Experimental Study of A Domestic Refrigerator/Freezer Using Variable Condenser Length

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Abstract— The condenser design plays a very important role in the performance of a vapour compression refrigeration system. Optimized design is 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.

In my experimental work, it is proposed to optimize condenser length for domestic refrigerator of 165 litres capacity. It may give a chance to find a different length other than existing length will give better performance and concluded that the optimum length of coil is 7.01m

Keywords — Vapour Compression Refrigeration System, Refrigerant, Optimized design.

I. INTRODUCTION

The first mechanically produced cooling system was developed in England in 1834. The process later became known as vapour compression. After availability of electricity automatic refrigeration system was developed in 1897. Basically a refrigeration or air conditioning is nothing more than a heat pump whose job is to remove heat from a lower temperature source and reject heat to high temperature sink. The Vapour Compression Refrigeration Cycle is a process that cools an enclosed space to a temperature lower than the surroundings. To accomplish this, heat must be removed from the enclosed space and dissipated into the surroundings. However, heat tends to flow from an area of high temperature to that of a lower temperature.

During the cycle refrigerant circulates continuously through four stages. The first stage is called Evaporation and it is here that the refrigerant cools the enclosed space by absorbing heat. Next, during the Compression stage, the pressure of the refrigerant is increased, which raises the temperature above that of the surroundings. As this hot refrigerant moves through the next stage, Condensation, the natural direction of heat flow allows the release of energy into the surrounding air. Finally, during the Expansion phase, the refrigerant temperature is lowered by what is called the auto refrigeration effect. This cold refrigerant then begins The Evaporation stage again, removing more heat from the enclosed space. Each of the four stages will now be revisited in detail, explaining the physical changes that occur in the refrigerant and the devices used to accomplish these changes. A visual representation of the cycle is displayed below with the explanation of each stage.

A. Refrigeration System: Working Principle and Construction

Refrigeration system is based upon the Clausius statement of second law of thermodynamics. This statement shows, "It is impossible to construct a device which, operating in a cycle, will produce no affect other than the transfer of heat from a cooler to a hotter body". The construction of

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vapour compression refrigeration system is illustrated in figure 1. A vapour compression cycle is used in most household refrigerators, refrigerator–freezers and freezers. In this cycle, a circulating refrigerant such as R134a enters a compressor as low-pressure vapour at or slightly above the temperature of the refrigerator interior. The vapour is compressed and exits the compressor as high-pressure superheated vapour. The superheated vapour travels under pressure through coils or tubes comprising "the condenser", which are passively cooled by exposure to air in the room. The condenser cools the vapour, which liquefies. As the refrigerant leaves the condenser, it is still under pressure but is now only slightly above room temperature. This liquid refrigerant is forced through a metering or throttling device, also known as an expansion valve (essentially a pin-hole sized constriction in the tubing) to an area of much lower pressure. The sudden decrease in pressure results in explosive-like flash evaporation of a portion (typically about half) of the liquid.

The latent heat absorbed by this flash evaporation is drawn mostly from adjacent still-liquid refrigerant, a phenomenon known as "auto-refrigeration". This cold and partially vaporized refrigerant continues through the coils or tubes of the evaporator unit. A fan blows air from the refrigerator or freezer compartment ("box air") across these coils or tubes and the refrigerant completely vaporizes, drawing further latent heat from the box air. This cooled air is returned to the refrigerator or freezer compartment, and so keeps the box air cold. Note that the cool air in the refrigerator or freezer is still warmer than the refrigerant in the evaporator. Refrigerant leaves the evaporator, now fully vaporized and slightly heated, and returns to the compressor inlet to continue the cycle.

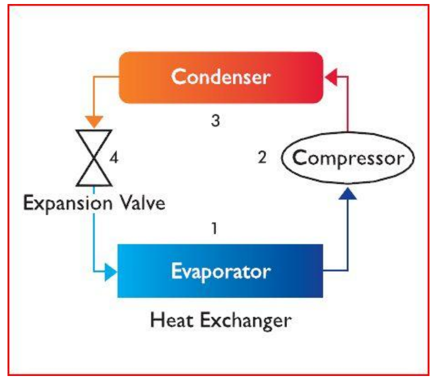


Figure 1: Basic cycle of domestic refrigeration system

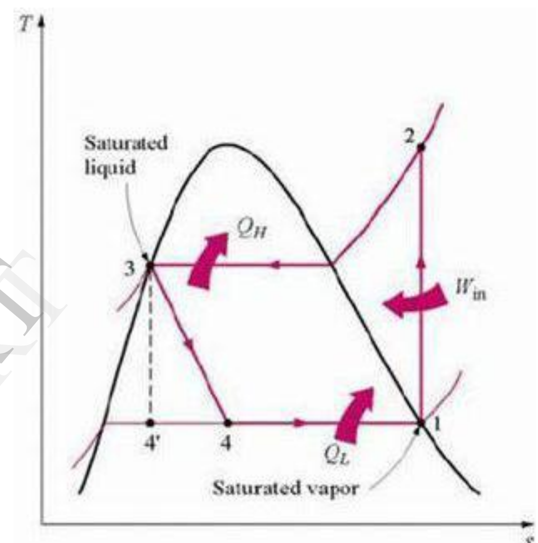


Figure 2 : T-S Diagram for the Ideal Vapor Compression Refrigeration Cycle

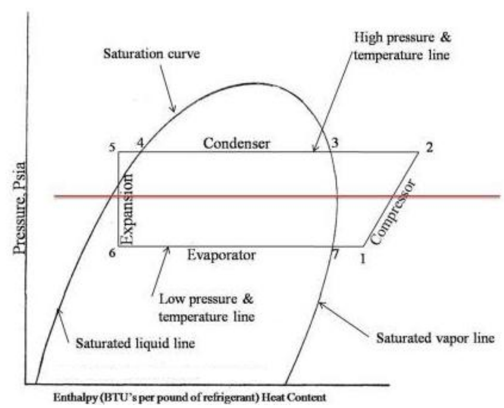


Figure 3: Pressure-enthalpy graph for vapour compression refrigeration system

Process 1–2: Isentropic compression in compressor.

Process 2–3: Constant pressure heat rejection in condenser.

Process 3–4: Isenthalpic expansion in expansion device.
Process 4–1: Constant pressure heat absorption in evaporator.

II. LITERATURE REVIEW

R. Cabello, E. Torrella and J. Navarro-Esbri [1], have analyzed the performance of a vapour compression refrigeration system using three different working fluids (R134a, R407c and R22). The operating variables are the evaporating pressure, condensing pressure and degree of superheating at the compressor inlet. They analyzed that the power consumption decreases when compression ratio increases with R22 than using the other working fluids.

B.O. Bolaji et al[2] investigated experimentally the performances of three ozone friendly Hydrofluorocarbon (HFC) refrigerants R12, R152a and R134a. R152a refrigerant found as a drop in replacement for R134a in vapour compression system.

B.O. Bolaji[3] discussed the process of selecting environmental-friendly refrigerants that have zero ozone depletion potential and low global warming potential. R23 and R32 from methane derivatives and R152a, R143a, R134a and R125 from ethane derivatives are the emerging refrigerants that are non toxic, have low flammability and environmental-friendly. These refrigerants need theoretical and experimental analysis to investigate their performance in the system.

James M. Calm [4], has studied the emission and environmental impacts of R11, R123, R134a due to leakage from centrifugal chiller system. He also investigated the total impact in form of TEWI and change in system efficiency or performance due to charge loss. He also summarized the methods to reduce the refrigerant losses by the system like design modifications, improvement in preventive maintenance techniques, use of purge system for refrigerant vapour recovery, servicing and lubricant changing in system.

Samira Benhadid-Dib and Ahmed Benzaoui [5], have showed that the uses of halogenated refrigerants are harmful for environment and the use of "natural" refrigerants become a possible solution. Here natural refrigerants are used as an alternative solution to replace halogenated refrigerants. The solution to the environmental impacts of refrigerant gases by a gas which contains no chlorine no fluorine and does not reject any CO2 emissions in the atmosphere. The researchers showed that emissions have bad effects on our environment. They also concerned by a contribution to the reduction of greenhouse gases and by the replacement of the polluting cooling fluids (HCFC).

Eric Granryd [6], has enlisted the different hydrocarbons as working medium in refrigeration system. He studied the different safety standards related to these refrigerants. He showed the properties of hydrocarbons (i.e. no ODP and negligible GWP) that make them interesting refrigeratingalternatives for energy efficient and environmentally friendly. But safety precautions due to flammability must be seriously taken into account.

Y. S. Lee and C. C. Su [7], have studied the performance of VCRS with isobutene and compare the results with R12 and R22. They used R600a about 150 g and set the refrigeration temperature about 4 °C and -10 °C to maintain the situation of cold storage and freezing applications. They used 0.7 mm internal diameter and 4 to 4.5 m length of capillary tube for cold storage applications and 0.6 mm internal diameter and 4.5 to 5 m length of capillary tube for freezing applications.

They observed that the COP lies between 1.2 and 4.5 in cold storage applications and between 0.8 and 3.5 in freezing applications. They also observed that the system with two capillary tubes in parallel performs better in the cold storage and air conditioning applications, whereas that with a single tube is suitable in the freezing applications.

III. EXPERIMENTAL STUDY

A. Calculations and Analysis with Existing Dimensions

Condenser Sizes

Length 6.1m (20feet)

Diameter 6.4mm

Evaporator Sizes

Length 7.26m

Diameter 6.4mm

Capillary tube Sizes

Length 3.35m

Diameter 0.8mm

Ambient Temperature = 31.5^0 C

Temperatures

Compressor suction temperature $T_1 = 29.5^0$ C

Compressor Discharge Temperature $T_2 = 72.2^0$ C

Condensing Temperature T3 = 39.4^0 C

Evaporator Temperature T_4 = 0.10 C

Pressure

Compressor Suction pressure P_1= 16.5 psi

Compressor Discharge Pressure p_2= 165 psi

Condensing Pressure P_3= 156 psi

Evaporator pressure P_4= 17 psi

Convert all the pressure in to Bar

Conversion pressure Unit -1.psi = 0.069 bar

P_1 =16.5 x 0.06 = 1.13 bar

P_2 =165 x 0.069 = 11.38 bar

P_3 = 156 x 0.069 = 10.76 bar

P_4 = 17 x 0.069 = 1.173 bar

Current and Voltage

Current = 1.1 Amps

Voltage = 230 Volts

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.

h_1 = 426.510 KJ/Kg

h_2 = 453.88 KJ/Kg

h_3 = 263.284 KJ/Kg

h_4 = 199.38 KJ/Kg

Calculations Performance Parameters

1. Net Refrigerating Effect (NRE)
   = $(h_1 - h_4)$
   = (426.51 – 199.38)
   = 227.13 KJ/Kg
2. Circulating rate to obtain one tone of Refrigeration, kg/min.
   m_r= 210/NRE = 210/227
   = 0.924Kg/min
3. Heat of compression = $(h_2 - h_1)$
   = (453.88 – 426.51)
   = 27.37 KJ/Kg
4. Heat Equivalent of work of compressor = m_r x $(h_2 - h_1)$
   = 0.924x (27.37)
   =25.31kJ/min
5. Compressor power =(25.31/60)
   =0.422 kW
6. Coefficient of performance (COP)
   $$= \frac{\text{Net refrigerating Effect}}{\text{Heat of Compression}}$$
   = (227.13/27.37) =8.298
7. Heat rejected in condenser
   = $(h_2 - h_3)$
   = (453.88-263.284) =190.59
8. Heat rejection Rate
   = (210/227.13) x 190.59 =176.105

9. Heat rejection factor = (176.105/210) =0.839
10. Specific volume of suction gas V_s = 0.19 m3/Kg
11. Volume of refrigerant to be handled bycompressor $V=m_r x V_s$=0.927x0.19=0.176 m3/min

12. Compression Pressure Ratio= P_d/P_s =11.38/1.13=10 bar

*B. Calculations and Analysis with Varying Dimensions (**Decreased Condenser Length**)*

Condenser Sizes

Length 5.1m (17feet)
Diameter 6.4mm

Evaporator Sizes

Length 7.26m
Diameter 6.4mm

Capillary tube Sizes

Length 3.35m
Diameter 0.8mm

Ambient Temperature = 31.5^0 C

Temperatures

Compressor suction temperature T_1= 29.1^0 C

Compressor Discharge Temperature T_2 = 70.2^0 C

Condensing Temperature T3 = 41.2^0 C

Evaporator Temperature T_4 = 0.8^0 C

Pressure

Compressor Suction pressure P_1= 16 psi

Compressor Discharge Pressure p_2= 162 psi

Condensing Pressure P_3= 157 psi

Evaporator pressure P_4= 21 psi

Convert all the pressure in to Bar

Conversion pressure Unit -1.psi = 0.069 bar

P_1 =16 x 0.06 = 1.104 bar

P_2 =162 x 0.069 = 11.17 bar

P_3 = 157 x 0.069 = 10.83 bar

P_4 = 21 x 0.069 = 1.44 bar

Current and Voltage

Current = 1.1 Amps

Voltage = 230 Volts

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.

h_1 = 426.190 KJ/Kg

h_2 = 451.567 KJ/Kg

h_3 = 260.560 KJ/Kg

h_4 = 200.566 KJ/Kg

Calculations Performance Parameters

1. Net Refrigerating Effect (NRE) = $(h_1 - h_4)$ = (426.19 – 200.566) = 225.624 KJ/Kg
2. Circulating rate to obtain one tone of Refrigeration, kg/min. m_r= 210/NRE = 210/225.624 = 0.931Kg/min

3. Heat of compression = $(h_2 - h_1)$ = (451.567 – 426.19) = 25.377 KJ/Kg
4. Heat Equivalent of work of compressor = m_r x $(h_2 - h_1)$ = 0.931x (25.377) =23.619kJ/min
5. Compressor power =(23.619/60) =0.394 kW
6. Coefficient of performance (COP)

   $$= \frac{\text{Net refrigerating Effect}}{\text{Heat of Compression}}$$

   = (225.624/25.377) =8.89
7. Heat rejected in condenser = $(h_2 - h_3)$ =(451.567-260.565) =191.007 kJ/Kg

8. Heat rejection Rate
 = (210/225.624) x 191.007 =177.78
9. Heat rejection factor =
 (177.78/210) =0.846
10. Specific volume of suction gas
 $V_s = 0.19\ m^3/Kg$
11. Volume of refrigerant to be handled bycompressor
 $V=m_r x V_s$=0.931x0.19=0.177 m3/min

12. Compression Pressure Ratio= P_d/P_s
 =11.17/1.104=10.117 bar

*C. Calculations and Analysis with Varying Dimensions (**Increased Condenser Length**)*

Condenser Sizes

Length **7.01m (23feet)**

Diameter 6.4mm

Evaporator Sizes

Length 7.26m

Diameter 6.4mm

Capillary tube Sizes

Length 3.35m

Diameter 0.8mm

Ambient Temperature = 31.5^0 C

Temperatures

Compressor suction temperature T_1= 36.5^0 C

Compressor Discharge Temperature T_2 = 75.6^0 C

Condensing Temperature T3 = 38.5^0 C

Evaporator Temperature T_4 = -4.5^0 C

Pressure

Compressor Suction pressure P_1= 9 psi

Compressor Discharge Pressure p_2= 157 psi

Condensing Pressure P_3= 149 psi

Evaporator pressure P_4= 13 psi

Convert all the pressure in to Bar

Conversion pressure Unit -1.psi = 0.069 bar

P_1 =9 x 0.069 = 0.621 bar

P_2 =157 x 0.069 = 10.83 bar

P_3 = 149 x 0.069 = 10.30 bar

P_4 = 13 x 0.069 = 0.897 bar

Current and Voltage

Current = 1.1 Amps

Voltage = 230 Volts

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.

h_1 = 430.77 KJ/Kg

h_2 = 458.676 KJ/Kg

h_3 = 258.6 KJ/Kg

h_4 = 193.23 KJ/Kg

Calculations Performance Parameters

1. Net Refrigerating Effect (NRE)
 = $(h_1 – h_4)$
 = (430.77 – 193.23)
 = 237.54 KJ/Kg
2. Circulating rate to obtain one tone of Refrigeration, kg/min.
 m_r= 210/NRE = 210/237.54
 = 0.884Kg/min

3. Heat of compression = $(h_2 – h_1)$
 = (458.676 – 430.77)
 = 27.906 KJ/Kg
4. Heat Equivalent of work of compression = m_r x $(h_2 – h_1)$
 = 0.884x (27.906)
 =24.667 kJ/min
5. Compressor power =(24.667 /60)
 =0.411 kW
6. Coefficient of performance (COP)

$$= \frac{\text{Net refrigerating Effect}}{\text{Heat of Compression}}$$

 = (237.54/27.906) =8.51

7. Heat rejected in condenser
 $= (h_2 - h_3)$
 $= (458.676 - 258.6) = 200.076$

8. Heat rejection Rate
 $= (210/237.54) \times 200.076 = 176.88$
9. Heat rejection factor $=$ $(176.88/210) = 0.842$
10. Specific volume of suction gas
 $V_s = 0.2$ m3/Kg
11. Volume of refrigerant to be handled bycompressor
 $V = m_r x V_s = 0.884 x 0.2 = 0.177$ m3/min

12. Compression Pressure Ratio= P_d/P_s
 $= 10.83/0.621 = 17.43$ bar

IV. THEORITICAL STUDY

Theoretical calculations for determining of length of condenser for air cooling cross flow type

Size of Condenser Tube

Outer diameter of tube = 6.4 mm
Inner diameter of tube = 4.6 mm
Condensing temperature= 39.4^0 C
Ambient temperature = 31.5^0 C

For air

Specific heat of air, $C_P = 1.007$ KJ/KgK
Density of air ρ = 1.1614 m^3/kg
Condenser design load

$q_{condenser} = m_r (h_2 - h_3)$
$= (0.924/60) \times (453.88 - 263.284)$
$= 2.94 = 3$ kW

Log mean temperature difference

$$LMTD = \frac{\Theta_i - \Theta_0}{\ln \frac{\Theta_i}{\Theta_0}}$$

Temperature rise of air

But $m = \rho v$

$q_{condenser} = mc_p \Delta t_a$

$3 \times 10^3 = 1.1614 \times 2 \times 1.007 \times t_a$

$t_a = 1.3^0$ C

LMTD $= 9.03^0$ C

Air side heat transfer coefficient (h_0)

Normally the air velocities over air-cooled condenser are between 2 to 6 m/sec depending upon the application.

Volume of flow = face area x Velocity of flow

Assuming a blower Capacity = 2 m^3/ sec
Face area = 2/6 = 0.333 m^2
Face diameter= $[0.333/ (\pi/4)]^{0.5} = 0.651$m
Mean temperature of air = 31. 5 + Δ ta / 2

$= 31.5 + (1.3/2) = 32.15^0$C

At 32.5^0C properties air at atmospheric pressure

ρ = 1.13 kg/m^3
μ = 18.93×10^{-3} NS / m^2
C_p = 1.009 k J/ Kg
K = 27.04 w/mK
R_e $= (\rho VD/ \mu)_{air}$

$$= \frac{(1.13 \times 6 \times 0.651)}{18.93 \times 10^{-6}}$$

$$P_r = \frac{\mu C_p}{K}$$

$$= \frac{18.93 \times 10^{-6} \times 1.009}{27.04 \times 10^{-3}} = 0.706$$

$(h_0 D/k) = N_u = 0.3 R_e^{0.6} P_r^{0.333}$
$= 0.3 (239642)^{0.6} \times (0.706)^{0.333}$
$= 440.36$

$$h_0 = \frac{440.36 \times 27.04 \times 10^{-3}}{0.651}$$

$h_0 = 18.29$ w/m^2 k

Condensing heat transfer coefficient (h_i)

Flow are of refrigerant through the tube
$= (II/4) \times (4.6/1000)^2$

$= 1.66 \times 10^{-5}$ m^2

$$Re = (\rho VD/\mu)$$

$$= \frac{m_r D_i}{\mu_i A_i}$$

Refrigerant at properties at 42^0 C

$\mu = 0.574 \times 10^{-3}$ NS/m^2

$C_p = 1510$ J/kg k

$K = 73.9 \times 10^{-3}$ w/m K

$$Re = \frac{(0.924/60) \times (4.6/1000)}{(0.574 \times 10^{-3}) \times (1.66 \times 10^{-5})}$$

$$= 7434$$

$$P_r = \frac{\mu Cp}{K} = \frac{0.574 \times 10^{-3} \times 1510}{73.9 \times 10^{-3}} = 11.73$$

$$(h_i D/k) = N_u = 0.026 \times R_e^{0.8} p_r^{0.4}$$

$$= 0.026 \times (7434)^{0.8} \times (11.73)^{0.4}$$

$$= 87$$

$$h_i = \frac{87 \times (73.9 \times 10^{-3})}{(4.6/1000)}$$

$h_i = 1398$ w/m^2 k

Overall heat transfer coefficient

Neglect metal resistance, and scale effect. Since refrigerant to condensing insider the pipe, denoting subscript "f" for fin;

$$\frac{1}{U_f A_f} = \frac{1}{h_0 A_f} + \frac{1}{h_f A_i}$$

Finned surface to outside bare are a ratio 20 (let us consider)

$(1/u_f) = (1/h_0) + (1/h_I) \times (A_f/A_0)(D_0/Di)$

$(1/u_f) = (1/18.29) + (1/1398) \times (20) \times (6.4/4.6)$

$u_f = (1/0.0745) = 13.408$ w/m^2 k

Calculations for Condenser Length

__Case: 1__

Q = UA (LMTD)

Surface area of fins:

$$= \frac{Q}{U_f(LMTD)} = \frac{224}{13.408 \times (9.03)} = 1.85 \text{ m}^2$$

Coil outside bare surface area of tube (A_s)

$A_s = (1.85/15) = 0.1233$ m^2

$$\text{Length of tube (L)} = \frac{A_s}{\Pi D} = \frac{0.1233}{\Pi (6.4/1000)}$$

$$= 6.13 \text{ m}$$

__Case: 2__

Q = UA (LMTD)

Surface area of fins:

$$= \frac{Q}{U_f(LMTD)} = \frac{225}{13.408 \times (9.03)} = 1.858 \text{ m}^2$$

Coil outside bare surface area of tube (A_s)

$A_s = (1.858/15) = 0.123$ m^2

$$\text{Length of tube (L)} = \frac{A_s}{\Pi D} = \frac{0.123}{\Pi (6.4/1000)}$$

$$= 6.16 \text{ m}$$

__Case: 3__

Q = UA (LMTD)

Surface area of fins:

$$= \frac{Q}{U_f(LMTD)} = \frac{227}{13.408 \times (9.03)} = 1.874 \text{ m}^2$$

Coil outside bare surface area of tube (A_s)

$A_s = (1.874/15) = 0.125$ m^2

$$\text{Length of tube (L)} = \frac{A_s}{\Pi D} = \frac{0.124}{\Pi (6.4/1000)}$$

$$= 6.16 \text{ m}$$

Case: 4

$Q = UA\,(LMTD)$

Surface area of fins:

$$= \frac{Q}{U_f\,(LMTD)} = \frac{237}{13.408 \times (9.03)} = 1.957\ m^2$$

Coil outside bare surface area of tube (A_s)

$A_s = (1.957/15) = 0.1304\ m^2$

$$\text{Length of tube (L)} = \frac{A_s}{\Pi D} = \frac{0.133}{\Pi\,(6.4/1000)}$$

$$= 6.49\ m$$

Case: 5

$Q = UA\,(LMTD)$

Surface area of fins:

$$= \frac{Q}{U_f\,(LMTD)} = \frac{246}{13.408 \times (9.03)} = 2.0318\ m^2$$

Coil outside bare surface area of tube (A_s)

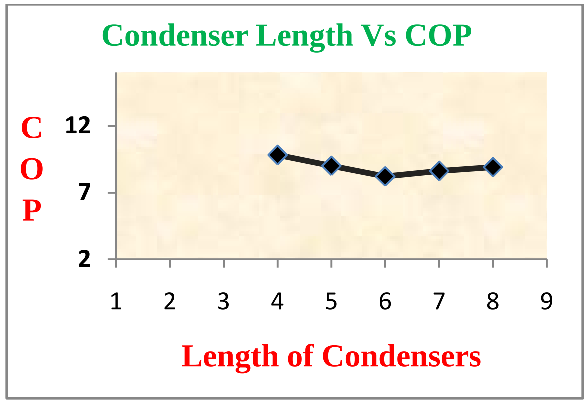
$A_s = (2.0318/15) = 0.135\ m^2$

$$\text{Length of tube (L)} = \frac{A_s}{\Pi D} = \frac{0.135}{\Pi\,(6.4/1000)}$$

$$= 6.74\ m$$

V. RESULTS AND DISCUSSIONS

The performance of Vapour Compression Refrigeration Cycle, operating the experimental domestic refrigerator varies considerably with the length of condenser. The results are plotted on graphs. The relationship between length of condenser and performance parameters have been compared are shown in the following Graphs.



Graph 6.1. Lengths of Condenser Vs COP

The length of condenser increases COP is gradually decreases. It starts to increase at 6.1 meter length of condenser.

Net Refrigerating Effect slightly increases as the of condenser length increases [Graph 6.2].

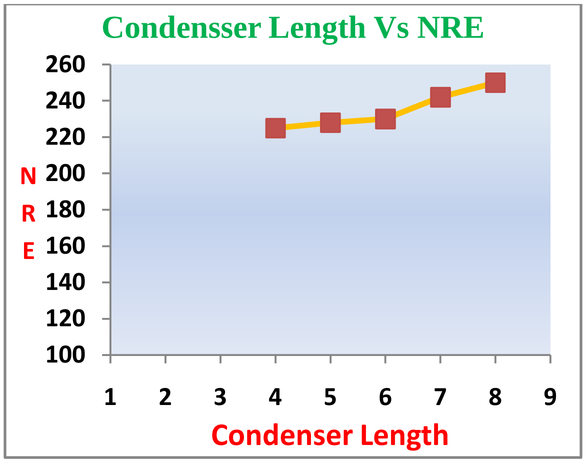
Heat Rejection in Condenser increases as the length of condenser increases. It starts to slightly increase at 6.1 meters length of condenser [Graph 6.3].

Compressor work increases as the length of condenser increases [Graph 6.4].

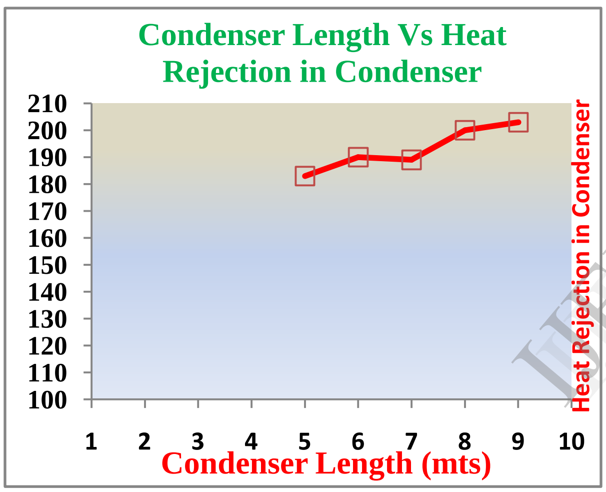
The mass flow rate of refrigerant decreases as the length of condenser increases. [Graph 6.5].

Heat rejection condenser increases as the length of condenser increases. It decreases at 6.1 meters of condenser [Graph 6.6].

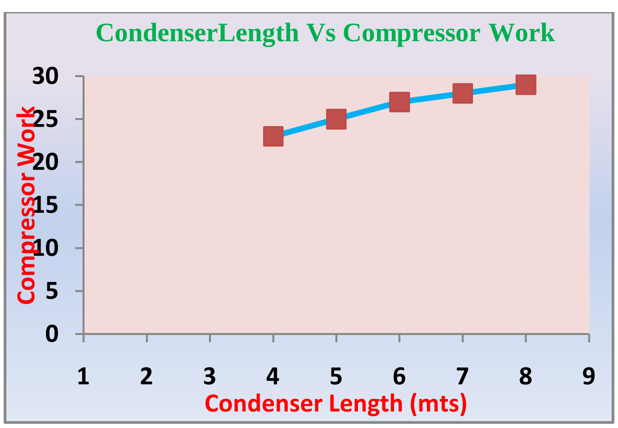
For lower condensing temperature, the length of condenser required is more. As the condensing temperature increases, the requirement of condenser length decreases sharply. Hence condensing temperature influences the length of the condenser [Graph 6.7].



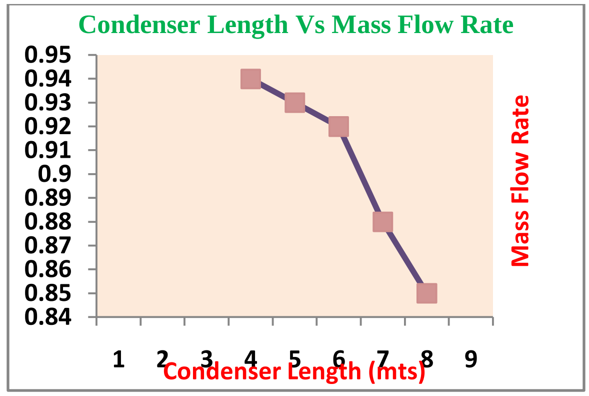
Graph 6.2. Length of Condenser Vs Net RefrigeratingEffect



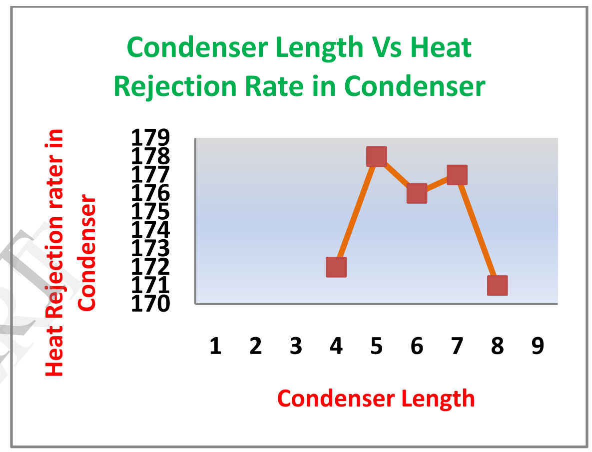
Graph 6.3. Length of Condenser Vs Heat Rejection in Condenser



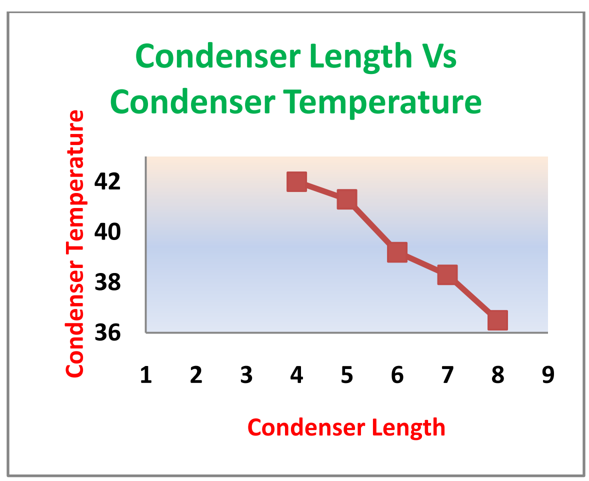
Graph 6.4. Length of Condenser Vs compressor work.



Graph 6.5. Length of Condenser Vs Mass flow rate of Refrigerant



Graph 6.6. Length of Condenser Vs Heat Rejection in Condenser:



Graph 6.7. Length of Condenser Vs Condensing Temperature:

VI. CONCLUSION

In the present work, the length of the condenser is optimized for a vapour compression refrigeration system used for a domestic refrigeration of 165 Litres capacities, through experimental investigation.

Theoretical computation are also made and compared and found that the optimum length of coil is 7.01 m instead of standard value 6.1m.

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