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EXPERIMENTAL ANALYSIS OF VAPOUR COMPRESSION REFRIGERATION SYSTEM WITH LIQUID LINE SUCTION LINE HEAT EXCHANGER BY USING R134A AND R404A

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ABSTRACT

Because of simplicity and low cost, capillary tubes are used as the expansion device in most small refrigeration and air conditioning systems. Another advantage is that capillary tubes allow high and low side pressures to equalize during the off-cycle, thereby reducing the starting torque required by the compressor. In this application the liquid line is usually placed in contact with the suction line, forming a counter flow heat exchanger. The liquid line is welded to the suction line in the lateral configuration. The temperature of the vapour refrigerant coming out from the evaporator is less than the temperature of the liquid coming out from the condenser. Before the expansion process, heat is transferred from the liquid line to the suction line. As a consequence this in turn reduces the refrigerant quality at the inlet of the evaporator and therefore increases the refrigerating capacity. The suction line exit temperature also increases, eliminating suction line sweating and preventing slugging of the compressor. The main objective of this project is to evaluate the performance of refrigerator with liquid line suction line heat exchanger for different lengths of heat exchanger by using R134a and R404a as refrigerants and compare with different lengths of liquid line- suction line heat exchanger.

KEYWORDS: *Refrigeration, COP, heat exchanger, refrigerant.*

I. INTRODUCTION

Vapour compression Refrigeration system is an improved type of air refrigeration system. 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 re-circulated over and over again to obtain refrigeration effect. Thus the vapor compression system employs a liquid refrigerant which evaporates and condenses readily. The System is a closed one since the refrigerant never leaves the system.
The coefficient of performance of a refrigeration system is the ratio of refrigerating effect to the compression work; therefore the coefficient of performance can be increased by increasing the refrigerating effect or by decreasing the compression work.
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.

Energy analysis of Refrigeration

Consider a boundary enclosing a space in which a refrigerator is placed. It is clear that some heat q_2 is given out at temperature higher than the surroundings. It is also clear that the foodstuff placed inside the refrigerator is cooled by giving out their heat to the refrigerator which in turn, so to say, absorbs heat q_1, of course at lower temperature than the surroundings. Every refrigerator is supplied with energy wither in the form of heat or electricity, that is, some work (w) is provided to it. The refrigerating device, thus is absorbing heat at lower temperature and giving out at higher temperature; this is usually not possible in our day to day life, since heat cannot flow from lower to higher temperature, but in case of a refrigerator this is achieved at the cost of energy supplied to it. For the boundary total heat given out (q_2) is equal to the total energy input in the form of heat absorbed (q_1) and the work absorbed (w) Balancing them.

For a refrigerator device, we are interested in how much heat is extracted from food stuff and how little electrical energy we spend, minimizing our power bill. The ratio of heat absorbed to the work input in the form of electric energy (w) is called coefficient of performance (COP). The ratio should be as high as possible.

$$COP = q_1/w = q_1/q_2\text{-}q_1$$

Theoretical COP is ratio of theoretical refrigerating effect (N), found from pressure heat content chart or temperature-entropy chart to the theoretical compressor work (W) or isentropic compressor work, found from the chart. Actual COP is the ratio of actual cooling effect, to the actual energy supplied to the compressor known from watt-hour reading.

II. SELECTION OF CONDENSER FOR A VCR SYSTEM

2.1 Condenser

Condenser is that component which is placed next to compressor in a vapor compression refrigeration system. It is a heat exchanger that affects heat transfer between refrigerant gas, vapor or super saturated vapor coming from compressor and cooling medium such as air or water. It removes heat absorbed by refrigerant in the evaporator and the heat of compression added in the compressor and condenses it back to liquid. The condenser abstracts the latent heat from high pressure refrigerant at the same pressure and constant temperature. For this purpose the condenser employs a cooling medium such as air or water.

2.1.1 Gross Heat Rejection

The refrigeration effect and the heat rejection rate of the system will vary depending on the actual balance to evaporator, compressor and condenser. Once the compressor and evaporator are selected to perform the required cooling it is essential that the condenser be selected on the basis of the capability of these components. Thus, the selection is made not on the heat gain, but rather on the actual load on the condenser. It is also important to anticipate overload conditions on the evaporator and compressor that may occur at start-up, pull down, or unusual loading so that these may be considered in the condenser selection.

Great variation in compression heat occurs with variation in the ratio of the suction pressure and the discharge pressure. Large compression ratio needs more compression work. However, the heat loss from the compressor body, discharge gas piping are neglected in this energy as these are very small.

Since 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 condensing medium ^{0}K

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.

2.1.2 Ambient Temperature

While selecting suitable ambient temperature in condenser sizing, it is kept in mind that the value of ambient temperature in working out the TD is not taken at a very conservative value. The realistic one is always found to be more accurate and economical in final selection of a condenser. In the normal course while sizing condenser the ambient air temperature of 38°C has been found satisfactory.

2.1.3 Condensing temperature

The condensing temperature depends on the type of refrigerant, the type of condenser, and the compressor capability. The type of refrigerant influences the condensing pressure, the superheated gas temperature, and mass flow required. The condensing pressure and the gas temperature in turn affect the power consumption and the cooling capacity of the system. It is, therefore, necessary to select a condensing temperature based on the recommendation of compressor manufactures more on realistic basis than on conservative basis in which case condenser size will work out to be too large.

2.1.4 Temperature Difference (TD)

Air cooled condensers are normally rated on the basis of initial temperature difference (TD) which is the difference between the saturated temperature of the condensing refrigerant and the entering dry bulb air temperature. The capacity of the condenser is proportional to the temperature difference so that an increase in temperature difference increases the capacity of the condenser. Therefore, a higher temperature difference means that a physically smaller condenser may be adequate for necessary heat dissipation.

The proof of a good condenser selection comes only during the operating season when satisfactory operation without problems at various conditions.

It has been the practice to select a TD of 12°C in sizing an air-cooled condenser. Still other establishes a nominal condensing temperature and subtracts the highest recorded ambient temperature to determine the temperature difference providing enough margin of safety (conservative method). In such cases 55°C condensing temperature is considered satisfactory design temperature.

A recommendation for improved application is to determine a reasonable ambient temperature (10% percent level), determine the maximum condensing temperature for the compressor's capacity, and utilize 6% of the difference as the design temperature difference:

$$TD = 0.6(CT_{max} - t_a)$$

Whereas CT_{max} = maximum condensing temperature for compressor selected, °C.

t_a=ambient air temperature at10% design level °C DB

III. HEAT EXCHANGER

Heat exchangers are devices used to transfer heat energy from one fluid to another. Typical heat exchangers experienced by us in our daily lives include condensers and evaporators used in air conditioning units and refrigerators. Boilers and condensers in thermal power plants are examples of large industrial heat exchangers. There are heat exchangers in our automobiles in the form of radiators and oil coolers. Heat exchangers are also abundant in chemical and process industries.

There is a wide variety of heat exchangers for diverse kinds of uses, hence the construction also would differ widely. However, in spite of the variety, most heat exchangers can be classified into some common types based on some fundamental design concepts. We will consider only the more common types here for discussing some analysis and design methodologies.

Basic Heat Exchanger Flow Arrangements

Two Basic flow arrangements are as shown in Figure 1. Parallel and counter flow provides alternative arrangements for certain specialized applications. In parallel flow both the hot and cold streams enter the heat exchanger at the same end and travel to the opposite end in parallel streams. Energy is transferred along the length from the hot to the cold fluid so the outlet temperatures asymptotically approach each other. In a counter flow arrangement, the two streams enter at opposite ends of the heat exchanger and flow in parallel but opposite directions. Temperatures within the two streams tend to approach one another in a nearly linearly fashion resulting in a much more uniform heating pattern. Shown below the heat exchangers are representations of the axial temperature profiles for each. Parallel flow results in rapid initial rates of heat exchange near the entrance, but heat transfer rates rapidly decrease as the temperatures of the two streams approach one another. This leads to higher energy loss during heat exchange. Counter flow provides for relatively uniform temperature differences and, consequently, lead toward relatively uniform heat rates throughout the length of the unit.

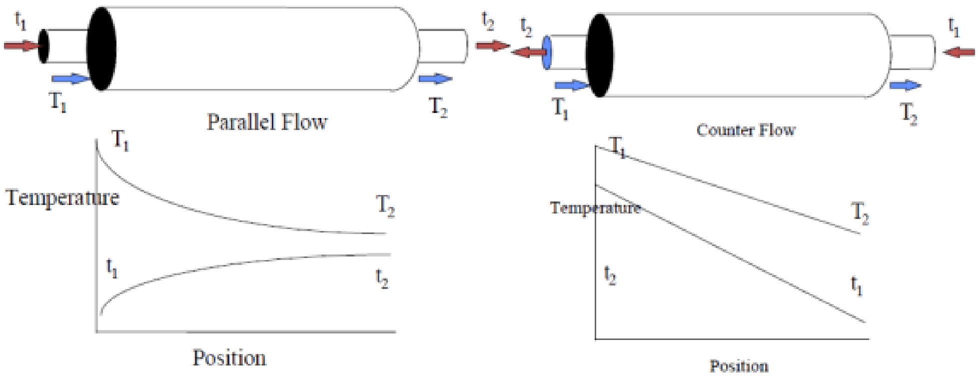


Fig 1: Basic flow arrangements for tubular heat exchangers.

Note in the Figures shown above that the hot stream may be cooled to t1 for counter flow, but may only be cooled to t2 for parallel flow. Counter flow allows for a greater degree of energy recovery. Similar arguments may be made to show the advantage of counter flow for energy recovery from refrigerated cold streams.

IV. REFRIGERANT (R404A) PROPERTIES

R-404A (HFC-404A) is a non-ozone depleting compound designed to serve as a long-term alternative to R-502 (CFC-502) and R-22 (HCFC-22) in low- and medium-temperature commercial refrigeration applications. Applications where R-404A is a suitable retrofit refrigerant include supermarket freezer cases, reach-in coolers, display cases, transport refrigeration and ice machines. R-404A has been designed as a substitute for R-502, but it is not a drop-in replacement. Mineral oils and alkyl benzene lubricants, which have been used traditionally with R-502, are immiscible with R-404A. Service technicians should consult the original equipment manufacturer for the recommended lubricants.R-404A is a blend. For this reason it is essential that systems be charged only with liquid from the cylinder, not vapour. Vapour-charging R-404A may result in the wrong refrigerant composition and could damage the system.

Table 1: Properties of Refrigerants

Refrigerant	Chemical Name	Chemical Formulae	Weight %	Molecular weight	Boiling Point in C at atmpr	ODP	GWP
R-134a	Tetra fluoro ethane	CF2FCF3	4%	102.03	-15	0	1300
R-404a	(R125, R134a, R143a) BLEND	HFC BLEND	100%	97.6	-47.8	0	3300

V. EXPERIMENTAL SETUP

In vapor compression refrigerating system basically there are two heat exchangers. One is to absorb the heat which is done by evaporator and another is to remove heat absorbed by refrigerant in the evaporator and the heat of compression added in the compressor and condenses it back to liquid which is done by condenser.

This work focuses on heat rejection in the condenser this is only possible either by providing a fan or by extending the surfaces. The extended surfaces are called fins. The rate of heat rejection in the condenser depends upon the number of fins attached to the condenser.

This work investigated the performance of condenser using condenser in the present domestic refrigerator galvanized iron steel material fins are used. In this project mild steel material fins are replaced and galvanized iron steel is used for the condensers.

The performance of the condenser will also help to increase COP of the system as the sub cooling region .incurred at the exit of the condenser. The performance of the condenser is also investigated by existing and modification condenser. In general domestic refrigerators have no fans at the condenser and hence extended surfaces like fins play a very vital role in the rejection of heat.

In order to know the performance characteristics of the vapor compression refrigerating system the temperature and pressure gauges are installed at each entry and exit of the component. Experiments are conducted on condenser having fins.

Different types of tools are also used like snips to cut the plated fins to required sizes, tube cutter to cut the tubes and tube bender to bend the copper tube to the required angle. Finally the domestic refrigerator is fabricated as for the requirement of the project. All the values of pressures and temperatures are tabulated.

The figure 4 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 condenser with coil spacing of the condenser on a refrigerator of capacity 215liters.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: 160 liters

Compressor capacity: 0.16 H.P.

Condenser Sizes

Length - 8.5 m

Diameter - 6.4 mm

Evaporator

Length - 7.62 m

Diameter - 6.4 mm

Capillary tube

Length - 2.428 m

Diameter - 0.8 mm

Among many possible variations of the basic refrigeration cycle, the cycle with the liquid-line/suction-line heat exchanger (LLSL-HX) is probably used most often. As a result of employing this intra cycle heat exchange, the high pressure refrigerant is sub cooled at the expense of superheating the vapour entering the compressor. Schematics of hardware arrangement for the basic cycle and cycle with LLSL-HX are shown in figure 3; the realized cycles are outlined on the pressure-enthalpy diagram shown in figure 2.

The use of liquid line/suction line heat exchangers is widespread in commercial refrigeration. The heat exchangers are often employed as a means for protecting system components, by helping to ensure single-phase liquid to the expansion device and single phase vapour to the compressor. in residential refrigerators, Capillary tube /suction-line heat exchanger is used to heat the suction line above the dew-point temperature of ambient air, thus preventing condensation of the water vapour on the outside of the water vapour on the outside of the suction line.

Employing an intra-cycle heat exchanger alters refrigerant thermodynamic states in the cycle, which may have significant (positive or negative) performance implications. For any fluid and system, an LLSL-HX increases refrigerant temperature at the compressor inlet and outlet, which is shortcoming. The coefficient of performance (COP) and volumetric capacity may increase for some fluid-application combinations, while for others they may decrease.

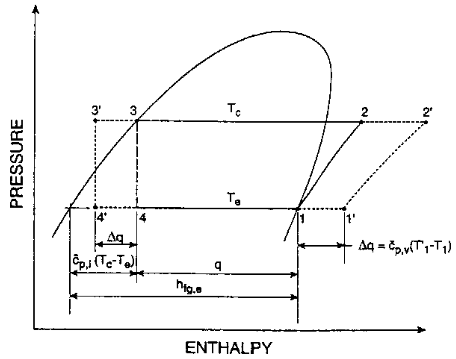


Figure 2 key refrigerant state points in the basic cycle and LLSL-HX cycle

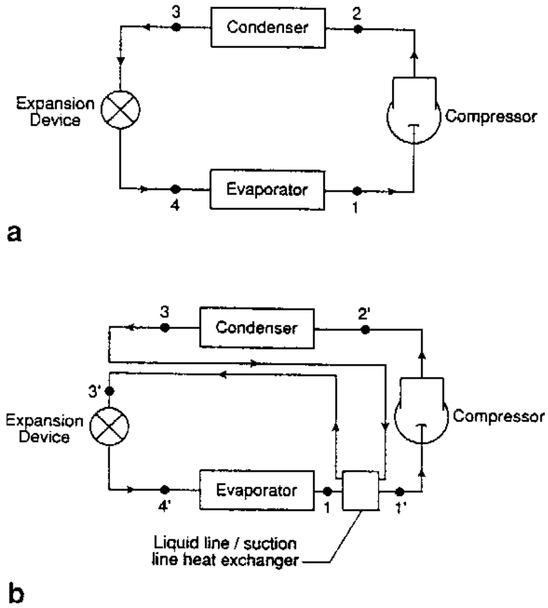


Figure 3hardware arrangements for a)The basic cycle b)cycle with the liquid-line/suction-line heat exchanger

Fig 4 New System with heat exchanger length of 30cms

VI. EXPERIMENTAL PROCEDURE

The following procedure is adopted for experimental setup of the vapor compression refrigeration system

1. The domestic refrigerator is selected, working on vapor compression refrigeration system.
2. Pressure and temperature gauges are installed at each entry and exit of the components.
3. Flushing of the system is done by pressurized nitrogen gas.
4. R 134a refrigerant is charged in to the vapor compression refrigeration system by the following process:

The systematic line diagram for charging is shown in the fig 5. it is necessary to remove the air from the refrigeration unit before charging. First the valve V_2 is closed and pressure gauge P_2, vacuum gauge V are fitted as shown in the fig. the valve V_5 is also closed and valves V_1, V_4, V_6 and V_3 are opened and the motor is started thus the air from the condenser receiver and evaporator is sucked through the valve V_1 and it is discharged in to atmosphere through the valve V_6 after compressing it in the compressor the vacuum gauge V indicates sufficiently low vacuum when most of the air is removed in the system. The vacuum reading should be at least 74 to 75 cm of Hg. If the vacuum is retained per above an hour it may be concluded that the system is free from the air. After removing the air the compressor is stopped and valves V_1 and V_6 are closed, the valves V_5, V_2 and V_7 of the refrigerant cylinder are opened and then the compressor is started whenever the sufficient quantity of refrigerant is taken in to the system which will be noted in the pressure gauges. The compressor is stopped. The valves V_7 and V_5 are closed and valve V_1 is opened the refrigerant cylinder is disconnected from the system the pressure gauge is used to note the pressure during the charging the system. Same procedure is followed for R 404a.

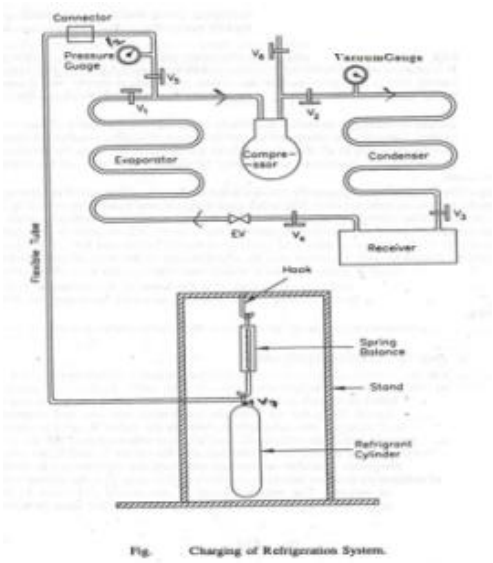


Fig.5 charging of refrigeration system

5. Leakage tests are done by using soap solution, In order to further test the condenser and evaporator pressure and check purging daily for 12 hours and found that there is no leakages which required the absolutely the present investigation to carry out further experiment.
6. Switch on the refrigerator and observation is required for 1 hour and take the pressure and temperature readings at each section.
7. The performance of the existing system is investigated, with the help of temperature and pressure gauge readings.
8. The refrigerant is discharged out and condenser is located at the inlet of the capillary tube.
9. Temperature and pressure gauge readings are taken and the performance is investigated.
10. The readings are tabulated for the length of heat exchanger of 10cm, 20cm&30cm.

The following tests are conducted and calculations are shown below.

VII. PERFORMANCE CALCULATIONS

The temperature, pressure and enthalpy at state points for the system using R134a by adopting heat exchanger with different lengths is shown in following table.

Table 2: Temperature, Pressure and enthalpy readings at state points using R134a with and without heat exchanger

Parameter	Heat Exchanger length(R134a)			
	Exist	10	20	30
Compressor Discharge Temperature T2(°C)	59.2	60	60	60
Condensing Temperature T3(°C)	44	44	44	43
Evaporator Temperature T4(°C)	-15	-15	-15	-15
Compressor suction pressure P1(bar)	1.03	1.03	1.03	1.02
Compressor discharge pressure P2(bar)	12.75	12.75	12.75	12.75
Condenser pressure P3 (bar)	12.75	12.75	12.75	12.75
Evaporator pressure P4(bar)	1.03	1.03	1.03	1.02
Enthalpy,h1 (kJ/kg)	395	395	396	396
Enthalpy,h2 (kJ/kg)	436	436	436	436
Enthalpy,h3 (kJ/kg)	262	262	262	260
Enthalpy,h4 (kJ/kg)	262	262	262	260

Existing system(r134a)

Calculation of Performance Parameters

1. Net Refrigerating Effect (NRE) = h_1-h_4 = 395-262 = 131 kJ/kg
2. Mass flow rate to obtain one TR, kg/min.
   m_r= 210/NRE = 210/142 = 1.603 kg/min.
3. Work of Compression = h_2-h_1 = 436-395 = 41 kJ/kg
4. Heat Equivalent of work of compression per TR

m_r x (h_2-h_1) = 1.603 x 41 = 65.723 kJ/min

5. Theoretical power of compressor = 65.723/60 = 1.0953 kW
6. Coefficient of Performance (COP) = h_1-h_4 / h_2-h_1 = 131/41 = **3.24**
7. Heat to be rejected in condenser = h_2-h_3 = 436-262 = 174 kJ/kg
8. Heat Rejection per TR = (210/NRE) x (h_2-h_3)= 1.603x 174 = 2782.92 kJ/min
9. Heat Rejection Ratio = 272.65/210 = 1.298
10. Compression Pressure Ratio = $\frac{\boldsymbol{Dishcarge\ \ Pressure}}{\boldsymbol{Suction\ \ Pressure}}$ =$\frac{\boldsymbol{P_d}}{\boldsymbol{P_s}}$ =12.72/1.03= 12.349

Repeating the experimentation using the same refrigerant R134a, by adopting heat exchanger with different lengths, the performance parameters are calculated that are tabulated as follows.

Table 3: Performance parameters using R134a with and without heat exchanger

Tim	PARAMETERS	Heat Exchanger length(R134a)			
		Exist	10	20	30
1	(COP)	3.24	3.24	3.35	3.4
2	Net refrigerating effect , kJ/kg	131	131	134	136
3	Work of Compression, kJ/kg	41	41	40	40
4	Compressor Power, kW	1.0953	1.09538	1.0446	1.0293
5	Mass flow rate to obtain one TR, kg/min	1.603	1.603	1.567	1.544
6	Heat Equivalent of work of compression per TR, kJ/kg	65.723	65.723	62.68	61.76
7	Heat rejected in condenser , kJ/kg	174	174	174	176
8	Heat Rejection per TR, kJ/min	2782.92	278.92	272.65	271.744
9	Heat Rejection Ratio	1.298	1.328	1.298	1.294
10	Compression Pressure Ratio	12.349	12.378	12.349	12.5

The temperature, pressure and enthalpy at state points for the system by using R404a by adopting heat exchanger with different lengths is shown in following table.

Table 4: Temperature, Pressure and enthalpy readings at state points using R404a with and without heat exchanger

Parameter	Heat Exchanger length(R404a)			
	Exist	10	20	30
Compressor Suction Temperature T_1(°C)				
Compressor Discharge Temperature T2(°C)	52	52	53	52
Condensing Temperature T3(°C)	38	38	39	36
Evaporator Temperature T4(°C)	-15	-15	-15	-15
Compressor suction pressure P1(bar)	1.24	1.24	1.24	1.24
Compressor discharge pressure P2(bar)	18.62	18.62	18	17.655
Condenser pressure P3 (bar)	18.62	18.62	18	17.655
Evaporator pressure P4(bar)	1.24	1.24	1.24	1.24
Enthalpy,h1 (kJ/kg)	366	366	366	366
Enthalpy,h2 (kJ/kg)	400	400	400	400
Enthalpy,h3 (kJ/kg)	254	254	252	248
Enthalpy,h4 (kJ/kg)	254	254	252	248

Repeating the experimentation by using R404a and by adopting heat exchanger with different lengths, the performance parameters are calculated that are tabulated as follows.

Table 5: Performance parameters using R404a with and without heat exchanger

S.No	PARAMETERS	Heat Exchanger length(R404a)			
		Exist	10	20	30
1	(COP)	3.294	3.294	3.35	3.47
2	Net refrigerating effect , kJ/kg	112	112	114	118
3	Work of Compression, kJ/kg	34	34	34	34
4	Compressor Power, kW	1.0625	1.0625	1.043	1.0081
5	Mass flow rate to obtain one TR, kg/min	1.875	1.875	1.842	1.779
6	Heat Equivalent of work of compression per TR, kJ/kg	63.75	63.75	62.628	60.486
7	Heat rejected in condenser , kJ/kg	146	146	148	152
8	Heat Rejection per TR, kJ/min	273.75	273.75	272.616	270.408
9	Heat Rejection Ratio	1.3035	1.3035	1.2981	1.287
10	Compression Pressure Ratio	15.016	15.016	14.564	14.2379

VIII. RESULTS AND DISCUSSIONS

Effect on the Net refrigerating effect by adopting liquid line –suction line heat exchanger

The refrigerating effect means heat absorbed or extracted. Referring to chart 1 in the present work, system with Liquid line suction line heat exchanger using R134a as a refrigerant. There is 3.81% increment in refrigerating effect compared with existing. When R404a taken as a refrigerant, there is 5.35% increment in refrigerating effect compared with existing system.

Effect on compressor work by adopting liquid line –suction line heat exchanger

Referring to chart 2 in the present work system with Liquid line suction line heat exchanger using R134a as a refrigerant, there is 2.5% decrement in compressor work compared with existing. When R404a taken as a refrigerant, there is no change in compressor work compared with existing system.

Effect on compressor power by adopting liquid line –suction line heat exchanger

Referring to chart 3 in the present work system with Liquid line suction line heat exchanger using R134a as a refrigerant, there is 6.4% decrement in compressor power compared with existing. When R404a taken as a refrigerant, there is 5.39% decrement in compressor power compared with existing system.

Effect on coefficient of performance of the system by adopting liquid line –suction line heat exchanger

Referring to chart 4 in the present work system with Liquid line suction line heat exchanger using R134a as a refrigerant, there is 4.93% increment in co-efficient of performance of the system compared with existing system. When R404a taken as a refrigerant, there is 5.34% increment in co-efficient of performance of the system compared with existing system.

Effect on Heat rejection by adopting liquid line –suction line heat exchanger

Referring to chart 5 in the present work system with Liquid line suction line heat exchanger using R134a as a refrigerant, there is 1.14% increment in heat rejection compared with existing system. When R404a taken as a refrigerant, there is 4.10% increment in heat rejection compared with existing system.

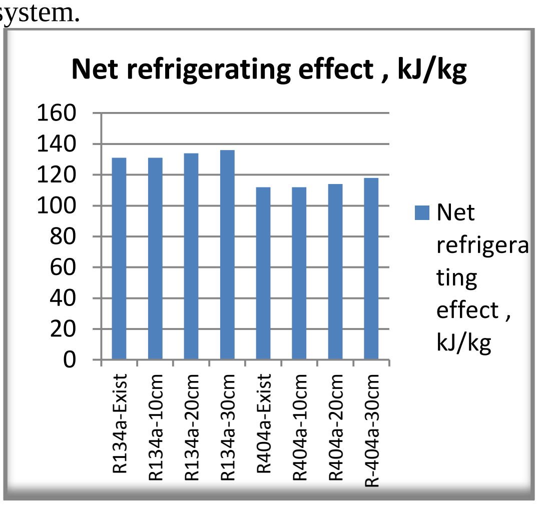


Chart 1: Effect of Heat exchanger length and different refrigerants on Net Refrigerating effect

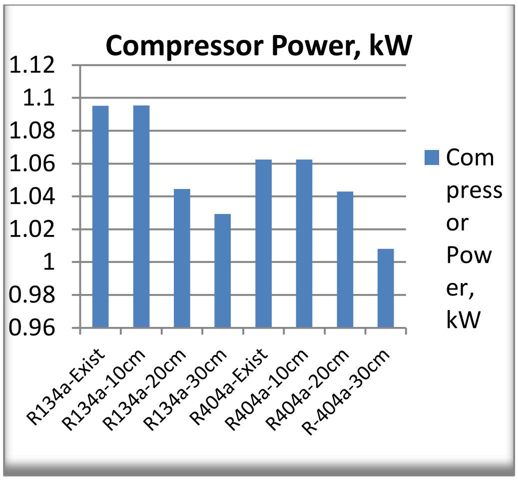


Chart 3: Effect of Heat exchanger length and different refrigerants on compressor power

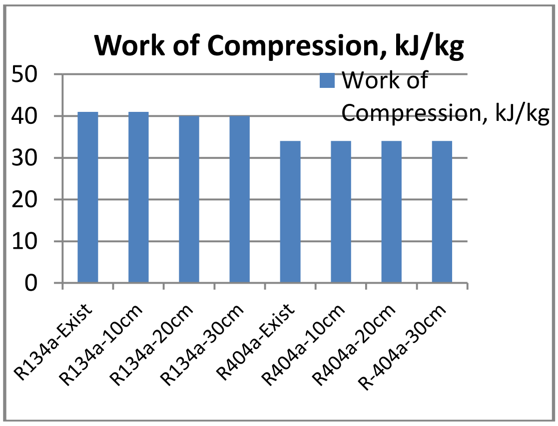


Chart 2: Effect of Heat exchanger length and different refrigerants on compression work

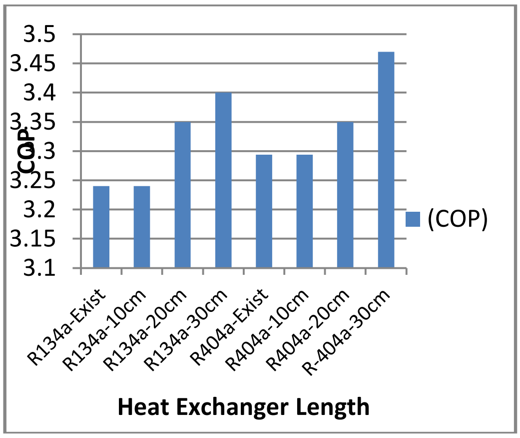


Chart 4: Effect of Heat exchanger length and different refrigerants on COP

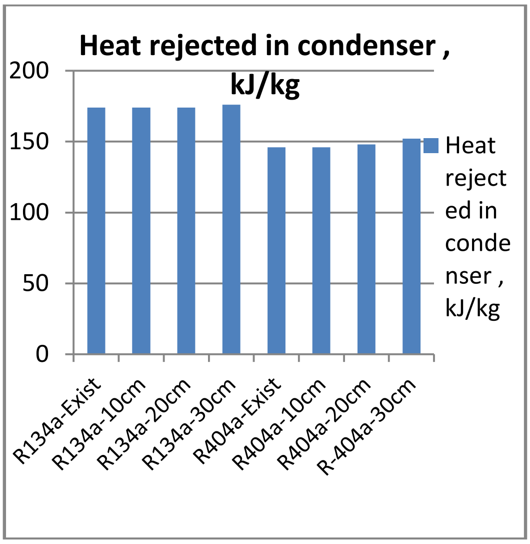
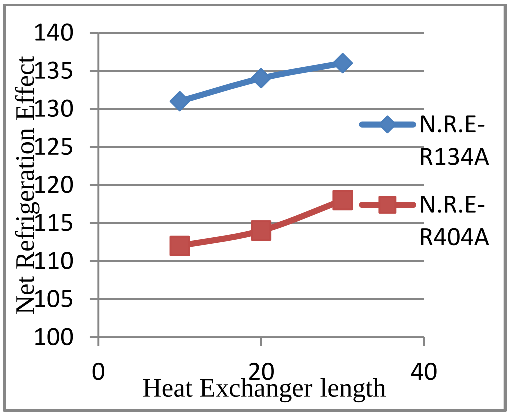


Chart 5: Effect of Heat exchanger length and different refrigerants on Heat rejected

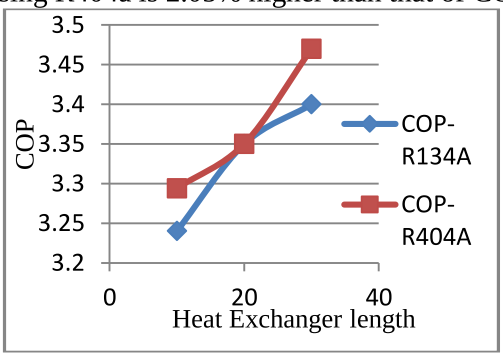
Already it is found from above results that providing heat exchanger gives better refrigeration effect and now comparison has been made for the effect of heat exchanger length and refrigerants R134a and R404a as follows.

Effect of heat exchanger length on refrigeration effect using refrigerants R134a and R404a is shown graph 1. It is observed that as heat exchanger length increases the net refrigeration effect increases for both the refrigerants under test. Also it is found that refrigeration effect is superior in case of R134a than that of R404a. The maximum refrigeration effect obtained is 136Kj/Kg using R134a whereas the same is 118Kj/kg for R404a.



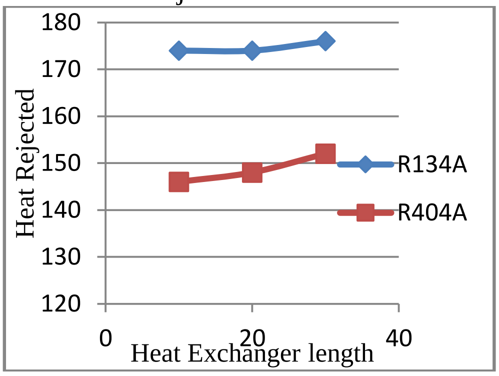
Graph 1: Effect of Heat exchanger length on net Refrigeration effect using R134a and R404arefrigerants**.**

From the following graph 2 it is observed that as heat exchanger length increases COP of the system also increases for both the refrigerants R134a and R404a. COP is superior for R404a than that of R134a. The maximum COP using R404a is 2.05% higher than that of COP using R134a.



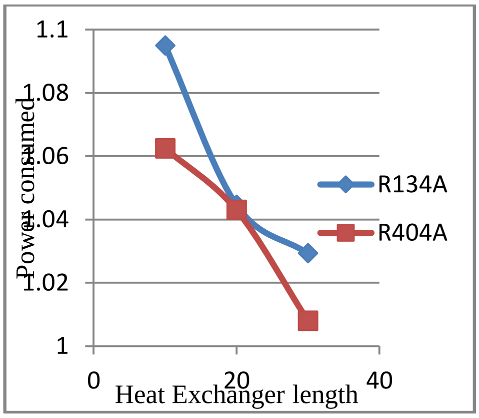
Graph 2: Effect of Heat exchanger length on COP using R134a and R404arefrigerants**.**

From the graph 3 it is found that increase of heat exchanger leads to increase of heat rejected for both the refrigerants R134a and R404a. Heat rejection is higher for R134a than that of R404a. As the length increases surface area increases, which in turn results in increase of heat rejection. Thou even pressure drop occurs which opposes the heat rejection but its effect is small compare to the sensibility of surface area increment. So ultimately slight improvement in found in heat rejection with increase of heat exchanger length. The maximum heat rejected for R134a is 15.7% higher than that of R404a.



Graph 3: Effect of Heat exchanger length on heat rejected using R134a and R404arefrigerants**.**

From the graph 4 it is observed that increase of heat exchanger length decreases the power consumption in both the cases of using R134a and R404a. Considering COP, refrigeration effect and heat rejected the optimum length of heat exchanger is 30cm which saves power up to 2.1% for R134a than that of using R404a.



Graph 4: Effect of Heat exchanger length on power consumption using R134a and R404arefrigerants**.**

IX. CONCLUSIONS

In the present work experimental investigation is carried out to investigate the performance of vapour compression refrigeration system of a domestic refrigerator of 160 liters capacity, with R-134a and R-404a as refrigerants by adopting different lengths of liquid line-suction line heat exchanger for domestic refrigerator.

After conducting the experiments, the following conclusions are drawn.

- Net refrigerating effect is increased for different lengths of liquid line-suction line heat exchanger and at the length of heat exchanger of 30 cms is high for R134a is 136 and for R404a is 118
- Coefficient of performance is increased for different lengths of liquid line-suction line heat exchanger and at the length of heat exchanger of 30 cms is high for R404a is 3.47 and for R134a is 3.4

- Heat to be rejected in condenser is increased for different lengths of liquid line-suction line heat exchanger and at the length of heat exchanger of 30 cms is high for R134a is 176 and for R404a is 152
- From the above discussions, it can be concluded that the performance of vapour compression refrigeration system of domestic refrigerator can be increased by increasing the length of a heat exchanger for different refrigerants. But there is a limitation to increase length due to the fabrication difficulty.

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