator Temperature	°C	-1.7	-1.9	-1.9	-2.0	-1.9	-2.0	-2.2
3	h_1	Evaporator Enthalpy	Kj/kg	560	558	558	555	558	555	552
4	P_2	Compressor Pressure	Bar	6.8565	6.8565	6.9254	6.891	6.994	7.029	7.0425
5	T_{com}	Compressor Temperature	°C	36.8	36.8	36.9	37.1	37.1	36.9	37.1
6	h_2	Compressor Enthalpy	Kj/kg	610	610	615	617	617	615	617
7	P_3	Condenser Pressure	Bar	6.8565	6.8565	6.9254	6.891	6.994	7.029	7.0425
8	T_{cond}	Condenser Temperature	°C	26.4	26.3	26.0	25.9	26.3	26.2	26.4
9	h_3	Condenser Enthalpy	Kj/kg	270	267	264	262	267	265	270
10	$\text{COP}_{(\text{Theo})}$	Coefficient of Performance		5.8	5.6	5.16	4.73	4.93	4.83	4.34
11	$\text{COP}_{(\text{Act})}$	Coefficient of Performance		4.433	4.49	4.53	4.52	4.49	4.47	4.35

Now we are going to determine the data of R-134a refrigerants

DATA TABLE 4.2 : For refrigerant R-134a

No.	Calc. symbol	Description	Unit	Time(min)						
				30	35	40	45	50	55	60
1	P_1	Evaporator Pressure	Bar	1.00689	1.00689	1.006201	1.005512	1.000689	1.000689	0.99311
2	T_{evp}	Evaporator Temperature	°C	-16.9	-17.7	-16.1	-18.4	-17.1	-19.1	-18.7
3	h_1	Evaporator Enthalpy	Kj/kg	294	288	290	289	289.5	287	288.8
4	P_2	Compressor Pressure	Bar	9.957	9.99145	10.0948	10.3015	10.1637	9.8881	9.269
5	T_{com}	Compressor Temperature	°C	41.9	42.8	43.9	44.1	44.4	44.8	44.8
6	h_2	Compressor Enthalpy	Kj/kg	324	322	324	324.5	323	325	325
7	P_3	Condenser Pressure	Bar	9.957	9.99145	10.0948	10.3015	10.1637	9.8881	9.269
8	T_{cond}	Condenser Temperature	°C	30.6	29.8	29.9	29.3	28.8	28.6	28.3
9	h_4	Condenser Enthalpy	Kj/kg	143	144	142	141	140	139.7	140
10	$COP_{(Theo)}$	Coefficient of Performance		5.03	4.24	4.353	4.169	4.463	3.876	4.105
11	$COP_{(Act)}$	Coefficient of Performance		3.9957	3.81	3.9164	3.9164	3.9561	3.898	3.9322

CHAPTER - 05

RESULTS AND DISCUSSIONS

We have done the experiment by recording evaporator, compressor and condenser temperature for both LPG(R-290) and Freon (R-134a). Both theoretical and actual COP of these two refrigerants are calculated with the help of various charts and p-h diagrams. After collecting these data, we have plotted these data or value in different graphs. Examining these graph, we can have some decisions about using LPG (R-290) as refrigerant as an alternative of former popular refrigerant Freon (R-134a).

5.1 COOLING EFFECT OF LPG

Our main concern is to examine the properties of LPG and use it as a refrigerant. According to the value we have got in our project, we have come to a decision that LPG can be used as a refrigerant. LPG has run through our system perfectly. It can decrease the temperature of the substance that we have used to be cooled. It has decreased cooling chamber temperature to -2.2°C which is enough to cool the load we have used.

Now, we plot the chamber temperature with respect to time to see the cooling capacity of LPG as refrigerant

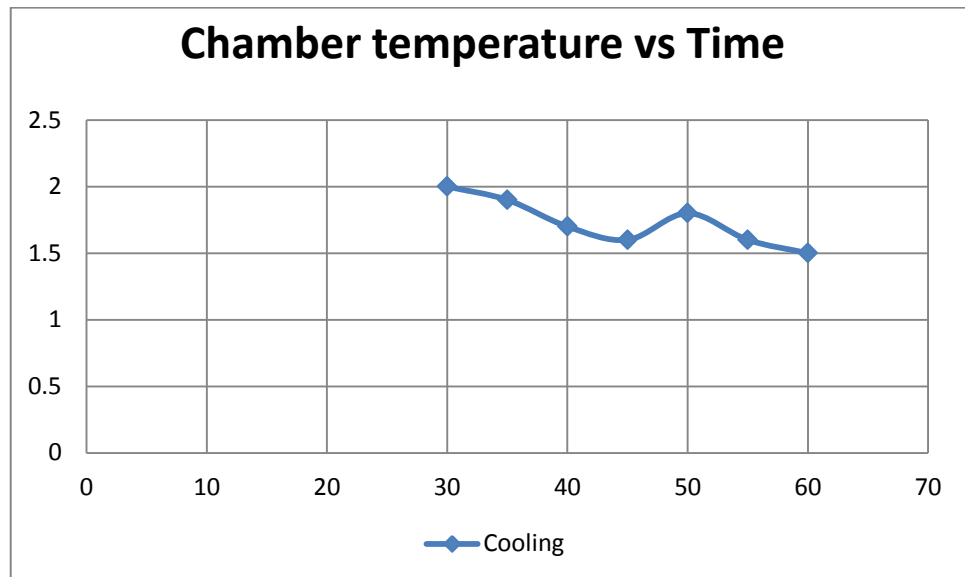


Fig 5.1 : Cooling temperature vs time

From the graph, we can see that, the temperature of cooling chamber is gradually decreasing. This indicates that, LPG is working perfectly as a refrigerant. We can also

notice a deflection between the times 40 minutes to 50 minutes. This is because of the difference of voltage and electricity that is supplied to the system.

5.2 COEFFICIENT OF PERFORMANCE WITH TIME

Now, we are going ahead to our main perspective. We can put the values of theoretical and actual coefficient of performance of LPG. After putting that, we can get the graph,

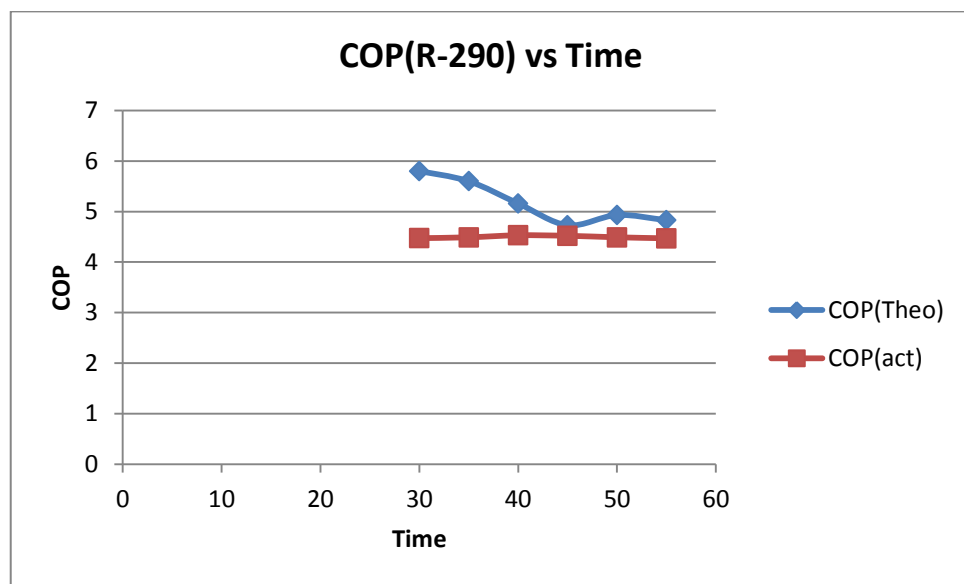


Fig 5.2 : COP(R-290) vs Time

Here, we have plotted time in X-axis and both theoretical and actual COP in Y-axis. We can see that the line of theoretical COP and actual COP comes to a similar point after a certain time. The major difference between these two lines comes at the first few minutes after the system has started. Gradually the lines come closer to each other and that indicates a good characteristic of LPG as a refrigerant.

Now, we can have a look on the graph of R-134a. We have similarly plotted the values of COP of R-134a. There we can see the similar case. Both of the lines of theoretical and actual COP come closer and after some time they have met one other at a point. We know, Freon (R-134a) is the refrigerant that is usually used as refrigerant in refrigeration system now a days. Here is the graph of theoretical and actual COP of R-134a

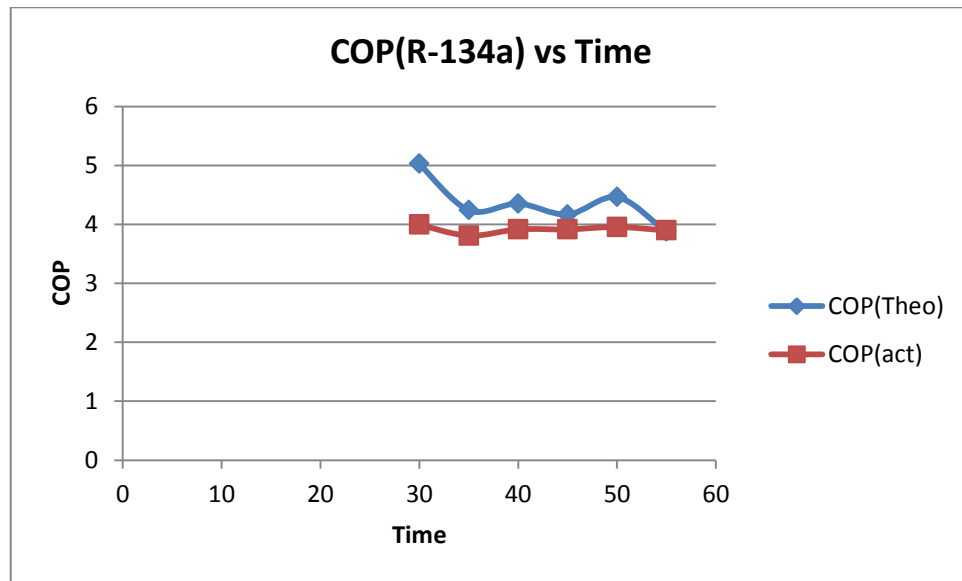


Fig 5.3 : COP(R-134a) vs time

This means the characteristics of LPG are very close or we can say similar to the refrigerant that we are using now a days. So we can easily replace LPG with the usual R-134a refrigerant. For more conformation, we can go to our next steps.

5.3 COP vs DIFFERENCE OF EVAPORATOR AND CONDENSER TEMPERATURE

Now we are putting the values coefficient of performance with respect to temperature difference of evaporator and compressor. Determining of the values of difference of coefficient of performance is important because it helps us to know how efficiently the refrigeration system is removing heat from the load. With these values we can we can measure the value of "Q" the useful heat supplied or removed by the refrigeration system. Coefficient of performance is proportional to the difference of enthalpy of evaporator and condenser. Enthalpy of evaporator and condenser is determined by the temperature of evaporator and condenser. So, we are determining the coefficient of performance graph with respect to the difference of evaporator and condenser.

We are now putting the values of difference of temperature of evaporator and condenser in X-axis and the values of coefficient of performance in Y-axis. We are determining the graph for both LPG(R-290) and Freon(R-134a). Though the values of difference of temperature will not be the same, but we can easily understand the comparison by the shape of the line from the graph.

Here is graph of difference of temperature of evaporator and condenser and coefficient of performance of LPG(R-290)

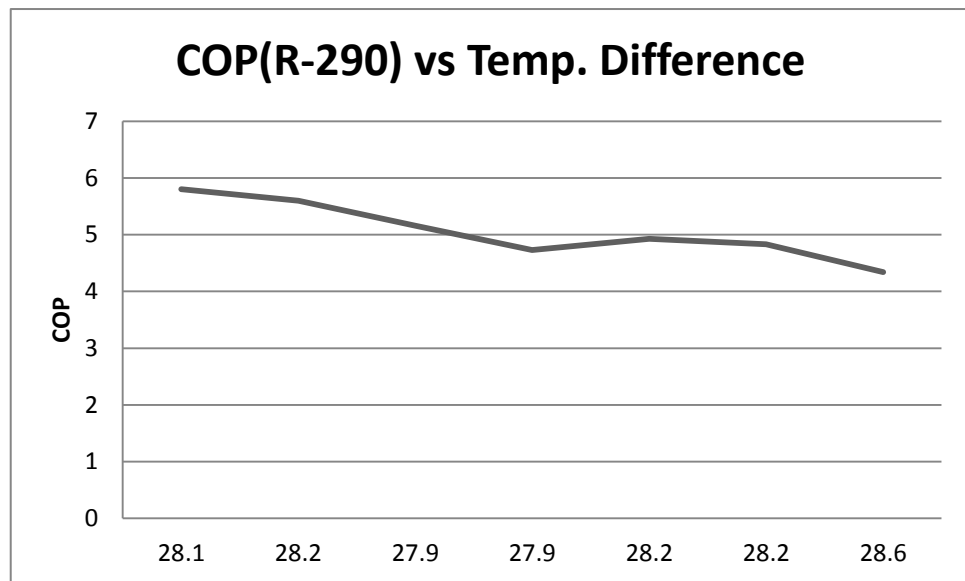


Fig 5.4 : COP(R-290) vs Temperature Difference

And here is the graph of difference of temperature and condenser with respect to coefficient of performance of Freon(R-134a)

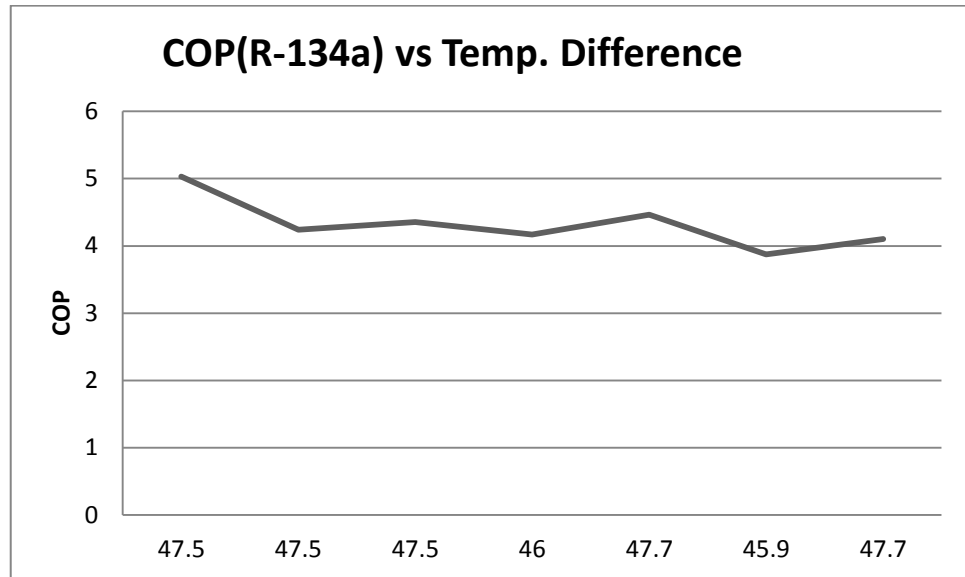


Fig 5.5 : Difference of temperature of evaporator and condenser vs COP(R-134a)

From these two graphs, we can say they are closely similar. Both of the graph lines are going downwards with a slight deflection. From these graphs, we can say that, the characteristics of LPG as a refrigerant is closely similar to usual refrigerant Freon in these aspects. So we can easily replace Freon with LPG to use as refrigerant.

5.4 DIFFERENCE OF THEORETICAL COP WITH TIME

Here now we are going to put the values of theoretical coefficient of performance of both LPG(R-290) and Freon (R134a). This is important to determine because through these values and graph we will be able to know which one gives the better coefficient of performance.

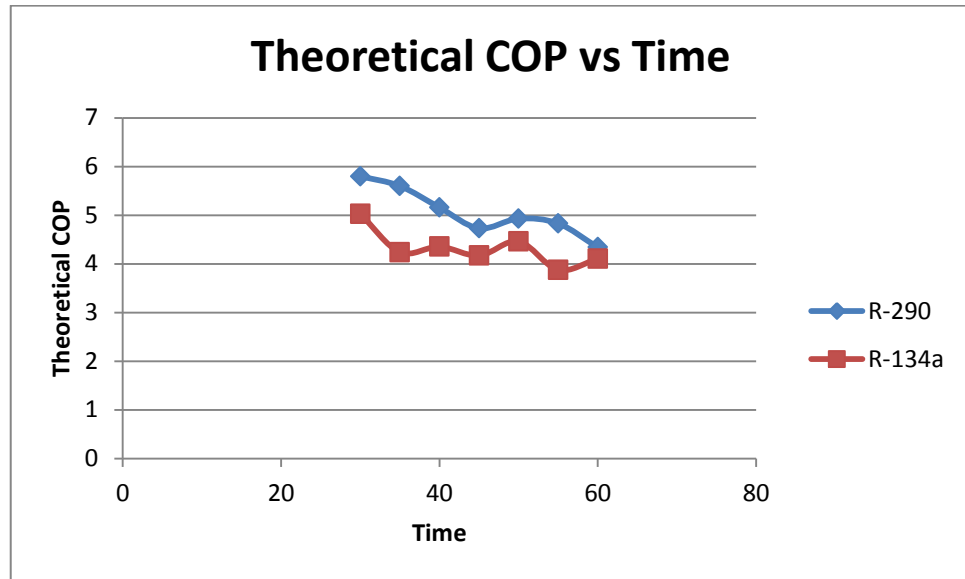


Fig 5.6 : Theoretical COP vs Time

We have put time in X-axis and the values of theoretical coefficient of performance of both refrigerants in Y-axis. After putting the values, we have got that, LPG(R-290) is giving even better coefficient of performance than Freon (R-134a). Both of the graph lines are going downwards with deflections. These are because of temperature and power supply. At some points, the lines met each other. But, the values of R-290 are greater than R-134a. This means LPG is comparatively more efficient than Freon (R-134a). So, we can replace LPG with the Freon (134a).

5.5 DIFFERENCE OF ACTUAL COP WITH TIME

Now, we are putting actual values of coefficient of performance for both LPG (R-290) and Freon (R134a) with respect to time. Actual coefficient of performance is actually the ratio of the heat that is supplied or removed from the system to the work required by the system.

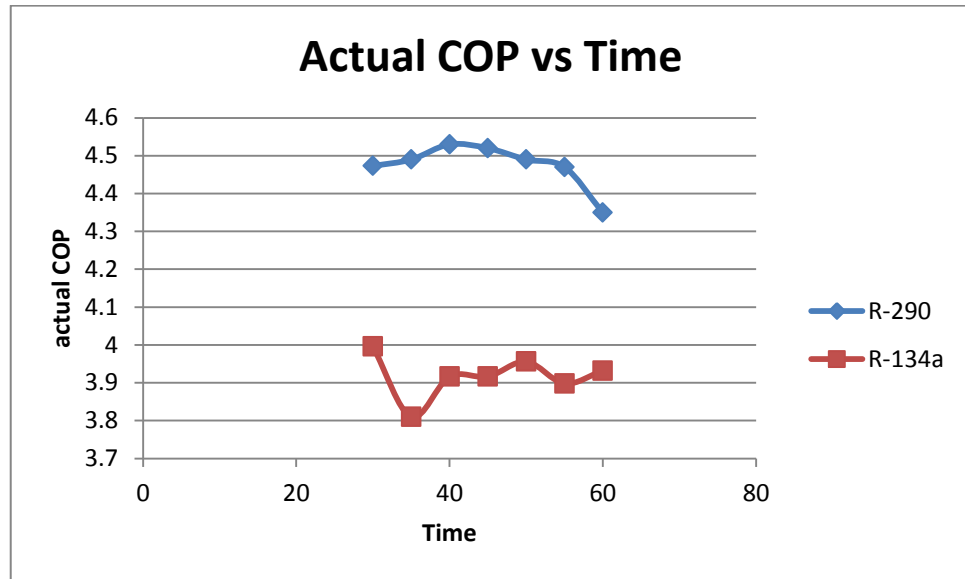


Fig 5.7 : Actual COP vs Time

We have put time in X-axis and the values of actual coefficient of performance of both refrigerants in Y-axis. After putting the values, we have got that, LPG(R-290) is giving better coefficient of performance than Freon (R-134a).The lines of LPG (R-290) is going downwards with deflections. On the other hand, the line of actual COP R-134a is going irregularly with a deflection at first. These are because of room temperature and power supply. But, the values of R-290 are greater than R-134a. This means LPG is comparatively more efficient than Freon (R-134a). So, we can replace LPG with the Freon (134a).

From all these graphs and data stated above, we can come to a decision that in spite of being having flammability, it will be a wise decision to use Liquefied Petroleum Gas (LPG) as a useful refrigerant. Besides of cooling effect, it gives better coefficient of performance and more importantly it is not harmful to our environment as it is non-toxic with zero ODP (Ozone Depletion Potential) and very low GWP (Global Warming Potential). So we can use LPG in spite of Freon (R-134a) as a refrigerant and create a new era of environment friendly and efficient refrigeration system.

CHAPTER – 06

CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSION

This experiment was carried out in order to observe and find COP of simple vapor compression refrigeration cycle with Liquefied petroleum gas and R-134a as refrigerants. Temperature and pressure measurements were taken with special care with gages critical points of the system. With the properties of both refrigerants at all the stages of the refrigeration cycle known, the COP of refrigeration system was calculated. The conclusions that could be taken from this experiment are as follows:

- With the increase of compressor work COP decrease while change of refrigeration effect is negligible.
- Theoretical COP decrease with the increase of condenser temperature.
- As the condensing temperature is lowered, the compressor power input decreases.
- As evaporating temperature increases, so too does the compressor power input; however, this power increase is less than the capacity increase.
- As the temperature lift is reduced, the refrigeration compressor capacity increases.
- Both high pressure and low pressure reading of pressure gauge changes due to ambient temperature during the experiment.

6.2 RECOMMENDATION

- May contain sensor based doors for auto open or close like lift door.
- Some chamber LED lights can be added.
- Electronic touch controls may be incorporated in the system.
- More efficient chilling technology.
- Out looking can be developed by using proper tools and architectural designs.
- Models with three doors and a freezer on the bottom may be introduced.
- Thinner, more effective insulation.
- Radio devices that make them Internet and smart-grid ready to take advantage of future improvements in convenience and efficiency.
- Identifying cooling load automatically and serve properly based on that.
- Advanced defrost sensor may be added which minimize frost build-up to maintain evaporator coil efficiency.

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APPENDIX - A

CALCULATION FOR R-134a(here, $m = 12.1 \text{ m}^3 / \text{s}$)

For Obs.1,

Evaporator Pressure, $P_{\text{EVP}} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 0.1 \times .0689 \text{ bar}$$

$$= 1.00689 \text{ bar}$$

$$T_{\text{EVP}} = -16.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 294 \text{ kJ/kg}$

Compressor Pressure, $P_{\text{com}} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 130 \times .0689 \text{ bar}$$

$$= 9.957 \text{ bar}$$

$$T_{\text{com}} = 41.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 324 \text{ kJ/kg}$

Condenser Pressure, $P_{\text{con}} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 130 \times .0689 \text{ bar}$$

$$= 9.957 \text{ bar}$$

$$T_{\text{con}} = 30.6 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 143 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{294 \text{ kJ/kg} - 143 \text{ kJ/kg}}{324 \text{ kJ/kg} - 294 \text{ kJ/kg}} \\ &= 5.03 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{12.1 \times (294 \text{ kJ/kg} - 143 \text{ kJ/kg})}{\sqrt{3} \times 220\text{v} \times 1.5 \text{ A} \times 0.8} \\ &= 3.9957 \end{aligned}$$

For Obs.2,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 0.1 \times .0689 \text{ bar}$$

$$= 1.00689 \text{ bar}$$

$$T_{EVP} = -17.7 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 288 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 130.5 \times .0689 \text{ bar}$$

$$= 9.99145 \text{ bar}$$

$$T_{com} = 42.8 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 322 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 130.5 \times .0689 \text{ bar}$$

$$= 9.99145 \text{ bar}$$

$$T_{con} = 29.8 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 144 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{288 \text{ kJ/kg} - 144 \text{ kJ/kg}}{322 \text{ kJ/kg} - 288 \text{ kJ/kg}} \\ &= 4.24 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{12.1 \times (288 \text{ kJ/kg} - 144 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ V} \times 1.5 \text{ A} \times 0.8} \\ &= 3.81 \end{aligned}$$

For Obs.3,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 0.09 \times .0689 \text{ bar}$$

$$= 1.006201 \text{ bar}$$

$$T_{EVP} = -16.1 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 290 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 132 \times .0689 \text{ bar}$$

$$= 10.0948 \text{ bar}$$

$$T_{com} = 43.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 324 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 132 \times .0689 \text{ bar}$$

$$= 10.0948 \text{ bar}$$

$$T_{con} = 29.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 142 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{290 \text{ kJ/kg} - 142 \text{ kJ/kg}}{324 \text{ kJ/kg} - 290 \text{ kJ/kg}} \\ &= 4.353 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{12.1 \times (290 \text{ kJ/kg} - 142 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ V} \times 1.5 \text{ A} \times 0.8} \\ &= 3.9164 \end{aligned}$$

For Obs.4,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 0.08 \times .0689 \text{ bar}$$

$$= 1.005512 \text{ bar}$$

$$T_{EVP} = -18.4 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 289 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 135 \times .0689 \text{ bar}$$

$$= 10.3015 \text{ bar}$$

$$T_{com} = 44.1 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 324.5 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 135 \times .0689 \text{ bar}$$

$$= 10.3015 \text{ bar}$$

$$T_{con} = 29.3 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 141 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{289 \text{ kJ/kg} - 141 \text{ kJ/kg}}{324.5 \text{ kJ/kg} - 289 \text{ kJ/kg}} \\ &= 4.169 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{12.1 \times (289 \text{ kJ/kg} - 141 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ v} \times 1.5 \text{ A} \times 0.8} \\ &= 3.9164 \end{aligned}$$

For Obs.5,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 0.01 \times .0689 \text{ bar}$$

$$= 1.000689 \text{ bar}$$

$$T_{EVP} = -17.1 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 289.5 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 133 \times .0689 \text{ bar}$$

$$= 10.1637 \text{ bar}$$

$$T_{com} = 44.4 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 323 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 133 \times .0689 \text{ bar}$$

$$= 10.1637 \text{ bar}$$

$$T_{con} = 28.8 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 140 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{289.5 \text{ kJ/kg} - 140 \text{ kJ/kg}}{323 \text{ kJ/kg} - 289.5 \text{ kJ/kg}} \\ &= 4.463 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{12.1 \times (289.5 \text{ kJ/kg} - 140 \text{ kJ/kg})}{\sqrt{3} \times 220\text{v} \times 1.5 \text{ A} \times 0.8} \\ &= 3.9561 \end{aligned}$$

For Obs.6,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 0.01 \times .0689 \text{ bar}$$

$$= 1.000689 \text{ bar}$$

$$T_{EVP} = -19.1 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 287 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 129 \times .0689 \text{ bar}$$

$$= 9.8881 \text{ bar}$$

$$T_{com} = 44.8 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 325 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 129 \times .0689 \text{ bar}$$

$$= 9.8881 \text{ bar}$$

$$T_{con} = 28.6 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 139.5 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{287 \text{ kJ/kg} - 139.5 \text{ kJ/kg}}{325 \text{ kJ/kg} - 287 \text{ kJ/kg}} \\ &= 3.876 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{12.1 \times (287 \text{ kJ/kg} - 139.5 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ v} \times 1.5 \text{ A} \times 0.8} \\ &= 3.898 \end{aligned}$$

For Obs.7,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + (-0.1) \times .0689 \text{ bar}$$

$$= 0.99311 \text{ bar}$$

$$T_{EVP} = -18.7 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 288.8 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 120 \times .0689 \text{ bar}$$

$$= 9.269 \text{ bar}$$

$$T_{com} = 44.8 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 325 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 120 \times .0689 \text{ bar}$$

$$= 9.269 \text{ bar}$$

$$T_{con} = 28.3 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 140.2 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{288 \text{ kJ/kg} - 140.2 \text{ kJ/kg}}{325 \text{ kJ/kg} - 288 \text{ kJ/kg}} \\ &= 4.105 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{12.1 \times (288 \text{ kJ/kg} - 140.2 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ v} \times 1.5 \text{ A} \times 0.8} \\ &= 3.9322 \end{aligned}$$

CALCULATION FOR R-290(here, $m = 7.0527 \text{ m}^3 / \text{s}$)

For Obs.1,

Evaporator Pressure, $P_{\text{EVP}} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 2 \times .0689 \text{ bar}$$

$$= 1.1378 \text{ bar}$$

$$T_{\text{EVP}} = -1.7 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 560 \text{ kJ/kg}$

Compressor Pressure, $P_{\text{com}} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 85 \times .0689 \text{ bar}$$

$$= 6.8565 \text{ bar}$$

$$T_{\text{com}} = 36.8 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 610 \text{ kJ/kg}$

Condenser Pressure, $P_{\text{con}} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 85 \times .0689 \text{ bar}$$

$$= 6.8565 \text{ bar}$$

$$T_{\text{con}} = 26.4 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 270 \text{ kJ/kg}$

Now,

$$\text{Theoretical, COP} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{560 \text{ kJ/kg} - 270 \text{ kJ/kg}}{610 \text{ kJ/kg} - 560 \text{ kJ/kg}}$$

$$= 5.8$$

$$\text{Actual, COP} = \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{7.0527 \times (560 \text{ kJ/kg} - 270 \text{ kJ/kg})}{\sqrt{3} \times 220\text{v} \times 1.5 \text{ A} \times 0.8}$$

$$= 4.473$$

For Obs.2,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 1.8 \times .0689 \text{ bar}$$

$$= 1.12402 \text{ bar}$$

$$T_{EVP} = -1.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 558 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 85 \times .0689 \text{ bar}$$

$$= 6.8565 \text{ bar}$$

$$T_{com} = 36.8 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 610 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 85 \times .0689 \text{ bar}$$

$$= 6.8565 \text{ bar}$$

$$T_{con} = 26.3 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 267 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{560 \text{ kJ/kg} - 267 \text{ kJ/kg}}{610 \text{ kJ/kg} - 560 \text{ kJ/kg}} \\ &= 5.6 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{7.0527 \times (560 \text{ kJ/kg} - 267 \text{ kJ/kg})}{\sqrt{3} \times 220\text{v} \times 1.5 \text{ A} \times 0.8} \\ &= 4.49 \end{aligned}$$

For Obs.3,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 1.7 \times .0689 \text{ bar}$$

$$= 1.11713 \text{ bar}$$

$$T_{EVP} = -1.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 558 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 86 \times .0689 \text{ bar}$$

$$= 6.9254 \text{ bar}$$

$$T_{com} = 36.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 615 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 86 \times .0689 \text{ bar}$$

$$= 6.9254 \text{ bar}$$

$$T_{con} = 26 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 264 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{558 \text{ kJ/kg} - 264 \text{ kJ/kg}}{615 \text{ kJ/kg} - 558 \text{ kJ/kg}} \\ &= 5.16 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{7.0527 \times (558 \text{ kJ/kg} - 264 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ v} \times 1.5 \text{ A} \times 0.8} \\ &= 4.53 \end{aligned}$$

For Obs.4,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 1.5 \times .0689 \text{ bar}$$

$$= 1.10335 \text{ bar}$$

$$T_{EVP} = -2.0 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 555 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 85.5 \times .0689 \text{ bar}$$

$$= 6.891 \text{ bar}$$

$$T_{com} = 37.1 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 617 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 85.5 \times .0689 \text{ bar}$$

$$= 6.891 \text{ bar}$$

$$T_{con} = 25.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 262 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{555 \text{ kJ/kg} - 262 \text{ kJ/kg}}{617 \text{ kJ/kg} - 555 \text{ kJ/kg}} \\ &= 4.73 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{7.0527 \times (555 \text{ kJ/kg} - 262 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ v} \times 1.5 \text{ A} \times 0.8} \\ &= 4.52 \end{aligned}$$

For Obs.5,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 1.6 \times .0689 \text{ bar}$$

$$= 1.11024 \text{ bar}$$

$$T_{EVP} = -1.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 558 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 87 \times .0689 \text{ bar}$$

$$= 6.9943 \text{ bar}$$

$$T_{com} = 37.1 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 617 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 87 \times .0689 \text{ bar}$$

$$= 6.9943 \text{ bar}$$

$$T_{con} = 26.3 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 267 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{558 \text{ kJ/kg} - 267 \text{ kJ/kg}}{617 \text{ kJ/kg} - 558 \text{ kJ/kg}} \\ &= 4.93 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{7.0527 \times (558 \text{ kJ/kg} - 267 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ v} \times 1.5 \text{ A} \times 0.8} \\ &= 4.49 \end{aligned}$$

For Obs.6,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 1.2 \times .0689 \text{ bar}$$

$$= 1.08268 \text{ bar}$$

$$T_{EVP} = -2.0 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 555 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 87.5 \times .0689 \text{ bar}$$

$$= 7.02875 \text{ bar}$$

$$T_{com} = 36.9 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 615 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 87.5 \times .0689 \text{ bar}$$

$$= 7.02875 \text{ bar}$$

$$T_{con} = 26.2 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 265 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{555 \text{ kJ/kg} - 265 \text{ kJ/kg}}{615 \text{ kJ/kg} - 555 \text{ kJ/kg}} \\ &= 4.83 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{7.0527 \times (555 \text{ kJ/kg} - 265 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ V} \times 1.5 \text{ A} \times 0.8} \\ &= 4.83 \end{aligned}$$

For Obs.7,

Evaporator Pressure, $P_{EVP} = \text{Atmospheric pressure} + \text{Gauge pressure}$

$$= 1 \text{ bar} + 1.1 \times .0689 \text{ bar}$$

$$= 1.07579 \text{ bar}$$

$$T_{EVP} = -2.2 \text{ }^\circ\text{C}$$

Enthalpy, $h_1 = 552 \text{ kJ/kg}$

Compressor Pressure, $P_{com} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 87.7 \times .0689 \text{ bar}$$

$$= 7.04253 \text{ bar}$$

$$T_{com} = 37.1 \text{ }^\circ\text{C}$$

Enthalpy, $h_2 = 617 \text{ kJ/kg}$

Condenser Pressure, $P_{con} = \text{Atmospheric Pressure} + \text{Gauge Pressure}$

$$= 1 \text{ bar} + 87.7 \times .0689 \text{ bar}$$

$$= 7.04253 \text{ bar}$$

$$T_{con} = 26.4 \text{ }^\circ\text{C}$$

Enthalpy, $h_3 = 270 \text{ kJ/kg}$

Now,

$$\begin{aligned} \text{Theoretical, COP} &= \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{552 \text{ kJ/kg} - 270 \text{ kJ/kg}}{617 \text{ kJ/kg} - 552 \text{ kJ/kg}} \\ &= 4.34 \end{aligned}$$

$$\begin{aligned} \text{Actual, COP} &= \frac{m(h_1 - h_4)}{\sqrt{3} VI \cos\phi} = \frac{7.0527 \times (552 \text{ kJ/kg} - 270 \text{ kJ/kg})}{\sqrt{3} \times 220 \text{ v} \times 1.5 \text{ A} \times 0.8} \\ &= 4.35 \end{aligned}$$

APPENDIX - B

THERMODYNAMIC PRPERTIES OF REFRIGERANTS

Refrigerant	Chemical Formula	Molecular Mass (g/mol)	Boiling point (°C)	Ozone Depletion Potential	Global Warming Potential
R-290	C ₃ H ₈ F ₄	44.09	-44	0	3
R-134a	C ₂ H ₂ F ₄	102	-29.8	0	1430

APPENDIX - C

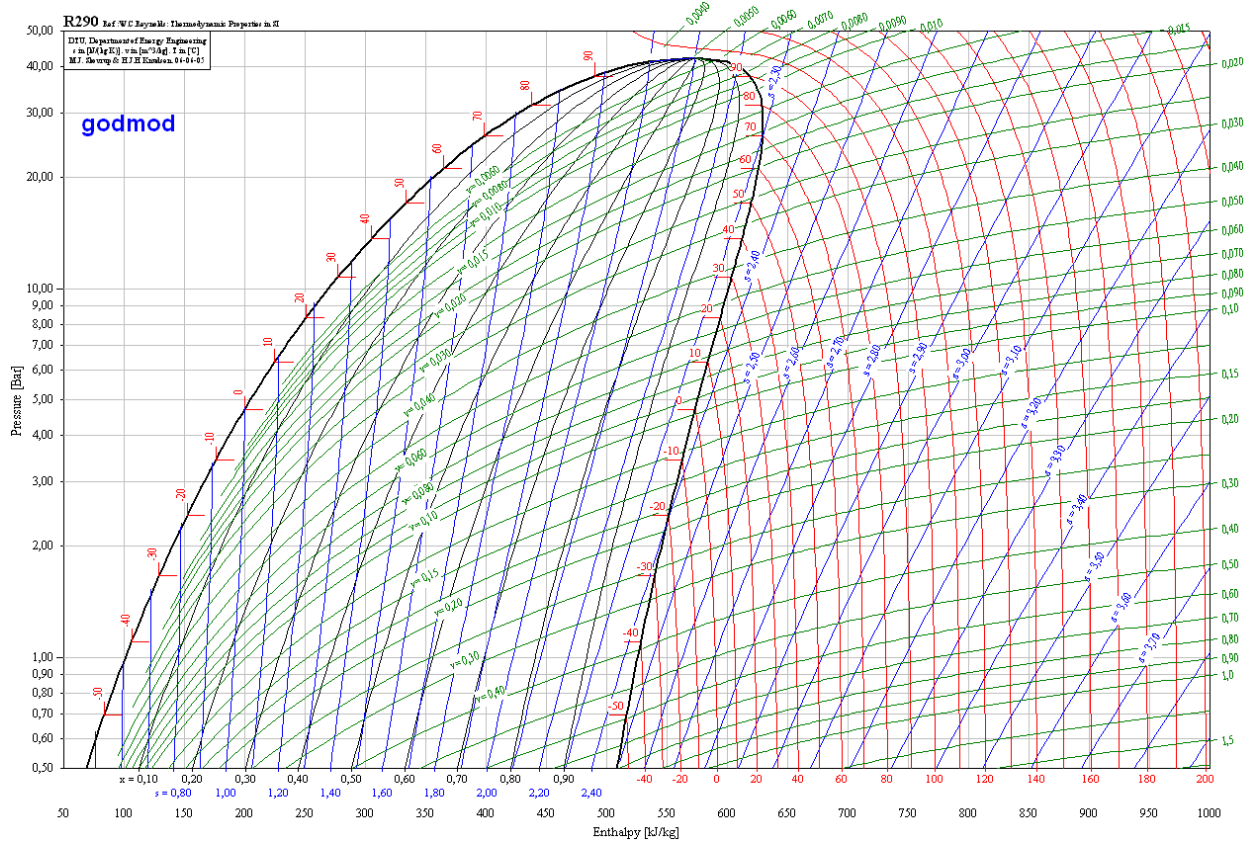


Fig 6: Pressure-enthalpy diagram of R-290 vapor.

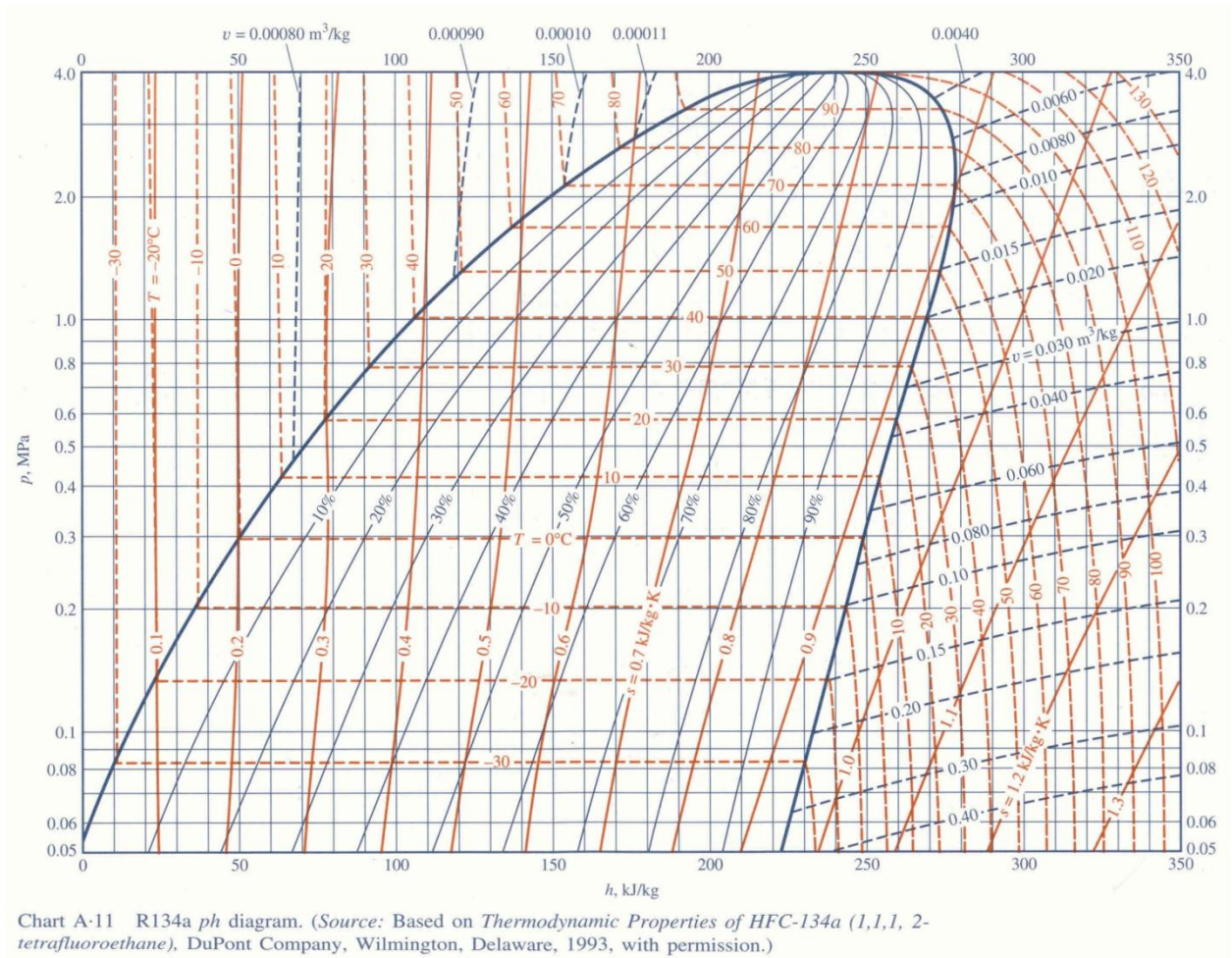


Fig 7: Pressure-enthalpy diagram of R-134a vapor.