# Performance Analysis of VCR System by Varying Diameters of Helical Condenser Coil Using R-600a as Refrigerant

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ABSTRACT

Vapour Compression Refrigeration system is now a days used for all purposes of refrigeration in domestic as well as commercial applications which has high coefficient of performance. The system performance depends on the performance of all the components of the system. By changing design and design parameters of various components, we can enhance the performance of the system.The present work deals with experimental analysis of vapour compression refrigeration system with helical condenser coil varying in diameters using R-600A as refrigerant, which is modification to normal condenser of domestic refrigerator working on vapour compression refrigeration system using R-134A as refrigerant. An attempt has been made to increase the performance of the refrigeration system with a change in shape of condenser coil from grid to helical. The pressure gauges and temperature indicators are integrated with the system at appropriate positions. A series of experiments were conducted on grid shape condenser coil (existing) and helical shape condenser coil. Various parameters like Net Refrigeration Effect, Mass flow rate, Work of compression, Heat equivalent of work of compression per TR etc were evaluated. It was observed that the helical shaped condenser is more efficient than the grid type condenser coil.

## 1. INTRODUCTION

Basic Components of a vapor compression system
Basic components of a vapor compression refrigeration system are shown in Figure
They are,
Compressor: It is motor driven; it sucks vapor refrigerant from evaporator and compresses.
Condenser: High pressure vapor refrigerant is condensed into liquid form in the condenser        using cooling medium such as water.
Expansion Valve: High pressure refrigerant is throttled down to evaporator pressure; rate of flow is metered.
Evaporator: A cooling chamber in which products are placed; low pressure liquid refrigerant flows in the coils of evaporator and absorbs heat from products; the refrigerant vaporizes and leaves for compressor.
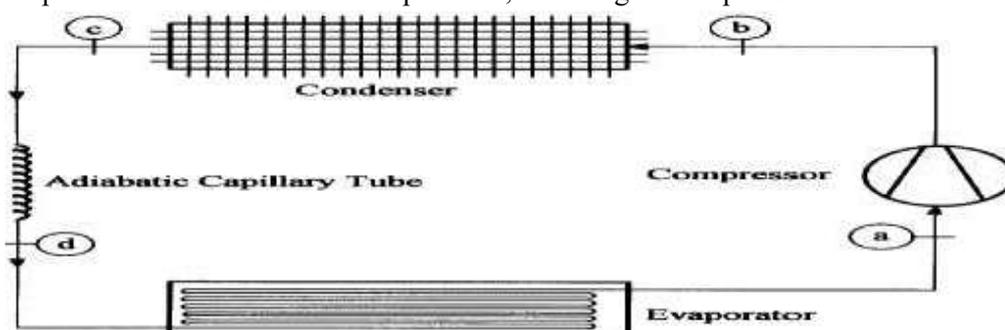


*Fig.1.1 Schematic diagram of a vapor compression refrigeration system*

Condensers and evaporators are basically heat exchangers in which the refrigerant undergoes a phase change. Next to compressors, proper design and selection of condensers and evaporators is very important for satisfactory performance of any refrigeration system. Since both condensers and evaporators are essentially heat exchangers, they have many things in common as far as the design of these components is concerned. However, differences exist as far as the heat transfer phenomenon is concerned. In condensers the refrigerant vapour condenses by rejecting heat to an external fluid, which acts as a heat sink. Normally, the external fluid does not undergo any phase change, except in some special cases such as in cascade condensers, where the external fluid (another refrigerant) evaporates. In evaporators, the liquid refrigerant evaporates by extracting heat from an external fluid (low temperature heat source).

The external fluid may not undergo phase change, for example if the system is used for sensibly cooling water, air or some other fluid. There are many refrigeration and air conditioning applications, where the external fluid also undergoes phase change. For example, in a typical summer air conditioning system, the moist air is dehumidified by condensing water vapour and then, removing the condensed liquid water. In many low temperature refrigeration applications freezing or frosting of evaporators takes place. These aspects have to be considered while designing condensers and evaporators. Evaporator and condenser temperatures only and is independent of the nature of the working substance. This is the reason why exactly the same expression was obtained for air cycle refrigeration systems operating on Carnot cycle The Carnot COP sets an upper limit for refrigeration systems operating between two constant temperature thermal reservoirs (heat source and sink). From Carnot's theorems, for the same heat source and sink temperatures, no irreversible cycle can have COP higher than that of Carnot COP

## 2. LITERATURE REVIEW

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Vapour Compression Refrigeration Systems:

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 process involves two thermodynamic concepts, the vapour 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 vapours can condense at a temperature greater than that of the surroundings.
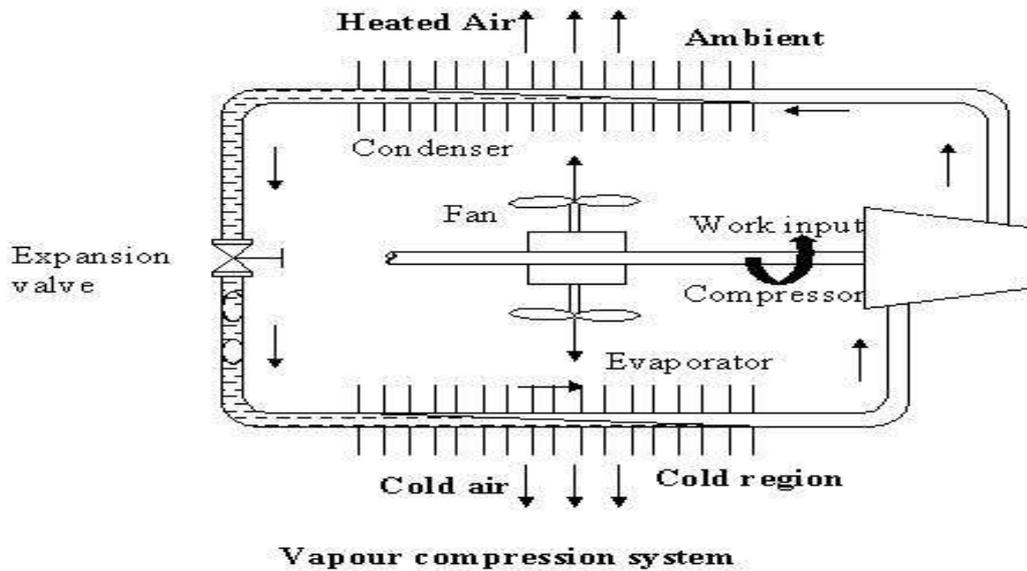
Vapour compression system

Fig:2.2 shows the basic components of a vapour compression refrigeration system

As shown in the figure the basic system consists of an evaporator, compressor, condenser and an expansion valve. The refrigeration effect is obtained in the cold region as heat is extracted by the vaporization of refrigerant in the evaporator. The refrigerant vapour from the evaporator is compressed in the compressor to a high pressure at which its saturation temperature is greater than the ambient or any other heat sink. Hence when the high pressure, high temperature refrigerant flows through the condenser, condensation of the vapour into liquid takes place by heat rejection to the heat sink.

To complete the cycle, the high pressure liquid is made to flow through an expansion valve. In the expansion valve the pressure and temperature of the refrigerant decrease. This low pressure and low temperature refrigerant vapour evaporates in the evaporator taking heat from the cold region. It should be observed that the system operates on a closed cycle. The system requires input in the form of mechanical work. It extracts heat from a cold space and rejects heat to a high temperature heat sink.

A refrigeration system can also be used as a heat pump, in which the useful output is the high temperature heat rejected at the condenser. Alternatively, a refrigeration system can be used for providing cooling in summer and heating in winter. Such systems have been built and are available now.

## 3. REFRIGERANTS

Safe refrigerants

These are the non-toxic, non-flammable refrigerants such as R11, R12, R13, R14, R21, R22, R113, R114, methyl chloride, carbon dioxide, water etc. Toxic and moderately flammable: Di-chloroethylene methyl format, ethyl chloride, sulphur dioxide, ammonia etc. come under this category. Highly flammable refrigerants: The refrigerants under this category are butane, isobutene, propane, ethane, methane, ethylene etc.

CHEMICAL COMPOSITIONS ;They are further sub-divided as Halocarbon compounds: Refrigerants these are obtained by replacing one or more hydrogen atoms in ethane or methane with halogens.

Azeotropes: These are the mixtures of two or more refrigerants and behave as a compound.

Oxygen and Nitrogen Compounds: Refrigerants having either oxygen or nitrogen molecules in their structure, such as ammonia, are grouped separately and have a separate nomenclature from the halogenated refrigerants.

Cyclic organic Compounds: The compounds coming under this class are R316, R317 and R318.

Inorganic Compounds: These are further divided into two categories:

Cryogenic and Non cryogenic.Cryogenic fluids are those which are applied for achieving temperatures as low as – 160 0C to – 273 0C. Above this temperature range, we can use a multi-stage refrigeration system to realize the desired temperature. But below – 160 0C, this is not possible since the COP of the cycle becomes very low. To attain temperatures below – 160 0C, we use refrigerants such as nitrogen, oxygen, helium, hydrogen etc. and for

temperatures close to – 273 0C, magnetic cooling is employed. The inorganic compounds which are employed above the cryogenic temperature ranges come under the remaining sub-division of inorganic refrigerants.

**Refrigerant R-600A**     The refrigerant R-600A is the chemical formula of C4H10. It is an isomer of butane. It is the simplest alkane with a tertiary carbon. Isobutane is used as a precursor molecule in the petrochemical industry, for example in the synthesis of isooctane.The molecular weight of refrigerant R-600A is 133.4

3.5.3 Physical Properties:

Miscibility with Oil: The refrigerant should not be miscible with the oil else the lubricating strength will be reduced.

Viscosity: It should be as small as possible to ensure that the pressure drop in the system is as small as possible. A low viscosity refrigerant will require less energy for its circulation through the refrigeration system

## COMMON REFRIGERANTS

Sulphur Dioxide: Sulphur dioxide (molecular weight 64) is a colorless, suffocating and irritating gas and is twice as heavy as air at atmospheric conditions. It was mostly used as a household refrigerant in the older days, but has since been discarded for better refrigerants. It suffers from a lot of disadvantages. Sulphur dioxide reacts with water forming sulphurous acid, which in presence of oxygen becomes sulphuric acid, a corrosive compound for metals. It is non-flammable but attacks foodstuff on coming in contact with it. It is also partially miscible with the lubricating oil.

## TYPES OF CONDENSERS

.                                   In actual refrigeration systems with a finite pressure drop in the condenser or in a system using a zoetrope refrigerant mixture, the temperature of the refrigerant changes during the condensation process also. However, at present for simplicity, it is assumed that the refrigerant used is a pure refrigerant (or an Azeotropes mixture) and the condenser pressure remains constant during the condensation process. Process 3-4 is a sensible, sub cooling process, during which the refrigerant temperature drops from $T_3$ to $T_4$.
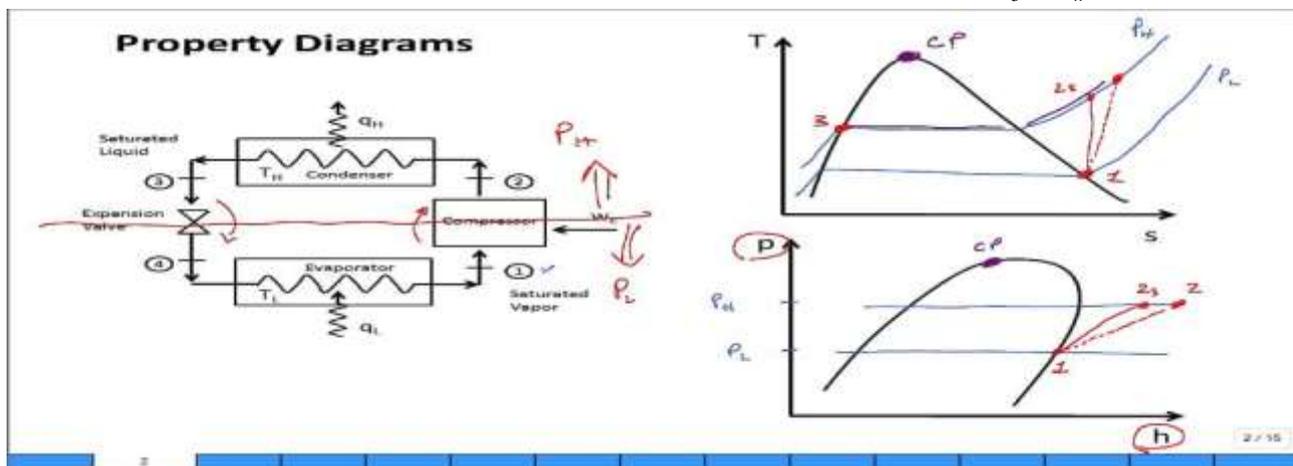


## CLASSIFICATION OF CONDENSERS:

Based on the external fluid, condensers can be classified as:

a) Air cooled condensers

b) Water cooled condensers, and

c) Evaporative condensers

**Air-cooled condensers:**

As the name implies, in air-cooled condensers air is the external fluid, i.e., the refrigerant rejects heat to air flowing over the condenser. Air-cooled condensers can be further classified into natural convection type or forced convection type.

**i) Natural convection type**:

In natural convection type, heat transfer from the condenser is by buoyancy induced natural convection and radiation. Since the flow rate of air is small and the radiation heat transfer is also not very high, the

combined heat transfer coefficient in these condensers is small. As a result a relatively large condensing surface is required to reject a given amount of heat. Hence these condensers are used for small capacity refrigeration systems like household refrigerators and freezers. The natural convection type condensers are either plate surface type or finned tube type. In plate surface type condensers used in small refrigerators and freezers, the refrigerant carrying tubes are attached to the outer walls of the refrigerator. The whole body of the refrigerator acts like a fin. Insulation is provided between the outer cover that acts like fin and the inner plastic cover of the refrigerator. It is for this reason that outer body of the refrigerator is always warm. Since the surface is warm, the problem of moisture condensation on the walls of the refrigerator does not arise in these systems. These condensers are sometimes called as flat back condensers.

The finned type condensers are mounted either below the refrigerator at an angle or on the backside of the refrigerator. In case, it is mounted below, then the warm air rises up and to assist it an air envelope is formed by providing a jacket on backside of the refrigerator. The fin spacing is kept large to minimize the effect of fouling by dust and to allow air to flow freely with little resistance.

In the older designs, the condenser tube was attached to a plate and the plate was mounted on the backside of the refrigerator. The plate acted like a fin and warm air rose up along it. In another common design, thin wires are welded to the serpentine tube coil. The wires act like fins for increased heat transfer area. Figure shows the schematic of a wire-and-tube type condenser commonly used in domestic refrigerators. Regardless of the type, refrigerators employing natural convection condenser should be located in such a way that air can flow freely over the condenser surface.
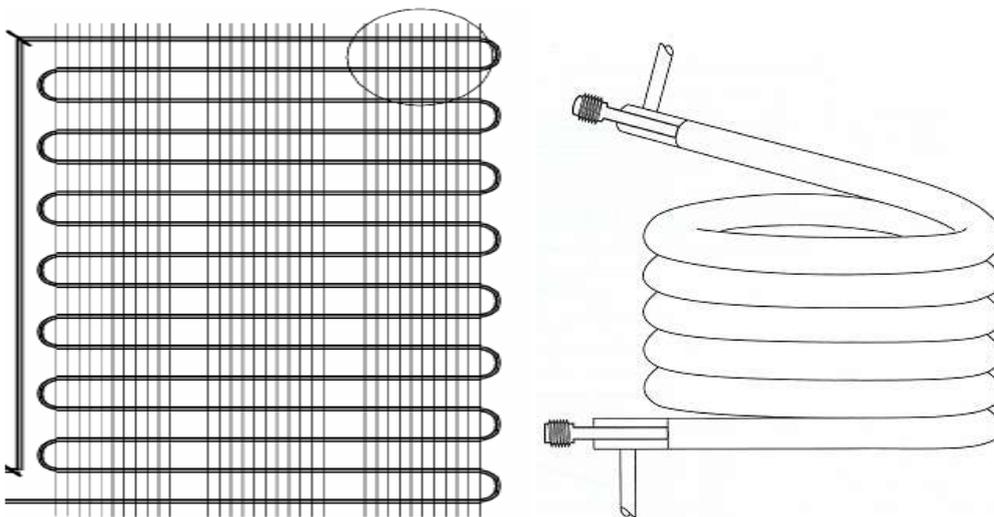
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Fig :4.3 forced convention Type

**SELECTION OF CONDENSER FOR A VCR SYSTEM**

**Condenser:** Condenser is the component which is placed next to compressor in a vapor compression refrigeration system. It removes heat absorbed by refrigerant in the evaporator and the heat of compression added in the compressor and condenses it back to liquid.

**Selection of condenser:** The condenser is one of the most important components of refrigeration system. Its function is to dissipate heat absorbed by the refrigerant during evaporation (refrigeration effect) and compression (Heat of compression).There are three different type of condensers classified on the basis of cooling used to dissipate heat. These are.

Air cooled

Water cooled

Evaporative type

Air-cooled condenser can be natural convection type or forced convection type. In this project air-cooled condenser is used which is the most common type in use. Before sizing a condenser, careful evaluation should include, consideration of initial cost, operating cost , service life and type of load. A condenser that is too large can be expensive and create operating problems in lower ambient conditions; an undersized condenser can cause operating problems in higher ambient conditions. It is therefore

4.3.3 Important parameters to consider before sizing a condenser:

1 Gross heat rejection

2 Ambient temperatures

3 Condensing temperature

4 Temperature difference (TD)

5 Air flow

The heat transfer through the condenser is by conduction, condenser capacity is a function of the fundamental heat transfer equitation.

Qc = U.A. (LMTD)

WhereQc = Condenser capacity in KJ/Sec. (Ref. Effect Heat of Comp. +Motor Wdg. Heat)

U = Overall heat transfer coefficient KJ/h-m2 0K

A = Effective surface area in m2

LMTD = the log mean temperature difference between the condensing refrigerant and        medium.

From the above equation it is evident that for any fixed value of 'U' the capacity of condenser is directly proportional to the surface area of the condenser and to the temperature difference between the condensing refrigerant and condensing medium.

## EXPERIMENTAL SETUP

The figure shows the experimental setup of the refrigerator. In order to know the performance characteristics of the vapor compression refrigeration system the temperature and pressure gauges are installed at each entry and exit of the components. Experiments are conducted on existing and helical condenser coils having the refrigerator capacity of 166liters. All the values of pressures and temperatures are tabulated.

**DESIGN OF HELICAL CONDENSER:**       The helical condenser is the applications for helical tubing coils range from copper helical coil with end fixture the aerospace industry to the refrigeration (ACR), petroleum, and brewing industry. In this present work remove the existing condenser and install the helical design condenser to the refrigerator (166 liters).To taken the temperatures and pressure readings and calculate the performance of the system.
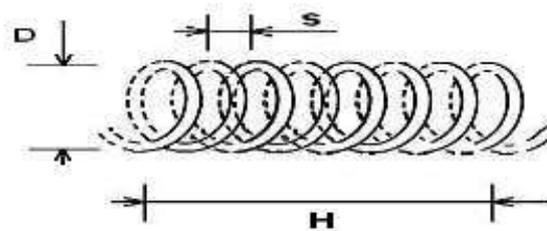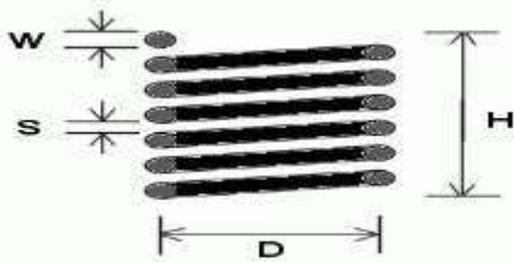


*Fig.5.2side view of the helical shape.*                    *Fig.5.1 Helical shape.*

| Parameter | dimension |
|---|---|
| Diameter of design coil D [mm] | 250 |
| Diameter of tube W [mm] | 6.35 |
| Spacing S [mm] | 60 |
| Turns | 12 |
| Length of tube[mm] | 9420 |
| Height H [mm] | 740 |

*Tabular column of Helical Condenser coil Parameters*.

Domestic refrigerator selected for the project has the following specifications:

Refrigerant used: R-600A

Capacity of The Refrigerator: 166 liters

Compressor capacity: 0.16 H.P.

*Fig.5.3 existing condenser                    modeled condenser*

Dimensions Condenser Evaporator Expansion valve

| Dimensions | Condenser | Evaporator | Expansion valve |
|---|---|---|---|
| Length (m) | 9.5 | 7.62 | 3.6 |
| Diameter (mm) | 6.4 | 6.4 | 0.9 |

## PERFORMANCE CALCULATIONS:

**Analysis of the condenser:**

Thermal analysis in the heat exchangers can be done in two ways.

1. LMTD Method (Logarithmic Mean Temperature Difference)

2. NTU Method (Number of Heat transfer Units)

LMTD Method is useful when the inlet and outlet fluid temperatures of condenser and air are known. NTU Method is useful when the heat exchanger is designed for the particular mass flow rate. For the given conditions LMTD Method is suitable.

1. LMTD Method:

In a heat exchanger, the temperature of the heating and cooling fluids do not in general, remain constant, but vary from point to point along the length of the heat exchanger. Since the temperature difference between the two fluids keeps changing, the rate of heat transfer also changes along the length of the heat exchanger as shown.
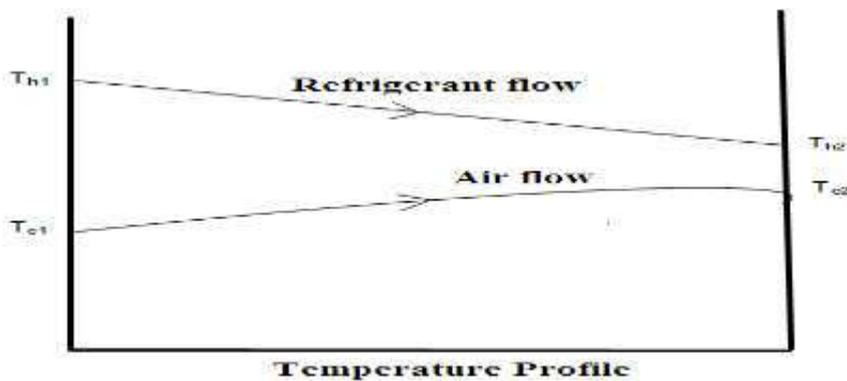


*Fig 5.4Temperature Profile*

The rate of heat transfer can be calculated from the relation

$Q = U A \Delta T$

Since $\Delta T$ changes from point to point in a heat exchanger, we propose to use $\Delta T_m$, a suitable mean temperature difference between the two ends of a heat exchanger. The rate of heat transfer can be rewritten as

Where

$\Delta Tm$ = Log Mean Temperature Difference (LMTD)

A = surface area of condenser in $m^2$ = $\Pi DL$

$\Delta Tm = \dfrac{\Delta T1 - \Delta T2}{ln\frac{\Delta T1}{\Delta T2}}$

$\Delta T_1$ = $Th_1$-$TC_1$  $\Delta T_2$ = $Th_2$-$TC_2$

U = overall heat transfer coefficient in $w/m^2k$   $A_0$ = outside tube Area in $m^2$

$A_i$ = inside tube Area in $m^2$ $h_i$ = convective heat transfer coefficient of R-134(a) in $w/m^2k$

$h_0$ = convective heat transfer coefficient of Air in $w/m2k$ = 10 $w/m^2k$

$r_0$ = outside radius of pipe in m   $r_i$= inside radius of pipe in m

K = thermal conductivity of copper in w/m-k

If $A_o$ = $A_i$ the above equation can be reduced to

U = 1/ (1/ $h_i$ + 1/$h_o$)

Properties of R-134(a) at bulk mean temperature at various condenser speeds are taken

Bulk mean temperature of condenser can be calculated by

= (Condenser inlet temp + Condenser outlet temp.)/2

In order to calculate convective heat transfer coefficient of R-134(a) the following steps are to be followed and the convection is of forced convection.

ReD = (pvD)/$\mu$

Pr = ($\mu$ Cp)/K

Where  $Re_D$ = Reynolds number

p = Density of R-410A in $kg/m^3$

v = velocity in m/sec = 3 to 4 m/sec

D= Diameter of the pipe in m

$\mu$ = viscosity in pa.s

$C_p$= specific heat in j/kgk

K= thermal conductivity in w/mk

Forced convection correlations in turbulent pipe flow are given by Dittus-Boelter

NUD = $0.023ReD^{4/5}Pr^n$

NUD = $h_iD/K$

Where

D= Diameter of the pipe = $6.35 \times 10^{-3}$m

Pr = Prandtle number

n = 0.4 for heating of the fluid and 0.3 for cooling of the fluid

The Dittus-Boelter equation is valid for

0.7< Pr <160 and ReD >10000

The Dittus-Boelter equation is good approximation where temperature differences between bulk fluid and heat transfer surface are minimal.


**RESULTS AND DISCUSSIONS**

**CALCULATIONS**

6.1.1 Existing System Temperatures with actual condenser

Compressor Suction Temperature T1 =18.1°CCompressor Discharge Temperature T2 =56.2°C

Condensing Temperature T3 =32.8°CEvaporator Temperature T4 =-14.1°C

Pressures

Compressor suction pressure P1 =0.98 bar  Compressor discharge pressure P2 =11.28bar

Condenser pressure P3 =11.28 bar  Evaporator pressure P4 =0.98 bar

Enthalpies

From pressure-enthalpy chart for R-134a, enthalpy values at state points 1,2,3,4. The state points are fixed using pressure and temperature and each point.

h1 = 620 kJ/kg  h2 = 642 kJ/kg

h3 = 640 kJ/kg  h4 = 580 kJ/kg

**Calculation Performance Parameters**

☐☐Net Refrigerating Effect (NRE) = Enthalpy of refrigerant leaving the evaporator –

Enthalpy of refrigerant entering the evaporator

= h1-h4= 620-580 = 40 kJ/kg

☐☐Mass flow rate to obtain one TR, kg/min.

Mr = 210/NRE = 210/40 = 5.25 kg/min.

☐☐Work of Compression = Enthalpy of refrigerant leaving the compressor –

Enthalpy of refrigerant entering the compressor.

=h2-h1 =642-620 = 22 kJ/kg

☐☐Heat Equivalent of work of compression per TR= mass flow rate X  work of compression

$M_r$ x (h2-h1) = 5.25 x 22 = 115.5 kJ/min

☐☐Theoretical power of compressor =115.5/60 = 1.925 kW

☐☐Coefficient of Performance (COP)= $\frac{net\ efrigeration\ effect}{compression\ work}$ = h1-h4 /h2-h1 = 40/22 = 1.81


**6.1.2 Helical condenser with 200 Diameter:**

Compressor Suction Temperature T1 =19.2°C

Compressor Discharge Temperature T2 =47.8°C

Condensing Temperature T3 =31.2°C

Evaporator Temperature T4 =-9°C

Pressures

Compressor suction pressure P1 =1.05 bar

Compressor discharge pressure P2 =10.2bar

Condenser pressure P3 =10.2 bar

Evaporator pressure P4 =1.05 bar

Enthalpies

From pressure-enthalpy chart for R-134a, enthalpy values at state points 1,2,3,4. The state points are fixed using pressure and temperature and each point.

h1 = 629 kJ/kg

h2 = 649kJ/kg

h3 = 606kJ/kg

h4 = 591 kJ/kg

**Calculation Performance Parameters**

☐☐Net Refrigerating Effect (NRE) = Enthalpy of refrigerant leaving the evaporator –

Enthalpy of refrigerant entering the evaporator

= h1-h4= 629-591 = 38 kJ/kg

☐☐Mass flow rate to obtain one TR, kg/min.

Mr = 210/NRE = 210/38 = 5.52kg/min.

☐☐Work of Compression = Enthalpy of refrigerant leaving the compressor –

Enthalpy of refrigerant entering the compressor.

=h2-h1 =649-629 = 20kJ/kg

☐☐Heat Equivalent of work of compression per TR

☐☐$M_r$ x (h2-h1) = 5.52 x 20 = 110.4 kJ/min

☐☐Theoretical power of compressor =110.4/60 = 1.84 kW

☐☐Coefficient of Performance (COP) = $\frac{net\ efrigeration\ effect}{compression\ work}$ = h1-h4 /h2-h1 = 38/20= 1.90


 **Helical Condenser with 250mm diameter**

Compressor Suction Temperature T1 =17.3°C

Compressor Discharge Temperature T2 =49.5°C

Condensing Temperature T3 =29.4°C

Evaporator Temperature T4 =-17.2°C

Pressures

Compressor suction pressure P1 =1.28 bar

Compressor discharge pressure P2 =12.85bar

Condenser pressure P3 =12.85 bar

Evaporator pressure P4 =1.28 bar

Enthalpies

From pressure-enthalpy chart for R-134a, enthalpy values at state points 1,2,3,4. The state points are fixed using pressure and temperature and each point.

h1 = 648 kJ/kg

h2 = 681 kJ/kg

h3 = 623kJ/kg

h4 = 575 kJ/kg

Calculation Performance Parameters

☐☐Net Refrigerating Effect (NRE) = Enthalpy of refrigerant leaving the evaporator –

Enthalpy of refrigerant entering the evaporator

h1-h4= 648-575 = 73 kJ/kg

☐☐Mass flow rate to obtain one TR, kg/min.

☐Mr = 210/NRE = 210/73 = 2.876kg/min.

☐☐Work of Compression = Enthalpy of refrigerant leaving the compressor –

Enthalpy of refrigerant entering the compressor.

h2-h1 =681-648 = 33kJ/kg

☐☐Heat Equivalent of work of compression per TR= mass flow rate X work of compression

☐☐$M_r$ x (h2-h1) = 2.876 x 33 = 94.908 kJ/min

☐☐Theoretical power of compressor =94.908/60 = 1.58 kW

☐☐Coefficient of Performance (COP) = $\frac{net\ Refrigeration\ effect}{compression\ work}$ = h1-h4 /h2-h1 = 73/33 = 2.21

| S no | Parameter | Existing system (R134A) | Helical condenser with design change (R-600A) | |
| --- | --- | --- | --- | --- |
| | | | 200 Diameter | 250 Diameter |
| 1 | Net Refrigerating effect(KJ/Kg) | 40 | 38 | 73 |
| 2 | Mass flow rate to obtain 1TR(KJ/min) | 5.25 | 5.52 | 2.876 |
| 3 | Work of compression(KJ/kg) | 22 | 20 | 33 |
| 4 | Compressor power, KW | 1.92 | 1.84 | 1.58 |
| 5 | Co efficient of performance | 1.81 | 1.90 | 2.21 |

**Tabular column of results**

**PERFORMANCE OF A VAPOR COMPRESSION REFRIGERATION   CYCLE**

The performance of vapor compression refrigeration cycle with helical condenser and the existing condenser and variation in coil diameter has the considerable effect. To illustrate these effects the calculated values of helical condenser and various diameter of the coil have been plotted on graphs.

6.2.1. Effect of helical condenser coil diameter (D) on net refrigeration effect.

From the calculations it is observed that the net refrigeration effect of condenser is to be varied at different diameters of helical coil condenser is shown in bellow fig. The net refrigeration effect of helical condenser is more than the net refrigeration effect of existing condenser.
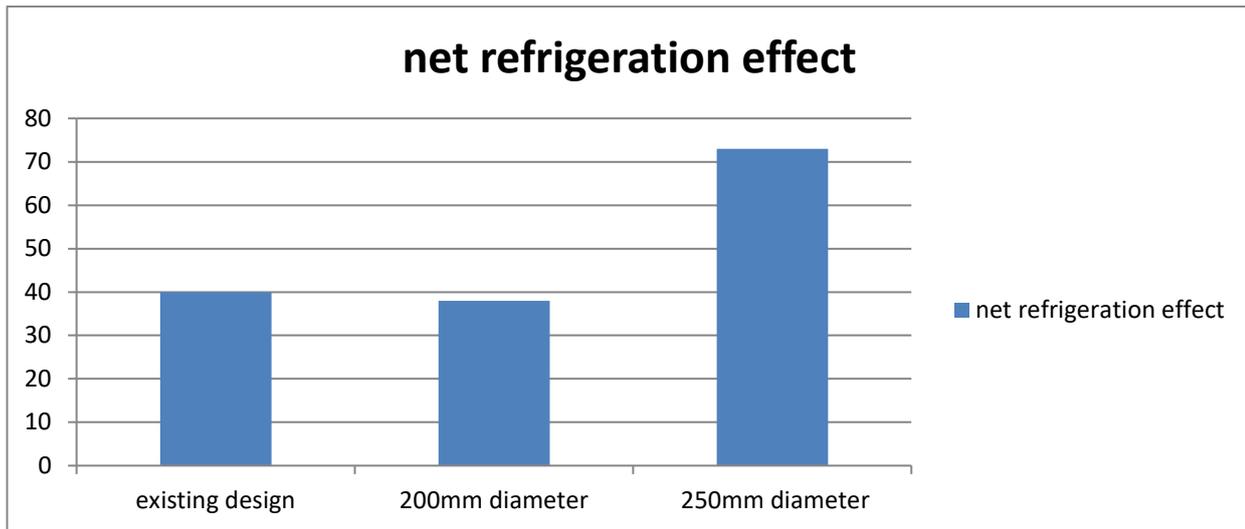
*Fig.7 Effect of helical condenser coil diameter (D) on net refrigeration effect.*

**Effect of helical condenser coil diameter (D) on coefficient of performance.**

From the calculations it is observed that the performance of the refrigeration system increases as the diameter of the coil increases and it is maximum at the 250 mm. After that the cop of system is stat to decreasing the. Due to more heat transfer sub cooling occurs at the exit of the condenser and hence the performance of the system increases.
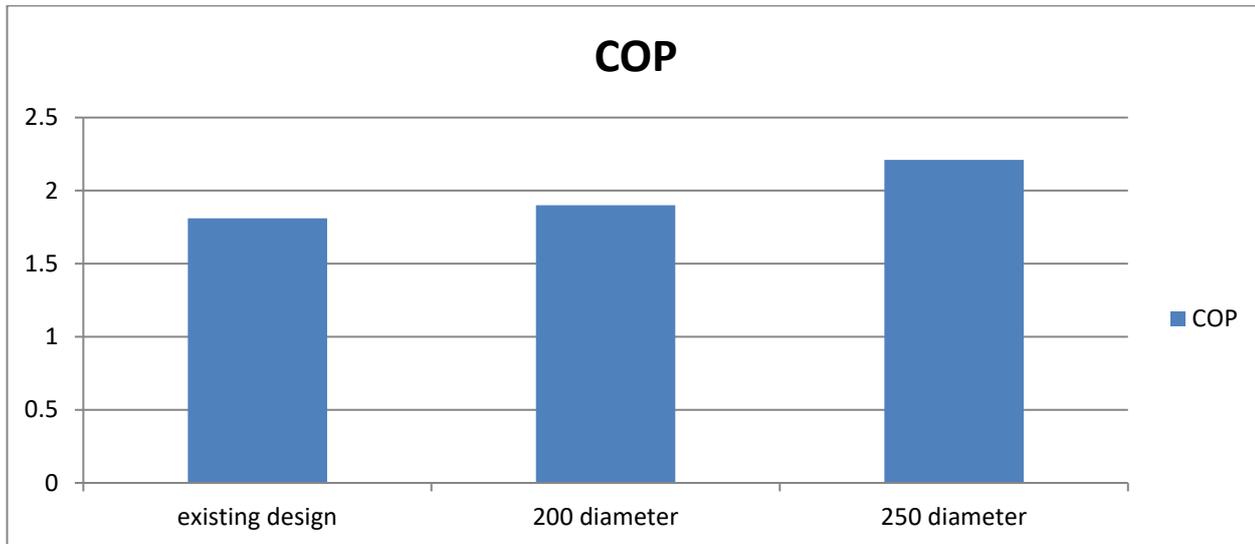


*Fig.8 Effect of helical condenser coil diameter (D) on coefficient of performance.*

**. Effect of condenser on coefficient of performance**

From the above results it is observed that the net refrigeration effect of helical condenser is more than the net refrigeration effect of existing condenser and work of compression of helical condenser is less than the existing condenser than the COP of helical system is to be more than the existing condenser as shown in bellow fig.
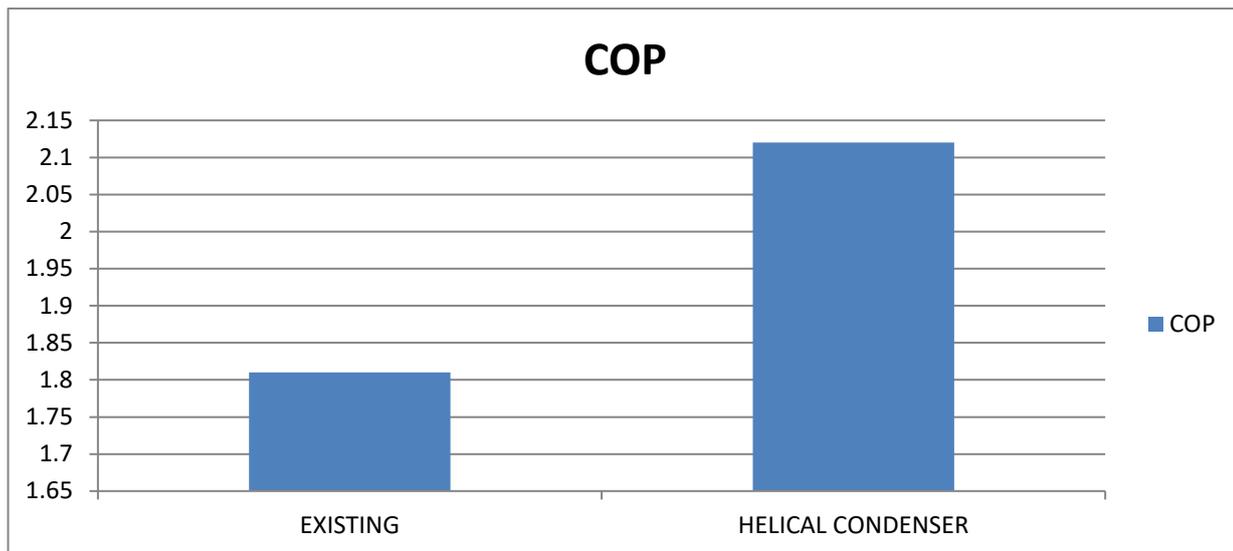
*Fig.9 Effect of condenser on coefficient of performance*

**CONCLUSION**

In the present work, experiments are conducted for the helical design condenser by taking different Diameter (D) of the condenser coil for a domestic refrigerator of 166 liters capacity.

By incorporating the helical condenser in the refrigerator, it was observed that

The COP enhanced by 0.4, as a result of 33 kj/kg increase in refrigeration effect.

Reduction in the compressor work is 2.37 kj/kg

Increase in heat rejection 11kj/kg.

The performance of helical condenser is also changed at different diameters so, design of diameter helical condenser coil plays a prominent role. It is advantageous to provide a helical condenser at the inlet of the capillary tube and maintain the condenser pressure and the performance of vapor compression refrigeration system can be enhanced with the help of the helical condenser.

Finally, it is concluded that the change in shape of existing design to helical condenser, the coefficient of performance is increased and heat transfer rate is also increased and maximum value of heat transfer is attain at 250mm coil diameter

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