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Analysis and Improvement of the Rate of Heat Transfer by varying Diameter

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Abstract: In the rate of heat transfer is a critical factor in various engineering applications, ranging from thermal management in electronic devices to efficient heat exchanger design. This study focuses on evaluating and improving the rate of heat transfer by varying the diameter of the heat transfer medium. The objective is to investigate how changes in diameter impact heat transfer and identify strategies for optimizing this process. The experimental setup involves a cylindrical heat transfer medium with variable diameters. The heat transfer rates are measured under controlled conditions, including constant temperature differentials and controlled thermal conductivity of the medium. The experimental results are then analyzed to understand the relationship between diameter and heat transfer efficiency.

Keywords: Heat transfer, diameter variation, thermal conductivity, convective heat transfer, optimization, experimental analysis.

I. INTRODUCTION

The efficient transfer of heat is a fundamental consideration in numerous engineering applications, ranging from electronic devices to industrial processes. The rate of heat transfer plays a pivotal role in determining the performance and reliability of these systems. One intriguing avenue for enhancing heat transfer efficiency is the variation of the diameter of the heat transfer medium. This study aims to evaluate and improve the rate of heat transfer by systematically varying the diameter and exploring the associated thermal dynamics [1], [2]. Heat transfer is a complex phenomenon influenced by several factors, including the thermal conductivity of the material, the surface area available for heat exchange, and the temperature differential. The diameter of the heat transfer medium emerges as a critical parameter that can significantly impact these factors. Understanding the interplay between diameter variations and heat transfer rates is essential for optimizing the design of heat exchangers, cooling systems, and other thermal management devices. The objective of this research is to systematically investigate how changes in the diameter of a cylindrical heat transfer medium affect the rate of heat transfer. Through controlled experiments and computational simulations, we aim to elucidate the underlying mechanisms governing heat transfer at different diameters. Additionally, we seek to identify strategies for improving heat transfer efficiency, ultimately contributing to the development of more effective thermal management systems. In this introductory phase, we acknowledge the existing body of knowledge on heat transfer and diameter effects. Previous studies have touched upon the impact of geometry on heat transfer, but a comprehensive understanding of the specific influence of diameter variations remains elusive[3], [4]. By addressing this gap, our research seeks to provide valuable insights for engineers and researchers working on systems where precise control of heat transfer rates is paramount. The structure of this study includes an experimental setup with a heat transfer medium of variable diameters, controlled environmental conditions, and comprehensive data collection. The obtained results will be rigorously analyzed, and computational simulations will be employed to model the heat transfer processes at different diameters. The subsequent sections will delve into the methodology, results, and discussions, ultimately leading to practical recommendations for optimizing heat transfer by varying diameter. As we embark on this investigation, we anticipate that the outcomes will not only advance our understanding of heat transfer phenomena but also offer tangible strategies for improving the efficiency of thermal management systems in diverse engineering applications[3]–[6].

1.1 Refrigeration

Refrigeration are the important branch of science which are producing & maintaining temperatures below that of the surrounding environment. This means the elimination of heat from a one body (hot body) to be another body (cooled body). Refrigeration may be defined as the process of achieving & maintaining a temperature below that of the surrounding's, the aim being to cool some body to the required temperature. In simple, refrigeration means the cooling or removal of heat from a one system to another system.

1.2 History of Refrigeration

The history of refrigeration are very interesting, the availability of refrigerants, the prime movers & the developments in compressors & the methods of refrigeration all are a part of it. Air refrigeration machine, Linde developed a machine in 1856 which working on ammonia refrigerant. With the development of motors & consequent higher speeds of the compressors, the scope of applications of refrigeration very large. The pace of development was considerably in the 1920 decade, the fluoro-chloro derivatives of methane, ethane, etc.- popularly known as choro fluorocarbons or CFCs-that are Freon. Recent developments involve finding atoms in Freon's are responsible for the depletion of ozone layer in the upper atmosphere.

1.3 Classification of refrigeration

1.3.1. Natural refrigeration

In natural refrigeration heat removed of the body are naturally such as evaporative cooling. In this refrigeration heat are removed by concentration of air density.

Ice was either:

1. Transported from colder regions,
2. Made during night by cooling of water by radiation to stratosphere.

In Europe, America & Iran & many countries a number of icehouses were built to store ice. Insulation materials like sawdust or wood shavings were utilized as in these icehouses. Later on another cork was utilized as insulating material.

1.3.2. Artificial refrigeration

In artificial refrigeration, refrigerant are use as heat absorber which circulates over the body & gets the heat & create the cooling effect. Now these days cooling are produced by artificial means. Though it are very difficult to make a clear demarcation between natural & artificial refrigeration, history of artificial refrigeration are began in the year 1755, when the Scottish professor William Cullen invented the first refrigerating machine, which could produce a small qquantity of ice in the laboratory [7]–[11].

1.4 Types of Refrigeration System

i. Vapor compression refrigeration system
ii. Vapor absorption refrigeration system
iii. Air refrigeration system etc

1.5 Domestic Refrigeration Systems

The domestic refrigerator using natural ice was developed in 1803 & was utilized for al- most 150 years without much alteration. The domestic ice box utilized to be made of wood with proper insulation. Ice utilized to be kept at the above of the box, & low temperatures are produced in the box due to heat transfer from ice. , it appears that warm winters caused se- vere shortage of natural ice in U.S.A. The working principle of a domestic refrigerator are exactly the same as that of an air conditioner. A schematic diagram of the refrigerator are shown in fig.1.1 Like the air conditioner, it also consists of the following four basic components:

(i) Evaporator
(ii) Compressor
(iii) Condenser
(iv)Expansion device (capillary tube)

But there are some design features which are very typical of a refrigerator. For example, the evaporator are located in the freezer compartment of the top of refrigerator. The freezer forms the coldest part of the cabinet with a temperature of about -20°C, while the refriger- ant corporate inside the evaporator tube -30°C. below the freezer, there are a chiller tray. Further below are compartments with lower temperatures. The bottom-most compartment which are meant for vegetable are the least cold one. The cold air being heavier flows down from the freezer to the bottom of the refrigerator the warm air being lighter rises from the vegetable compartment to the freezer gets cooled & flows down again[12]–[17]. Thus, a natural convection current are set up which maintains a temperature gradient between the top & the bottom of the refrigerator. The temperature maintained in the freezer are about - 20°C,whereas the mean inside temperature of the cabinet are 5°C. The design of the con- denser are also a little different. It are usually a wire & tube or plate & tube type mounted at the back side of the refrigerator. There are no fan. The refrigerant vapor are condensed with the help of surrounding air which rises above by natural convection.

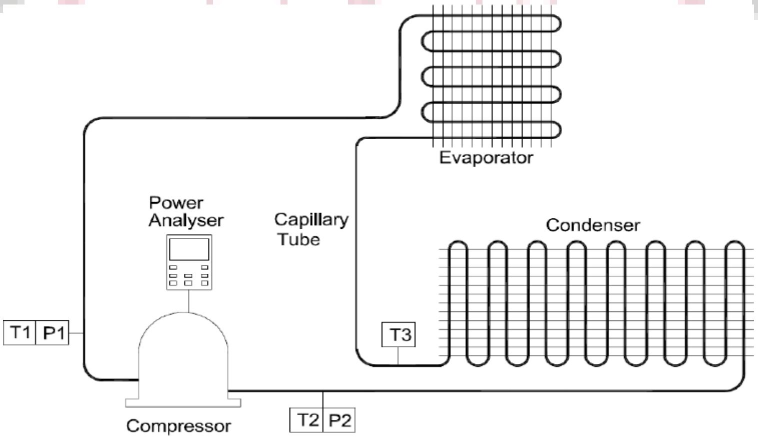


Fig.1 Block diagram of domestic refrigerator

as it gets heated after receiving the latent heat of condensation from the refrigerant. The standard condensing temperature are 50°C.

1.6 Applications of Refrigeration

Important refrigeration applications are given below

1. For transportation of foods above & below freezing temperature
2. For ice making
3. In medical & surgical aids
4. For comfort air-conditioning
4. In chemical & related industries
6. For industrial air-conditioning
7. For processing food products & beverages
8. Oil refining & synthetic rubber manufacturing
9. Manufacturing & treatment of metals
10. Freezing food products

1.7 Environmental Concerns

Due to environmental concern on the depletion of ozone layer & global warming, C.F.C (chlorofluorocarbon) & H.C.F.C (hydro chlorofluorocarbon) are being phase out from refrigeration industry. As a result H.F.C (hydro fluorocarbon), H.C (hydrocarbon) & mixture have emerged an alternative to R-12 & R-22. In fact, the earth are enveloped in a thin cell of ozone layer which prevent the earth from harmful ultraviolet radiation coming from the sun the chlorine resent in conventional refrigerant are responsible for the deletion of protective layer of ozone in the upper atmosphere this lead to increase the level of ultraviolet radiation reaching the earth surface which result in higher rated of skin cancer eye contact & damage to people in immune system. As the free molecule of conventional refrigerant reached the upper atmosphere, the strong solar radiation break down the conventional refrigerant molecule freeing chlorine atom from the structure, this chlorine reacts with ozone & convert it to oxygen. The conversion of ozone into oxygen will ultimately case thinning of the layer to the extent the hole are formed[18]–[22]. This deletion of the ozone lay- er are term as ozone hole. The conventional refrigerants have varying degree of ozone deletion potential. In the lower atmosphere the molecule of conventional refrigerant ab- sorb infrared radiation which may contribute the warming of the earth that are conventional refrigerant also act as greenhouse gases & thus having a global warming potential. The ex- tent to which a greenhouse gas contribute to global warming depend on the amount of that are emitted, the length of time which he elapses before it are urged from the atmosphere & the infrared energy of absorption property of the gas[23]–[25]. Thus global warming from a green- house gas are connected to a particular time scale known as time horizon. The global warming potential of greenhouses gases are the ratio of global warming caused from one unit mass of a greenhouse gas to that of one unit of carbon dioxide a period of time .nowadays increase the global warming are mainly because of the raid industrialization which produced large amount of green house gas.

II. LITERATURE REVIEW

Eiamsa-ard et al. [9] investigated the effects of twisted tapes on the heat transfer of HE (oblique and straight delta winglet, O-DWT, and S-DWT). The O-DWT also has a higher Nu and f than the S-DWT. Across the range studied, the performance factor in tubes fitted with the O-DWT and S-DWT was determined to be roughly 0.92-1.24 and 0.88-1.21, respectively. When it comes to heat transfer improvement, the DWT outperforms ordinary twisted tape. It means that a DWT-equipped HE was more condensed than a traditional twisted tape-equipped HE. Again, any of the TT can effectively replace the DWT.

Wang et al. [10] used a novel baffle type in the STHE, with experiments performed on both the novel baffle and the SB. The operational performances of the two HEs were also compared. According to the findings, the improved model's overall performance was 20-30% higher than the SB in HE under similar conditions. The results of the tests revealed that, because the Re number in the tube and shell sides was the same, the Nu for flower baffles was roughly half that of SBs, and the P of the former was roughly one-third that of the latter. When building HEs to save energy, heat transfer improvement and flow friction growth would be taken into account.

Bhatta and colleagues [11] investigated how CFD could be used in various types of HEs, including fluid flow, P, heat transfer, and existing turbulence models for HEs. They discovered that the k-turbulence model was the most commonly used for HE simulation, and they proposed a method for comparing the research's experimental and numerical findings.

You et al. [12] performed a shell-side thermal and hydraulic analysis of a STHE with trefoil hole baffles. According to the test results, the q on the shell side was effectively increased; however, the flow resistance increased dramatically. Furthermore, as Re increases, both heat transfer performance and P improve. As a result, the q has significantly improved, and the flow resistance has increased significantly. The trefoil-hole baffle may cause a high-speed flush against the down flow tube wall, intense recirculating flow, and a high level of turbulence intensity, according to a numerical simulation of the unit channel. As a result, the temperature boundary thickness near the wall was greatly reduced, and the q improved significantly, resulting in an increase in flow resistance.

Gowthaman and Satish [13] investigated two distinct baffles in a STHE quantitatively. HEs who meet specific criteria have a strong hand and a low P. When compared to an SB for a new installation, helical baffles reduce shell side P, pumping cost, mass, fouling, and other factors. In comparison to an SB, the increased cross-flow area results in less mass flux across the shell, resulting in a larger P. Because of the reduced bypass impact and less shell side fouling, the helical baffle was significantly larger than the SB. The friction factor and heat transfer enhancement of different STHEs with discontinuous helical baffles were investigated by

Gao et al. [14]. The findings revealed that the HE with Lower helix angles have higher shell-side P and h than higher helix angles. The irreversibility of a HE was calculated using entropy production and entrance dissipation theories in second-law thermodynamic comparisons. According to an experimental study, the STHE with lower helix angle baffles produces less irreversibility in the heat exchange procedure in the same heat transfer area and under the same operating conditions. Furthermore, HEs with helical baffles were more efficient under certain shell-side Re conditions.

Wen et al. [15] investigated the THP and revised the ladders pattern fold baffle construction to eliminate the triangle leakage regions in the original STHE. The results showed that in the improved HE, axial short circuit flow was eliminated, and the fluid velocity and temperature distribution inside the shell were more uniform.

Yahiya et al. [16] simulated fluid flow fields through HEs over a wide temperature, Res, and geometry range. The tube side Nu and tube side f increased as the MFR increased, while the thermal enhancement factor decreased marginally. The tube side Nu and f increase as the diameter of the inserted vane twirlers decreases as the blade angle decreases, but the tube side thermal enhancement decreases. The factor decreases. When compared to the plain tube scenario, the resulting Nu, f, and thermal enhancement factor were 2.3, 19.02, and 0.86, respectively. Increased swirl vanes improve heat transmission and the thermal enhancement factor, resulting in a more effective HE with less heat transfer area and volume, and thus lower costs.

Wang et al. [17] investigated the effect of rod baffle on thermal performance and P in a double-shell side rod baffle HE (DS-RBHX) using CFD. According to the results, the DS-RBHX has a higher q and P than the SS-RBHX by 34.5-42.7% and 41.6-40.6%, respectively. The efficiency estimation criterion, which was the ratio of the increase in q to the cost of power consumption, was used to evaluate overall performance in this study.

Gomaa et al. [18] looked into the triple concentric tube HE with inserted THP parameters for ribs. We used both experimental and numerical methods. Correlations for Nu, f, and efficacy were also calculated using dimensionless design parameters. At various flow configurations, the Nu and efficacy of the triple tube HE with ribs were greater than those of the triple tube HE without ribs. Without ribs, by 21.48% and 16.74%, respectively. When the flow pattern was countercurrent, the Nu and HE efficacy were higher.

Lei and Jing [19] compared two types of reformed STHEs with louver baffles to STHEs with traditional SBs in order to reduce pumping power and increase overall shell performance. The STHE-LV1 and STHE-LV2 h per P were found to be approximately 94.6-118.2% and 73.3-89.7% higher than the STHE-SG, respectively. Louvre The flow pattern produced by baffles on the shell side of HEs is gentler than that produced by SBs on the shell side of HEs. Because the new STHEs have fewer dead areas and recirculation zones than HEs with SBs, heat transfer efficiency has improved. On the shell side of the two new STHEs, abrupt changes in flow direction were avoided, resulting in a smaller P.

Labbadlia et al. [20] investigated four possible tube arrangement types. The results showed that the tube characteristics had a significant impact on the flow pattern. A 60° design was found to have a 21% more homogeneous flow distribution than a 90° configuration. In comparison to the other designs, the 45° layout provides superior pressure distribution.

Mellal et al. [21] investigated 3D numerical simulations of turbulent water flow and heat transfer in the shell of a STHE. Baffle spacing's of 106.6, 80, and 64 mm were investigated, as well as six orientation orientations of 45, 60, 90, 120, 150, and 180°. The simulation was carried out with the COMSOL package and the finite element procedure with Re ranging from 3000 to 10,000. Many numerical outcomes were compared to experimental data and kept close together. When compared to STHE without baffles, the findings demonstrated the importance of the investigated parameters in improving shell-side thermal performance, with the 180° baffle arrangement at 64 mm baffle spacing being the best that ensures mixing flow, yielding a thermal performance criteria of 3.55.

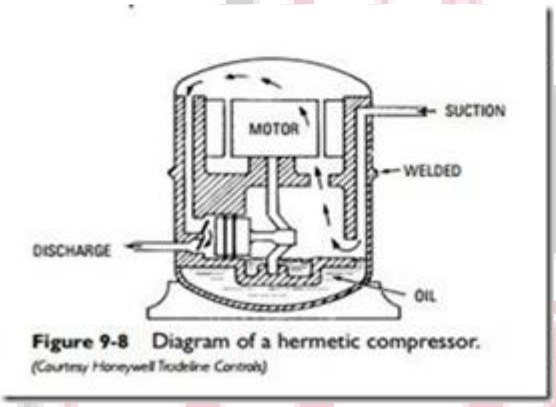
Dizaji et al. [22] used corrugated shell and corrugated tube. The researchers investigated various concave and convex corrugated tube configurations. Corrugations, according to the data, cause an increase in NTU as well as an increase in exergy loss. When both the tube and the shell were corrugated, the exergy loss and NTU increased by about 17-81 and 34-60%, respectively. Exergy loss was greatest in the HE with the concave corrugated shell and convex corrugated tube.

III. RESEARCH METHODOLOGY

3.1 Fabricating Material

3.1.1 Compressor

Reciprocating compressor are utilized for increase the pressure of refrigerant in domestic refrigerator. This compressor are hermetically sealed type. In hermetic or sealed compressor both compressor & motor are confined in a single outer welded steel shell.



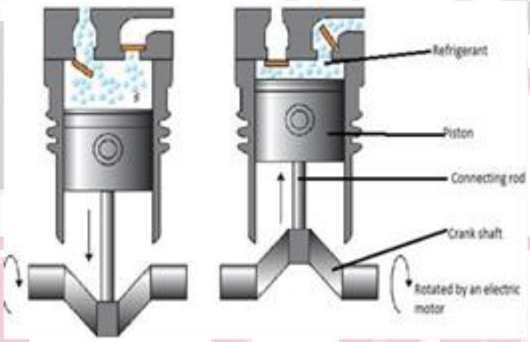


Fig.3.1 Block diagram of compressor Fig.3.2 Piston movement of compressor

In this compressor, Electric motor & reciprocating mechanism are laced inside this housing. The motor & compressor are directly attached on the same shaft, with the motor inside the refrigeration circuit. A reciprocating compressor consists of connecting rod, piston, crank shaft. Crank shaft are rotated through an electric motor. During downward & upward motion of the piston refrigerant are sucked & compressed respectively.. This process are continues in cyclic manner & deliver at high pressure refrigerant.

3.1.2. Condenser

Condenser are an very important of any refrigeration system. Condensation changes vapour to a liquid form. Its main purpose are to liquefy the refrigerant gas sucked by the compressor from evaporator. As condensation start, the heat will flow from the condenser into the air, only if the condensation temperature are higher than that of the atmosphere. From compressor high pressure refrigerant enters the condenser in a superheated state. It are first de-superheated & then condensed by removing heat from refrigerant to an external medium. The refrigerant may leave the condenser either a saturated or a sub-cooled liquid, depend on the temperature of the external medium & design of the condenser. In actual refrigeration systems with a limited pressure drop in the condenser or in a system using a R134a or mix- ture of R134a+H.C refrigerant, the temperature of the refrigerant changes during the condensation process also.

3.1.2.1 Classification of condensers:

Condensers can be classified as:

A) Air Cooled Condensers

B) Water Cooled Condensers,

C) Evaporative Condensers

3.1.3 Dryer cum filter (liquid receiver)

This are also known as the liquid receiver or pressure vessel which design for store liquid re- frigerant. It serves two functions:

1. The cooling capacity of the fridge can be varied as required. So liquid receiver are use as governor of refrigerator which connect with thermosetting valve. When required more cool- ing rate we can increase the mass flow rate of refrigerant. It are possible to 'pump down' a refrigeration system such that all the refrigerant can be stored in the liquid receiver. And
2. This are use for removing the moisture of refrigerant.

Fig. 3.3 Dryer cum filter

3.1.4 Expansion device

An expansion device are another very important part of a refrigeration system. The basic functions of an expansion device utilized in refrigeration systems are to:

1. Decrease the pressure from condenser pressure to evaporator pressure.
2. Regulate the flow of refrigerant from the high-pressure liquid line to the evaporator at a rate adequate the evaporation rate within the evaporator.

There are basically seven kinds of refrigerant expansion devices. These are:

1. Hand (manual) expansion valves
2. Capillary tubes
3. Orifice
4. Constant pressure or automatic expansion valve (aev)

4. regulator expansion valve (tev)

6. Float kind expansion valve
   A) high side float valve
   B) low side float valve
7. Electronic expansion valveAn expansion device are another very important part of a refrigeration system. the basic functions of an expansion device utilized in refrigeration systems are to:

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2. Regulate the flow of refrigerant from the high-pressure liquid line to the evaporator at a rate adequate the evaporation rate within the evaporator.

The expansion devices utilized in refrigeration systems may be divided in- to fastened opening type or variable opening type. asthe name implies, in fastened opening type the flow space remains fastened, whereas in varia- ble opening type the flow space changes with changing mass flow rates. There are basically seven kinds of refrigerant expansion devices. These are:

1. Hand (manual) expansion valves
2. Capillary tubes
3. Orifice
4. Constant pressure or automatic expansion valve (AEV)

4. regulator expansion valve (TEV)

6. Float kind expansion valve
   A) high side float valve
   B) low side float valve
7. Electronic expansion valve

Given above seven types expansion device, capillary tube & orifice belong to the fixed opening type, while the rest belong to the variable opening type. Of the above 7 types, the hand operated expansion valve (HOEV) are not utilized when an automatic control are required.

. 3.2. Important measuring & fabricating instruments

3.2.1 Pressure gauge:

Pressure gauge are a mechanical instrument which are designed to measure the inter- nal pressure & vacuum of a vessel or system. This pressure gauges are offered in a variety of styles, sizes, & wetted part materials to meet the demands of standard & special applications. We are using two type pressure gauge.

(a) High Pressure Gage: the range of high pressure gauge are ranges between 0 to 500 psi. High pressure gauge installed between compressor & condenser because exit of compressor pressure are too high.

(b)

Fig.3.3 Photographic & schematic view of high pressure Gauge

3.3 Experimental set-up & procedure

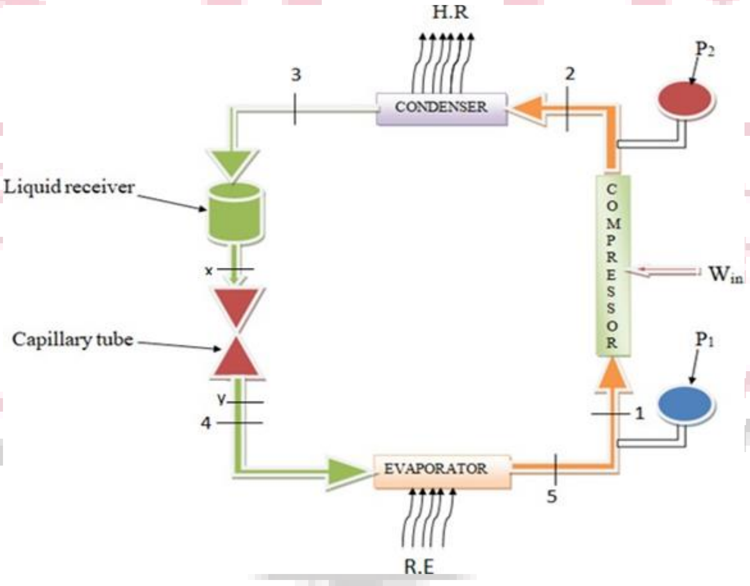

Fig. 3.4 Block diagram of Experimental set-up

In this experiment, we use a wrought iron stand as a structure for installing compressor & condenser. Wrought iron are easily available in the market & very cheap. The frame are square shape created 30×30 cm2.after creating frame I installed the compressor between 1 & 2. The compressor are fitted at the bottom of the stand. Using drilling machine create the hole for the fitting nut. Now installed the compressor with the help of nut bolt .but for prevention of movement use gas welding & get strengthen of the compressor. The wood sheet are utilized for support the evaporator & hold of the capillary. It also balances the structure. The condenser are installed back side of the compressor by a screw. Location of the condenser are between 2 to 3. The condenser are made in cast iron & quarter size diameter. After installing condenser now fit the dryer cum filter or liquid re- ceiver which are located between 3 to x. A hand operated valve was also provided after the liquid receiver situated at x point. Hand valve controls the refrigerant flow in the capillary tube by bypassing the refrigerant. In this experiment, many time change the capillary so must be required the hand operated the valve. A number of hand shut-off valves were pro- vided between the major components of the experimental setup, in case of leak or repair; the damaged component could be repaired with ease. The temperature measured at different locations by means of k-type thermocouples which are connected with the six-channel

perature indicator. While the pressure of the refrigerant was measured with pressure gauges. The pressure gauge are located on the set-up 1 & 2 points.

Now going to the fitting of the capillary which are the major part of our experiment. Before installation of the capillary, we are creating the spiral shape of the capillary. Spiral tube test-section was formed by first producing a spiral. The shape of the required pitch on the face of the wooden plate of 310mm×310mm. & 10mm thickness. For the making of a spiral required two points ‘c1’ & ‘c2’ offset by a distance of ‘x’ mm. With the help of a compass, a semicircle of ‘x’ mm radius was drawn taking ‘c1’ as the center & second inverted semicir- cle are twice of the offset distance (i.e., 2x) taking ‘c2’ as the center. This procedure was repeated until the curve reached the diameter are equal to 30 cm. The step-by-step procedure for generating spiral has been shown in fig. 3.3

The relation between the offset distance & the pitch of the spiral are as follows: p = 2x. Af- ter marking in the wooden plate overlaps the capillary tube in the marked spiral diagram. & do the fix using the pin. Pitch of the spiral are 2cm & the diameter of the spiral are 30 cm. Similarly, two other spiral capillaries created with the fallow same procedure. In this pro- cess three different diameters with spiral capillary tube get.

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IV. RESULT ANALYSIS

4.1 Performance of VCRS using R134a as a refrigerant

Here
T1 =inlet temperature of compressor in °C

T2 = exit temperature of compressor in °C

T3 = exit temperature of condenser in °C

T4 = exit temperature of capillary or inlet temperature of evaporator in °C

T5 = exit temperature of evaporator in °C

T0= surrounding temperature in °C

P1 = exit pressure of capillary or inlet pressure of compressor in bar

P2= inlet pressure of capillary or exit pressure of compressor in bar

h1= inlet enthalpy of compressor in kJ/kg

h2= exit enthalpy of compressor in kJ/kg

h3= exit enthalpy of condenser in kJ/kg

h4= exit enthalpy of capillary or inlet enthalpy of evaporator in kJ/kg

h5 = Exit enthalpy of evaporator in kJ/kg

4.1.1 Pressure ratio & Carnot COP for 1.12 mm diameter capillary tube

Spiral capillary, d =1.12 mm, L = 4.4M, T0 =25 °C

Table 4.1 Pressure ratio & Carnot COP for 1.12 mm diameter capillary tube

S. No.	Time (min)	T_1 (°C)	T_2 (°C)	T_3 (°C)	T_4 (°C)	T_5 (°C)	P_1 (bar)	P_2 (bar)	P_2/P_1	Carnot COP
1	15	20	108	33	-31	14	0.89	13.72	14.41	3.78
2	30	19	109	34	-34	14	0.8	13.1	16.37	3.51
3	45	19	107	34	-35	10	0.79	12.69	16.06	3.45
4	60	19	109	34	-35	9	0.79	13.031	16.49	3.45

This table gives the data for the diameter of capillary 1.12mm. This table represents the temperature of all point of the system. In this table, we calculate pressure ratio

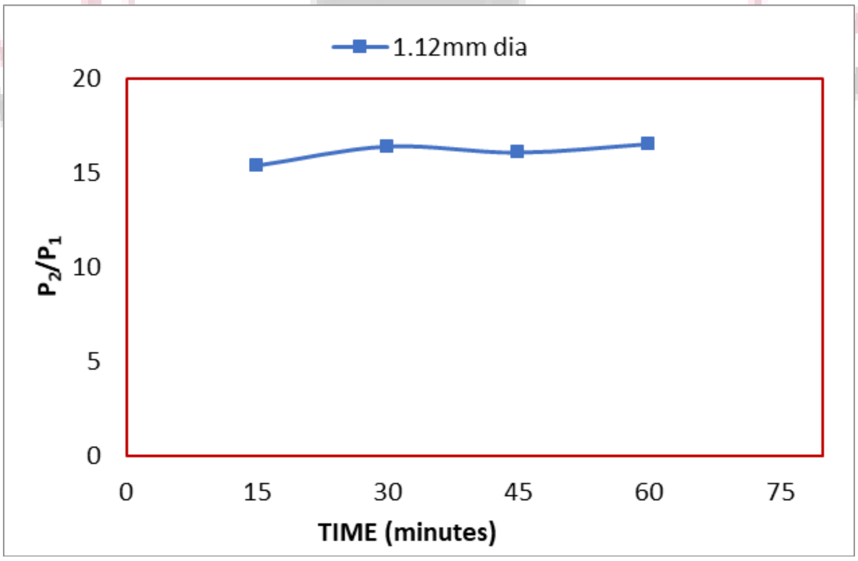

Fig. 4.1 Variation of pressure ratio with time using 1.12 mm diameter capillary tube

In Fig.4.1 show the variation of pressure ratio. For dia. 1.12mm minimum pressure ratio are14.41 & maximum pressure ratio gets 16.49. The Fig. 4.1shows pressure ratio increase with time up to 15 minutes. After 15 minute time pressure ratio are decreasing up to 30 minutes & again after 30-minute pressure ratio are increased up to 60 minutes & reach 16.49.

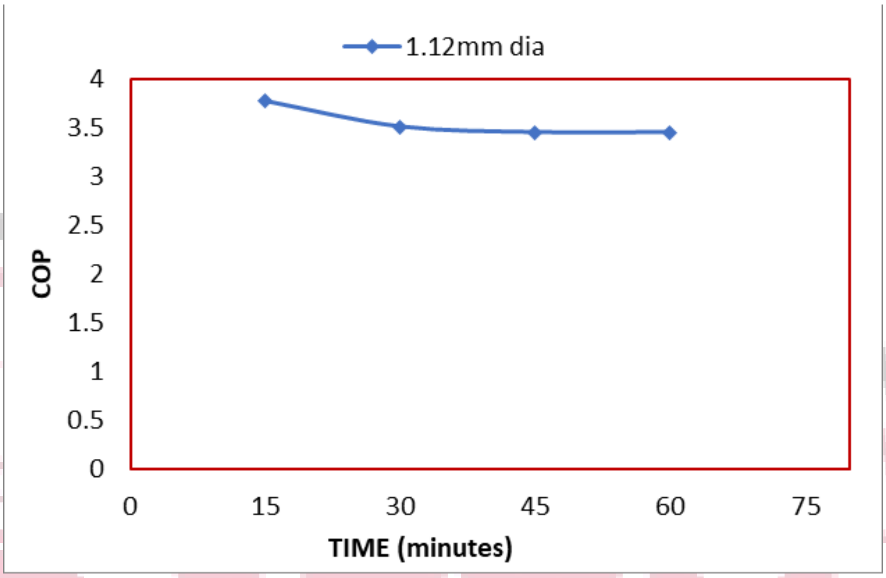

Fig. 4.2 Variation of Carnot COP with time using 1.12 mm diameter capillary tube

Fig. 4.2 shows the variation of Carnot COP with time. At starting COP occur maxi- mum, after some time COP decreases continuously. Maximum COP found at 15 minutes which are 3.78 & minimum COP found at 60 minutes are 3.45.

4.1.2 Mass flow rate, refrigeration capacity & COP for 1.12 mm diameter capillary

Table 4.2 Mass flow rate, refrigeration capacity & COP for 1.12 mm diameter

S.No.	h_1	h_2	$h_3 = h_4$	h_5	m	W	R.C	$(COP)_{actual}$
1	420	490	240	383	2.86	200	408	2.04
2	420	490	238	380	2.86	200	405	2.03
3	419	489	242	379	2.86	200	391	1.96
4	418	490	245	380	2.78	200	375	1.88

Table 4.2 represents the value of enthalpy corresponding to table 4.1 temperatures. In this table, we also calculate mass flow rate

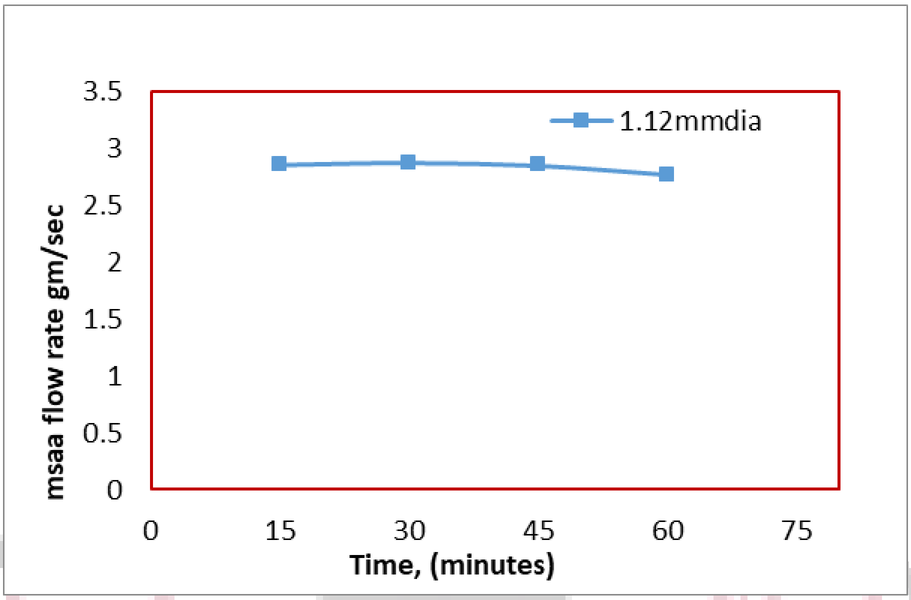

Fig. 4.3 Variation of mass flow rate with time using 1.12 mm diameter capillary tube

Show the Fig. 4.4 variation of C.O.P with respect to time. In starting C.O.P occur maximum after some time C.O.P are decreasing continuously. Maximum C.O.P get in 15 minutes are 2.04 & minimum C.O.P get at 60 minutes are 1.87

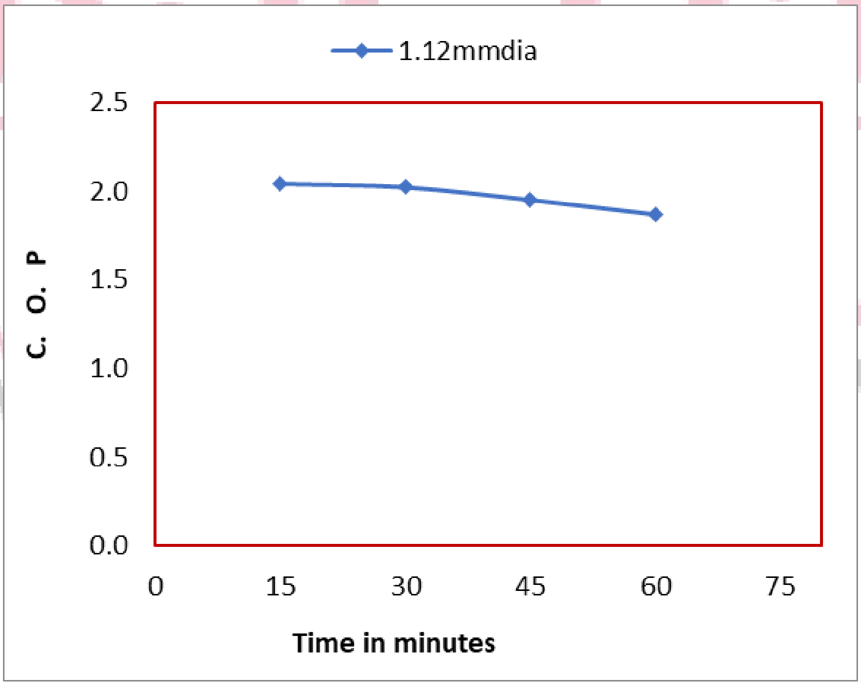

Fig. 4.4 Variation of actual COP with time using 1.12 mm diameter capillary tube

V. CONCLUSION

In this experiments we are studying in the spiral capillary tube with three different diame- ters 1.12mm,1.4mm & 1.52mm & two type of refrigerant one are an R134a & second are a mixture of R134a & hydrocarbon. & the conclusion of the experiment are discussed below:

- Maximum pressure ratio for R134a are 16.49 for 1.12mm diameter capillary tube, & for the mixture are 12.7 for 1.4 mm diameter capillary tube. The maximum pressure ratio of R134a are 29.8% greater than the mixture.
- Minimum pressure ratio for R134a are 8.61 for 1.4mm diameter capillary tube, & for the mixture are 8.07 for 1.52 mm diameter capillary tube. The minimum pressure ratio of R134a are 6.69% greater than the mixture.
- Vapor pressure characteristics of the mixture closely match with R134a refrigerant thus same R134a compressor can be utilized.
- Mass flow rate occur maximum at 1.4mm diameter of the capillary tube are 3.63 g/sec, & average mass flow rate for 1.4 mm diameter of the capillary tube are 3.48 g/sec which are 4.13% lower than maximum mass flow rate.

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