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COMPARATIVE PERFORMANCE ANALYSIS OF VAPOUR COMPRESSION REFRIGERATION SYSTEM WITH R-134a AND BLENDS OF HYDROCARBON R290/600a

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ABSTRACT

In domestic refrigerators and refrigeration system the most widely used refrigerant is R134a.It must be phased out soon according to Kyoto protocol due to its high Global Warming Potential (GWP) of 1300. Hydrocarbon mixture (HCM) is an alternative refrigerant for Hydro Fluorocarbon (HFC) and Chlorofluorocarbon (CFC) compounds due to their lower GWP and zero Ozone Depletion Potential (ODP). The impact on the environment is also reduced due to usage of hydrocarbon mixture in different mass ratio. In the present work, an experimental investigation has been made with hydrocarbon refrigerant mixture composed of R290 (Propane) and R600a (Isobutane) as an alternative to R134a in a single evaporator domestic refrigerator. The primary object of the proposed work is to evaluate the performance parameters like compressor power consumption, Discharge temperature, Refrigeration effect, Coefficient of performance (COP), cycle efficiency and Ton of Refrigeration with optimized mixture of R290 and R600a refrigerant and compare it with the performance of R 134a refrigerant. The investigation aims to find out the optimized blending mixture of R290 and R600a refrigerant and validate it towards REFPROP software. The hermetically sealed compressor (Capacity 1/4 T) to be used to analyze the performance of refrigeration cycle. From comparative analysis of both refrigerants it was found that hydrocarbon blend of R-290 and R-600a in 30%-70% on mass basis given better performance than R-134a refrigerant..

Keywords- *Global Warming Potential, Ozone Depletion Potential, REFPROP, VCRS, HCM..*

I. INTRODUCTION

Thermodynamic cycles can be categorized into Gas cycles and Vapour cycles. In a typical Gas cycle, the working fluid (a gas) does not undergo phase change; consequently the operating cycle will be away from the Vapour dome. In a Gas cycles, Heat rejection and Refrigeration take place as the gas undergoes sensible cooling and heating. Whereas in a Vapour cycle the working fluid undergoes phase change and refrigeration effect is due to the Vaporization of Refrigerant liquid. If the refrigerant is a pure substance then its temperature remains constant during the phase change processes. However, if a Zeotropic mixture is used as a refrigerant, then there will be a temperature glide during Vaporization and Condensation. Since the refrigeration effect is produced during phase change, large amount of heat (latent heat) can be transferred per kilogram of refrigerant at a near constant temperature. Hence, the required mass flow rate for a given refrigeration capacity will be much smaller compared to a Gas cycle. Vapour cycles can be subdivided into Vapour compression systems, Vapour absorption systems and Vapour jet systems etc. Among these the Vapour compression refrigeration systems (VCRS) is predominant.

Table No.1 Environmental Impact of Refrigerants

Refrigerant	R12	R22	R134a	R290	R600a
Class	CFC	HCFC	HFC	HC	HC
Atmospheric Life (In Years)	130	15	16	<1	<1
ODP	1	0.07	0	0	0
GWP	8500	1700	1300	8	8

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II. EXPERIMENTAL SETUP

2.1 Vapour Compression Refrigeration Cycle

Vapour compression refrigeration system is the most widely used in the refrigeration process. It is adequate for most refrigeration applications. The ordinary vapour compression refrigeration systems are simple, inexpensive, reliable and practically maintenance free. Most of the domestic refrigerators today are running based on the vapour compression refrigeration system. It is somewhat analogous to a reverse Rankine cycle. The vapour compression refrigeration system contains four main components which are compressor, condenser, expansion device(capillary tube), and evaporator. Compressor is used to compress the low pressure and low temperature refrigerant from the evaporator to high pressure and high temperature state. After the compression process the refrigerant is then discharge into the condenser for condensation which requires heat rejection to the surroundings (heat sink). The refrigerant can be condensed at atmospheric temperature by increasing the refrigerant's pressure and temperature above the atmospheric temperature. After the condensation process, the condensed refrigerant will flow into the capillary tube, where the temperature of refrigerant will be dropped to lower temperature than surrounding caused by the reducing pressure inside the expansion device. When the pressure drops, the refrigerant vapour will expand. As the vapor expands, it draws the energy from its surroundings or the medium in contact with it and thus produces refrigeration effect to its surroundings. After this process, the refrigerant is ready to absorb heat from the space to be refrigerated. The heat absorption process is to be done in the evaporator. The heat absorption process is normally being called as evaporation process. The cycle is completed when the refrigerant returns to the suction line of the compressor after the evaporation process. Low temperature refrigeration, at temperatures below 0^0C, affects everyday life; it is mostly used for food preservation, such as in the freezer of a refrigerator.

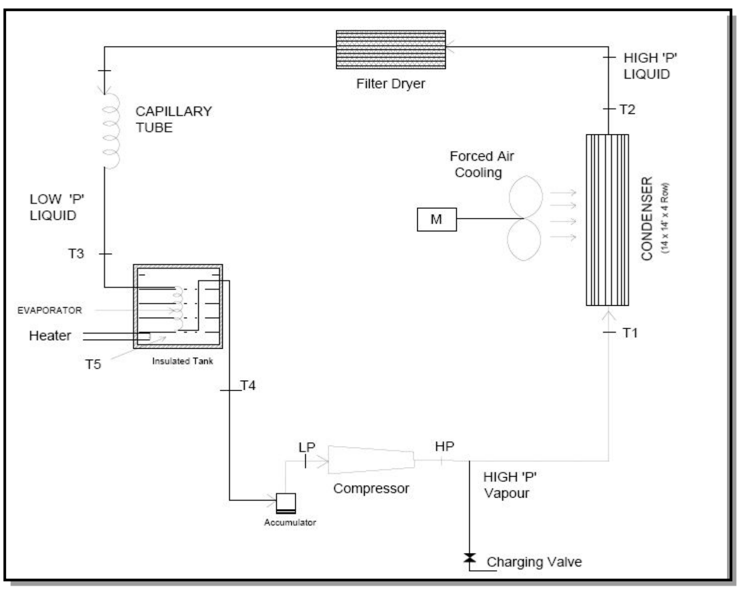

Fig. 2.1 Experimental Setup Cycle

2.2 Experimental Setup

A test rig is designed and fabricated for the experimentation purpose and it consists of ¼ T capacity hermetically sealed compressor of Kirloskar make suitable for 134a refrigerant as well as hydrocarbon refrigerant. Forced draft finned type copper coiled air cooled condenser with fan motor, constant pressure capillary tube as expansion device, a single evaporator coil made of copper. For filling or emptying the refrigerant into the system we provide a charging or maintenance valve. With the help of energy meter we measure the power consumption of compressor or refrigeration system. Control panel is fitted on the apparatus itself for measuring various parameters during experimentation. Whole unit is fitted on strong and sturdy supporting stand with appreciate colour combination powder coating.

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2.3 Components of Experimental Setup:

1) Compressor: Hermetically sealed type, Kirloskar Make, ¼ T Capacity
2) Air Cooled Condenser: 14 x 14'x 4 Row copper coils with aluminium fins & cooling fan.
3) Filter Dryer: 1 No. for removing moisture from refrigerant. The high pressure liquid leaves the condenser through the "liquid line" and travels to the "metering device". Sometimes running through a filter dryer first, to remove any dirt or foreign particles.
4) Expansion Device- Capillary Tube, 5000 mm in length. HCM demanded lengthening of capillary tube by about 25% to achieve maximum COP. He was found that the maximum COP is achieved at 5000 mm capillary tube length. [1]

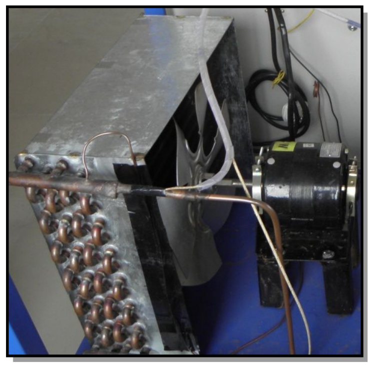

Fig.2.2 Air Cooled Condenser

5) Evaporator-Diameter-3/8", Length-8,300mm, Copper pipe. The evaporator is installed from outside to prevent heat loss. To vary the evaporator load or to calculate evaporator load, evaporator coil is immersed in water tank and with the help of electric heater (1.5 KW) we change the temperature of water hence the evaporator load.

Fig.2.3 Evaporator Tube

To avoid the heat losses from tank surface and to established thermally insulated evaporator we provide the insulation to tank.

6) Energy Meter: For measuring power supply to the Evaporator Heater.
7) Dimmer stat-To vary the voltage hence power supply of heater.
8) Pressure Gauge-One each for the measurement of high/discharge line and low/suction line pressure.
9) Electric Heater-Immersion type 1.5 KW
10) Digital Temperature Indicator-To Measure the temperatures at different Points.(Evaporator Inlet & Outlet, Condenser Inlet & Outlet, Evaporator Bath)
11) Accumulator- To supply saturated dry vapour refrigerant to the compressor suction.

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12) HP & LP Cutout-Safety device suitable for the LP and HP of compressor.
13) Service Valve- Needle type valve for changing the refrigerant.
14) Ammeter-Range 0 to 5 ampere for measuring power consumption of compressor.
15) Voltmeter-230 V, AC for measuring power consumption of compressor.
16) Switches: For various controls. The refrigerant circuit is mounted on a board. The unit is supported on a frame.

Table 2.1 Refrigeration Cycle Test Apparatus

No.	Components	Description
1	**Compressor Unit**	Hermetically sealed compressor, Kirloskar Make, ¼ T Capacity
2	**Condenser Unit**	Forced draft finned type copper coiled air cooled condenser with fan motor
3	**Expansion Unit**	Constant pressure Capillary Tube
4	**Evaporator Unit**	Sufficient length copper tube coil is merged in water evaporator Bath. Heater placed inside the evaporator to maintained cooled effect.
5	**Control Panel Unit**	Digital temperature Indicator, Cam operated On-Off switch, Voltmeter, Ammeter, Energy meter, Auto Transformer, Pressure Gauges, Heater

Fig.2.4 Experimental Setup Panel Board

III. EXPERIMENTATION

3.1 Preparation and Charging of HCM 30/70

For charging the hydrocarbon mixture in set up, prepared 30% propane and 70% Isobutane mixture in separate cylinder on mass basis up to the 60% of charge required of R-134a. As propane (R-290) has highest saturation pressure comparing to R-600a , hence while charging the refrigerant into cylinder first of all charged the 30%

R-600a mass refrigerant into cylinder and then charged the R-290. After charging the cylinder of total mass 60% (30% R-290 and 70% R-600a) to that of R-134a charge, slowly charge our system. When observed defrosting at evaporator inlet and outlet line stop the charging. Calculate the total charged of HCM in to the system.

3.2 Experimental Procedure

Connect the plug to main supply. Before ON the supply, confirm that all the switches on panel are in off position. See the dimmer stat zero position. Then put ON the heater switch & give power to heater. This will heat the water in evaporator & this can be seen in on temperature indicators on control board. Adjust the heater voltage such that the temperature of bath reaches up to 25-30 ^{0}C. Now ON the D.P. switches. Put ON the condenser fan switch & wait for 2 to 3 minutes. Now switch ON the solenoid valve switch & the compressor switch. The refrigeration flow will start. This can be confirmed by pressure gauge reading on suction line (LP) and discharge line (HP). Now the ammeter, voltmeter will show the current & voltage for compressor and cooling fan. Note down the time (Th) for 10 revolutions of energy for compression. After some time we will see that the temperature of water bath in the evaporator tank slowly goes down & reaches steady state.

After the steady state note down the readings as follow-

1. HP Condenser pressure and LP Evaporator Pressure in (psi) and convert it into the MPa.

2. Condenser Inlet Temperature (T1) and Condenser outlet Temperature (T2) in ^{0}C.

3. Evaporator Inlet Temperature (T3) and Evaporator Outlet Temperature (T4) in ^{0}C.

4. Time (Th) for 10 Pulses of heater energy meter in sec.

5. Ammeter reading in ampere (A).

6. Voltmeter reading in volt (V).

7. Evaporator Bath Temp (T5) in ^{0}C.

IV. RESULTS AND DISCUSSION

Experimental results obtained for continuous running mode at different heating load at ambient temperature of 29^{0}C are discussed in this section. From results it was found that R-134a refrigerant maintained steady temperature (T5) for the heating load ranging from 296 W to 698 W, whereas HCM 30/70 maintained 296 W to 800 W heating loads.

4.1 Optimized HCM as a Refrigerant

With the help of REFPROP version 7.0 we found that optimized hydrocarbon mixture from different composition is 30% Propane (R-290) and 70% Isobutane (R-600a).For finding out this HCM we were used constant reference pressure of HP, LP and Discharge Temperature (T1) for all the composition. By plotting the vapour compression refrigeration cycle on P-h diagram we found that maximum theoretical COP was achieved by using optimized blend mixture that is R-290 in 30% and R-600a in 70% on mass proportion.

This optimized HCM were given Theoretical COP higher than R-134a, 100% Propane and 100% Isobutane. Hence this gives the software validation for our experimentation.

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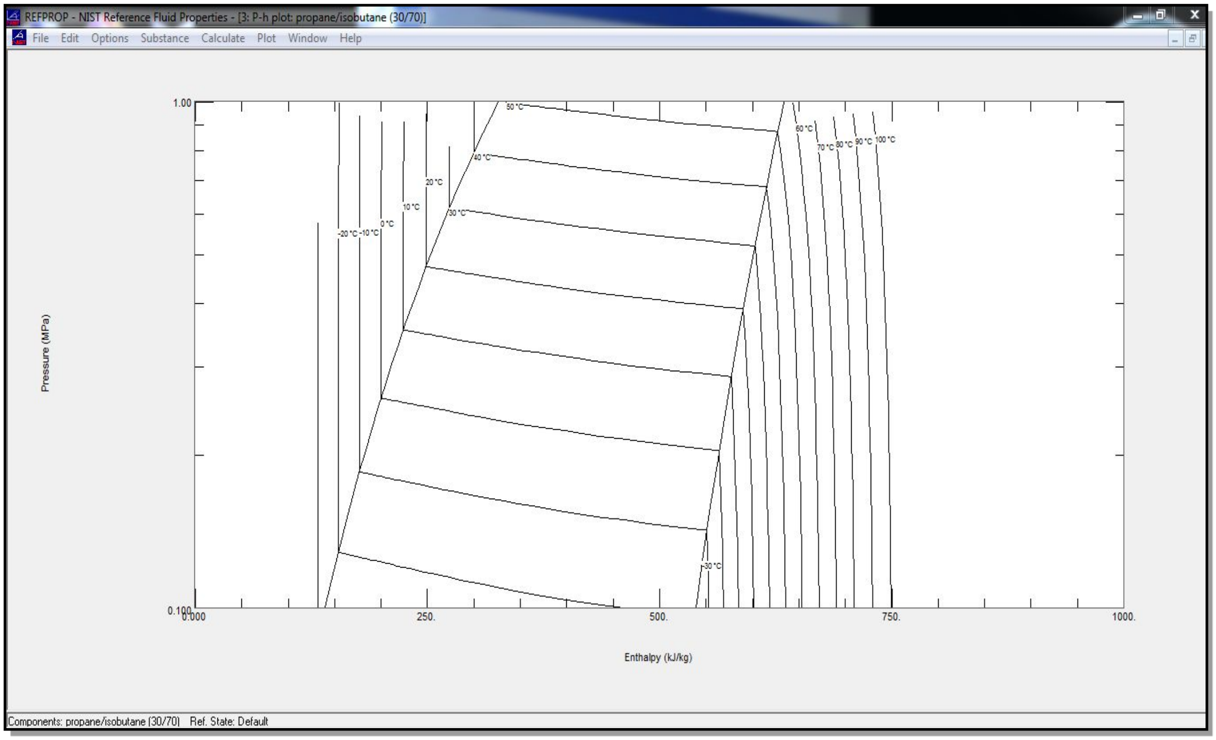
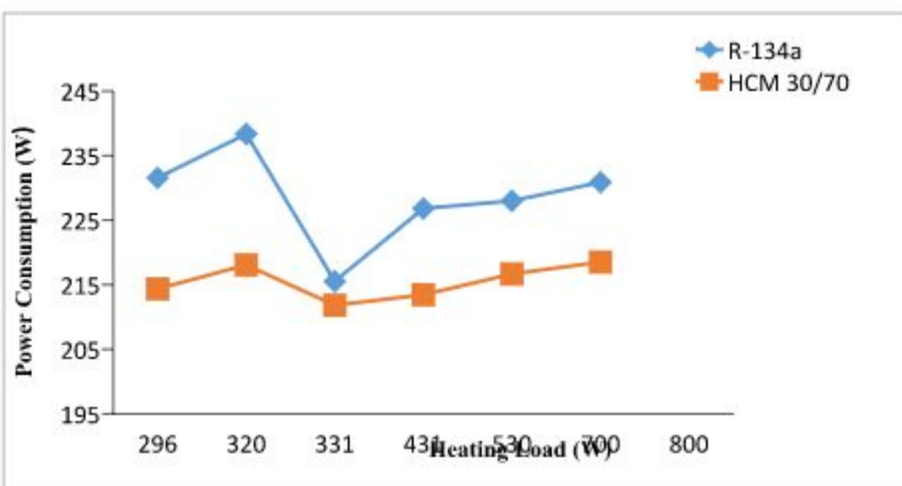


Fig. 4.1 P-h Diagram of HCM 30/70 in REFPROP

4.2 Performance Characteristics

The Power Consumption, Discharge temperature, Actual COP, Theoretical COP, Efficiency of Plant and Tons of Refrigeration (TR) are important parameter considered for choosing an alternative refrigerant.

4.2.1 Power Consumption against Heating Load:



Graph 4.1 Heating Load vs. Power Consumption

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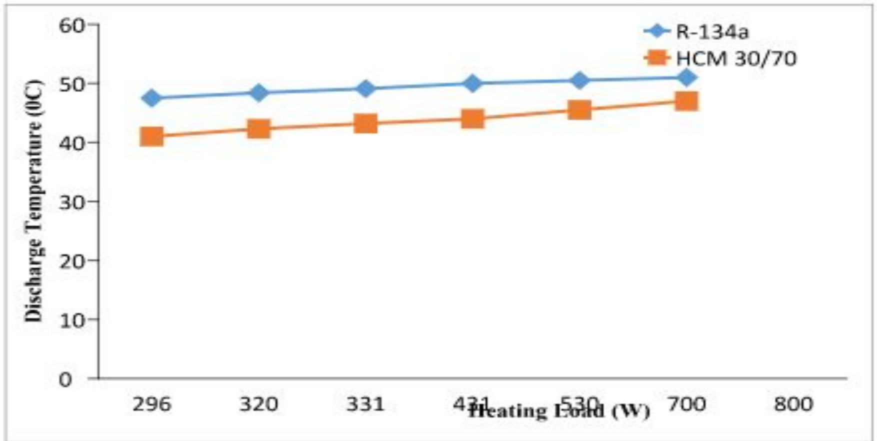
From the graph and results it was founded that power consumption of compressor get reduced when we used HCM 30/70 for comparing the R-134a as a refrigerant. Also when we used HCM 30/70 as a refrigerant compressor body temperature was lower than the R-134a as a refrigerant.

4.2.2 Discharge Temperature against heating Load:

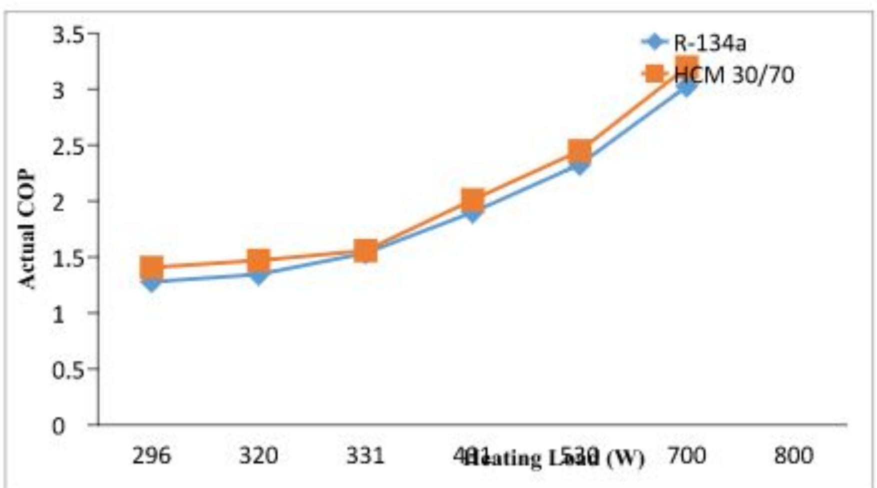


Graph 4.2 Heating Load vs. Discharge Temperature

The discharge temperature influences the stability of the lubricants and compressor components. Graph 4.2 reveals that discharge temperature of optimized HCM 30/70 was found to be lower than that of R-134a.

HCM has lower impact on compressor components and stability of lubricants. Hence, longer compressor life time can be expected when HCM is used as an alternative. [12]

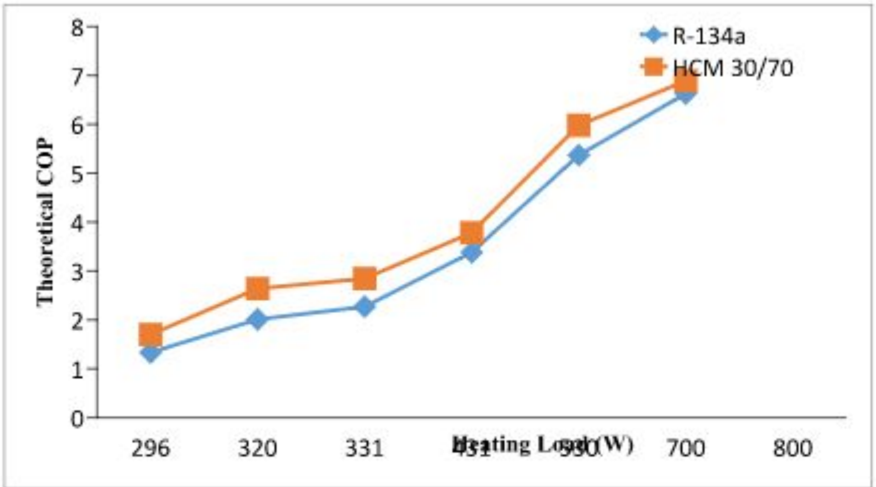
4.2.3 COP against Heating Load:



Graph 4.3 Heating Load vs. Actual COP

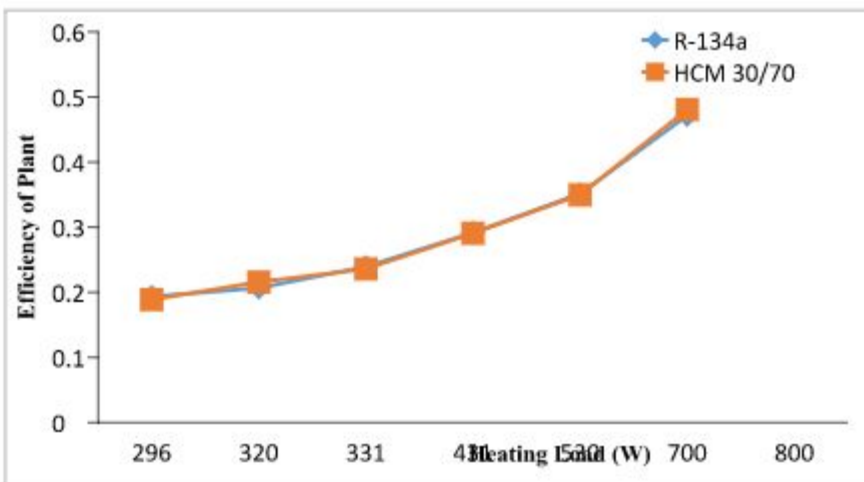
Maximum actual COP for R-134a was found to be 3.0267 whereas for HCM 30/70 it was recorded 3.6501. Same way maximum theoretical COP for R-134a was recorded to be 6.629 and same to be 7.36 for HCM 30/70.

Apart from maximum COP, for the same heating load (698 W) too COP of HCM 30/70 was found to be higher than that of R-134a refrigerant.



Graph 4.4 Heating Load vs. Theoretical COP

4.2.4 <u>Efficiency of plant vs. Heating Load:</u>

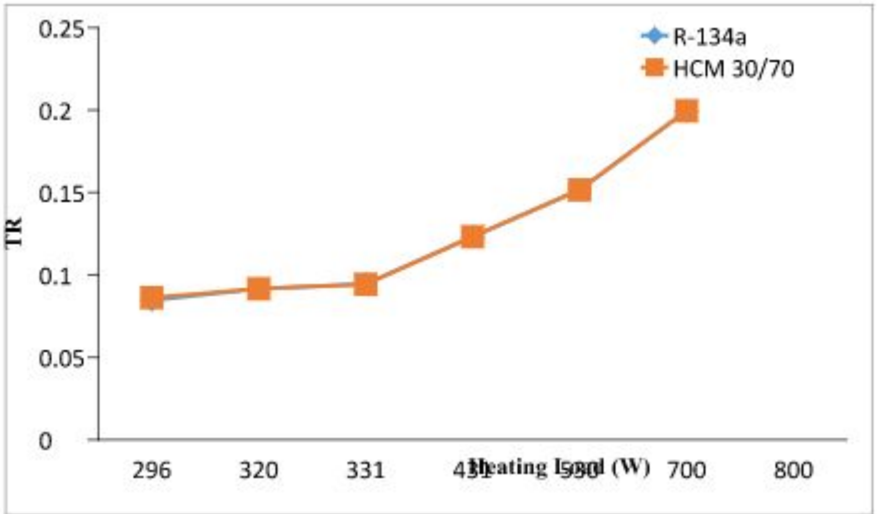


Graph 4.5 Heating Load vs. Efficiency of Plant

Efficiency of VCRS using HCM 30/70 was found to be very closer to that of R-134a. Whereas maximum plant efficiency using HCM 30/70 as a refrigerant was found to be 0.5016.

Hence HCM 30/70 is to be best alternative to that for R-134a as a refrigerant.

4.2.5 <u>Ton of Refrigeration against Heating Load:</u>



Graph 4.6 Heating Load vs. Tons of Refrigeration

Maximum TR (cooling capacity) obtained from HCM mixture was 0.2296 and same for R-134a was recorded 0.1996. For the same heating load TR for HCM 30/70 also coincided with that of R-134a results.

V. CONCLUSION

1) Theoretical study using REFPROP software for various Hydrocarbon mixtures (on mass basis) on VCRS cycle were done and from analysis it was found that HCM with 30% Propane and 70% R-600aon mass basis is optimized amongst the all composition.

2) For the experimentation charged required for R-134a was 522 g whereas for HCM it was found to be 272 g only. Hence HCM reduced the charge of refrigerant in system by 48 %.

3) Power consumption of compressor with the HCM 30/70 as a refrigerant was found to be lower than that of R-134a. Also discharge and suction pressure of HCM 30/70 as a refrigerant is lower than when we used R-134a as a refrigerant.

4) Discharge pressure and Discharge temperature for HCM 30/70 were found to be lower than that of R-134a, hence longer the life of Compressor components.

5) Actual COP and Theoretical COP for HCM 30/70 were greater than that of R-134a by 5.35% and 3.79% respectively for the heating load of 698 W.

6) Efficiency of plant (system) using HCM 30/70 as a refrigerant was much more comparable to R-134a. When we compared cooling capacity of the refrigerants it was found that HCM 30/70 given maximum TR of 0.5016 whereas R-134a given 0.4715.

7) At no load condition time required for lowering the water bath temperature (T5) from ambient temperature for HCM 30/70 is lower than R-134a.

8) Hydrocarbon refrigerants are flammable in nature hence to work on its safety to accomplish better environment is future need to phase out R-134a as early as possible.

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