Design and Fabrication of Vapour Compression Refrigeration System's Condenser for Blended Refrigerants to Improve the System Performance

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ABSTRACT

Flow condensing experiments for refrigerant R-134a, and R-600a mixed with the NANO GASES (TiO_2, SiO_2, AL_2O_3 etc) in serpentine small-diameter (3 mm) U-tubes are to be reported. The tests will be run at the saturation temperature of 40 ^{0} C, vapor qualities of 0.41–0.82, mass flux of 300–600 (kg/m^2s) and NANO GAS concentrations from 0 to 5 mass % refrigerant mixture. It was found that the condensation heat-transfer coefficients increased as mass flux values, vapor quality and the number of tube bends increased, but it decreased as the TiO_2 concentration increased. In addition, the two-phase pressure drops increased with increases in mass flux values, the number of tube bends and the TiO_2 concentration. In this investigation the level of COP will be increased. The existing readings will be taken for R134a and readings for mixture will be used the proportion range will calculate based on the REFPROP 7.0 software result .The better COP proportion will and the effective performance of the system with modified Condenser will calculate.

Key words-
Design; Fabrication; VAR; Condenser; performance

1. INTRODUCTION

The vapour compression cycle is presently the most widely used method of cooling for domestic, commercial, and mobile air conditioning and refrigeration. Vapour compression technology has been developed to its present level of maturity by using chloro fluoro carbon (CFC) and hydro chloro fluoro carbon (HCFC) refrigerants. These refrigerants have excellent thermodynamic properties for cooling cycles. They are inexpensive, stable, non-toxic, and were thought to be environmentally safe. This naturally occurring ozone in the upper atmosphere shields the Earth's surface from ultraviolet (UV)radiation emitted from the sun. Depletion of ozone results in the additional transmittance of UV band electromagnetic radiation to the Earth. Overexposure to UV radiation has been linked to skin cancer and other medical problems in humans and other animals. Focus of finding replacement refrigerants has typically been on volumetric flow rates and operating pressures. In these replacement studies, the compressor efficiency was normally assumed constant in the evaluation of the system's COP. However, the efficiency of centrifugal compressors is a function of the chosen refrigerant, with particular sensitivity to the required enthalpy rise and the specific speed. It is therefore crucial to allow for this effect in any serious performance assessment of replacement refrigerants. In this project full assessment of replacement refrigerants for the current CFC compressors. This is an environment friendly approach in the field of pavement construction as almost all sorts of polymer waste can be recycled and used as a reinforcing admixture in the concrete pavements. As waste polymers which are produced in large quantities are non bio degradable they can cause immense environmental issues. Instead of disposing it we can efficiently make use of its properties in the pavement construction.

A theoretical performance study on a traditional vapour-compression refrigeration system with refrigerant mixtures based on HFC134a, HFC152a, HFC32, HC290, HC1270, HC600, and HC600a was done for various ratios and their results are compared with CFC12, CFC22, and HFC134a as possible alternative replacements. Theoretical results showed that all of the alternative refrigerants investigated in the analysis have a slightly lower performance coefficient (COP) than CFC12, CFC22, and HFC134a for the condensation temperature of 50 °C and evaporating temperatures ranging between −30 °C and 10 °C. Refrigerant blends of HC290/HC600a (40/ 60 by wt.%) instead of CFC12 and HC290/HC1270 (20/80 by wt.%) instead of CFC22 are found to be replacement refrigerants among other alternatives in this paper as a result of the analysis. The effects of the main parameters of performance analysis such as refrigerant type, degree of sub cooling, and superheating on the refrigerating effect, coefficient of performance and volumetric refrigeration capacity are also investigated for various evaporating temperatures [1].

Refrigerants are usually stored in cylinders during transport and while on site. During storage, the pressure of the liquid refrigerant must be periodically checked and adjusted. Excessive pressure may cause an explosion. According to Interstate Commerce Commission (ICC) regulations, liquid refrigerants must not be stored above 130°F (54.4°C), although the containers are designed to withstand up to 3 times the saturated pressure at 130°F (54.4 °C). If a container bursts, liquid refrigerant flashes into vapor. Such a sudden expansion in volume could cause a violent explosion inside a building, blasting out windows, walls, and roofs [2]. Experimental data for pure R22, R134a, R407C and their mixtures with polyester oil FUCHS Reniso/Triton SEZ 32 in a tube with porous coating and smooth, stainless steel reference tube are presented. Mass fraction of oil was equal to 1% or 5%. During the tests inlet vapour quality was set at 0 and outlet quality at 0.7. Mass velocity varied from about 250 to 500 kg/m2s. In the case of flow boiling of pure refrigerants, the application of a porous coating on inner surface of a tube results in higher average heat transfer coefficient and simultaneously in lower pressure drop in comparison with the flow boiling in a smooth tube for the same mass velocity. Correlation equation for heat transfer coefficient calculation during the flow boiling of pure refrigerants inside a tube with porous coating has been proposed [3]. Experimental results of heat transfer characteristic and pressure gradients of hydrocarbon refrigerants R-290, R-600a, R-1270 and HCFC refrigerant R-22 during evaporating inside horizontal double pipe heat exchangers are presented. The test sections have one tube diameter of 12.70 mm with 0.86 mm wall thickness; another tube diameter of 9.52 mm with 0.76 mm wall thickness was used for this study. The local evaporating heat transfer coefficients of hydrocarbon refrigerants were higher than those of R-22. The average evaporating heat transfer coefficient increased as the mass flux increased. It is showed the higher values in hydrocarbon refrigerants than R-22. Hydrocarbon refrigerants have higher pressure drop than R-22 in 12.7 mm and 9.52 mm. These results from the study can be used in the case of designing heat transfer exchangers using hydrocarbons as the refrigerant for the air-conditioning systems [4].

Two-phase heat transfer characteristics of R410A–POE oil mixture and R22-mineral oil mixture flow boiling inside a horizontal C-shape curved smooth tube with an outside diameter of 7.0 mm and a curvature ratio of 60 were investigated experimentally. The test results show that the curvature of C-shape curved smooth tube deteriorates the flow boiling heat transfer, and the ratios of the heat transfer coefficients in C-shape curved smooth tube to those in straight smooth tube for R410A–oil mixture and R22–oil mixture are within 0.46 – 0.74 and 0.74–0.90, respectively [5] Local heat transfer coefficients and pressure drops during condensation of pure Dimethyl Ether (DME) and non-azeotropic mixtures of CO_2 and DME inside a horizontal smooth tube have been measured experimentally. The mass fractions of CO_2 and DME in the mixtures have been varied as CO_2/DME (39/61, 21/79 mass %) and the refrigerant mass fluxes have been varied from 200 to 500 kgm_2 s_1. The increase of mass fraction of CO_2 in the mixture decreases the heat transfer coefficient and the pressure drop. At the high refrigerant mass flux, the effect of mass transfer resistance on the heat transfer is decreased [6]. An experimental investigation of the ester oil ISO VG10/refrigerant R134a mixture flashing flow in a 6.0 m long, 3.22 mm ID tube, which is one of the primary steps towards the construction of a methodology for the study of the lubrication and gas leakage in refrigeration compressors. In order to study this pressure drop, an experimental apparatus was designed to allow the measurement of both pressure and temperature profiles along the tube as well as the visualization of the flow patterns [7].

Presents single-phase and two-phase pressure drop data for R-134a/oil mixture flowing in a wavy tube with inner diameter of D = 5.07 mm and curvature ratio 2R/D = 5.18 and R-410A/oil mixture flowing in a wavy tube of D = 3.25 mm and 2R/D = 3.91.Both mixtures have oil concentration C = 0%, 1%, 3% and 5% for the tests. The influence of oil concentration on single-phase friction factor is negligible, provided that the properties are based on the mixture of lubricant and refrigerant. The influence of oil is augmented at a higher mass flux for liquid spreading around the periphery at an annular flow pattern. This is associated the induced swirled flow motion and an early change of flow pattern from stratified to annular flow pattern [8].

This paper presents a new refrigerant-mixture model for condensation in micro-fin tubes. Several modifications have been implemented in the original smooth-tube model to account for mass transfer thermal resistance between the liquid and vapor phases. The comparison shows that the new model is capable of producing consistent prediction results with a mean absolute deviation (MAD) less than 22% for most of the available data sets. The MAD values obtained with the new model are lower compared to the results obtained using another models [9]. Flow condensing experiments for refrigerant R-290, and R-600a mixed with the lubricating oil (EMKARATE RL 32H) in serpentine small-diameter (2.46 mm) U-tubes are reported. The tests were run at the saturation temperature of 40 _C, vapor qualities of 0.41–0.82, mass flux of 300–600 (kg/m^{2}s) and inlet oil concentrations from 0 to 5 mass% oil. It was found that the condensation heat-transfer coefficients increased as mass flux values, vapor quality and the number of tube bends increased, but it decreased as the oil concentration increased. In addition, the two-phase pressure drops increased with increases in mass flux values, the number of tube bends and the oil concentration [10].

The frictional two-phase multiplier for straight tube can be fairly correlated by using the Chisholm correlation for the data having Martinelli parameter X between 0.05 and 1.0. Fridel correlation also shows a good agreement with a mean deviation of 17.6% to all the straight tube data. For the two-phase pressure drop in U-bend, the revised Geary correlation agrees very well with the R-134a and R-410A oil–refrigerant data with a mean deviation of 16.4% [11].

Pressure and temperature distribution along the flow were measured for saturation pressure ranging from 450 to 650 kPa, mass flux ranging from about 2000 to 3000 kg/(m^2s), temperatures around 303 K, and inlet refrigerant concentration varying between 0.2 and 0.4 kg ref/kg mixt. An available correlation proposed to predict the frictional pressure drop for a mixture composed by the mineral oil SUNISO 1GS and refrigerant R12 flowing in small diameter tubes yielded large eviations in predicting the ester oil and refrigerant R134a mixture flow. A new correlation has been proposed that fitted the experimental data with rms deviations of 24% [12].

2. PROBLEM FORMULATION

a. Global warming and ozone depletion has now become the major concerns in selecting refrigerants for refrigeration and air conditioning equipments.
b. Chlorine containing fluorinated alkalies have a successful association with refrigeration industries. These fluorinated alkalies lead to the damage of ozone layer.
c. Power consumption is found to be high for the existing refrigerant mixtures, which is also a major concern.
d. Heat transfer in the air cooled condenser tube is less and this adversely affects the COP of refrigeration system.
e. Possibility of compressor oil mixing with the refrigerant is greater and this has the inverse effect in COP as well as heat transfer at condenser.

3. OBJECTIVES OF THIS STUDY

a. To investigate the effect of refrigerant mixture with the influence of compressor oil in water cooled condenser.
b. To identify the efficiency of refrigeration even in the influence of oil.
c. Obtain the characteristic curves which show the efficiency of refrigeration.
d. Compare the two refrigerants and its efficiency of refrigeration with the existing setup.

4. METHODOLOGY

Methodology of this study is shown in fig.1

5. DESIGN

Three dimensional model of the double type condenser design (nx8.0) is shown in figure.2

5.1 Specifications of double tube condenser

a. Inner Diameter of Refrigerant tube: 6.35mm
b. Outer diameter of Refrigerant tube: 8mm
c. Inner Diameter of water flow tube: 12.7mm
d. Outer diameter of water flow tube: 14mm

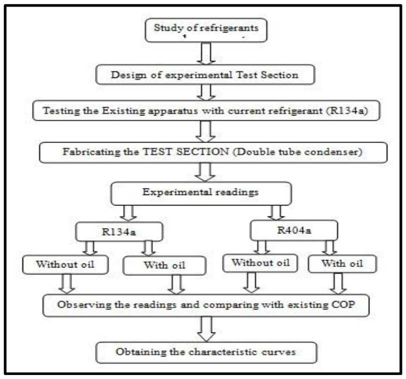

Fig.1. Methodology

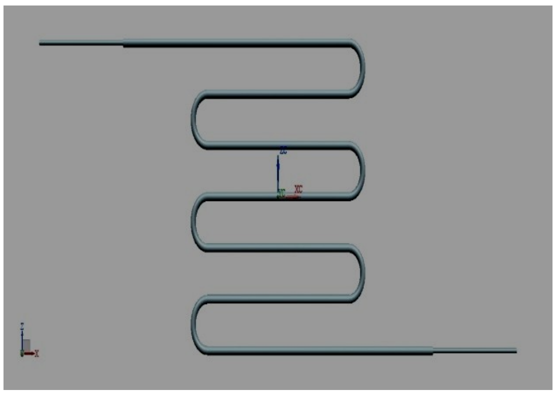

Fig.2. Double tube condenser design (nx8.0)

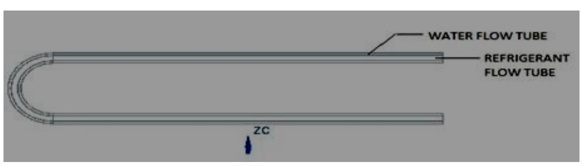

Fig.3. 2D view of double tube condenser

6. EXPERIMENTAL APPARATUS AND METHOD

The experimental apparatus consisted of compressor, condenser, expansion valve and Condenser. The system also consists of two main flow loops: a refrigerant loop and heat source water for condensing loop. The heat exchanger (test section) is shown in Fig. 2. The outer and inner diameter of the inner tube (copper) is 6.35 mm, 8 mm, and outer and inner diameters of the outer tube (copper) are 12.7 mm and 14 mm respectively. The experiment was performed on steady state after conditions control, temperature at the Condenser is the only parameter varied with respect to time. All the observations are taken for the corresponding temperature drop rate at the Condenser.

6.1 Experimental method

In this paper, we used R-134a (tetrafluro ethane) and R-404a (Pentafluoroethane/1, 1, 1-Trifluoroethane/1, 1, 1-Tetrafluoroethane (44/52/4% by weight)) as working fluids. To examine the condensation heat transfer characteristics, the data (temperature of refrigerant, heat source water and outer wall) are measured at the heat exchanger. In addition the pressure between inlet and outlet of heat exchanger are measured as well. Validating parameters are shown below.

7. TESTING IN THE EXISTING APPARATUS

7.1 Specifications of refrigeration system (existing)

a. Compressor-1/8 HP reciprocating compressor
b. Condenser-Fin type condenser
c. Expansion devices- Capillary tube diameter (0.036 inches)
d. Expansion devices-Capillary tube
e. Length-10 feet
f. Condenser coil diameter - ¼ inch length
g. Condenser - Length 11 feet
h. Condenser- 4 liters flask capacity
i. Heat exchanger -2 feet
j. Refrigerant used- R134a (80 gm)

7.2 Result of tetrafluro ethane (r134a) in the existing setup (observations)

a. Condenser outlet/Compressor inlet : 0.135 MPa

b. Compressor outlet/condenser inlet : 1.00 MPa
c. Water inlet temperature (T_o) : 30ºC
d. Final temperature of water (T_f) : -3 ºC
e. Amount of water taken (W) : 3 kg
f. Time taken (t) : 76.27 minutes
g. Energy meter initial reading (E_o) :0.0 KWh
h. Energy meter final reading (E_f) : 0.188 KWh
i. Refrigeration effect : 0.09KW
j. Energy Input : 0.15KW
k. Actual C.O.P : 0.6

8. FABRICATION & TESINGS IN DOUBLE TUBE CONDENSER

Fig.4 Experimental setup of double tube condenser

8.1 Result of R404a pure in double tube condenser (observations)

a. Condenser outlet/Compressor inlet : 0.264MPa
b. Compressor outlet/condenser inlet : 1.608MPa
c. Water inlet temperature (T_o) : 30ºC
d. Final temperature of water (T_f) : -3 ºC
e. Amount of water taken (W) : 3 kg
f. Time taken (t) : 38.13 minutes
g. Energy meter initial reading (E_o) :0.0 KWh
h. Energy meter final reading (E_f) : 0.101 KWh
i. Refrigeration effect : 0.180KW
j. Energy Input : 0.158KW
k. Actual C.O.P : 1.14
l. Condensation heat transfer coefficient(h) : 6997.78W/m^{o}C
m. Pressure difference(ΔP) : 0.0527

9. RESULTS AND DISCUSSION

Temperature of the water at the Condenser is dropped from 30^{o}C to -3^{o}C. Temperature drop at various points are noted and their corresponding time is recorded which is shown in Fig.5. In the existing setup with R134a as refrigerant, it takes about 75 minutes for reaching -3^{o}C. These values are recorded for further comparison with the double tube setup.

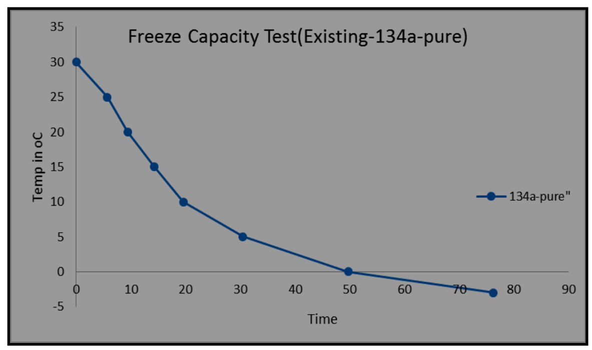

Fig.5. Freeze capacity test (existing)

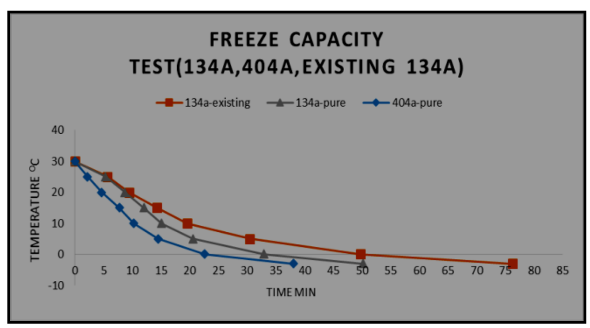

Fig.6. Freeze Capacity Test- R404a-pure & R134a-pure comparing with the existing setup

This test shows the time at which various refrigerants reach at -3^{o}C. The refrigerants used are R134a-pure, R404a-pure and the result is compared with the readings taken in the existing setup. It has been found that the freezing action of R404a (Refrigeration effect) takes place almost 10 minutes before R134a-pure and 35 minutes before the existing air cooled condenser. The freeze capacity tests conducted for R134a & R404a pure (without oil) Fig.7. Shows that R404a has higher freezing capability comparing withR134a at same time period.

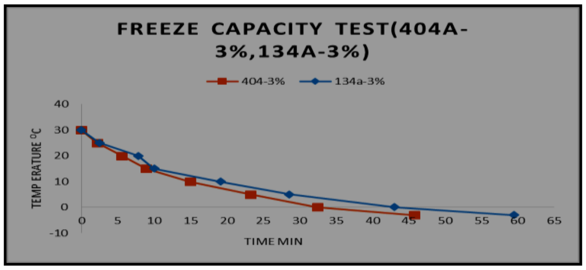

Fig.7 Freeze Capacity Test for R404a-3% oil & R134a-3% oil

Temperature drop of the refrigerants were recorded individually and their corresponding time periods are also determined. Thus the result displays that; refrigerant 404a provides better cooling capacity with 3% oil concentration than R134a with the same oil concentration which is shown in Fig.8.

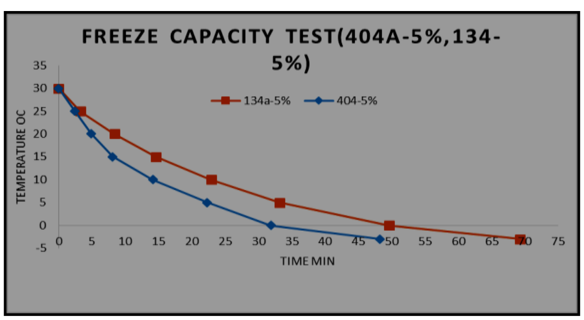

Fig.8. Freeze Capacity Test for R404a-5% oil & R134a-5% oil

Influence of oil concentration in the refrigerants (Fig.9 & 10) decreases the freezing capacity proportional to mass fraction of oil adding to it. R404a with oil has better freezing capacity than R134a with oil. Hence it can be said that the effect of oil on R404a is comparatively less than R134a. Refrigeration effect of R134a greatly reduces with the increase in oil concentration (i.e., 5%). R404a has only lesser reduction in refrigeration effect with higher oil concentration comparing with the Refrigerant 134a. With 5% oil concentration, R134a reaches -3^{o}C at almost 70 minutes but, R404a at 48 minutes. This shows that the efficiency of 404a is more even in the influence of oil.

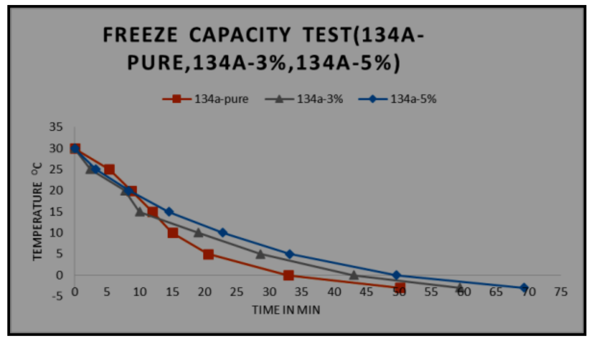

Fig.9 Freeze Capacity Test for R134a-pure, 3%oil & 5% oil

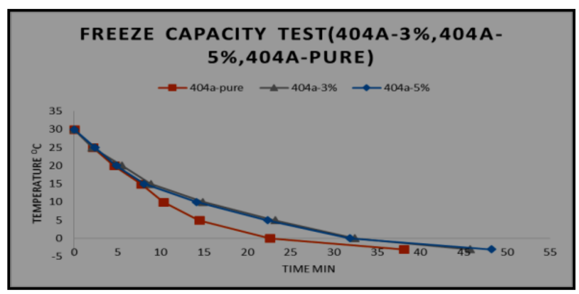

Fig.10.Freeze Capacity Test for R404a-pure, 3% & 5% oil

The results obtained from the COP tests ensure that R404a has consistent COP at various temperature ranges. Also this is retained proportionally with the oil concentration is increased up to 5%. Whereas R134a is not so consistent at different temperature ranges. Also, it reaches smaller values with greater minimum temperatures. Even in case of COP, refrigerant 404a serves better than

Refrigerant 134a. The COP results of R134a-pure in the existing setup were compared with the R134a-pure and R404a-pure in the double tube condenser. Thus the result is very clear that the performance of R404a in the double tube is far greater than R134a in the existing and R134-pure in double tube. Even at the maximum oil concentration (5%), 404a performs better than R134a.

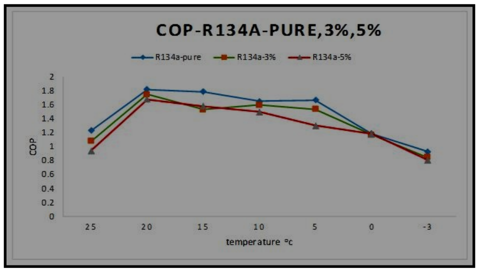

Fig.11. COP Test-R134a-pure, 3%&5% oil

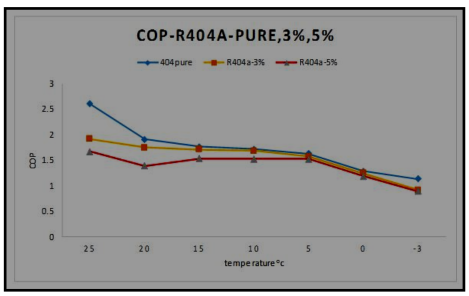

Fig.12. COP Test-R404a-pure, 3%&5% oil

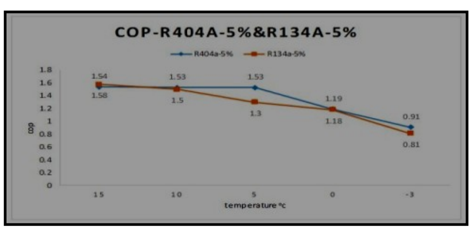

Fig.13. COP Test-R404a-5%, R134a-5%

Flow condensation heat-transfer coefficients of R-134a and R-404a are shown in the Fig.14&15. Tests were conducted with different lubricant concentrations (3% & 5% polyester oil with R134a & R404a) for 6 bend serpentine double tube condenser. The result thus obtained from the condensation heat transfer coefficient test shows that the heat transfer coefficient of refrigerants 134a and 404a drops with increase in percentage of oil addition.

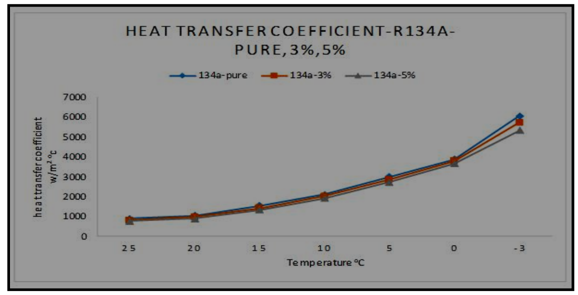

Fig.14. Heat transfer Test-R134a-pure, 3%, 5%oil

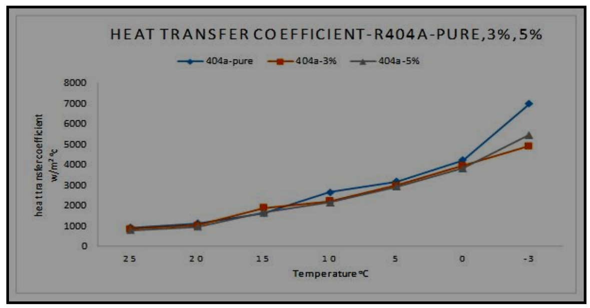

Fig.15. Heat transfer Test-R404a-pure, 3%, 5% oil

It is very clear that R404a provides better heat transfer than R134a even in the higher oil concentration (5% oil). The Pressure drop penalty factor is found for both the refrigerants, R134a & R404a at various oil concentrations (i.e 3% & 5%). From the graph it is clear that PF values of R134a & R404a is greater than one. Hence it is clear that the pressure difference is caused between the inlet and outlet of the condenser. Due to the greater ideal pressure of 404a, pressure drop occurring at the condenser outlet is comparatively lesser than R134a. But, greater the concentration of oil, greater the pressure drop.

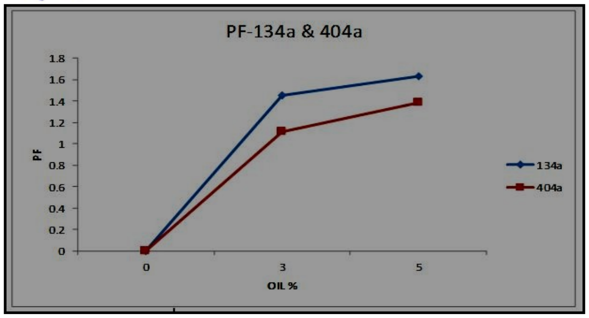

Fig.16. PF-R404a & R134a

SUMMARY

Compressor oil is usually added for the effective lubricity of compressor. Addition of compressor oil is hence, unavoidable in case of vapour compression refrigeration system. The results are very clear that the compressor oil has a major influence in the COP, heat transfer rate, and pressure drop in the system. The current study covered the effect of compressor oil in the refrigerants namely, R134a & R404a. The result shows that R404a works better in case of COP, heat transfer coefficient & freezing capacity than R134a, which is the current and commonly using refrigerant in home appliances. Due to the consistent performance of R404a, it can be preferred to R134a for the domestic appliances as well as commercial applications.

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