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Calibration & Comparison of Coefficient of Performance of Fridge with Normal & Fan Fitted Condenser

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Abstract—*The domestic refrigerators are generally designed on the basis of natural convection .The natural convection occurs by the condenser placed backside of refrigerator in the form of net or fins. For this a open space is require so that heat can be transferred from condenser to atmosphere effectively but it is often seen that the domestic refrigerators are generally located in kichten or just near the wall. Thus in kitchen due to large temperature of surrounding the effectiveness gets decreased due to low heat transfer rate also if the refrigerator is placed near the wall the heat transfer rate will also further decrease due to lack of open area since heat transfer rate depend upon the area thus due to following reason the refrigerator cannot work effectively and results more electricity consumption due to more work done by compressor and further decrease of refrigerator component's life keeping in mind the following problem I convert the natural convection into forced convection for this, I used separate condenser fitted with fan, the arrangement of this condenser and fan can be installed as per the location and need . But it is further suggested that the condenser should be placed on the exhaust fan of the kitchen since in all the kitchens there is a portion of exhaust fan to exhaust the fumes of cooking out. Thus no extra electricity and fan is required. Due to this forced convection heat transfer rate increase rapidly because output temperature of condenser is 20%-30% less than the natural convection whole result to increase the C.O.P. of the refrigerator further due to following effects effective cooling occurs inside the refrigerator and due to more cooling the refrigerator will not run continuously because the sensor attached will cut off the supply of compressor at desired set value of temperature by this the electricity consumption will decrease up to 20%-30% and the life of refrigerator will increase.*

1. INTRODUCTION

The system is not present in use due to arrangement of its various parts which are using additionally in it. In present, domestic refrigerators work on vapour compression cycle. In this cycle the heat is rejected through condenser or heat exchanger. In this way the rejection of heat occurs as a natural convection, due to this, refrigerating effect will obtain. Our modification in this system is that, if we perform through an arrangement of heat exchanger and exhaust fan the flow rate of heat should increase. Thus, in this way at same energy input the refrigerating effect may improve. Our future plan is to modify the domestic refrigerator that it could work on both natural convection and forced convection. In this way we can compare the Coefficient of performance for both processes and may do comparative study about both systems at different-different conditions.

2. EXPERIMENTAL PLANNING AND PROCEDURE

To prepare experimental setup various additional parts are used on the 65 liter refrigerator as listed below.

2.1 List of Parts

Following are the essential parts for the modification of our project-

1- Fan fitted condenser (Fig. 1)

Fig. (1)

2- Duct (Fig. 2)

Fig. (2)

3- Double manifold valves (Fig. 3)

Fig. (3)

4- Copper pipe
5- Flayers
6- Insulators
7- Copper constatine sensors (Fig. 4)

Fig. (4)

8- Temperature display meter (Fig. 5)

9- Selector switch (Fig. 6)

Fig. (6)

3. CALCULATION OF HEAT TRANSFER THROUGH CONDENSOR

Considering natural convection:

Following empirical correlations and dimentionless numbers are important for free or natural convection:

(i) Nusselt number, Nu = $\frac{hL}{k}$

(ii) Grasshoff number, Gr = $\frac{L^3 \beta g \Delta t}{v^2}$

(iii) Prandlt number, Pr = $\frac{\mu C_p}{k}$

Following correlations are for nusselt number laminar and turbulent flow on the basis of Gr.Pr:

Laminar Flow, Nu = 0.59 $(Gr.Pr)^{1/4}$ $(10^4 < Gr.Pr < 10^9)$

Turbulent Flow: Nu = 0.10 $(Gr.Pr)^{1/3}$ $(10^9 < Gr.Pr < 10^{12})$

Calculations of average convective heat transfer coefficient and net heat transfer rate[1] :

Considering Forced Convection:

Following observations are obtained during forced convection:

t_∞ = 32˚C, U = 3 m/s, Ts = 56˚C,

Properties of air at t_f = (32+56)/2 = 44˚C are

ρ = 1.1374 Kg/m^3, k = 0.02732 W/m˚C,

Cp = 1.005 Kj/Kg K, $\mathcal{V}$ = 16.768 × 10^{-6} m^2/s,

Pr = 0.7, X = 20 × 15.7 = 314 mm = 0.314 m,

L = 255 mm = 0.255 m

Now, Reynold number,

$$Re = \frac{U.X}{v}$$
$$= 3 \times 0.314/16.768 \times 10^{-6}$$
$$= 5.6 \times 10^4$$

Now, Average convective heat transfer coefficient,

$$\bar{h} = 0.664\ (\frac{k}{L})\ (Re)^{1/2}\ .(Pr)^{1/3}$$
$$= 0.664\ (\frac{0.02732}{0.255})\ (5.6 \times 10^4)^{1/2}\ .(0.7)^{1/3}$$
$$= 0.664 \times 0.107 \times 236.6 \times 0.887$$
$$= 14.92\ W/m^2{}^\circ C$$

Thus, Rate of heat transfer by forced Convection,

$$Qconv. = \bar{h} \times As \times (t_s - t_\infty)$$
$$= 14.92 \times 0.314 \times 0.225 \times (56 - 32)$$
$$= 46.41\ Watt$$

Considering Natural or free Convection:

Following observations are obtained during natural convection [2]:

t_∞ = 32˚C, U = 3 m/s, Ts = 56˚C,

Properties of air at t_f = (32+56)/2 = 44˚C

ρ = 1.1374 Kg/m^3, k = 0.02732 W/m˚C,

Cp = 1.005 Kj/Kg K, v = 16.768 × 10^{-6} m^2/s,

P =0.7, X=20 × 15.7 = 314 mm = 0.314 m,

Total length of tubes, L = 255 mm = 0.255 m

Now,

$$\beta = \frac{1}{T} = \frac{1}{273+t_f} = 0.00315\ K^{-1}$$

Now, Grasshoff number,

$$Gr = \frac{L^3 \beta g \Delta t}{v^2}$$
$$= \frac{8.6^3 \times 9.8 \times 0.00315 \times 44}{(16.76 \times 10^{-6})^2}$$
$$= 863.9 \times 10^{12}/280.89$$
$$= 3.07 \times 10^{12}$$

Thus the product of Gr and Pr,

$$Gr.Pr = 3.07 \times 10^{12} \times 0.7$$
$$= 2.15 \times 10^{12}$$

Now, According to the value of Nusselt number flow will be turbulent thus from empirical relation:

Nusselt Number,

$$Nu = \frac{\bar{h}L}{k} = 0.10\ (Gr.Pr)^{1/3}$$
$$= 0.10\ (2.15 \times 10^{12})^{1/3}$$
$$= 1290$$

Thus,

$$\bar{h} = \frac{k}{L} \times Nu = \frac{0.02732}{8.6} \times 1290$$
$$= 4.09\ W/m^2{}^\circ C$$

Thus, Rate of heat transfer by forced Convection,

$$Qconv. = \bar{h} \times As \times (t_s - t_\infty)$$
$$= 4.09 \times 0.314 \times 0.225 \times (56 - 32)$$
$$= 14.4\ Watt$$

Thus it is clear from that the rate of heat transfer is more in forced convection than free or natural convection and it is shown on bar diagram Fig. (7).

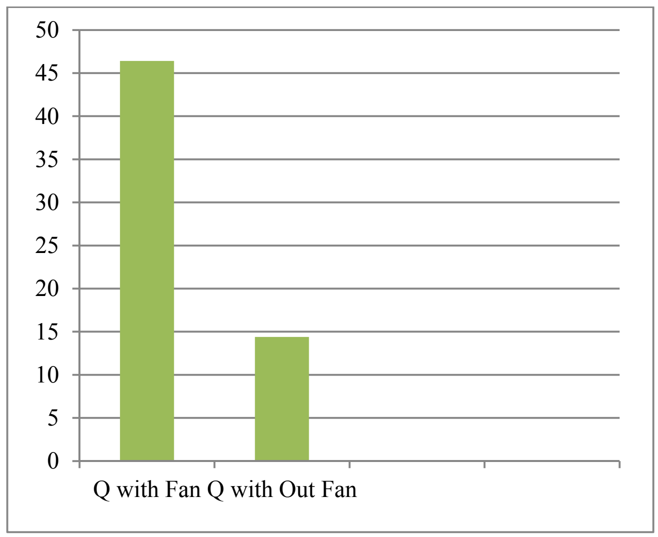

Fig. (7)

4. ANALYSIS OF COP WITH FORCED AND NATURAL CONVECTION:

4.1 Vapour Compression Cycle with Superheated Vapour after Compression:

A vapour compression cycle with superheated vapour after compression is shown on T-s and p-h diagram in Fig. 8(a) and 8(b) respectively. In this cycle the enthalpy at point 2 is found out with the help of degree of superheat. The degree of superheat may be found out by equating the entropies at points 1and 2.[3]

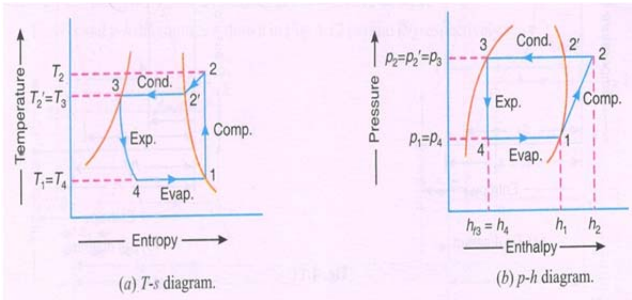

Fig. 8[5]

Now the coefficient of performance may be found out as usual from the relation,

$$\text{C.O.P.} = \frac{Refrigerating\ Effect}{Work\ Done}$$

$$= \frac{h_1 - h_{f3}}{h_2 - h_1}$$

A little consideration will show that the super heating increases the refrigerating effect and the amount of work done in the compressor. Since the amount of refrigerating effect is less than work done in the compressor, therefore, the net effect of superheating is to have low coefficient of performance.

In this cycle the cooling of superheated vapour will take place in two stages. Firstly it will be condensed to dry saturated stage at constant pressure (shown by graph 2-2') and secondly it will be condensed at constant temperature (shown by graph 2'-3).

4.2 Calculations of coefficient of performance :

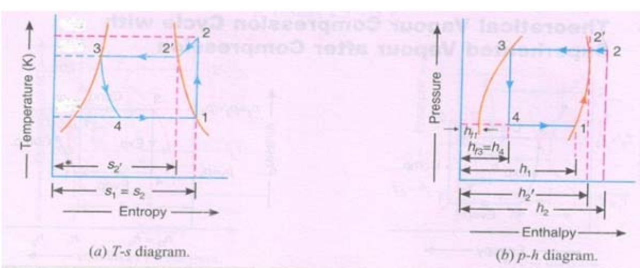

Fig. 9[5]

Considering Forced Convection:

Following observations are obtained during forced convection:

$T_1 = T_4 = -4^\circ C = -4 + 273 = 269$ K

$T_2' = T_3 = 32^\circ C = 32 + 273 = 310$ K

$T_2 = 49 + 273 = 322$ K

By the tables of properties of refrigerant R-134a

Table 1[4]

Saturation Temperature	Enthalpy (KJ/Kg)		Entropy (KJ/KgK)	
	Sat. Liquid	Sat. Vapour	Sat. Liquid	Sat. Vapour
-4˚C	44.75	244.9	0.1777	0.9213
32˚C	94.39	264.48	0.3490	0.9064

h_{f1}=44.75KJ/Kg, h_{f3} = 94.39 KJ/Kg

h_1=244.9KJ/Kg, h_2' = 264.48 KJ/Kg

S_{f1}= 0.1777 KJ/KgK S_{f3} = 0.3490 KJ/KgK

$S_1 = S_2 =$ 0.9213 KJ/KgK, S_2'= 0.9064 KJ/KgK

Let, Cp = Specific heat at constant pressure for superheated vapour,

Thus, We know that Entropy at point 2,

$$S_2 = S_2' + 2.3\ Cp \log\left(\frac{T_2}{T_1}\right)$$

$$0.9213 = 0.9064 + 2.3\ Cp \log\left(\frac{322}{310}\right)$$

2.3 Cp log1.083 = 0.0149

Cp = 0.4 KJ/KgK

Now, Enthalpy at point 2,

$$h_2 = h_2' + \text{Cp X Degree of super heat}$$

$$= h_2' + \text{Cp X } (T_2 - T_2')$$

$$= 264.48 + 0.4 \text{ X } (322\text{-}310)$$

$$= 268.8 \text{ KJ/KgK}$$

Thus, coefficient of performance of the refrigeration system,

$$\text{C. O. P.} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

$$= \frac{244.9 - 94.39}{268.8 - 244.9}$$

$$= \frac{150.51}{23.9}$$

$$= 6.29$$

4.3 Considering Free or natural Convection:

Following observations are obtained during free convection:

$T_1 = T_4 = 4^\circ C = 4 + 273 = 277$ K
$T_2' = T_3 = 48^\circ C = 48 + 273 = 321$ K
$T_2 = 70 + 273 = 343$ K

By the tables of properties of refrigerant R-134a

Table: 2[4]

Saturation Temperature	Enthalpy (KJ/Kg)		Entropy (KJ/KgK)	
	Sat. Liquid	Sat. Vapour	Sat. Liquid	Sat. Vapour
4˚C	55.35	249.53	0.2162	0.9169
48˚C	118.35	271.68	0.4243	0.9017

h_{f1}= 55.35 KJ/Kg, , h_{f3} = 118.35 KJ/Kg

h_1= 249.53 KJ/Kg, h_2' = 271.68 KJ/Kg

S_{f1}= 0.2162 KJ/KgK S_{f3} = 0.4243 KJ/KgK

$S_1 = S_2 =$ 0.9169 KJ/KgK, S_2' = 0.9017 KJ/KgK

Let, Cp = Specific heat at constant pressure for superheated vapour,

Thus, We know that Entropy at point 2,

$$S_2 = S_2' + 2.3\ Cp \log\left(\frac{T_2}{T_1}\right)$$

$$0.9169 = 0.9017 + 2.3\ Cp \log\left(\frac{343}{321}\right)$$

2.3 Cp log1.06 = 0.0152

Cp = 0.23 KJ/KgK

Now, Enthalpy at point 2,

$$h_2 = h_2' + Cp \text{ X Degree of super heat}$$
$$= h_2' + Cp \text{ X } (T_2 - T_2')$$
$$= 271.68 + 0.23 \text{ X } (343 - 321)$$
$$= 276.74 \text{ KJ/KgK}$$

Thus, coefficient of performance of the refrigeration system,

$$\text{C. O. P.} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$
$$= \frac{249.53 - 118.35}{276.74 - 249.53}$$
$$= \frac{131.18}{27.21}$$
$$= 4.82$$

Thus it is clear that Greater coefficient of performance is obtained in forced convection. Comparision of COP of natural and forced convection is as shown in bar diagram Fig. (10) below-

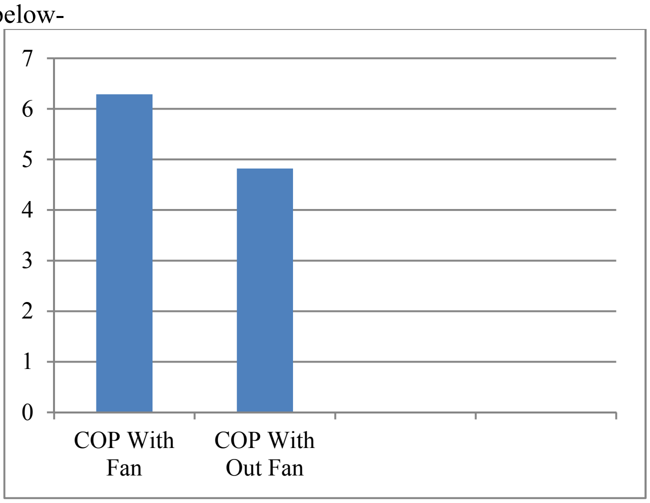

Fig. (10)

5. VIEW OF SETUP

Fig. (11)

6. CONCLUSION

This is clear from the analysis of Condenser and coefficient of performance that the heat transfer rate increased form 14.4 Watt to 46.41 Watt and coefficient of performance is improved from 4.82 to 6.29 due to forced convection as shown in bar diagrams 7 and 10.

Due to improvement in coefficient of performance the running time of unit will reduced to obtain the desired refrigerating effect thus the life of unit will also increased.

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