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. Optimised 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.


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
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 refrigerating alternatives 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

<table>
<thead>
<tr>
<th>Condenser Sizes</th>
<th>Length</th>
<th>Diameter</th>
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<tbody>
<tr>
<td></td>
<td>6.1 m (20 feet)</td>
<td>6.4 mm</td>
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<tr>
<th>Evaporator Sizes</th>
<th>Length</th>
<th>Diameter</th>
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<tbody>
<tr>
<td></td>
<td>7.26 m</td>
<td>6.4 mm</td>
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<thead>
<tr>
<th>Capillary tube Sizes</th>
<th>Length</th>
<th>Diameter</th>
<th>Ambient Temperature = 31.5°C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>3.35 m</td>
<td>0.8 mm</td>
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</table>

Temperatures
Compressor suction temperature \( T_1 = 29.5°C \)
Compressor Discharge Temperature \( T_2 = 72.2°C \)
Condensing Temperature \( T_3 = 39.4°C \)
Evaporator Temperature \( T_4 = 0.10°C \)

Pressure
Compressor Suction pressure \( P_1 = 16.5 \text{ psi} \)
Compressor Discharge Pressure \( P_2 = 165 \text{ psi} \)
Condensing Pressure \( P_3 = 156 \text{ psi} \)
Evaporator pressure \( P_4 = 17 \text{ psi} \)

Convert all the pressure in to Bar

<table>
<thead>
<tr>
<th></th>
<th>( P_1 )</th>
<th>( P_2 )</th>
<th>( P_3 )</th>
<th>( P_4 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conversion pressure Unit -1 psi = 0.069 bar</td>
<td>16.5 x 0.06 = 1.13 bar</td>
<td>165 x 0.069 = 11.38 bar</td>
<td>156 x 0.069 = 10.76 bar</td>
<td>17 x 0.069 = 1.173 bar</td>
</tr>
</tbody>
</table>

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.
\( h_1 = 426.510 \text{ KJ/Kg} \)
\( h_2 = 453.88 \text{ KJ/Kg} \)
\( h_3 = 263.284 \text{ KJ/Kg} \)
\( h_4 = 199.38 \text{ KJ/Kg} \)

Calculations Performance Parameters

1. Net Refrigerating Effect (NRE) \( = (h_1 - h_4) \)
\( = (426.51 - 199.38) \)
\( = 227.13 \text{ KJ/Kg} \)

2. Circulating rate to obtain one tone of Refrigeration, kg/min.
\( m_r = \frac{210}{NRE} = \frac{210}{227} \)
\( = 0.924 \text{ Kg/min} \)

3. Heat of compression \( = (h_2 - h_1) \)
\( = (453.88 - 426.51) \)
\( = 27.37 \text{ KJ/Kg} \)

4. Heat Equivalent of work of compressor \( = m_r \times (h_2 - h_1) \)
\( = 0.924 \times (27.37) \)
\( = 25.31 \text{ KJ/min} \)

5. Compressor power \( = \frac{25.31}{60} \)
\( = 0.422 \text{ kW} \)

6. Coefficient of performance (COP)
Net refrigerating Effect \( = \frac{h_1 - h_4}{h_2 - h_1} \)
\( = \frac{227.13}{27.37} \)
\( = 8.298 \)

7. Heat rejected in condenser \( = (h_2 - h_3) \)
\( = (453.88 - 263.284) \)
\( = 190.59 \text{ KJ/Kg} \)

8. Heat rejection Rate \( = (\frac{210}{227.13}) \times 190.59 \)
\( = 176.105 \text{ KJ/min} \)
9. Heat rejection factor = \frac{176.105}{210} = 0.839
10. Specific volume of suction gas \( V_s = 0.19 \text{ m}^3/\text{Kg} \)
11. Volume of refrigerant to be handled by compressor
   \[ V = m_r \times V_s = 0.927 \times 0.19 = 0.176 \text{ m}^3/\text{min} \]

12. Compression Pressure Ratio = \frac{P_d}{P_s} = \frac{11.38}{1.13} = 10 \text{ bar}

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

**Condenser Sizes**
- Length: 5.1 m (17 feet)
- Diameter: 6.4 mm

**Evaporator Sizes**
- Length: 7.26 m
- Diameter: 6.4 mm

**Capillary tube Sizes**
- Length: 3.35 m
- Diameter: 0.8 mm

Ambient Temperature = 31.5^0 \text{ C}

**Temperatures**
- Compressor suction temperature \( T_1 = 29.1^0 \text{ C} \)
- Compressor Discharge Temperature \( T_2 = 70.2^0 \text{ C} \)
- Condensing Temperature \( T_3 = 41.2^0 \text{ C} \)
- Evaporator Temperature \( T_4 = 0.8^0 \text{ C} \)

**Pressure**
- Compressor Suction pressure \( P_1 = 16 \text{ psi} \)
- Compressor Discharge Pressure \( P_2 = 162 \text{ psi} \)
- Condensing Pressure \( P_3 = 157 \text{ psi} \)
- Evaporator pressure \( P_4 = 21 \text{ psi} \)

**Convert all the pressure in to Bar**
- Conversion pressure Unit -1 psi = 0.069 bar
  - \( P_1 = 16 \times 0.069 = 1.104 \text{ bar} \)
  - \( P_2 = 162 \times 0.069 = 11.17 \text{ bar} \)
  - \( P_3 = 157 \times 0.069 = 10.83 \text{ bar} \)
  - \( P_4 = 21 \times 0.069 = 1.44 \text{ bar} \)

**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.

\[ h_1 = 426.190 \text{ KJ/Kg} \]
\[ h_2 = 451.567 \text{ KJ/Kg} \]
\[ h_3 = 260.560 \text{ KJ/Kg} \]
\[ h_4 = 200.566 \text{ KJ/Kg} \]

**Calculations Performance Parameters**

1. Net Refrigerating Effect (NRE) = \( (h_1 - h_4) = (426.19 - 200.566) = 225.624 \text{ KJ/Kg} \)

2. Circulating rate to obtain one tone of Refrigeration, kg/min.
   \[ m_r = \frac{210}{225.624} = 0.931 \text{ Kg/min} \]

3. Heat of compression = \( (h_2 - h_1) = (451.567 - 426.19) = 25.377 \text{ KJ/Kg} \)

4. Heat Equivalent of work of compressor = \( m_r \times (h_2 - h_1) = 0.931 \times 25.377 = 23.619 \text{ KJ/min} \)

5. Compressor power = \( \frac{23.619}{60} = 0.394 \text{ kW} \)

6. Coefficient of performance (COP)
   Net refrigerating Effect
   \[ = \frac{225.624}{25.377} = 8.89 \]

7. Heat rejected in condenser
   \[ = (h_2 - h_3) = (451.567 - 260.565) = 191.007 \text{ kJ/Kg} \]
8. Heat rejection Rate
\[ = \frac{210}{225.624} \times 191.007 = 177.78 \]
9. Heat rejection factor = \( \frac{177.78}{210} = 0.846 \)
10. Specific volume of suction gas \( V_s = 0.19 \text{ m}^3/\text{Kg} \)
11. Volume of refrigerant to be handled by compressor
\[ V = m_r \times V_s = 0.931 \times 0.19 = 0.177 \text{ m}^3/\text{min} \]
12. Compression Pressure Ratio = \( P_d/P_s \)
\[ = \frac{11.17}{1.104} = 10.117 \text{ bar} \]

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

Condenser Sizes
Length 7.01m (23 feet)
Diameter 6.4mm

Evaporator Sizes
Length 7.26m
Diameter 6.4mm

Capillary tube Sizes
Length 3.35m
Diameter 0.8mm
Ambient Temperature = 31.50°C

Temperatures
Compressor suction temperature \( T_1 = 36.5^0\text{C} \)
Compressor Discharge Temperature \( T_2 = 75.6^0\text{C} \)
Condensing Temperature \( T_3 = 38.5^0\text{C} \)
Evaporator Temperature \( T_4 = -4.5^0\text{C} \)

Pressure
Compressor Suction pressure \( P_1 = 9 \text{ psi} \)
Compressor Discharge Pressure \( P_2 = 157 \text{ psi} \)
Condensing Pressure \( P_3 = 149 \text{ psi} \)
Evaporator pressure \( P_4 = 13 \text{ psi} \)

Convert all the pressure in to Bar
Conversion pressure Unit -1.psi = 0.069 bar
\( P_1 = 9 \times 0.069 = 0.621 \text{ bar} \)
\( P_2 = 157 \times 0.069 = 10.83 \text{ bar} \)
\( P_3 = 149 \times 0.069 = 10.30 \text{ bar} \)
\( P_4 = 13 \times 0.069 = 0.897 \text{ bar} \)

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.
\( h_1 = 430.77 \text{ KJ/Kg} \)
\( h_2 = 458.676 \text{ KJ/Kg} \)
\( h_3 = 258.6 \text{ KJ/Kg} \)
\( h_4 = 193.23 \text{ KJ/Kg} \)

Calculations Performance Parameters
1. Net Refrigerating Effect (NRE)
\[ = (h_1 - h_4) \]
\[ = (430.77 - 193.23) \]
\[ = 237.54 \text{ KJ/Kg} \]

2. Circulating rate to obtain one tone of Refrigeration, kg/min.
\[ m_r = \frac{210}{237.54} = 0.884 \text{Kg/min} \]

3. Heat of compression = \( h_2 - h_1 \)
\[ = (458.676 - 430.77) \]
\[ = 27.906 \text{ KJ/Kg} \]

4. Heat Equivalent of work of compression = \( m_r \times (h_2 - h_1) \)
\[ = 0.884 \times (27.906) \]
\[ = 24.667 \text{ KJ/min} \]

5. Compressor power = \( 24.667 / 60 \)
\[ = 0.411 \text{ kW} \]

6. Coefficient of performance (COP)
\[ \text{Net refrigerating Effect} = \frac{(237.54 / 27.906)}{8.51} \]
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
    \[ V_s = 0.2 \text{ m}^2/\text{Kg} \]

11. Volume of refrigerant to be handled by compressor
    \[ V = m_r \times V_s = 0.884 \times 0.2 = 0.177 \text{ m}^3/\text{min} \]

12. Compression Pressure Ratio
    \[ = \frac{P_d}{P_s} = 10.83/0.621 = 17.43 \text{ bar} \]

IV. THEORETICAL 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°C
Ambient temperature = 31.5°C

**For air**

Specific heat of air, \( C_p = 1.007\ \text{KJ/KgK} \)
Density of air \( \rho = 1.1614 \text{ m}^3/\text{Kg} \)

Condenser design load
\[ q_{\text{condenser}} = m_r \times (h_2 - h_3) \]
\[ = (0.924/60) \times (453.88 - 263.284) \]
\[ = 2.94 = 3 \text{ kW} \]

Log mean temperature difference
\[ \Delta \Theta = \frac{\Theta_i - \Theta_0}{\ln \frac{\Theta_i}{\Theta_0}} \]

**LMTD**

\[ \text{LMTD} = \frac{\Theta_i}{\ln \frac{\Theta_i}{\Theta_0}} \]

**Temperature rise of air**

\[ \Delta t_a = 1.3^0 \text{C} \]

**LMTD** = 9.03°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 \times \text{Velocity of flow}

Assuming a blower Capacity = 2 m³/sec
Face area = 2/6 = 0.333 m²
Face diameter = \( [0.333/ (\pi/4)]^{0.5} = 0.651 \text{m} \)
Mean temperature of air = 31.5 + \Delta t_a / 2

\[ = 31.5 + (1.3/2) = 32.15^0 \text{C} \]

At 32.5°C properties air at atmospheric pressure
\( \rho = 1.13 \text{ kg/m}^3 \)
\( \mu = 18.93 \times 10^{-3} \text{ NS/m}^2 \)
\( C_p = 1.009 \text{ kJ/Kg} \)
\( K = 27.04 \text{ w/mK} \)

\( R_e = \frac{\rho V D}{\mu} = \frac{1.13 \times 6 \times 0.651}{18.93 \times 10^{-6}}\]
\[ = (1.13 \times 6 \times 0.651) \]
\[ \frac{\mu C_p}{K} = 18.93 \times 10^{-6} \]

\( 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_{a} = 0.3 \times R_e^{0.6} \times P_r^{0.333} \]
\[ = 0.3 \times (239642)^{0.6} \times (0.706)^{0.333} \]
\[ = 440.36 \times 27.04 \times 10^{-3} \]
\[ h_0 = \frac{440.36 \times 27.04 \times 10^{-3}}{0.651} \]
\[ = 18.29 \text{ w/m}^2 \text{ k} \]

**Condensing heat transfer coefficient \( (h_i) \)**

Flow area of refrigerant through the tube
\[ = \frac{(II/4) \times (4.6/1000)^2}{1.3} \]
\[ = 1.66 \times 10^{-5} \text{ m}^2 \]
Re = \( \frac{\rho V D}{\mu} \)

\[ m_r D_i = \frac{1}{\mu_i A_i} \]

Refrigerant at properties at 420 C
\( \mu = 0.574 \times 10^{-3} \text{ NS/m}^2 \)
\( C_p = 1510 \text{ J/kg k} \)
\( K = 73.9 \times 10^{-3} \text{ w/m K} \)

\[ (0.924/60) \times (4.6/1000) \]
Re = \( \frac{\mu C_p}{K} \times \frac{0.574 \times 10^{-3} \times 1510}{73.9 \times 10^{-3}} \)

\[ P_r = \frac{87 \times (73.9 \times 10^{-3})}{(4.6/1000)} \]

\[ h_i = \frac{1398 \text{ w/m}^2 \text{ k}}{1398 \times 0.026 \times 0.8 \times 0.4} = 87 \times (73.9 \times 10^{-3}) \]

Overall heat transfer coefficient
Neglect metal resistance, and scale effect. Since refrigerant to condensing insider the pipe, denoting subscript “f” for fin;

\[ U_f A_f = h_0 A_f + h_f A_i \]

Finned surface to outside bare are a ratio 20 (let us consider)
\( (1/\mu) = (1/ h_0) + (1/ h_f) \times (A_f/ A_0) (D_0/ D_i) \)

\[ u_f = (1/ 0.0745) = 13.408 \text{ w/m}^2 \text{ k} \]

Calculations for Condenser Length

Case: 1
\[ Q = UA \ (\text{LMTD}) \]
Surface area of fins:
\[ Q = \frac{224}{U_f (\text{LMTD})} = 13.408 \times (9.03) \]

Coil outside bare surface area of tube \( (A_s) \)
\[ A_s = \frac{1.85/15}{0.1233} \]

Length of tube \( (L) = \frac{224}{U_f (\text{LMTD})} = 13.408 \times (9.03) \]

\[ = 6.13 \text{ m} \]

Case: 2
\[ Q = UA \ (\text{LMTD}) \]
Surface area of fins:
\[ Q = \frac{225}{U_f (\text{LMTD})} = 13.408 \times (9.03) \]

Coil outside bare surface area of tube \( (A_s) \)
\[ A_s = \frac{1.85/15}{0.1233} \]

Length of tube \( (L) = \frac{225}{U_f (\text{LMTD})} = 13.408 \times (9.03) \]

\[ = 6.16 \text{ m} \]

Case: 3
\[ Q = UA \ (\text{LMTD}) \]
Surface area of fins:
\[ Q = \frac{227}{U_f (\text{LMTD})} = 13.408 \times (9.03) \]

Coil outside bare surface area of tube \( (A_s) \)
\[ A_s = \frac{1.874/15}{0.1233} \]

Length of tube \( (L) = \frac{227}{U_f (\text{LMTD})} = 13.408 \times (9.03) \]

\[ = 6.16 \text{ m} \]
Case: 4

\[ Q = UA \text{ (LMTD)} \]

Surface area of fins:

\[ \frac{Q}{U_f \text{ (LMTD)}} = \frac{237}{13.408 \times (9.03)} = 1.957 \text{ m}^2 \]

Coil outside bare surface area of tube \((A_s)\):

\[ A_s = \frac{1.957}{15} = 0.1304 \text{ m}^2 \]

Length of tube \((L)\):

\[ L = \frac{A_s}{\text{IID II (6.4/1000)}} = \frac{0.1304}{6.49} = 6.49 \text{ m} \]

Case: 5

\[ Q = UA \text{ (LMTD)} \]

Surface area of fins:

\[ \frac{Q}{U_f \text{ (LMTD)}} = \frac{246}{13.408 \times (9.03)} = 2.0318 \text{ m}^2 \]

Coil outside bare surface area of tube \((A_s)\):

\[ A_s = \frac{2.0318}{15} = 0.135 \text{ m}^2 \]

Length of tube \((L)\):

\[ L = \frac{A_s}{\text{IID II (6.4/1000)}} = \frac{0.135}{6.49} = 6.74 \text{ m} \]

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 length of condenser 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 Refrigerating Effect

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 Rate 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.

VII. REFERENCES


