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Experimental Assessment of Energy Efficiency of Microchannel and Conventional Condensers using R134a and R290/R600a Refrigerant Blend

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Abstract

An experimental performance study of a vapour compression refrigeration system using R134a and a drop-in hydrocarbon refrigerant mixture of R290/R600a (50/50% by mass) was conducted with both a conventional condenser and an aluminium microchannel condenser. The experiments were performed at condenser temperatures of 44°C, 48°C, 55°C, and 58°C, while the evaporating temperature was varied from −10°C to 15°C in increments of 5°C (−10°C, −5°C, 0°C, 5°C, 10°C, and 15°C). The refrigerant charge was reduced by 10% for R134a and by 60% for the R290/R600a (50/50% by mass) mixture when the microchannel condenser was used instead of the conventional condenser. At a condenser temperature of 44°C and over the evaporating temperature range of −10°C to 15°C, the compressor power consumption decreased by 4.61% to 7.14% for R134a with the microchannel condenser compared with the conventional condenser. In contrast, the compressor power increased by 13.26% to 18.46% for the R290/R600a (50/50% by mass) mixture. The coefficient of performance (COP) increased by 5.89% to 15.78% for R134a and by 8.73% to 11.60% for the R290/R600a mixture when the microchannel condenser was employed. The refrigeration capacity of the system equipped with the microchannel condenser increased by 1.22% to 17.00% for R134a and by 3.07% to 20.65% for the R290/R600a (50/50% by mass) mixture compared with the conventional condenser. Similarly, the condenser heat rejection capacity increased by 6.74% to 18.33% with the microchannel condenser. Furthermore, the overall heat transfer conductance (UA) increased by 13.68% to 17.42% for R134a and by 4.34% to 26.5% for the R290/R600a mixture when the aluminium microchannel condenser replaced the conventional condenser.

1. Introduction

Vapour compression refrigeration (VCR) systems are extensively used worldwide in heating, ventilating, air-conditioning, and refrigeration (HVACR) applications. However, these systems consume a significant amount of high-grade electrical energy, making the improvement of their

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energy efficiency a major research objective. In addition, the widely used refrigerant R134a possesses a global warming potential (GWP) of approximately 1400, which has accelerated the search for environmentally sustainable alternatives with lower environmental impact. One of the effective approaches for improving the performance of VCR systems is to enhance the heat transfer characteristics of heat exchangers while minimizing refrigerant-side pressure drop. The use of smaller diameter tubes and optimized refrigerant flow distribution through multiple passes has been reported to improve condenser performance and system efficiency (Shigeru Koyama et al., 2003; Primal Fernando et al., 2008). Microchannel heat exchangers have emerged as a promising technology for refrigeration and air-conditioning applications due to their compactness, high heat transfer coefficients, large surface area-to-volume ratio, and reduced refrigerant inventory. Initially developed for high heat flux applications such as electronic cooling, microchannel technology has gained considerable attention in HVACR systems because of its potential to reduce material consumption, improve energy efficiency, and minimize greenhouse gas emissions (Satish G. Kandlikar et al., 2006; Marko Matkovic et al., 2009; Wei Li and Zan Wu, 2010). The increasing demand for energy-efficient thermal systems, compact equipment, and sustainable manufacturing has encouraged the development of high-performance heat exchangers. Properly designed microchannel heat exchangers provide uniform refrigerant distribution, shorter flow paths, predominantly laminar flow, lower pressure drop, and significantly enhanced heat transfer performance. These characteristics make them suitable for next-generation thermal management systems (Rin Yun et al., 2006; Chang Y. et al., 2008; M. Ohadi et al., 2013). Their successful implementation in the automotive industry has further encouraged their adoption in residential and commercial HVACR systems (Hee Sung Park and Jeff Punch, 2008; Liang et al., 2009; Sung-Min Kim and Issam Mudawar, 2013).

Several researchers have reported that aluminium microchannel condensers not only improve heat transfer performance but also substantially reduce refrigerant charge compared with conventional round tube plate fin condensers (Bjorn Palm, 2007; F. Poggi et al., 2008; Akhil Agarwal et al., 2010; Zuhaogang et al., 2010). The reduced internal volume of microchannel tubes makes them particularly suitable for applications employing environmentally friendly hydrocarbon refrigerants. Hydrocarbon refrigerants such as propane (R290) and isobutane (R600a) have negligible ozone depletion potential and very low global warming potential. Their latent heat of vaporization is considerably higher than that of hydrofluorocarbon (HFC) refrigerants, resulting in improved refrigeration performance. However, their primary limitation is flammability, and therefore they are classified as A3 refrigerants according to ASHRAE Standard 34 (Dongsoo Jung et al., 2000; Bjorn Palm, 2008; Xi Lu and A. G. Agwu Nnanna, 2008; J. B. Copetti et al., 2009). Owing to their lower liquid density, hydrocarbon refrigerants require only about 40–42% of the refrigerant charge required by HFC or HCFC refrigerants (K. Mani and V. Selladurai, 2008; 2013), making them attractive alternatives when refrigerant charge can be minimized. Previous studies have demonstrated that the use of microchannel condensers significantly reduces refrigerant charge and enhances system safety. It has been reported that when the hydrocarbon refrigerant charge is maintained below approximately 150 g, the refrigerant concentration remains below the lower flammability limit even in relatively small enclosed spaces, thereby considerably reducing the explosion risk (A.

S. Dalkilic and S. Wongwises, 2010). Consequently, aluminium microchannel condensers have become an effective solution for improving system efficiency while enabling the safe application of hydrocarbon refrigerants in small-capacity refrigeration and air-conditioning systems. In the present study, an aluminium microchannel condenser with an integral receiver and filter drier was designed and fabricated for a 1 TR vapour compression refrigeration system, and its geometrical specifications are presented in Table 1 (Bernard Thonon, 2008). The refrigerant charge was reduced by approximately 10% for R134a and nearly 60% for the hydrocarbon mixture R290/R600a (50/50% by mass) when the microchannel condenser was used instead of the conventional condenser. Pure R290 exhibits relatively high operating pressures, whereas pure R600a operates at comparatively lower pressures. However, a binary mixture of R290/R600a (50/50% by mass) possesses thermodynamic characteristics similar to those of R134a, making it a suitable drop-in replacement (Akio Miyara, 2008; M. Rasti et al., 2012; V. W. Bhatkar et al., 2013a; 2013b). The objective of the present investigation is to experimentally evaluate the performance of a vapour compression refrigeration system using conventional and aluminium microchannel condensers with R134a and the hydrocarbon mixture R290/R600a (50/50% by mass). Experiments were conducted at condenser temperatures of 44°C, 48°C, 55°C, and 58°C, while the evaporator temperature was varied from −10°C to 15°C at intervals of 5°C. The system performance was evaluated in terms of refrigeration effect, compressor work, condenser heat rejection, coefficient of performance (COP), condenser pressure drop, and the overall heat transfer conductance (UA). The experimental results provide a comprehensive comparison of the thermal performance of conventional and microchannel condensers using both refrigerants (Ching-Song Jwo et al., 2009; Amr O. Elsayed and A. S. Hariri, 2011). Microchannel heat exchangers have emerged as one of the most promising technologies for next-generation refrigeration and air-conditioning systems because of their compact size, high heat transfer coefficients, reduced refrigerant inventory, and lower material consumption. The large surface-area-to-volume ratio of microchannels enhances heat transfer while reducing the overall size and weight of the condenser. Recent studies have reported that microchannel heat exchangers significantly improve thermal performance, reduce pressure drop, and contribute to higher system energy efficiency, making them suitable for compact and environmentally sustainable HVACR applications (Satish G. Kandlikar et al., 2006; Marko Matkovic et al., 2009; Wei Li and Zan Wu, 2010; Başaran and Benim, 2024).

Growing concerns regarding energy conservation, compact equipment design, and refrigerant charge reduction have accelerated research on microchannel condensers. Properly designed microchannel condensers provide uniform refrigerant distribution, shorter flow paths, enhanced condensation heat transfer, and improved refrigerant-side pressure characteristics. These advantages result in higher heat transfer effectiveness and reduced refrigerant charge compared with conventional fin-and-tube condensers. Recent review studies have highlighted that microchannel technology is becoming a key component in the development of high-performance thermal management systems and advanced refrigeration equipment (Rin Yun et al., 2006; Chang et al., 2008; M. Ohadi et al., 2013; Yu et al., 2024). The successful application of aluminium microchannel heat exchangers in the automotive industry has encouraged their adoption in residential and commercial HVACR systems. Aluminium microchannel condensers offer superior corrosion resistance, reduced refrigerant inventory, lower manufacturing cost, and improved heat transfer performance. Experimental and review investigations have demonstrated that microchannel condensers provide better thermal performance than conventional round tube plate-fin condensers while supporting the transition towards low-GWP refrigerants (Hee Sung Park and Jeff Punch, 2008; Liang et al., 2009; Sung-Min Kim and Issam Mudawar, 2013; Sayfetdinov et al., 2024). Recent investigations have also focused on enhancing the operational reliability of microchannel heat exchangers under frosting conditions and improving condensation characteristics in mini- and microchannels. Advanced designs have shown considerable potential for improving heat transfer performance while reducing refrigerant charge and enhancing system safety. Furthermore, systematic reviews have identified microchannel condensers as one of the most effective technologies for next-generation refrigeration and air-conditioning systems employing environmentally friendly refrigerants (Ye et al., 2024; Başaran and Benim, 2024). Recent studies have also examined the influence of lubricating oil on flow and heat transfer characteristics in microchannels, providing useful insights for optimizing condenser design and improving overall system performance under practical operating conditions (International Journal of Heat and Mass Transfer, 2024).

2. Materials and Methods

Experimental setup was prepared for 1TR vapour compression refrigeration system with the forced convection air cooled condenser with three air heaters of 2.2 kW capacity in the condenser compartment, water cooled evaporator with stirrer (12W motor, 2880 RPM) and two 2.2 kW capacity water heaters, manually controlled expansion valve and the measuring instruments. In the present experimental set up, condenser pressure was controlled with the help of air heater supply and multi speed fan regulator. Conventional condenser with the round copper tubes and aluminium fins with the diameter of 9.525 mm was used**.** Microchannel condenser was having total aluminium as the material for both for rectangular tubes as well as for fins. The hydraulic diameter for the rectangular microchannels was 1.1mm and one rectangular tube contains 10 microchannels. Microchannel condenser was designed with the integrated receiver and drier. Evaporator temperature was controlled with the help of manually controlled expansion valve and setting the suction temperature of the refrigerant with the help of proportional, integral and derivative (PID) controller. Ethylene glycol (25% by mass) was mixed with the water in the evaporator in order to reduce the glycol temperature to achieve the desired evaporator temperature. Temperature and pressure indicators, rotameter, air heaters, expansion valve, drier, accumulator and the positions of the vapour compression refrigeration system components are as shown in the Fig. 1. Table 1 shows the components used in the setup.

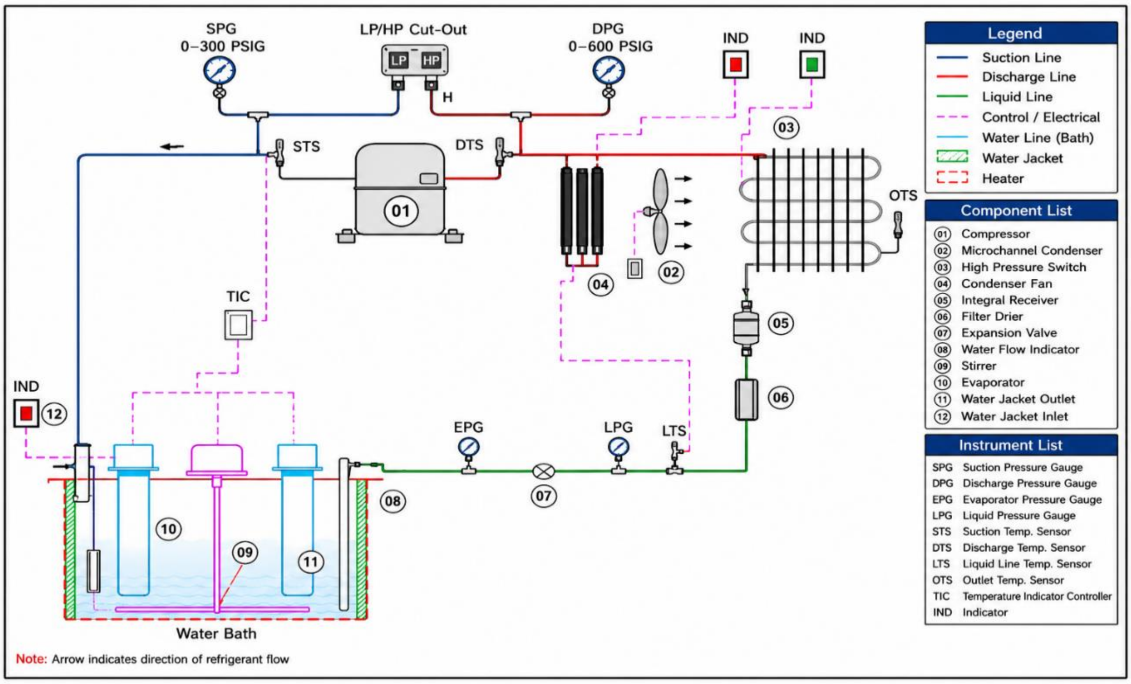
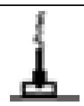
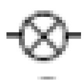
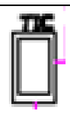


Fig. 1- Schematic diagram of experimental setup

Table 1 Experimental setup components

SN	Description	Quantity
1	Compressor	1
2	Air Cooled Condenser	1
3	Condenser Fan	1
4	Air Heater	3
5	Drier	1
6	Rotameter	1
7	Expansion Valve	1
8	Evaporator	1
9	Stirrer	1
10	Water Heater	2
11	S.S. Tank	1

Description	Symbol	Quantity
Gauges		4
Temperature Sensors		5
Expansion valve		1
PID		1
TIC		5

12	TIC	1		EPG	Evaporator Pressure Gauge	STS
13	PID	1		LPG	Liquid Pressure Gauge	DTS
14	Pressure Gauge	4		SPG	Suction Pressure Gauge	LTS
15	Temperature Sensor	5		DPG	Discharge Pressure Gauge	ETS
16	HP/LP Cut-Out	1		IND	Indicator	OTS

3. Experimental procedures

The objective of the present study was to experimentally evaluate and compare the performance of a vapour compression refrigeration (VCR) system using R134a and a hydrocarbon refrigerant mixture of R290/R600a (50/50% by mass) with a conventional round tube plate-fin condenser and an aluminium microchannel condenser. The performance of both systems was assessed in terms of refrigeration capacity (RC), compressor power consumption, coefficient of performance (COP), condenser heat rejection capacity, and the overall heat transfer conductance (UA). Experiments were conducted at condenser saturation temperatures of 44°C, 48°C, 55°C, and 58°C. For each condenser temperature, the evaporator temperature was varied from −10°C to 15°C at intervals of 5°C (i.e., −10°C, −5°C, 0°C, 5°C, 10°C, and 15°C). Pressure measurements were carried out using Bourdon-type pressure gauges installed at the inlet and outlet of the evaporator and condenser to determine the pressure drop across the heat exchangers. The pressure gauges had a measurement accuracy of ±1 psi. The refrigerant mass flow rate was measured using a calibrated rotameter with an accuracy of ±0.5 kg h^{-1}. Refrigerant temperatures at the compressor suction and discharge, condenser outlet (subcooled liquid), evaporator glycol bath, ambient air, and condenser air outlet were measured using PT-100 resistance temperature detectors (RTDs) with an accuracy of ±0.1°C. The surface temperatures of the condenser tubes and fins were measured using a calibrated infrared thermometer with an accuracy of ±0.5°C. The air velocity across the condenser was measured using a digital anemometer with an accuracy of ±0.1 m s^{-1}. The electrical power consumed by the compressor, condenser fan, air heater, water heater, and stirrer was measured independently using digital energy meters with an accuracy of ±0.01 W. All temperature sensors were installed in thermowells to ensure good thermal contact and minimize measurement errors. Prior to experimentation, all measuring instruments were calibrated to reduce experimental uncertainty and improve the reliability of the measured data. The specifications, operating ranges, and measurement accuracies of all instruments used in the experimental setup are summarized in Table 2 while Table 3 shows the comparison of Conventional and microchannel condenser.

Table 2 Range and accuracy of the equipment

Items	Range	Accuracy
Temperature sensor	-100 to 100 ^{0}C	±0.1 ^{0}C
Pressure gauge 1	0-300 psi	± 1 psi
Pressure gauge 2	0-600 psi	± 1 psi
Refrigerant rotamter	0-138 kg/hr	± 2 kg/hr
Power meter	8-digit display	±0.1 W
Anemometer	0.4-25 m/s	±0.1 m/s
Infra-red thermometer	-20-500 ^{0}C	±0.5 ^{0}C
Electronic balance weight	100 kg	± 1 gm
Voltage	0-500 V	± 1 V
Current	0-200 A	± 1 A

Table 3 conventional and microchannel condenser

Geometrical Parameters	Conventional Condenser	Microchannel Condenser
Face area (m^2)	470x360mm²=0.16	470x310mm²=0.146
Tube material	Copper	Aluminum
Fin material	Aluminum	Aluminum
Overall dimensions	470x360x90mm	470x310x15mm
Internal volume	3334	4939
Weight	5.610 Kg	1.6 Kg
Fin density	5 fins/10 mm	10 fins/10 mm
Fin height	25 mm	8 mm
Fin width	90 mm	15mm
Fin thickness	0.5 mm	0.1 mm
Hydraulic diameter	9.525 mm	1.1 mm
Total no. of rows	(7x4) x2=56	31
Air side area	11.741 m²	4.066 m²
Refrigerant side area	1.0495 m²	0.457 m²
No of passes	2 Rows, 28 turns	4 (14,10,4,3)
R134a and	1000 gm	900 gm
R290 & R600a (50:50)	423 gm	378 gm

The experimental test facility was initially charged with R134a, and performance tests were conducted using both the conventional round tube plate-fin condenser and the aluminium microchannel condenser. After completing the experiments with R134a, the system was evacuated, leak-tested, and subsequently charged with the hydrocarbon refrigerant mixture R290/R600a (50/50% by mass) to perform the drop-in replacement experiments under identical operating conditions.

The experiments were carried out sequentially in the following order:

1. R134a with the conventional condenser,
2. R134a with the aluminium microchannel condenser,
3. R290/R600a (50/50% by mass) with the conventional condenser, and
4. R290/R600a (50/50% by mass) with the aluminium microchannel condenser.

For each experimental condition, the system was allowed to attain steady-state operation before recording the measurements. The condenser saturation temperature was maintained at the desired value (44°C, 48°C, 55°C, and 58°C) by regulating the inlet air temperature using electrical air heaters and by controlling the condenser fan speed. The evaporator temperature was adjusted by controlling the refrigerant flow through the manually operated expansion valve and by regulating the compressor suction conditions. Throughout the experimentation, the degree of liquid subcooling at the condenser outlet was maintained at 7°C, while the compressor suction superheat was maintained at 10°C for both refrigerants to ensure identical operating conditions and facilitate a fair performance comparison. After the desired operating conditions were achieved, all temperatures, pressures, refrigerant mass flow rate, electrical power consumption, and air velocity were recorded. Each experiment was repeated to ensure repeatability, and the average values of the measured parameters were used for subsequent performance analysis. The thermophysical properties of R134a and the R290/R600a (50/50% by mass) refrigerant mixture required for the performance calculations were obtained using the NIST REFPROP thermodynamic property database. Figure 2 shows conventional condenser and Figure 3 shows microchannel condenser.

Fig. 2 Conventional condenser

Fig. 3 Aluminium microchannel condenser

4. Results and discussion

The experimental results obtained from the performance analysis of R134a and hydro carbon mixture of R290/R600a (50/50 % by mass) by using both the conventional and microchannel condensers are discussed with respect to the parameters such as compressor work consumption, coefficient of performance (COP), refrigeration capacity, condenser capacity and the product of overall heat transfer coefficient and area of condenser (UA).

4.1. Compressor energy consumption

Fig. 4 shows the consumption of compressor power for conventional and microchannel condenser with R134a and hydro carbon mixture of R290/R600a (50/50 % by mass) at 44^0C and 48^0C, 55^0C, 58^0C respectively. It was seen from the Fig. 4 that the work of compression increases as the condensation temperature increases. Work of compression is reduced from 4.61% to 7.14% for microchannel condenser with R134a than the conventional condenser at 44^0C for evaporator temperature ranging from -10^0C to 15^0C. It is seen that the work of compression increases by 18.46 % at -10 ^{0}C and then reduces by 13.26 % at 15^{0}C evaporation temperature with hydro carbon mixture of propane and isobutane (50/50 % by mass). It is also seen that the compressor power consumption increases from -10^{0}C to 15^{0}C for a condensation temperature of 44^{0}C, 48^{0}C, 55^{0}C & 58^{0}C. Work of compression with microchannel condenser with hydro carbon mixture of propane and isobutane (50/50 % by mass) is more than both the microchannel and conventional condenser with R134a as refrigerants.

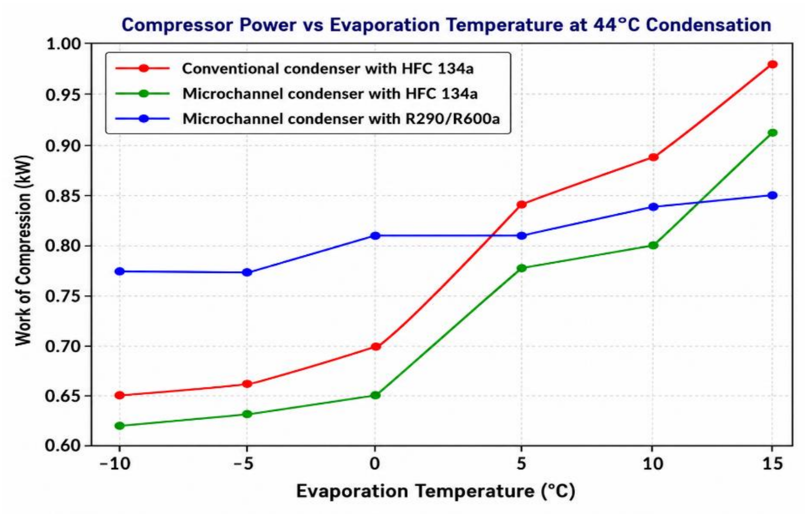


Fig. 4 Compressor power vs evaporation temp. at Tc=44^{0}C

4.2. Coefficient of Performance (COP)

Fig. 5 shows the coefficient of performance for evaporation temperature from -10^{0}C to 15^{0}C for condenser temperature of 44^{0}C and 48^{0}C, 55^{0}C & 58^{0}C respectively for conventional as well as microchannel condenser with the R134a and hydro carbon mixture of R290/R600a (50/50 % by mass). COP increases from -10^{0}C to 15 ^{0}C for all the condensation temperatures considered. It is seen that as the condensation temperature increases, COP decreases. COP is more for all the condensation temperature for hydro carbon mixture of R290/R600a (50/50 % by mass) than R134a with conventional as well as for microchannel condenser. It is seen from the Fig. 7 that for evaporation temperature from -10^{0}C to 15^{0}C, COP for the microchannel condenser was from 5.89 % to 15.78 % more than the conventional condenser with R134a and 8.73% to 11.6% more with the microchannel condenser using hydro carbon mixture of R290/R600a (50/50 % by mass) for condensation temperature of 44^{0}C.

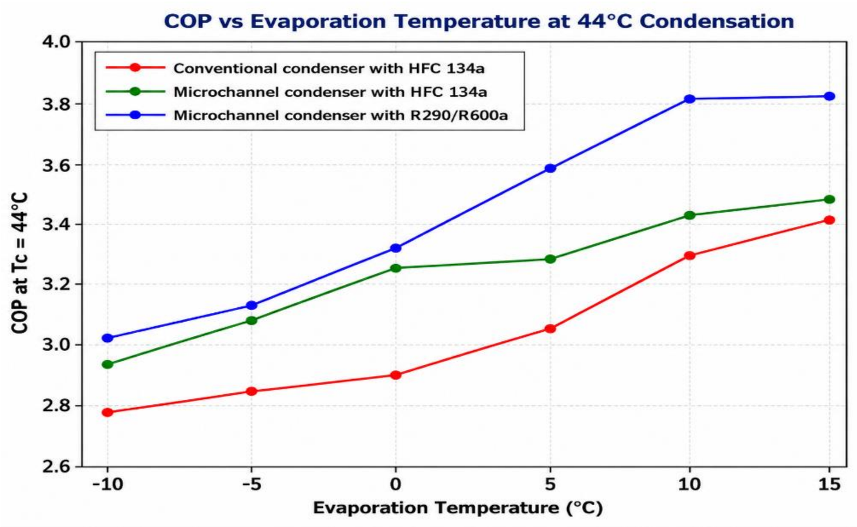


Fig. 5 COP vs evaporation temp. at Tc=44^{0}C

4.3. Condenser capacity

Fig. 6 shows the condenser capacity for conventional as well as microchannel condenser with the R134a and hydrocarbon mixture of R290/R600a (50/50 % by mass)) at 44^{0}C and 48^{0}C, 55^{0}C, 58^{0}C for evaporation temperature from -10^{0}C to 15^{0}C respectively. As the latent heat of hydrocarbon is nearly twice than that of the R134a, the condenser capacity is more with hydrocarbons for both the condensers at all condensation temperatures. Hence size of the condenser for the same tonnage capacity of the refrigeration system can be reduced. This is beneficial for the automobile air conditioning as the size as well as weight will be reduced.

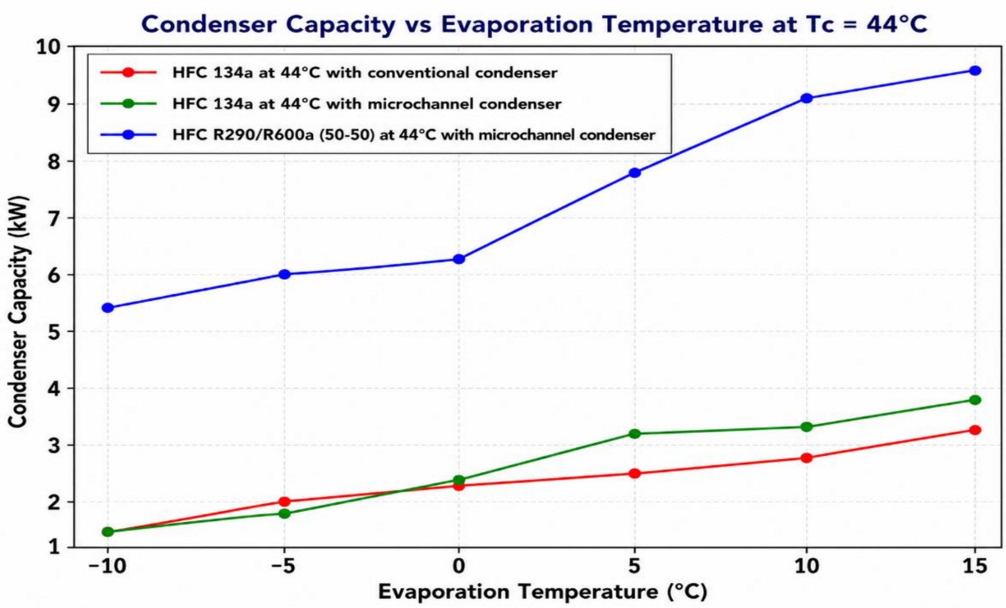


Fig.6 Condenser capacity vs evaporation temp at Tc=44 ^{0}C

4.4. Refrigerant capacity

Fig. 7 shows the variation of the refrigeration capacity against the evaporator temperature ranging from -10^{0}C to 15^{0}C for condensing temperature of 44^{0}C & 48^{0}C, 55^{0}C, 58^{0}C respectively. It is observed that the refrigerating capacity of hydro carbon mixture of R290/R600a (50/50 % by mass) is higher than R134a for both the conventional and microchannel condenser. Refrigeration capacity is more for microchannel condenser by 1.22% to 17% from -10^{0}C to 15^{0}C with the R134a and 3.07% to 20.65% with hydro carbon mixture of R290/R600a (50/50 % by mass) more than the conventional condenser. It was observed that the refrigerating capacity increases from condensation temperature of 44 ^{0}C to 58^{0}C for evaporation temperature ranging from -10^{0}C to 15^{0}C. It is also observed that the cooling is fast for the hydro carbon mixture of R290/R600a (50/50 % by mass) than the R134a for both the condensers.

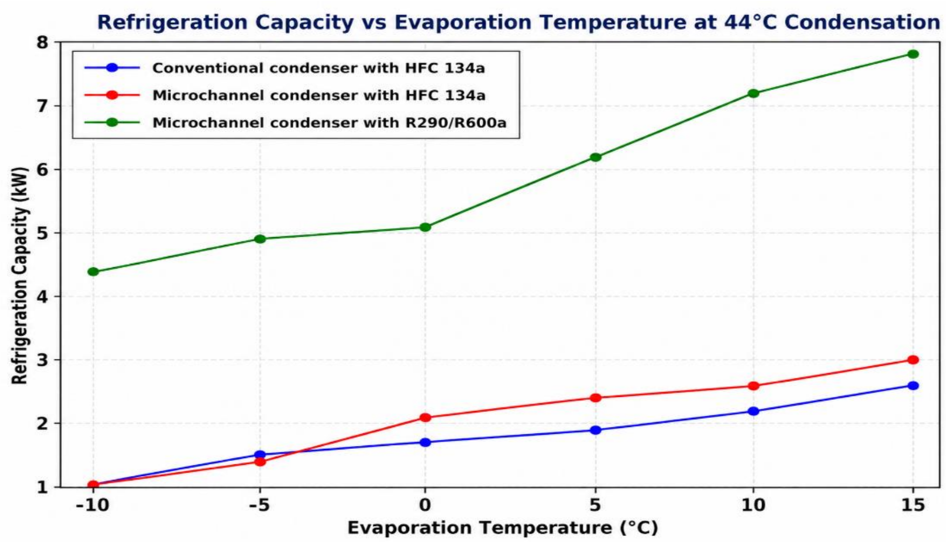


Fig.7 Refrigerating Capacity vs evaporation temp at Tc=44 ^{0}C

4.5. Product of overall heat transfer coefficient and area of condenser (UA)

Fig. 8 shows the product of overall heat transfer coefficient (U) and the area of the condenser (A) for conventional and microchannel condenser with the R134a and the hydro carbon mixture of R290/R600a (50/50 % by mass) at 44^{0}C for evaporation temperature range from -10^{0}C to 15^{0}C. It is observed that the product UA increases by13.68% to 17.42% for microchannel condenser with 134a, over the conventional condenser and 4.34 % to 200% for hydrocarbon mixture of R290/R600a (50/50 % by mass) using microchannel condenser over the conventional condenser.

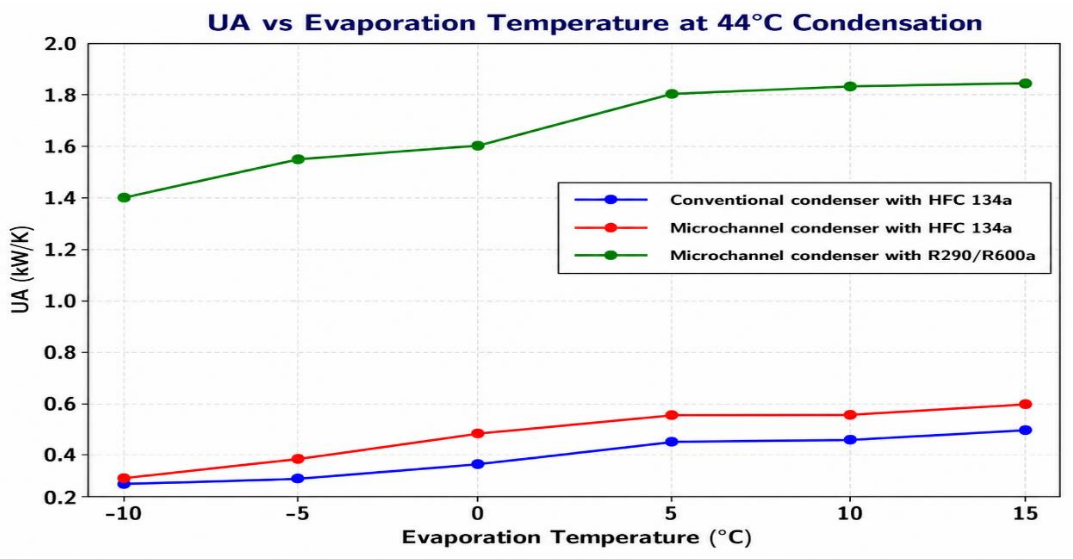


Fig. 8 UA vs evaporation temp at Tc=44 ^{0}C

5. Conclusions

A performance analysis on vapour compression refrigeration system was made using conventional condenser, copper round tube with plate fins and aluminimum microchannel condenser with aluminium fins using R134a and hydrocarbon mixture of R290/R600a (50/50 % by mass) for condensation temperature of 44^{0}C, 48^{0}C, 55^{0}C, 58^{0}C and for each condensation temperature, evaporator temperature was -10^{0}C, -5^{0}C, 0^{0}C, 5^{0}C, 10^{0}C, 15^{0}C. Following conclusions were drawn-

- Compressor power consumption reduces from 4.61% to 7.14% for microchannel condenser with R134a than the conventional condenser. It is also seen that the work of compression increases by 18.46 % at -10 ^{0}C and then reduces by 13.26 % at 15^{0}C evaporation temperature.
- COP increases by 5.89 % to 15.78 % for evaporation temperature of -10^{0}C to 15^{0}C by using microchannel condenser over the conventional condenser with R134a and COP increases by 8.73% to 11.6% by using the microchannel condenser with hydro carbon mixture of R290/R600a (50/50 % by mass) for condensation temperature of 44^{0}C. It is seen that as the condensation temperatures increases from 44^{0}C to 58^{0}C the COP increases.
- Refrigeration capacity was more for microchannel condenser by 1.22% to 17% from -10^{0}C to 15 ^{0}C with the R134a and 3.07% to 20.65% with hydro carbon mixture of R290/R600a (50/50 % by mass) than the conventional condenser.
- Condenser capacity of microchannel condenser increases by 6.74% to 18.33% over the conventional condenser. For R290/R600a the same condenser becomes oversize as the latent heat of hydrocarbon is nearly twice that of the R134a.
- The product of overall heat transfer coefficient (U) and the condenser area (A) increase by 13.68% to 17.42% for microchannel condenser with the R134a and 4.34% to 265% for

hydrocarbon mixtit with the microchannel condenser over the conventional condenser for condensation temperature of 44 ^{0}C and evaporation temperature of -10 ^{0}C to 15 ^{0}C.

Keeping in view the superior thermodynamic properties of the hydro carbon mixture, energy efficiency and two major environmental impact like ozone depletion potential and global warming potential, hydrocarbon mixture of R290/R600a (50/50 % by mass) can be used with slight modification as a replacement for R134a and microchannel condenser. Further the refrigerant charge, weight and size can be reduced with the aluminium microchannel condenser and material cost can be reduced.

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