Experimental Study on Auto Cascade Refrigeration Cycle using Mixed Refrigerant R410A/R134a

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Abstract— **An experimental investigation is presented in this study to evaluate the performance of three stages auto cascade refrigeration cycle with zeotropic mixture. Three stages auto cascade refrigeration system of one ton capacity is fabricated in current work as a test rig to perform the experimental work. The performance of a three stages auto-cascade refrigeration system is compared at two operation modes, the first one when the system is charged by 2100g of R-410A and R-134a zeotropic mixture with mass ratio (70/30), and the second when the system is charged by ?? g R-134a. The results showed that, the mixed refrigerant displayed a higher performance compared with R-134a. The enhancement in coefficient of performance (COP) and refrigeration effect were in the range of 24% and 41% respectively. While, the reduction in compression ratio and discharge temperature were in the range of 12% and 39% respectively as compared with that for R-134a. A lower evaporator inlet temperature of (−43.3°C) was achieved for mixed refrigerant compared with (-31.5°C) for refrigerant R-134a under the same operating conditions.**

Keywords— Auto-Cascade system; Zeotropic refrigerant; Coefficient of performance

I. INTRODUCTION

The simple vapor-compression cycle may not be suitable for some applications in the refrigeration field, and more advanced and innovative refrigeration systems may have to be used. Other motivations include the search for improved performance and efficiency as well as requirements to achieve a very low temperatures [1, 2]. The auto cascade refrigeration system represents one of such advanced systems which is used in the present work with zeotropic mixed refrigerants. In many industrial applications that require moderately low temperatures (with a considerably higher temperature and pressure difference), single vapor-compression refrigeration cycles become impractical. One of the solutions for such cases is to perform the refrigeration in two or more stages (two or more cycles) which operate in series. These refrigeration cycles are called cascade refrigeration cycles. Therefore, cascade systems are used to obtain high-temperature differentials between the heat source and heat sink and are applied in temperatures ranging from −70 to 100°C. Application of a three-stage compression system for evaporating temperatures below −70°C is limited, because of difficulties with refrigerants reaching their freezing temperatures. The impropriety of multi-stage vapor-compression systems can be avoided by applying a cascade vapor compression refrigeration system [3]. The auto-cascade refrigeration system uses mixed refrigerants to achieve multilevel cascade through one compressor, which can obtain a low temperature of -60 °C and therefore greatly simplify the system [4]. An auto-cascade refrigeration system operates with lower evaporating temperature, smaller compression ratio and higher compressor volumetric efficiency when compared with a single-stage one. The performance of auto cascade refrigeration system has been studied by many researchers. S.G. Kim and M.S. Kim [5] investigated the performance of an auto cascade refrigeration system using zeotropic refrigerant R744/R134a and R744/R290. It was found that the auto cascade refrigeration cycle has a merit of low operating pressure as low as that in a conventional vapor compression refrigeration cycle. Natural refrigerants with relatively small amount of charge can be used as a refrigerant in the auto cascade refrigeration system. Gong et al. [6] analyzed the thermodynamic characteristics of components and the whole refrigeration cycle of an auto-cascade system. Extensive comparison was made between the single stage Joule-Thomson (J-T) cycle that consist of compressor, heat exchanger, expansion valve and evaporator, and a typical auto-cascade cycle under the same conditions. The results showed that, using appropriate auto-cascade cycle can improve the performance of the refrigerator. Kai Du et al. [4] studied the comparison between the single refrigeration system and the auto-cascade refrigeration system using zeotropic mixture as refrigerant, which achieves cascade between high and low boiling point components by an evaporative condenser for the purpose of obtaining a lower evaporation temperature.

The coefficient of performance, cooling capacity, evaporation temperature and pressures and temperatures of the refrigerant at the inlet and outlet were measured and parameters analyses were conducted with different charging concentrations, and different operating conditions. Nayak et al. [7] studied the performance of an ARC refrigerator operating in the liquid refrigerant supply mode and operating with optimized hydrocarbon mixtures and different cascade heat exchangers. The optimum number of cascade heat exchangers (stages) to be used for different operating temperatures was suggested. Yan Gang et al. [8] proposed an internal auto-cascade refrigeration cycle (IARC) operating with the zeotropic mixture of R-290/R-600a or R-290/R-600 for domestic refrigerator-freezers. Performances of the auto-cascade refrigeration were evaluated using a developed mathematical model, and then compared with that of the conventional refrigeration cycle (CRC). According to the simulation results, the IARC with R-290/R-600a has 7.8–13.3% improvement in coefficient of performance, 10.2–17.1% improvement in volumetric refrigeration capacity and 7.4–12.3% reduction in pressure ratio of compressor compared with those of the CRC under the same given operating conditions. In this work the performance of a three stages auto-cascade refrigeration cycle was investigated under two refrigerant types. The first one was zeotropic refrigerant consist of R- 410A and R- 134a of mass ratio of (70/30), and the seconde one is pure R-134a.

II. AUTO-CASCADE CYCLE ANALYSIS

The performance of the auto-cascade refrigeration system depends on various factors, such as the effect of, gas–liquid separation, heat transfer of the heat exchanger and mixed working fluids. Performance parameters of the auto-cascade refrigeration system are evaluated in terms of energy aspects based on the first law of thermodynamics. The following assumptions are made for the analysis of the auto-cascade refrigeration system: all components are assumed to be a steady-state and steady flow process, the compression process in the compressor is irreversible, the evaporator exit vapor and condenser exit liquid are both saturated and refrigerant pressure drop and heat losses in the cycle are neglected. Refer to Fig. (1), the work of compressor can be determined by [9, 10]:

$$\dot{W}_{comp.} = \dot{m}(h_2 - h1) \quad (1)$$

$$\dot{m} = \dot{m}_{RI} + \dot{m}_{RII} \quad (2)$$

The heat rejected from the air cooled condenser is calculated as follows:

$$Q_{ACC} = \dot{m}(h_3 - h_4) \quad (3)$$

Heat absorbed by the evaporator can be calculated by:

$$Q_{evap.} = \dot{m}_{evap.}(h_{16} - h_{15}) \quad (4)$$

The overall performance of the three stage auto cascade refrigeration system is determined by evaluating the coefficient of performance as follows:

$$COP = \frac{Q_{evap.}}{\dot{W}_{comp.}} \quad (5)$$

Thermo physical properties at different state points were found for corresponding temperature and pressure readings taken during the experimentation. These values are calculated using the software ‘‘REFPROP-9’’ [11], which in turn is used to calculate the thermo physical properties of individual and mixture combinations at different state points.

III. EXPERIMENTAL SETUP

In present work, a test rig of one-ton capacity auto-cascade refrigeration system is built to investigate experimentally the performance of the system with zeotropic mixed refrigerant. The system setup consists of one rotary compressor, air cooled condenser, shell and coil evaporator, two tube-in-tube heat exchangers, two phase separators, expansion valves, flow meters, pressure gauges, thermocouples with readers, and other accessories. The system is at first charged with 2100 g of R-134a and operated to insure that the system works properly and to investigate the performance parameters for comparison purpose with mixed refrigerant. The test rig system is then charged with a mixed refrigerant R-410A/R-134a at mass ratio (70/30) and operated at different operating conditions. The schematic diagram of auto cascade refrigeration system used to achieve low temperature in the current work is shown in Fig. (1).

IV. RESULT AND DISCUSSIONS

Auto-cascade refrigeration system offer many benefits, such as a low compression ratio and relatively high volumetric efficiency as shown in the Fig. (2), which shows a variation of pressure ratio , discharge and suction pressures with operating time for ARS with mixed refrigerant R-134a/R410A at mass ratio (70/30) and a traditional vapour compression cycle with R-134a. It can be seen from this figure that, R-134a/R410A gives a higher discharge and suction pressures, as compared with that for pure refrigerant R-134a due to mixed refrigerant has a higher condensing and lower evaporation temperatures than that for R-134a as shown in Figs. (3) and (6). Although mixed refrigerant has higher discharge and suction pressures, but the difference between the discharge and suction pressures is less than that for R-134a, which leads to low pressure ratio for mixed refrigerant as compared with that for R-134a. Therefore, ARS with mixed refrigerant has displayed a lower compression ratio in range of 12% compared with R-134a.

The Comparison between discharge temperatures for the auto-cascade system uses R-410A /R-134a at mass fraction (70/30) and R-134a is shown in Fig. (3). The figure shows lower discharge temperature for mixed refrigerant in range of 39% compared with that for R-134a. This difference in temperature is due to the thermodynamic properties of R-134a that gives a relatively higher temperature at assigned pressure compared with the mixture R-410A /R-134a.

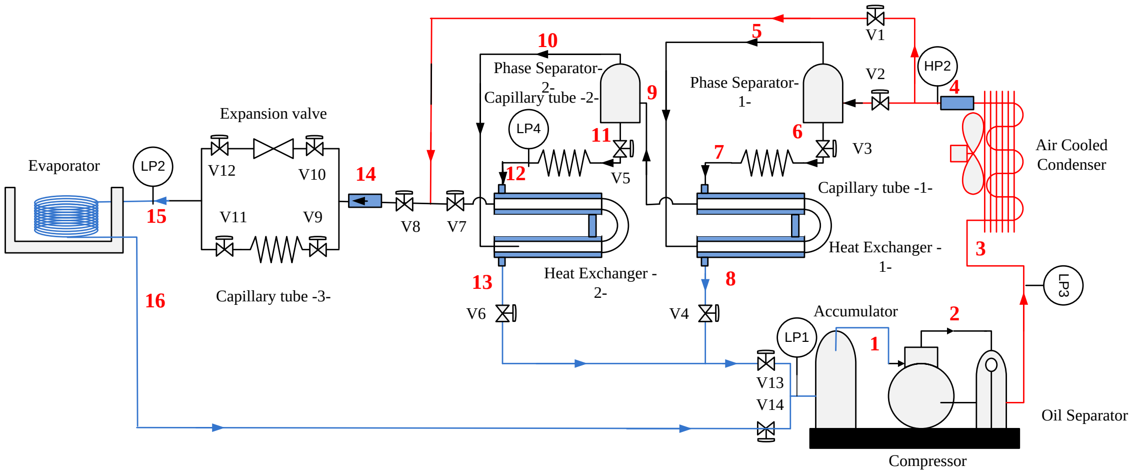

Fig. (1) Schematic diagram of the test rig for ARS.

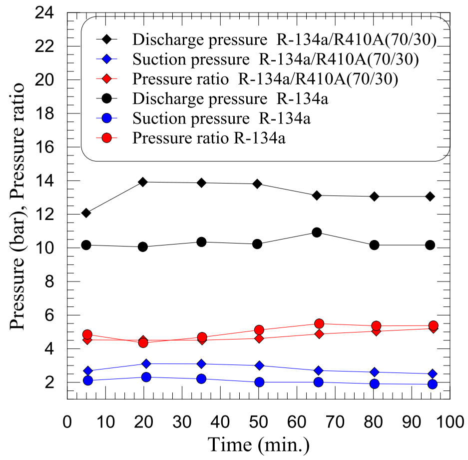

Fig.(2) Variation of pressure ratio , discharge and suction pressures with operating time of ARS for mixed refrigerant R-134a/R410A and R-134a.

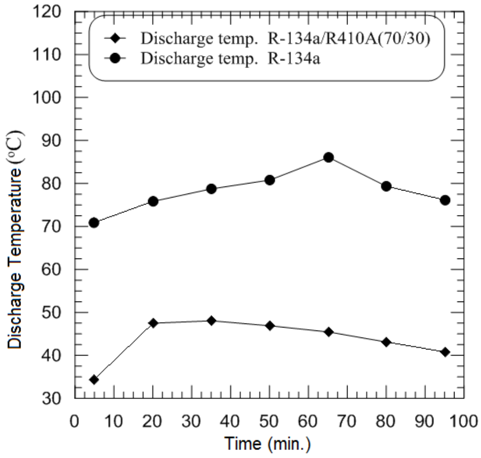

Fig.(3) variation of discharge temperature with operating time of ARS for mixed refrigerant R-134a/R410A and R-134a.

Fig.(4) shows the variation of refrigeration effect with operating time of ARS for mixed refrigerant R-134a/R410A and R-134a. It can be seen from the figure that, using mixed refrigerant improves the cycle refrigeration effect, due to the effect of glide temperature for zeotropic mixed refrigerant. This is due to the latent heat of evaporation of R-410A is much more than that for R-134a. As the refrigeration effect increases, the cycle COP increases also as shown in Fig. (5). The enhancement in the cycle COP and refrigeration effect for mixture R-410A /R-134a are in range of 24% and 41% respectively compared with R-134a.

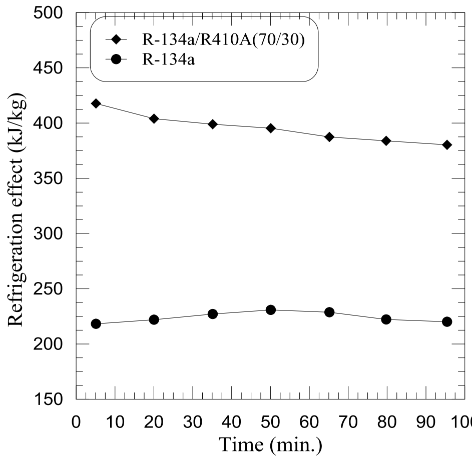

Fig.(4) Variation of refrigeration effect with operating time of ARS for mixed refrigerant R-134a/R410A and R-134a.

Fig.(6) shows the variation of evaporator inlet temperature with operating time of ARS for mixed refrigerant R-134a/R410A and R-134a. It is observed that, the evaporator inlet temperature decreases with time and approached a lower value (-43.3^{o}C) for mixed refrigerant compared with (-31.5°C) for R-134a, as a consequence of relatively low boiling point for R-410A /R-134a compared with R-134a.

The effect of evaporation temperature on both refrigeration effect and cycle COP for mixed refrigerant is shown in Fig..(7). It is observed that the values of cycle COP and refrigeration effect decrease with reduction in evaporator temperature, where from the definition, the value of COP depends on value of the refrigeration effect which decreases with reduction in evaporator temperature due to the reduction in thermal load applied in the evaporator within ARS operating period.

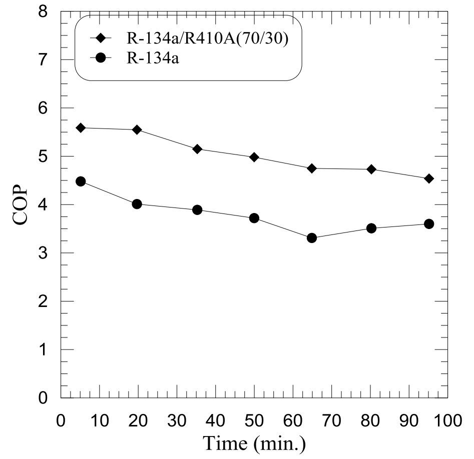

Fig.(5) Variation of coefficient of performance with operating time of ARS for mixed refrigerant R-134a/R410A and R-134a.

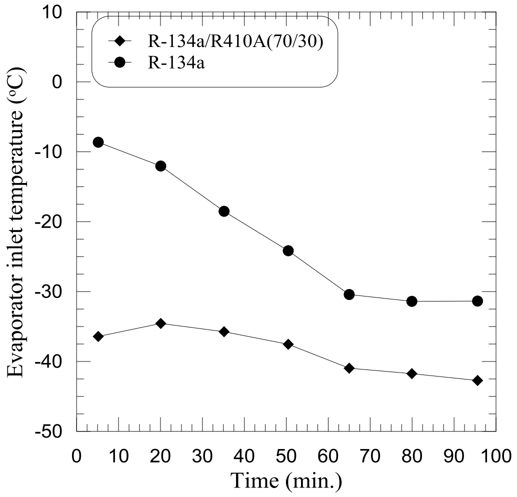

Fig.(6) Variation of evaporator inlet temperature with operating time of ARS for mixed refrigerant R-134a/R410A and R-134a.

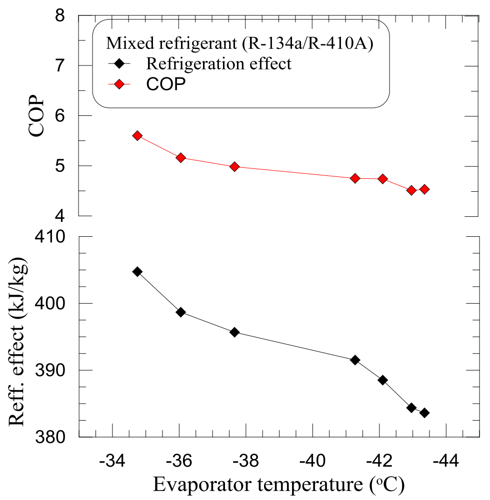

Fig.(7) Variation of refrigeration effect and COP with evaporator inlet temperature for ARS with mixed refrigerant R-410A /R-134a (70/30).

IV. CONCLUSION

1) The performance of the auto-cascade refrigeration system can be significantly improved with mixed refrigerant R-410A /R-134a at mass fraction (70/30) compared with pure refrigerant R-134a.

2) The enhancement in values of COP and refrigeration effect for the auto-cascade system with mixture R-410A /R-134a was in range of 24% and 41% respectively compared with that for R-134a.

3) Reduction in compression ratio and discharge temperature for mixed refrigerant was in range of 12% and 39% respectively compared with that for R-134a.

4) A low value of evaporator inlet temperature (-43.3°C) was achieved for mixed refrigerant R-410A /R-134a compared with (-31.5°C) for refrigerant R-134a under the same operating conditions.

SYMBOLS:

ACC: Air cooled condenser
ARS : Auto cascade refrigeration system
CRC: Conventional refrigeration cycle
COP: Coefficient of performance
h: Specific enthalpy (kJ/kg)
$\dot{m}$: Mass flow rate of mixed refrigerant (kg/s)
$\dot{m}_{evap.}$: Mass flow rate of refrigerant through evaporator (kg/s)
$\dot{m}_{RI}$: Mass flow rate of refrigerant I (kg/s)
$\dot{m}_{RII}$: Mass flow rate of refrigerant II (kg/s)
Q_{ACC}: Rate of heat transfer of condenser (kW)
$Q_{evap.}$: Rate of heat transfer of evaporator (kW)
$\dot{W}_{comp.}$: Power consumed (kW)

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