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PERFORMANCE ANALYSIS OF VCR SYSTEM WITH VARYING THE DIAMETERS OF HELICAL CONDENSER COIL BY USING R-134A REFRIGERANT

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ABSTRACT

Vapor compression machine is a refrigerator in which the heat removed from the cold by evaporation of the refrigerant is given a thermal potential so that it can gravitate to a natural sink by compressing the vapor produced. Majority of the refrigerators works on the Vapor compression refrigeration system. The system consists of components like compressor, condenser, expansion valve and evaporator. The performance of the system depends on the performance of all the components of the system.

The design of condenser plays a very important role in the performance of a vapor compression refrigeration system. Effective new designs are possible through theoretical calculations, however may fail due to the reason that the uncertainties in the formulation of heat transfer from the refrigerant inside the condenser tubes to the ambient air. Hence experimental investigations are the best in terms of optimization of certain design parameters. The main objective in the present work is an attempt is made to verify the performance of existing condenser design to helical shaped condenser design and varying the length of the helical condenser coil to verifying the effect on the performance of a domestic refrigerator capacity 165lts, R-134a as refrigerant, hermetic sealed compressor. It is expected that the helical shaped condenser installation may give optimum results. Finally it is observed that by changing the conventional design to Helical shaped condenser the performance of the refrigeration system is increased.

KEYWORDS: VCR system, Refrigerant, copper helical coil, COP.

INTRODUCTION

Vapor compression Refrigeration system is an improved type of air. The ability of certain liquids to absorb enormous quantities of heat as they vaporize is the basis of this system. Compared to melting solids (say ice) to obtain refrigeration effect, vaporizing liquid refrigerant has more advantages. To mention a few, the refrigerating effect can be started or stopped at will, the rate of cooling can be predetermined, the vaporizing temperatures can be governed by controlling the pressure at which the liquid vaporizes. Moreover, the vapor can be readily collected and condensed back into liquid state so that same liquid can be recirculated over and over again to obtain refrigeration effect. Thus the vapor compression system employs a liquid refrigerant which evaporates and condenses readily. The Vapor compression refrigeration system is now-a-days used for all purpose refrigeration. It is generally used for all industrial purposes from a small domestic refrigerator to a big air-conditioning plant.

Basic Components of a vapor compression system

Basic components of a vapor compression refrigeration system are shown in Fig.1 They are,

1.	Compressor	:	It is motor driven; it sucks vapor refrigerant from evaporator and compresses.
2.	Condenser	:	High pressure vapor refrigerant is condensed into liquid form in the condenser using cooling medium such as water.

3.	Expansion Valve	:	High pressure refrigerant is throttled down to evaporator pressure; rate of flow is metered.
4.	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.

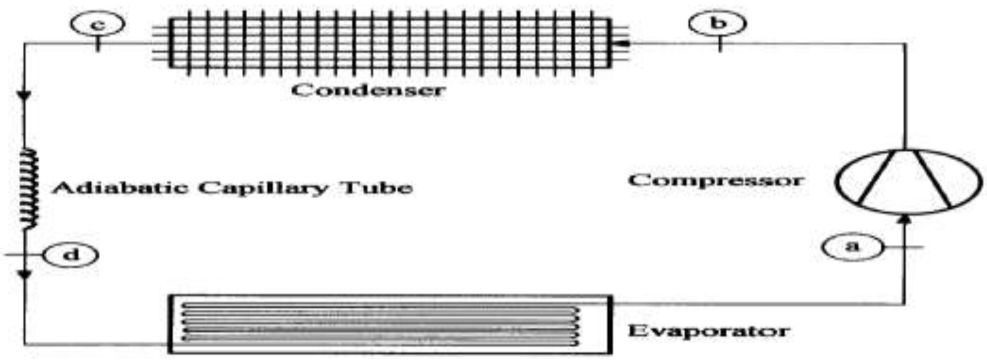

Fig.1 Schematic diagram of a vapor compression refrigeration system

P- H Diagram

The most convenient chart for studying the behavior of a refrigerant is the p-h chart, in which the vertical ordinates represent pressure and horizontal ordinates represent enthalpy. A typical chart is shown Fig., in which a few important lines of the complete chart are drawn. The saturated liquid line and saturated vapor line merge into one another at the critical point. A saturated liquid is one which has a temperature equal to the saturation temperature corresponding to its pressure. The spaces to the left of the saturated liquid line will, therefore, be sub cooled liquid region. The space between the liquid and the vapor lines is called wet vapor region and to the right of the saturated vapor line is a superheated vapor region

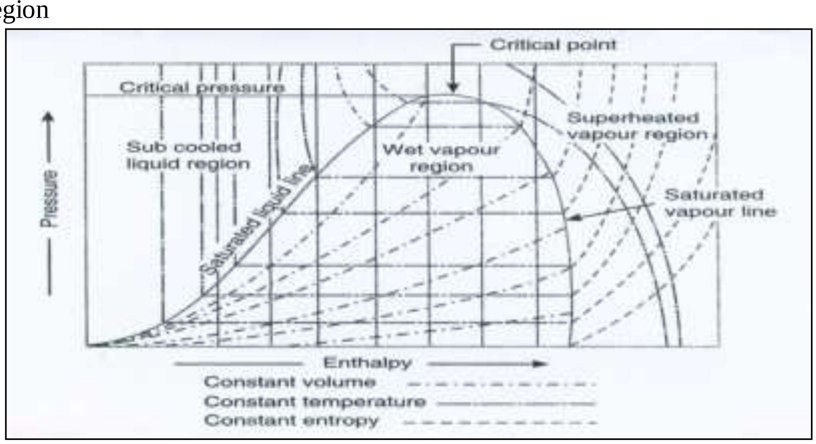

Fig.2 Pressure – Enthalpy (P-h) chart

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 component 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 important to consider the following factors before sizing a condenser.

1 Gross heat rejection
2 Ambient temperature
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.

$$Q_c = U.A. (LMTD)$$

Where

Q_c	=	Condenser capacity in KJ/Sec. (Ref. Effect Heat of Comp. + Motor Wdg. Heat)
U	=	Overall heat transfer coefficient KJ/h-m^2 ^{0}K
A	=	Effective surface area in m^2
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.

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 (165 liters).To taken the temperatures and pressure readings and calculate the performance of the system.

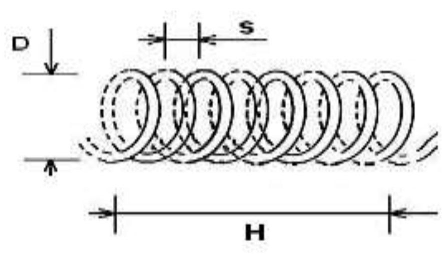

Fig.3 Helical shape.

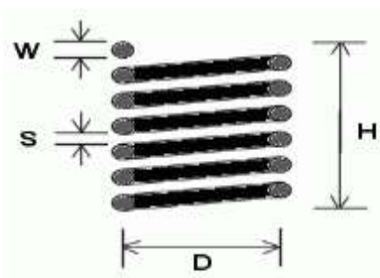

Fig.4 side view of the helical shape.

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Diameter of design coil D [mm]	262.5
Diameter of tube W [mm]	6.35
Spacing S [mm]	50
Turns	11
Length of tube[mm]	750
Height H [mm]	495

Tabular column of Helical Condenser coil Parameters.

EXPERIMENTAL SETUP

The figure 5 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 165liters. All the values of pressures and temperatures are tabulated.

Domestic refrigerator selected for the project has the following specifications:
Refrigerant used: R-134a
Capacity of The Refrigerator: 165 liters
Compressor capacity: 0.16 H.P.

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

Fig.5 Existing condenser

Fig.6 Helical condenser coil diameter of 225 mm

S.NO	PARAMETERS	EXISTING SYSTEM R-134A	HELICALCONDENSER SYSTEM WITH DIFFERENT DIAMETERS-R-134A			
			187	225	262	300
1.	Net refrigerating effect , kJ/kg	124	129	130	130	132
2.	Coefficient of performance (COP)	3.56	3.91	4.06	4.33	4.12
3.	Mass flow rate to obtain one TR, kg/min	1.69	1.63	1.61	1.61	1.59
4.	Work of compression, kJ/kg	35	33	32	30	32
5.	Heat equivalent of work of compression per TR, KJ/min	59.15	53.8	51.52	48.3	50.88
6.	Compressor power, KW	0.98	0.89	0.86	0.81	0.85
7.	Heat to be rejected in condenser , KJ/kg	159	162	162	160	164
8.	Heat rejection per TR, KJ/min	268.51	263.72	260.82	257.6	260.76
9.	Heat rejection ratio	1.28	1.26	1.24	1.24	1.24
10.	Compression pressure ratio	6.51	7.28	6.85	6.49	7

PERFORMANCE CALCULATIONS

ANALYSIS OF THE CONDENSER

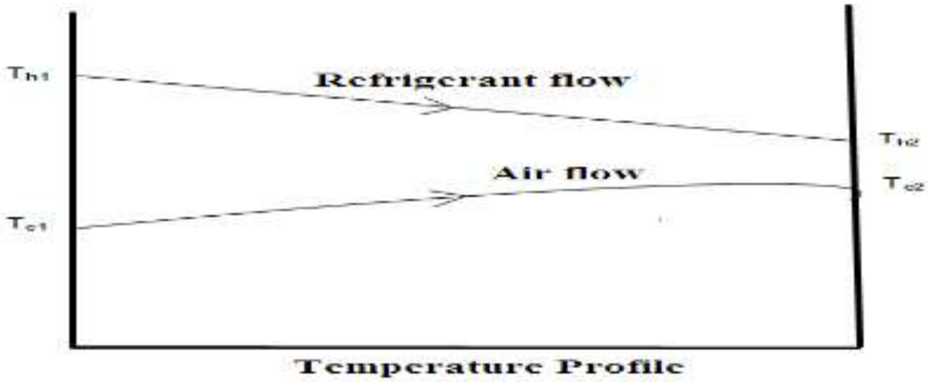
Thermal analysis in the heat exchangers can be done in two ways.

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.



Temperature Profile

The rate of heat transfer can be calculated from the relation

$Q = U\,A\,\Delta T$

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

Where

ΔT_m = Log Mean Temperature Difference (LMTD)

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

$\Delta T_m = (\square_1-\square_2)/\ln(\square_1/\square_2)$

Where $\square_1 = T_{h1}-T_{C1}$

$\square_2 = T_{h2}-T_{C2}$

$$U = \frac{1}{A_O/A_i \;\; 1/h_i + \dfrac{A_O \ln(r_o/r_i)}{2\Pi KL} + 1/h_o}$$

U = overall heat transfer coefficient in w/m^2k

A_O = 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_o = convective heat transfer coefficient of Air in $w/m^2k = 10\ w/m^2k$

r_o = 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.

$R_{eD} = (pvD)/\mu$

$P_r = (\mu\, C_p)/K$

Where

R_{eD} = Reynolds number

p = Density of R-134(a) in kg/m^3

v = velocity in m/sec = 3 to 4 m/sec
D= Diameter of the pipe in m
μ = 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
$N_{UD} = 0.023 R_{eD}^{4/5} Pr^n$
$N_{UD} = h_i D/K$
Where
D= Diameter of the pipe = $6.35 x 10^{-3}$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 $R_{eD} >10000$
The Dittus-Boelter equation is good approximation where temperature differences between bulk fluid and heat transfer surface are minimal.

Nusselt number:

In heat transfer at boundary (surface) within a fluid, the Nusselt number is the ratio of convective to conductive heat transfer across (normal to) boundary. Named after Wilhelm Nusselt, it is a dimensionless number.

A Nusselt number is close to one for slug or laminar flow. It varies for turbulent flow. For forced convection, the Nusselt number is generally a function of the Reynolds number and Prandtle number, or Nu = f (Re, Pr).

Heat Transfer Rate in Existing Condenser

Condenser entering temperature= 48^0 C
Condenser leaving temperature = 34^0 C
Air temperature at condenser inlet =28.1^0C
Air temperature at condenser outlet =29.1^0C
Mean temperature = (48.5+34)/2 = 41^0C
Mean temperature = 41^0C at this temperature properties are
From R-134(a) refrigerant property tables
p = $1141 kg/m^3$
D = $6.35 X 10^{-3}$ m
μ = $176 .1X 10^{-6}$ Pa.s
v = 3.5 m/s
K = $74.6 x 10^{-3}$ W/ m K
C_p= $1.507 x 10^{-3}$ J/kg K
$R_{e D} = (\rho v D)/\mu = (1141x 3.5 x 6.35 x 10^{-3}) / (176 .1x 10^{-6})= 1.44 x10^5$
$P_r = (\mu C_p)/K = (176.1 x10^{-6} x1.507x10^{-3})/ (74.6x10^{-3}) = 3.56$
$N_{UD} = 0.023 R_{eD}^{4/5} P r^n = 0.023x (1.44 x10^5)^{4/5} x (3.56)^{0.3}= 450.66$
$N_{UD} = h_i D/K$
$450.66 = (h_i 6.35 X 10^{-3})/ 74.6x10^{-3}$
$h_i = 5294.4 W/m^2K$
$U = 1/ (1/ h_i + 1/h_o) = 1/(1/5294.4 + 1/10) = 10 W/m^2$
$\Delta T_m = (\square_1-\square_2)/\ln (\square_1/\square_2)$
$\square_1 = T_{h1}-T_{C1} =48-28.1 =19.9$
$\square_2 = T_{h2}-T_{C2} =34-29.1 =4.9$
$\Delta T_m = (\square_1-\square_2)/\ln (\square_1/\square_2)$
=10.7
Q=Heat transfer rate through the condenser = U A ΔT_m
$Q= 10x\pi x6.35x10^{-3}x8.5x10.7= 18.14W$

Heat Transfer Rate for Helical Condenser Calculations

Inside heat transfer coefficient for condenser coil

Sizes of tube:
Outer diameter of tube = 6.4mm
For case -1(262mm)
Condenser entering temperature= 48 ^{0} C
Condenser leaving temperature = 40.5^0 C
Air temperature at condenser inlet =30.6^{0}C
Air temperature at condenser outlet=31.6^{0}C
Mean temperature = (48+40.5)/2= 44.25^{0}C
Mean temperature = 44.25^{0}C at this temperature properties are
From R-134(a) refrigerant property tables
$p = 1128.2 kg/m^3$
$D = 6.35 X 10^{-3} m$
$\mu = 169.2X 10^{-6}$ Pa.s
v = 3.5 m/s
$K = 73 x10^{-3}$ W/ m K
$C_p = 1.58x10^{-3}$J/kg K
$R_{e D} = (\rho vD)/\mu = (1128.2x3.5x6.35 x 10^{-3})/ (169.2x 10^{-6}) = 1.48 x10^5$
$P_r = (\mu C_p)/K = (169.2 x10^{-6} x1.58x10^{-3})/ (73x10^{-3}) = 3.66$
$N_{UD} = 0.023R_{eD}^{4/5} Pr^n = 0.023x (1.48x10^5)^{4/5} x (3.66)^{0.3} = 464.5$
$N_{UD} = h_iD/K$
$464.5 = (h_i 6.35 X 10^{-3})/ 73x10^{-3}$
$h_i = 5340 W/m^2K$
$U = 1/ (1/ h_i + 1/h_o) = 1/(1/5340 + 1/10) = 10 W/m^2$
$\Delta T_m = (\square_1-\square_2)/\ln (\square_1/\square_2)$
$\square_1 = T_{h1}-T_{C1}=48-30.6 =17.4$
$\square_2 = T_{h2}-T_{C2}=40.6-31.6 =9$
$\Delta T_m =(\square_1-\square_2)/\ln (\square_1/\square_2)$
=12.8
Q=Heat transfer rate through the condenser = U A ΔT_m
$Q= 10x\pi x6.35x10^{-3}x8.5x12.8=21.7W$
For case -2(300mm)
Condenser entering temperature= 47.3^0 C
Condenser leaving temperature = 38.3^0 C
Air temperature at condenser inlet =29^{0}C
Air temperature at condenser outlet=30^{0}C
Mean temperature = (47.3+38.3)/2= 42.8^{0}C
Mean temperature = 42.8^{0}C at this temperature properties are
From R-134(a) refrigerant property tables
$p = 1134.42kg/m^3$
$D = 6.35 X 10^{-3} m$
$\mu = 172.32X 10^{-6}$ Pa.s
v = 3.5 m/s
$K = 73 .7x10^{-3}$ W/ m K
$C_p = 1.518x10^{-3}$J/kg K
$R_{e D} = (\rho vD)/\mu = (1134.42x3.5x6.35 x 10^{-3})/ (172.32x 10^{-6}) = 1.46 x10^5$
$P_r = (\mu C_p)/K = (172.32x10^{-6} x1.518x10^{-3})/ (73x10^{-3}) = 3.55$
$N_{UD} = 0.023R_{eD}^{4/5} Pr^n = 0.023x (1.46x10^5)^{4/5} x (3.55)^{0.3} = 455.28$
$N_{UD} = h_iD/K$
$455.28 = (h_i 6.35 X 10^{-3})/ 73x10^{-3}$
$h_i = 5284W/m^2K$
$U = 1/ (1/ h_i + 1/h_o) = 1/(1/5284 + 1/10) = 10 W/m^2$
$\Delta T_m = (\square_1-\square_2)/\ln (\square_1/\square_2)$
$\square_1 = T_{h1}-T_{C1}=47.3-29 =18.3$

$\square_2 = T_{h2}-T_{C2}$=38.3-30 =8.3
ΔT_m =($\square_1$-$\square_2$)/ln ($\square_1$/$\square_2$)
=12.65
Q=Heat transfer rate through the condenser = U A ΔT_m
Q= 10xπx6.35x10^{-3}x8.5x12.65=21.4W

For case -3 (187mm)

Condenser entering temperature= 53.5 ^{0} C
Condenser leaving temperature = 36 C
Air temperature at condenser inlet =29.8^{0}C
Air temperature at condenser outlet=30.6^{0}C
Mean temperature = (53.5+36)/2= 44.75^{0}C
Mean temperature = 44.75^{0}C at this temperature properties are
From R-134(a) refrigerant property tables
p = 1125.86kg/m^3
D = 6.35 X 10^{-3} m
μ = 168.26X 10^{-6} Pa.s
v = 3.5 m/s
K = 72.76x10^{-3} W/ m K
C_p= 1.535x10^{-3}J/kg K
$R_{e\,D}$ = (ρvD)/μ = (1125.86x3.5x6.35 x 10^{-3})/ (168.26x 10^{-6})= 1.48 x10^5
P_r = (μ C_p)/K = (168.26x10^{-6} x1.535x10^{-3})/ (72.76x10^{-3}) = 3.55
N_{UD} = 0.023$R_{eD}^{4/5}$ Pr^n = 0.023x (1.48x10^5)$^{4/5}$ x (3.55)$^{0.3}$= 460.26
N_{UD} = h_iD/K
460.26 = (h_i x6.35 X 10^{-3})/ 72.76x10^{-3}
h_i = 5273.78 W/m^{2}K
U = 1/ (1/ h_i + 1/h_o) = 1/(1/5273.78 + 1/10) = 10 W/m^2
ΔT_m = ($\square_1$-$\square_2$)/ln ($\square_1$/$\square_2$)
$\square_1 = T_{h1}-T_{C1}$=53.5-29.8 =23.7
$\square_2 = T_{h2}-T_{C2}$=36-30.8 =5.2
ΔT_m =($\square_1$-$\square_2$)/ln ($\square_1$/$\square_2$)
=12.2
Q=Heat transfer rate through the condenser = U A ΔT_m
Q= 10xπx6.5x10^{-3}x8.5x12.2=20.68W

For case -4 (225mm)

Condenser entering temperature= 53 ^{0} C
Condenser leaving temperature = 36^0 C
Air temperature at condenser inlet =29.6^{0}C
Air temperature at condenser outlet=30.6^{0}C
Mean temperature = (48+40.5)/2= 44.5^{0}C
Mean temperature = 44.5^{0}C at this temperature properties are
p = 1126.97kg/m^3
D = 6.35 X 10^{-3} m
μ = 168.78X 10^{-6} Pa.s
v = 3.5 m/s
K = 72.89 x10^{-3} W/ m K
C_p= 1.529x10^{-3}J/kg K
$R_{e\,D}$ = (ρvD)/μ = (1126.97x3.5x6.35 x 10^{-3})/ (168.78x 10^{-6})= 1.48 x10^5
P_r = (μ C_p)/K = (168.78 x10^{-6} x1.58x10^{-3})/ (72.89x10^{-3}) = 3.54
N_{UD} = 0.023$R_{eD}^{4/5}$ Pr^n = 0.023x (1.48x10^5)$^{4/5}$ x (3.54)$^{0.3}$= 459.87
N_{UD} = h_iD/K
459.87 = (h_i 6.35 X 10^{-3})/ 72.89x10^{-3}
h_i = 5269.3W/m^{2}K
U = 1/ (1/ h_i + 1/h_o) = 1/(1/5269.3 + 1/10) = 10 W/m^2

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$\Delta T_m = (\theta_1 - \theta_2)/\ln(\theta_1/\theta_2)$
$\theta_1 = T_{h1} - T_{C1} = 53 - 29.6 = 23.4$
$\theta_2 = T_{h2} - T_{C2} = 36 - 30.6 = 5.4$
$\Delta T_m = (\theta_1 - \theta_2)/\ln(\theta_1/\theta_2) = 12.3$
Q=Heat transfer rate through the condenser = U A ΔT_m
Q= $10 \times \pi \times 6.5 \times 10^{-3} \times 8.5 \times 12.3 = 21.34$W

RESULTS AND DISCUSSIONS

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.

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.

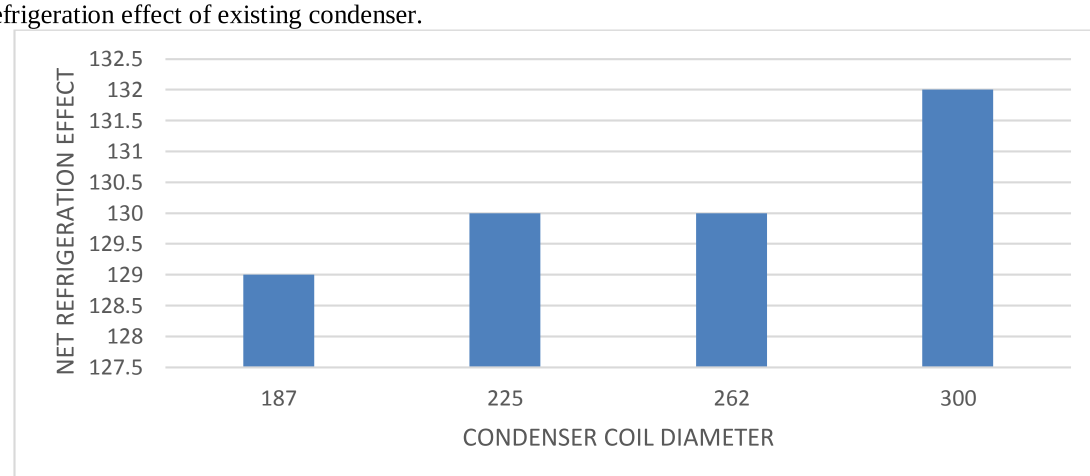

Fig.7 Effect of helical condenser coil diameter (D) on net refrigeration effect.

2. 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 **262 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.

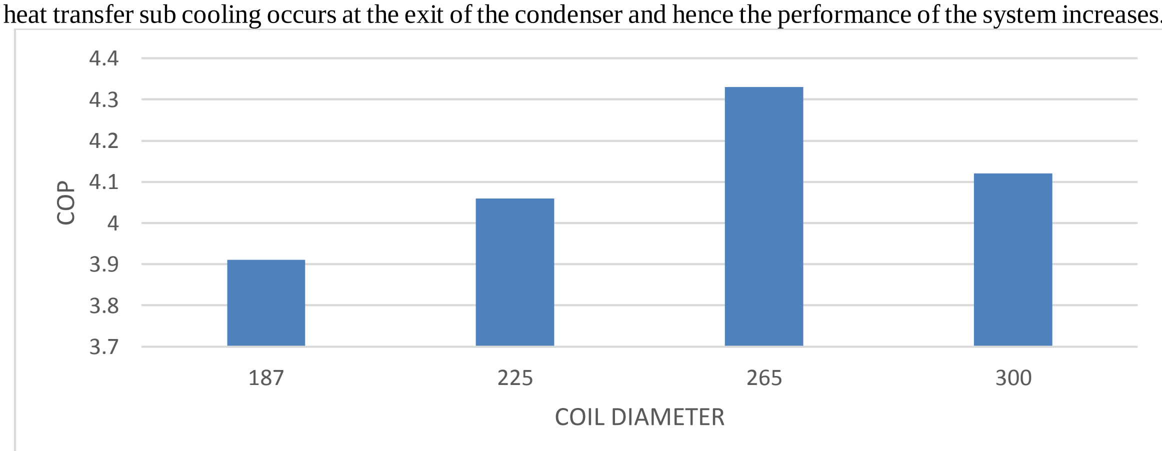

Fig.8 Effect of helical condenser coil diameter (D) on coefficient of performance.

3. 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.

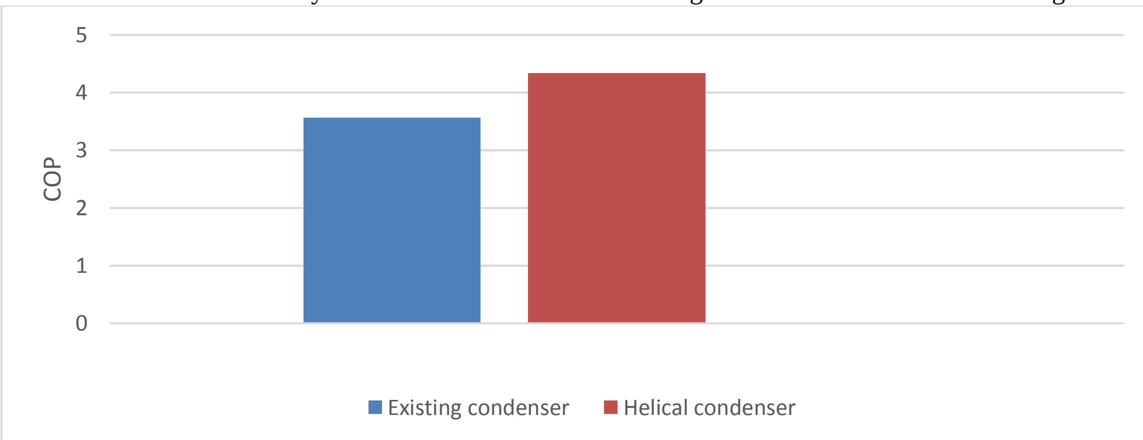

Fig.9 Effect of condenser on coefficient of performance

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 165 liters capacity.

By incorporating the helical condenser in the refrigerator the COP enhanced by 0.77, as a result of 6 kj/kg increase in refrigeration effect and 5 kj/kg reduction in compressor work and increase in heat rejection 1kj/kg. The performance of helical condenser is also changed at different diameters so, design of diameter helical condenser coil plays key 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 by change the shape of existing design to helical condenser the coefficient of performance is increased and heat transfer rate is increased and maximum value of heat transfer is attain at 10.5 inch (262mm) coil diameter (D).

REFERENCES

1. Ali M E (1998), "Laminar natural convection from constant heat flux helicoidally tubes", *International Journal of Heat and Mass Transfer*, Vol. 41, pp. 2175-2182.
2. Ashish Kumar Paharia and Gupta R C, *Effect of sub cooling and superheating on vapour compression refrigeration systems using r-22 alternative refrigerants.*
3. Bukola Olalekan Bolaji, *Effects of Sub-Cooling on the Performance of R12Alternatives in a Domestic RefrigerationSystem.*
4. Churchill S W (1977), "Friction factorequation spans all fluid flow regimes", *Chemical Engineering*, Vol. 84, pp. 91-92.
5. CP ARORA (2003), *Refrigeration and Air Conditining.*
6. Domkundhar and Domkundar, Refrigeration And Air conditioning.
7. Dalkilic A S and Wongwises S,A performance comparison of vapour compression refrigeration system using various alternative fuels.
8. Manohar Prasad, *Refrigeration and Airconditining Systems*.
9. Nagalakshmi K and Maruthiprasad YadavG, The Design and Performance Analysisof Refrigeration System Using R12 & R134aRefrigerants.

10. Tiwari A C and Shyam kumar Barode, Performance Analysis of a Vapour Compression Refrigeration Systems Using Hydro-fluorocarbon Refrigerants.
11. Akintunde, M.A. 2004b. Experimental Investigation of The performance of Vapour Compression Refrigeration Systems. Federal University of Technology, Akure, Nigeria.
12. Kays WM, London AL. Compact heat exchangers. New-York: McGraw-Hill, 1984.
13. Seshimo Y, Fujii M. An experimental study of the performanceof plate fin and tube heat exchangers at lowReynolds numbers. ASME/JSME Thermal Engineering Proceedings, ASME 1991;4:449–54..

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