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Experimental Analysis of Suction Line Heat Exchanger by Using R-134a

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Abstract: Now-a-days Vapour compression system is used for all purpose refrigeration. It is generally used for all industrial purpose from a small domestic refrigerator to a big air conditioning plant. Expansion device is one of the important components in the system that divides the high and low pressure sides. Because of simplicity and low cost, capillary tubes are used as the expansion device in most small refrigeration and air conditioning systems. Capillary tube maintains flow rate of refrigerant flowing through it at desired level, so capillary tube works as an automatic flow rate controller. 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.

The main objective of this project is to evaluate the performance of refrigerator with suction line heat exchanger of 35cm length by using R-134a as refrigerant. In the proposed system with suction line heat exchanger the pull down period is found to be less than the pull down period of existing system. The percentage of decrease in pull down period using R-134a is 1.74%. The percentage of increase in COP using R-134a in no load condition is 3.27% and in loaded condition is 3.10%.

The percentage of increase in frost collection using R-134a is 9.09%. From the above discussions, it can be concluded that the performance of vapour compression refrigeration system of domestic refrigerator can be increased by using 35cm length of a heat exchanger

I. INTRODUCTION

In this project the liquid line is usually placed in contact with the suction line, forming a counter flow heat exchanger. The heat exchanger may be of two kinds: lateral and double pipe. 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. By making this arrangement there are two advantages first there is delay in vaporization and second superheated vapour is going to compressor. This eliminates suction line sweating and preventing slugging of the compressor.

II. EXPERIMENTAL SETUP

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 a domestic refrigerator of 165 liters capacity, with R-134a as refrigerant and using liquid line-suction line heat exchanger of 35cm length.

Domestic refrigerator selected for the project has the following specifications

Compressor capacity: 0.16 H.P

Condenser sizes

Length - 8.5m

Diameter - 6.4mm

Evaporator Length - 7.62m

Diameter - 6.4mm

Capillary tube Length - 2.428m

Diameter - 0.8mm


FIG.1. Existing System
Fig.2. System with 35cms length of heat Exchanger

III. EXPERIMENTAL PROCEDURE

The vapour compression system is initially cleaned and the evacuation of the system is carried out with the help of a vacuum pump for nearly 30 min and then the refrigerant is charged into the system.

Initially the system is charged with refrigerant R-134a and then the following tests were carried out, Pull–down characteristics, No load performance, Performance with load, Frosting.

Frosting and Defrosting

For occurrence of frosting the system is switched on with a load of water in open trough is placed in the evaporator cabin and kept in continuous running condition for 72 hours and then the system is switched off. The system is allowed to defrost for certain period and the quantity of water is collected.


FIG.3. A view of refrigerator freezer box which is frosted
Fig.4.A view of refrigerator arrangement for frosting with bulb and load placed inside the cabin

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A. Data reduction

1. Net Refrigerating Effect (NRE) = h_1-h_4
2. Mass flow rate to obtain one TR, kg/min. m_f = 210/NRE
3. Work of Compression W_c = h_2-h_1
4. Heat Equivalent of work of compression per TR = m_f x (h_2-h_1)
5. Theoretical power of compressor T_c = m_f x (h_2-h_1) /60
6. Coefficient of Performance (COP) = h_1-h_4 / h_2-h_1
7. Heat to be rejected in condenser H_c = h_2-h_3
8. Heat Rejection per TR = (210/NRE) x (h_2-h_3)
9. Heat Rejection Ratio H R R = Heat Rejection per TR/210
10. Compression Pressure Ratio $= \frac{\text{Dishcarge Pressure}}{\text{Suction Pressure}} = \frac{P_d}{P_s}$

IV. RESULTS AND DISCUSSIONS

Tab.1

S.NO	PARAMETERS	EXISTING SYSTEM WITH NO LOAD	EXISTING SYSTEM WITH LOAD	NEW SYSTEM WITH NO LOAD	NEW SYSTEM WITH LOAD
1.	Net refrigerating effect kJ/kg	167	165	171	170
2.	Coefficient of Performance (COP)	6.1	5.15	6.3	5.31
3.	Mass flow rate to obtain one TR kg/min	1.25	1.27	1.228	1.23
4.	Work of Compression kJ/kg	27	32	27	32
5.	Heat Equivalent of work of compression per TR kJ/min	33.95	40.64	33.156	39.36
6.	Compressor Power KW	0.56	0.67	0.55	0.656
7.	Heat to be rejected in condenser kJ/kg	194	197	198	202
8.	Heat Rejection per TR kJ/min	243.95	250.19	243.144	248.46
9.	Heat Rejection Ratio	1.16	1.19	1.15	1.18
10.	Compression Pressure Ratio	8.90	9.11	8.06	8.6

Graphs

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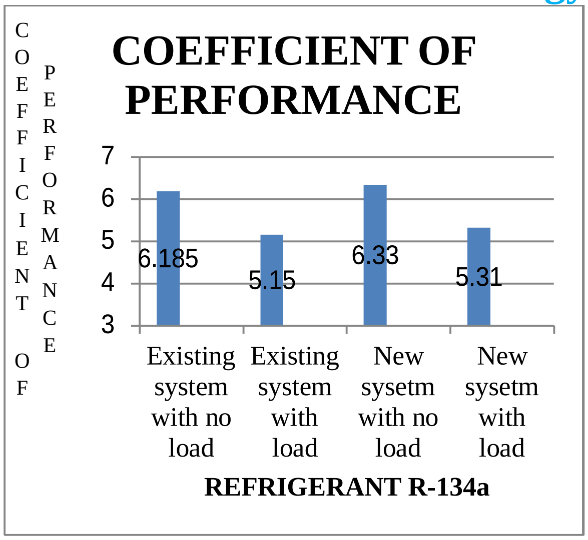
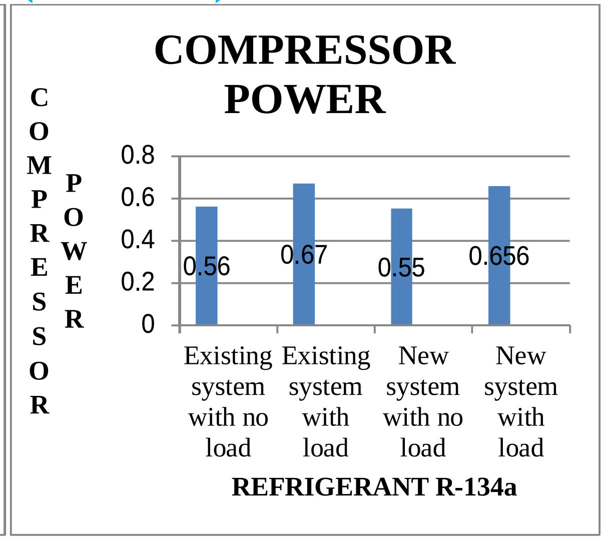
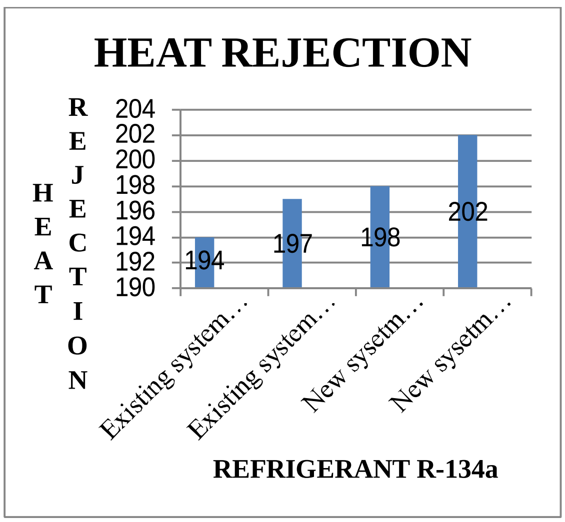


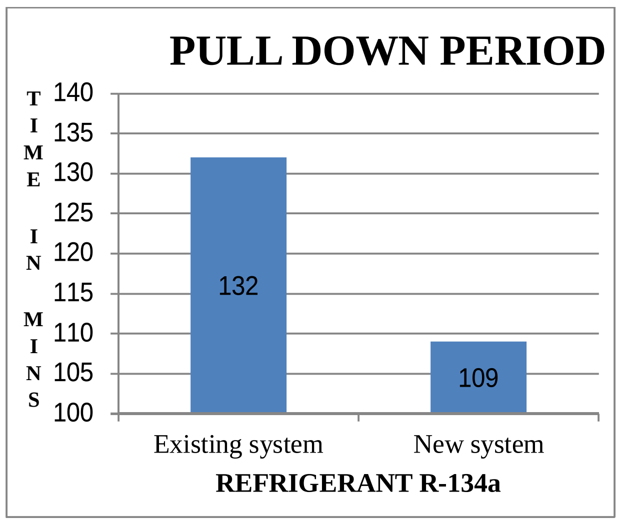
1.COP for R-134a



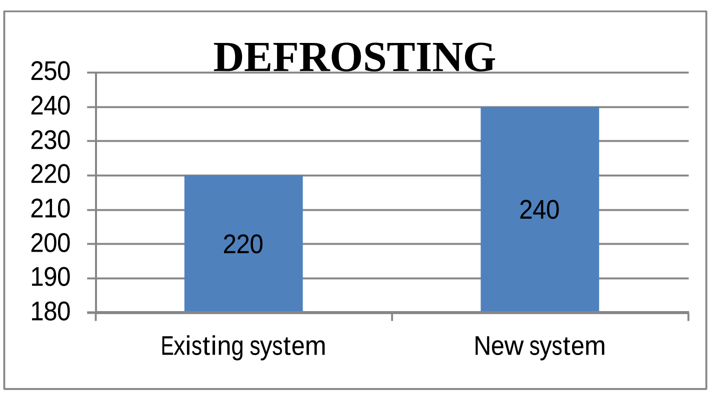
2. Compressor power for R-134a



3. Heat rejection for R-134a



4.Pull down period for R-134a



5. Defrosting water collected for refrigerant R-134a

Effect on coefficient of the performance of the system by adopting suction line heat exchanger using refrigerant R-134a

Graph .1. shows the coefficient of performance for refrigerant R-134a for both existing and new system with suction line heat exchanger at different load conditions.Net refrigeration effect is more with suction line heat exchanger so coefficient of performance is increased with proposed system.

Effect on compressor power by adopting suction line heat exchanger using refrigerant R-134a

Graph 2 shows the compressor power for refrigerant R-134a for both existing and new system with suction line heat exchanger at different load conditions.Mass flow rate is less with suction line heat exchanger so compressor power is decreased with proposed system.

Effect on heat rejection by adopting suction line heat exchanger using refrigerant R-134aGraph 3 shows the heat rejection for refrigerant R-134a for both existing and new system with suction line heat exchanger at different load conditions.Subcooling is more with suction line heat exchanger so heat rejection is increased with proposed system.

Effect on pull down period by adopting suction line heat exchanger using refrigerant R-134a

Graph 4 shows that pull down period of domestic refrigerator by using suction line heat exchanger is higher as compared to that of existing system by about 17.4%

Effect on defrosting by adopting suction line heat exchanger using refrigerant R-134a

Graph 5 shows that defrosting water collected in domestic refrigerator by using suction line heat exchanger is higher as compared to that of existing system by about 9.09% which shows that cooling is higher.

V. CONCLUSIONS

In the proposed system with suction line heat exchanger the pull down period is found to be less than the pull down period of existing system. The percentage of decrease in pull down period using R-134a is 1.74%.

In the proposed system the coefficient of performance is found to be greater than the coefficient of performance of existing system. The percentage of increase in COP using R-134a in no load condition is 3.27% and in loaded condition is 3.10%.

In the proposed system the frost collection is found to be greater than the frost collection of existing system. The percentage of increase in frost collection using R-134a is 9.09%.

From the above discussions, it can be concluded that the performance of vapour compression refrigeration system of domestic refrigerator can be increased by using 35cm length of a heat exchanger

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