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Comparative Analysis of Various Condenser in Vapour Compression Refrigeration System

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Abstract

The present work is to analyze performance of refrigeration system on three condensers viz. micro-channel, round tube and coil tube using R134a and R290 refrigerants. These three condensers are kept in parallel with other components of refrigerating unit while construction.The performance of refrigeration system is checked for each condenser at various cooling loads in the range from 175 W to 288 W.The performance of the condenser is measured for whole refrigeration unit in terms of coefficient of performance, efficiency of the system, heat rejection ratio, heat rejected from condenserand heat transfer coefficient.

The experimental data of heat transfer coefficient is validated with existing correlation.The result shows that for both refrigerants R134a and R290, coefficient of performance increases with increase in heating load. From the analysis of three condensers, coefficient of performance of refrigeration system using microchannel condenser is more compared to round tube and coil tube condenser. The coefficient of performance of the system with the microchannel condenser is found 15.3% higher than that with the round tube condenser and 8% higher than that with the coil tube condenser. AlsoR134a gives better cooling effect than the R290 for all operating condition.

Keywords: Microchannel, refrigerant, C.O.P., cooling load.

Introduction

Heat exchangers with multi-ported microchannel tubes are already used in mobile air-conditioning systems due to their compactness and high performance. For better understanding of the physical phenomena in microchannel tubes, the characteristics of heat transfer, pressure drop, and flow patterns have been studied by many researchers. The ability of micro channels to provide high surface area-to volume ratios, high heat transfer coefficients, high efficiencies and system compactness are among the major advantages of microchannel for use in a diverse range of industries. Condensation heat transfer in micro-channels and mini-channels is naturally of great practical importance in development of next generation ultra-compact and high performance two-phase flow thermal systems. But compared to the evaporation phenomenon, condensation in microchannel has been the subject of fewer studies. It can be argued that these two phenomena are essentially similar, and this may be true to some extent. The main aim of the current study was to characterize the condensation heat transfer performance of two selected refrigerants R134a (Tetrafluroethane) and R290 (Propane) in a single square micro-channel condenser, round tube condenser and coil tube condenser. The condenser heat exchanger plays a significant role in the structure and operation of the heat pump as it affects the system's coefficient of performance (COP). Two heat exchangers were used as condensers in the same air-conditioning system, one with round tubes and the other with flat microchannel tubes in a parallel-flow arrangement. The differences were recorded and are explained herein. This paper presents the difference measured in the performance for three condensers only as well as the effects on the system. The microchannel heat exchanger was made to have nearly an identical face area, depth and consequently volume, plus the same fin density as the baseline, round-tube heat exchanger with plate fins. The baseline condenser along with all other elements of the system was part of the very generously sized, off-the-shelf, air-conditioning system manufactured by one of the market and technology leaders.

The literature review is carried out in order to see the present research in this area which is elaborated under the present status. Many researchers have attempted experimental and theoretical work on micro channel condenser some of this work is focused on the use of micro channel condensers in refrigeration System exchangers.

D. A. Luhrs and W. E. Dunn (1994) [1], Presented Design and Construction of a Microchannel Condenser Tube Experimental Facility. A test facility was built for the purpose of performing heat transfer studies on microchannel heat exchangers. The studies will involve condensation of refrigerant 134a inside the enhanced tubes,' although no condensation results are presented in this document. The design and construction of the experimental facility is detailed with a description of each component and its function in the stand. The operation of the facility was verified using an energy balance analysis and the results are presented. The refrigerant and air side heat transfers agree within ±3% at high air flow rates but fall out of this error bound at lower flow rates. Also, a discussion of the method for determining the refrigerant and air side resistances for the tube is given along with the theory for future correlation development. Finally, future modifications to the stand are suggested in order to correct any problems with it, improving the ability of the stand to produce accurate, reliable heat transfer performance data.

Riehl et al. (1998) [2], reviewed single-phase and two-phase flow heat transfer coefficients of experimental data obtained for micro channels and compared them to the available analytical models. The comparisons showed large discrepancies. The models they examined were not able to predict the experimental data accurately. Furthermore, correlations of micro-channel convective flow also showed wide discrepancies. Later Riehl and Ochterbeck presented experimental results of condensation using methanol as the working fluid. The experiments were conducted for two different saturation temperatures, range of heat dissipation rate from 20 to 350W and four microchannel condensers with channel diameters between 0.5 and 1.5 mm. All the channels had aspect ratios of 1. Their results showed high heat transfer coefficients with Nusselt numbers ranging from 15 to 600. They also obtained a Nusselt number correlation which was able to predict 95% of the data within 25% error band.

Yin et al. (2001) [3], developed a CO2 microchannel gas cooler model. In their model, each pass was separated into 10 equal-length element. The model predicted the gas cooler capacity with good accuracy. A serpentine microchannel gas cooler model based on the microchannel gas cooler model presented by Yin et al. (2001) was used to simulate the serpentine gas cooler in their investigation. Tubes in each slab were divided into 10 elements. Thermal conductivity of the serpentine microchannel gas cooler was not considered in their model. The uniform air flow assumption was used in the serpentine gas cooler model.

Cavallini et al. (2002) investigated [4] ,condensation of R123a, R125, R410a, R32, R236ea and R22 inside a round tube with 8 mm inner diameter while varying the mass flux from 100 to750 kg m^{-2} s^{-1}. The study intended to improve Friedel's correlation (1979) in the annular regime. They also used the dimensionless vapor velocity to distinguish between the different flow regimes that exist in condensation. Then new constants were fitted to the Friedel's correlation from the study of the annular regime, and Due to the insignificant effect of gravitational forces in the annular flow regime, the Froude number was not accounted for in the two-phase multiplier correlation. However, these predictions cannot be applied to flow transitions.

Kim et al. (2003) [5], studied condensation in flat aluminum multi-channel tubes using R410A and R22. The tubes had two internal geometries: one with a smooth inner surface (D_h = 1.41 mm), the other with a micro-finned inner surface (D_h = 1.56 mm). Their results showed that for the smooth tube, the heat transfer coefficient of R410A was slightly larger than that of R22. For the micro-finned tube, however, the trend was reversed. They also compared their data with Webb's (1999), Koyama et al.'s (2003a, b), Akers et al.'s (1959) and Shah's (1979) correlations and concluded that for the smooth tube, Webb's correlation predicted the data reasonably well. For the micro-finned tube, they modified Yang and Webb's (1997) model to correlate with their data. The modified model predicted the data within 30%.

El Hajal et al. and Thome et al. (2003) [6], studied condensation of 15 different fluids amongst which were pure refrigerants and refrigerant blends. They used the studies of Kattan et al. (1998a, 1998b, 1998c) of evaporating refrigerants to develop a flow regime map and a heat transfer model. In this study they observed the following regimes: bubbly flow, intermittent, annular, stratified wavy, fully stratified and mist. However, the model did not include the bubbly flow regime. They suggested that heat transfer occurred due to two types of mechanisms:

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film and convective condensation. The regimes that contributed to convective condensation were annular, mist and intermittent flows, whereas stratified-wavy and stratified flows were governed by both mechanisms. The developed correlation of heat transfer coefficient was governed by the interfacial friction factor, Prandtl number and Reynolds number.

Baird et al. (2003) [7], experimentally investigated local heat transfer coefficient of condensation for R123 and of R11 inside 0.92 mm and 1.95 tube diameters, a range of mass fluxes 70-600 kgm^{-2} s^{-1}, heat fluxes 15-110 kWm^{-2}, and pressures 120-410 KPa. Their data showed a strong influence of mass flux and local quality on the heat transfer coefficient, with a weaker influence of system pressure. Then they developed a model that agreed with their experimental data more than other models by using a simple shear driven annular flow model to predict the condensation heat transfer coefficient.

Bandhauer et al. (2006) [8], implemented a thermal amplification technique for the accurate measurement of small heat duties over small refrigerant quality increments. They reported local heat transfer rates within 10% for 0.506, 0.761 and 1.520 mm circular multichannel tubes with R134a as the working fluid. Measurements were conducted over mass flux range 150-750 kg m2 s1 and refrigerant quality range about 0.15-0.85. In general, the data indicated an approximately linear trend between heat transfer coefficient and local quality over the range of qualities and mass fluxes measured. However, the proper distinction between heat transfer coefficients at different mass fluxes was difficult since the differences fell within the measurement uncertainty. Also there was no information about the possible effect of flow mal distribution among parallel tubes on measured parameters. The authors also developed a model for calculation of heat transfer coefficient that used their pressure drop model to compute the interfacial shear stress and the friction velocity. The resulting model predicts 86% of the data within 20% and also captured correctly the trends exhibited by the data.

Pega Hrnjak, 1, Andy D. Litch [9], Reported Microchannel heat exchangers for charge minimization in air-cooled ammonia condensers and chillers. They presented experimental results from a prototype ammonia chiller with an air-cooled condenser and a plate evaporator. The main objectives were charge reduction and compactness of the system. The charge is reduced to 20 g/kW (2.5 oz/Ton). This is lower than any currently available air-cooled ammonia chiller on the market. The major contribution comes from use of microchannel aluminum tubes. Two aluminum condensers were evaluated in the chiller: one with a parallel tube arrangement between headers and ‘‘microchannel’’ tubes (hydraulic diameter D_h = 0.7 mm), and the other with a single serpentine ‘‘macrochannel’’ tube (D_h = 4.06 mm). The performances of the chiller and condensers are compared based on various criteria to other available ammonia chillers. This prototype was made and examined in the Air Conditioning and Refrigeration Center in 1998, at the University of Illinois at Urbana-Champaign.

Qian Sub, Guang Xu Yua, Hua Sheng Wanga [2009] [10], reported short communication on Microchannel condensation: Correlations and theory Attention is drawn, to the fact that, while four different correlations for condensation in Micro channels are in fair agreement for the case of R134a (on which the empirical constants in the correlations are predominately based) they differ markedly when applied to other fluids such as ammonia. A wholly theoretical model is compared with the correlations for both R134a and ammonia.

Son and Lee (2009) [11], carried out experiments on condensation heat transfer of R22, R134a and R410A in single-channels with 1.77, 3.36 and 5.35 mm diameters, mass flux of 200-400 kgm^{-2} s^{-1} and saturation temperature of 40 °C. They observed that annular flow is almost the dominant flow regime for condensation in small diameter tubes and reported an earlier transition into the annular flow in their microchannel tubes. Also they concluded that the majority of the existing correlations failed to predict their condensation data accurately, and they proposed their own correlation. Some researchers (Kim et al. (2003a, b) and Wang et al. (2002)) suggested that the condensation phenomena in minichannels may be different from those in macro-channels.

Liang-Liang Shaoa, Liang Yanga,b, Chun-Lu Zhangb,, Bo Gua (2009)[12] , presented Numerical modeling of serpentine microchannel condensers Microchannel (or minichannel) heat exchangers are drawing more attention because of the potential cost reduction and the lower refrigerant charge. Serpentine microchannel heat exchangers are even more compact because of the minimized headers. Using the serpentine microchannel condenser, some thermodynamically good but flammable refrigerants like R-290 (Propane) can be extended to more applications. To well size the serpentine microchannel condensers, a distributed-parameter

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model has been developed in this paper. Model validation shows good agreement with the experimental data. The predictions on the heating capacity and the pressure drop fall into 10% error band. Further analysis shows the impact of the pass number and the airside maldistribution on the condenser performance.

Akhil Agarwal , Todd M. Bandhauer , Srinivas Garimella (2010)[13] , Reported measurement and modeling of condensation heat transfer in non-circular microchannel Heat transfer coefficients in six non-circular horizontal microchannel (0.424 < Dh < 0.839 mm) of different shapes during condensation of refrigerant R134a over the mass flux range 150 < G < 750 kg m2 s1 were measured in this study. The channels included barrel-shaped, N-shaped, rectangular, square, and triangular extruded tubes, and a channel with a W-shaped corrugated insert that yielded triangular microchannel. The thermal amplification technique developed and reported in earlier work by the authors is used to measure the heat transfer coefficients across the vapor-liquid dome in small increments of vapor quality. Results from previous work by the authors on condensation flow mechanisms in microchannel geometries were used to interpret the results based on the applicable flow regimes. The effect of tube shape was also considered in deciding the applicable flow regime. A modified version of the annular-flow-based heat transfer model proposed recently by the authors for circular microchannel, with the required shear stress being calculated from a non-circular microchannel pressure drop model also reported earlier was found to best correlate the present data for square, rectangular and barrel shaped microchannel. For the other microchannel shapes with sharp acute-angle corners, a mist-flow-based model from the literature on larger tubes was found to suffice for the prediction of the heat transfer data. These models predict the data significantly better than the other available correlations in the literature.

J.R. Garcı´a-Cascales, F. Vera-Garcı´a, J. Gonza´lvez-Macia (2010) [14], Presented Compact heat exchangers modeling: Condensation a model for the analysis of compact heat exchangers working as either evaporators or condensers is presented. This paper will focus exclusively on condensation modeling. The model is based on cell discretization of the heat exchanger in such a way that cells are analyzed following the path imposed by the refrigerant flowing through the tubes. It has been implemented in a robust code developed for assisting with the design of compact heat exchangers and refrigeration systems. These heat exchangers consist of serpentine fins that are brazed to multi-port tubes with internal microchannel. This paper also investigates a number of correlations used for the calculation of the refrigerant side heat transfer coefficient. They are evaluated comparing the predicted data with the experimental data. The working fluids used in the experiments are R134a and R410A, and the secondary fluid is air. The experimental facility is briefly described and some conclusions are finally drawn.

ZHANG Huiyong, LI Junming , LI Hongqi (2010) [15] , Presented Numerical Simulations of a Micro-Channel Wall-Tube Condenser for Domestic Refrigerators In recent years, microchannel heat exchangers have begun to be used in refrigeration and air conditioning systems. This paper introduces a microchannel condenser for domestic refrigerators with a theoretical model to evaluate its performance. The model was used to obtain the optimal design parameters for different numbers of tubes and tube lengths. The results show that the needed tube height of the downward section decreases with the number of tubes and the tube diameter. Compared with the original condenser, the present optimal design parameters can reduce the total metal mass by 48.6% for the two wall two side design and by 26% for the two wall one side design. Thus, the present condenser is much better than the condensers usually used in actual domestic refrigerators.

Gunda Mader, Georg P.F. Fosel, Lars F.S. Larsen(2013) [16] Presented Comparison of the transient behavior of microchannel and fin-and-tube evaporators The development of control algorithms for refrigeration systems requires models capable of simulating transient behavior with sensible computational time and effort. The most pronounced dynamics in these systems are found in the condenser and the evaporator, especially the transient behavior of the evaporator is of great importance when designing and tuning controllers for refrigeration systems. Various so called moving boundary models were developed for capturing these dynamics and showed to cover the important characteristics. A factor that has significant influence on the time constant and nonlinear behavior of a system is the amount of refrigerant charge in the evaporator which is considerably reduced when microchannel heat exchangers are utilized. Here a moving boundary model is used and adapted to simulate and compare the transient behavior of a microchannel evaporator with a fin-and-tube evaporator for a residential air-

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conditioning system. The results are validated experimentally at a test rig.

G.B. Ribeiro, J.R. Barbosa Jr. A.T. Prata(2013) [17] Presented Performance of microchannel condensers with metal foams on the air-side: Application in small-scale refrigeration systems the thermal-hydraulic performance of microchannel condensers with open-cell metal foams to enhance the air-side heat transfer is investigated in this paper. Three different copper metal foam structures with distinct pore densities (10 and 20 PPI) and porosities (0.893 and 0.947) were tested. A conventional condenser surface, with copper plain fins, was also tested for performance comparison purposes. The experimental apparatus consisted of a closed-loop wind tunnel calorimeter and a refrigerant loop, which allowed the specification of the mass flow rate and thermodynamic state of R-600a at the condenser inlet. The experiments were performed at a condensing temperature of 45 °C. The air-side flow rate ranged from 1.4 – 10.3 to 3.3 – 10.3 m3/s (giving face velocities in the range of 2.1e4.9 m/s). The heat transfer rate, the overall thermal conductance, the Colburn j-factor, the friction factor and the pumping power were calculated as part of the analysis.

Present Status of microchannel condenser

Presently microchannel condensers are used in electronics and automobile air conditioning. Extensive study of Measurement and modeling of condensation heat transfer in non circular microchannel has been carried out. Various correlations And Theories for condensation in microchannel are developed also experimental results for ammonia chillers with air cooled condensers and plate evaporators are presented which reduces charge and gives compactness to the system. The thermal hydraulic performance of microchannel condensers with open cell metal foam to enhance the air side heat transfer is investigated and heat transfer rate, the overall thermal conductance, pumping power is calculated for the same.

Objective of the study

Heat exchangers with multi-ported microchannel tubes are already used in mobile air-conditioning systems due to their compactness and high performance. To study the measurements and modeling of condensation heat transfer in microchannel. Also to apply various correlations and theories of condensations for the given setup.

Experimentation

A liquid boils and condenses – the change between the liquid and gaseous states at a temperature which depends on its pressure, within the limits of its freezing point and critical temperature. In boiling it must obtain the latent heat of evaporation and in condensing the latent heat must be given up again. The basic refrigeration cycle makes use of the boiling and condensing of a working fluid at different temperatures and, therefore, at different pressures. Heat is put into the fluid at the lower temperature and pressure and provides the latent heat to make it boil and change to a vapour. This vapour is then mechanically compressed to a higher pressure and a corresponding saturation temperature at which its latent heat can be rejected so that it changes back to a liquid.

Microchannel condenser:-Microchannel heat exchangers have begun to be used in refrigeration and air conditioning systems mainly consists of microchannel tubes, louvered fins, header tubes, baffles, receiver/dryer bottle, and inlet/outlet fittings. Parallel flow (PF) condenser, widely used in automotive A/C system, is a typical microchannel heat exchanger. In a PF condenser, refrigerant flows through microchannel tubes in parallel within the same pass while in series from pass to pass. In other words, the mass flow of refrigerant in any pass is a constant at a stable condition. The size and specification of microchannel condenser used in set up as given below.

Specification of miocrochannel condenser:-

Size of microchannel condenser:	325*325mm
Type:	Square port type
Number of channels:	32
Thickness of channel:	5mm
Diameter of refrigerant tube:	11mm
Fin material:	Aluminium
Refrigerant tube material:	Copper

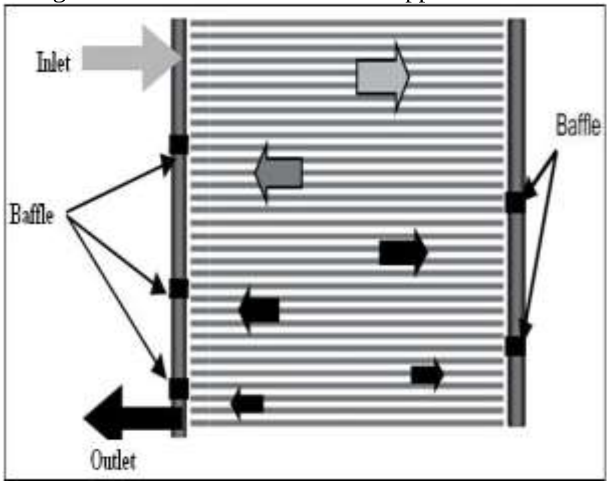

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Fig. 1 Profile of microchannel condenser

Round Tube condenser:- In Round type condensers, the circulation of air over the condenser surface is maintained by using a fan or a blower. These condensers normally use fins on air-side for good heat transfer. The fins can be either plate type or annular type. The red colour tubes indicate inlet and blue colour shows outlet of refrigerant from condenser. Actual view of round tube condenser shown in fig. 2.

Fig. 2 Profile of round tube condenser

The specification of Round tube condenser:
Diameter of refrigerant tube: 09 mm
Length of round tubes: 3600 mm
Number of round tubes: 36
Round tube: 12” *12” * 4 rows
Fins material: Aluminium
Refrigerant tube material: Copper

Shell and Coil tube condenser:- In these condensers the refrigerant flows through the shell while water flows through the tubes in single to four passes. The condensed refrigerant collects at the bottom of the shell. The coldest water contacts the liquid refrigerant so that some subcooling can also be obtained. The liquid refrigerant is drained from the bottom to the receiver. There might be a vent connecting the receiver to the condenser for smooth drainage of liquid refrigerant. The shell also acts as a receiver. Further the refrigerant also rejects heat to the surroundings from the shell. The most common type is horizontal shell type as shown in fig. 3

The specification of shell and coil tube condenser:
Diameter of shell: 100mm
Length of the shell: 400mm
Coil Diameter: 75mm
Refrigerant tube diameter: 6.25mm
Number of turns to refrigerant coil: 20

Fig. 3 Profile of shell and coil tube condenser

Evaporator:- The purpose of the evaporator is to receive low-pressure, low temperature fluid from the expansion valve and to bring it in close thermal contact with the load. The refrigerant takes up its latent heat from the load and leaves the evaporator as a dry gas. The charge from expansion device enters in evaporator bath and absorbs the heat from brine solution. Charge from evaporator again enters to compressor at a evaporator pressure (LP).

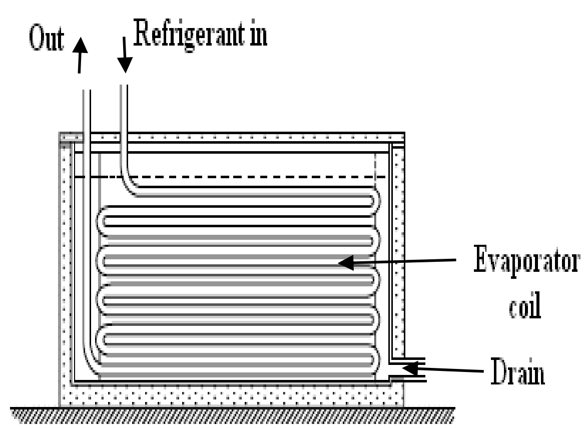

Fig.4 Line diagram of evaporator

Specification of evaporator:

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Refrigerant tube diameter: 9mm
Circular coil tube diameter: 200mm
Length of the tube: 600 mm
Volume capacity of brine solution: 10litre
Evaporator type: Wound coil
Evaporator coil material: Copper

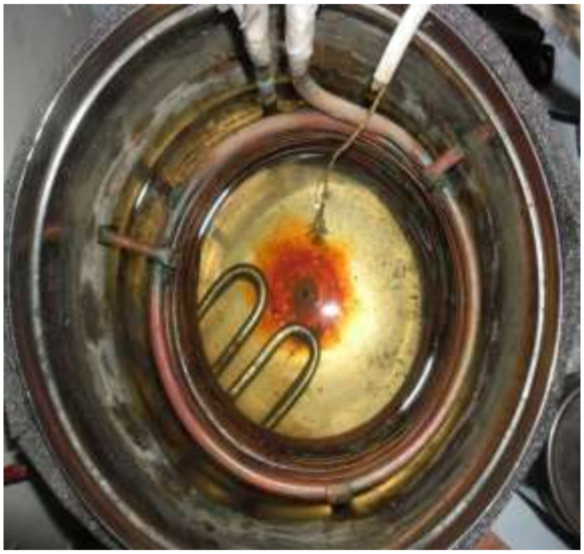

Fig. 5 Profile of evaporator

Experimentation

For the analysis of refrigeration system using microchannel condenser set up is built to find various parameters. The measurement parameters are actual coefficient of performance, theoretical coefficient of performance, mass flow rate of refrigerant, heat rejection ratio, heat rejected by condenser and heat transfer coefficient. From various operating conditions the data obtained from refrigeration system using microchannel condenser was compared with round tube and coil tube condenser. In experimental procedure, performance of microchannel condenser, round tube condenser and coil tube condenser was compared using two different refrigerants which are R134a (Tetrafluoroethane) and R290 ($CH_3CH_2CH_3$) propane. The three condensers are connected in series and operated by closing , opening of throttle valve shown in figure 6.

Operation procedure:-Connect the two plugs to main. Before ON the supply, conform that all the switches on panel are off position. See the dimmerstat is at zero position. Then put ON the heater switch & give power to heater. This will heat the water in evaporator & this can be seen to dial thermometer. Adjust the heater voltage such that the Temperature dial thermometer reading reaches 25 - 300 C. Now ON the D.P. switches. Put ON the condenser fan switch & wait for 2 - 3 minutes. Now switch ON the solenoid valve switch & the compressor switch. The refrigeration flow will start. This can be confirmed on the sight glass. Now the ammeter, voltmeter will show the current & voltage for compressor. Note down the time for 10 revolutions of energy for compression. After some time we will see that the Temperature of water in the evaporator slowly goes down & reaches steady state. (Adjust this temp. at 28 to 300 C). After the steady state note down the readings as follows:

1. HP Condenser pressure in Kg/cm^2. = Kg/cm^2
2. LP Evaporator Pressure in Kg/Cm^2 = Kg/cm^2
3. Rotameter in Reading LPH = LPH
4. Condenser Inlet Temperature in ^0C = Tci
5. Condenser Outlet Temperature ^0C = Tco
6. Evaporator Inlet Temperature in ^0C = Tci
7. Evaporator Outlet Temperature ^0C = Teo
8. Time for 10 Pulses of heater energy meter = in sec (EMC=3200imp. /KW-hr.)
9. Time for 10 Pulses of comp energy meter = in sec. (EMC=6400imp. /KW-hr.)
10. Ammeter reading = in Amp
11. Voltmeter reading = in V
12. Evaporator Bath Temp in ^0C = ^0C.

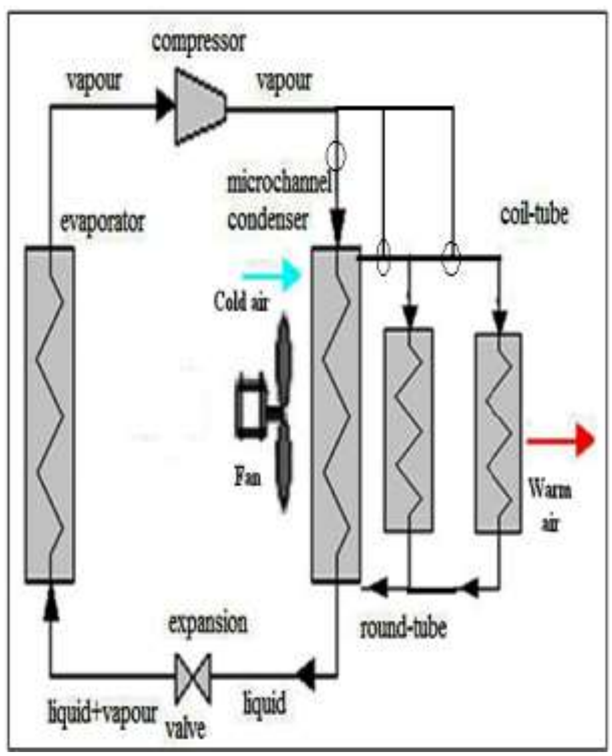

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Fig. 6 Profile of experimental set-up

Parameters calculated:-

1. Compressor Power (W_{act}) = $\frac{Nc*3600}{emc*tc}$
2. Heater power (N_{act}) = $\frac{Nh*3600}{emc*th}$
3. C.O.P $_{act}$ = $\frac{Wact}{Nact}$
4. C.O.P $_{theoretical}$ = $\frac{Heo-Hei}{Hci-Heo}$
5. HRR = $1 + \frac{1}{COP}$
6. Qc = $mC_p(\Delta T)$

Results and discussion

The experimental data obtained from three condensers and two refrigerants are presented in this chapter. To compare performance analysis of refrigeration system using microchannel condenser, round tube condenser and coil tube condenser with refrigerants R134a and R290 various graphs are plotted. The graphs are obtained from calculations shown in chapter 5 and results table as shown in this chapter.

1.Effect of cooling load on actual coefficient of performance using R134a:-

The coefficient of performance is an index of performance of a thermodynamic cycle or a thermal system. Because the COP can be greater than 1, COP is used instead of thermal efficiency.

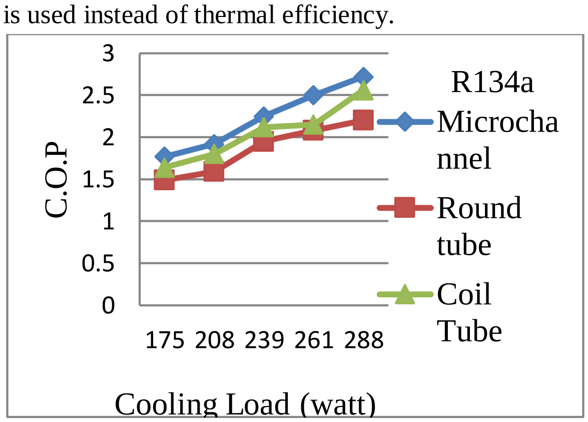

Fig.7 C.O.P Vs Cooling load for refrigerant R134a

The coefficient of performance can be used for the analysis of the following:

- A refrigerator that is used to produce a refrigeration effect only, that is, COP_{ref}
- A heat pump in which the heating effect is produced by rejected heat COP_{hp}
- A heat recovery system in which both the refrigeration effect and the heating effect are used at the same time, COP_{hr}

2. Effect of cooling load on efficiency using R134a

From figure 8 it can be seen that efficiency of microchannel condenser is more than round tube and coil tube condenser.

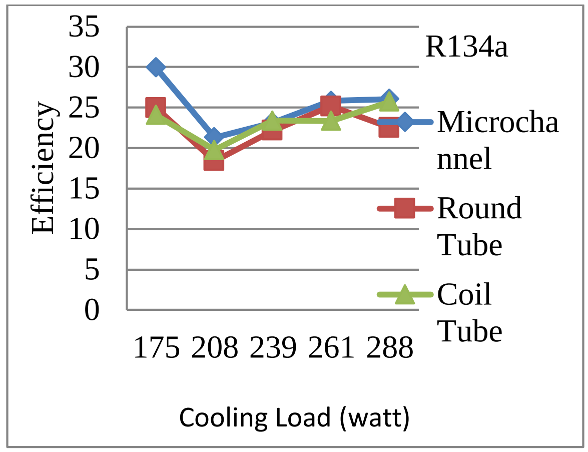

Fig. 8 Efficiency Vs cooling load for R134a

3. Effect of cooling load on theoretical coefficient of performance using R134a

Figure 9 shows Effect of cooling load on theoretical coefficient of performance using R134a . It can be seen from figure that as cooling load increases theorotical coefficient of performance increases for all condensers. The increase in coefficeint of performance is more for microchannel condenser.

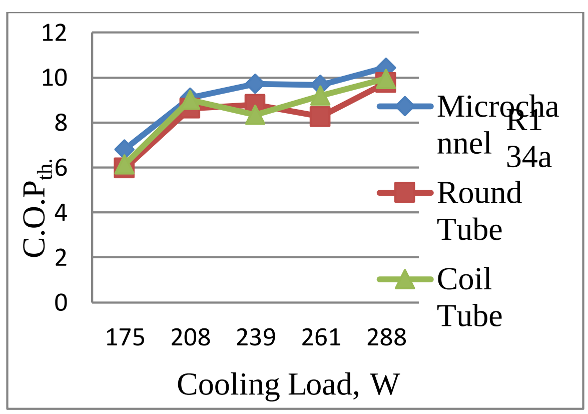

Fig. 9 C.O.Pth Vs cooling load for refrigerant R134a

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4. Effect of load on C.O.P using R290

Figure 10 shows effect of load on C.O.P and efficiency using R290. With increase in cooling load actual coefficient of performance increases. The increase in COP is more for microchannel condenser.

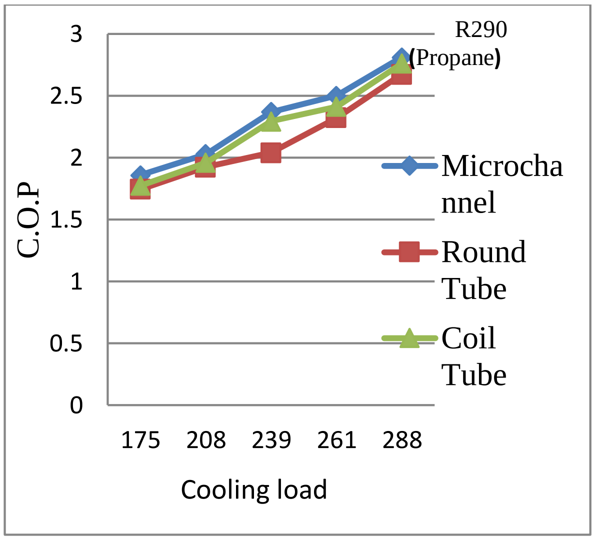

Fig. 10 C.O.P Vs cooling load for refrigerant R290

5 .Effect of cooling load on efficiency for R290

Figure 11 shows effect of cooling load on efficiency for R290 .It can be seen from figure that as cooling load increases efficiency of refrigeration system increase.

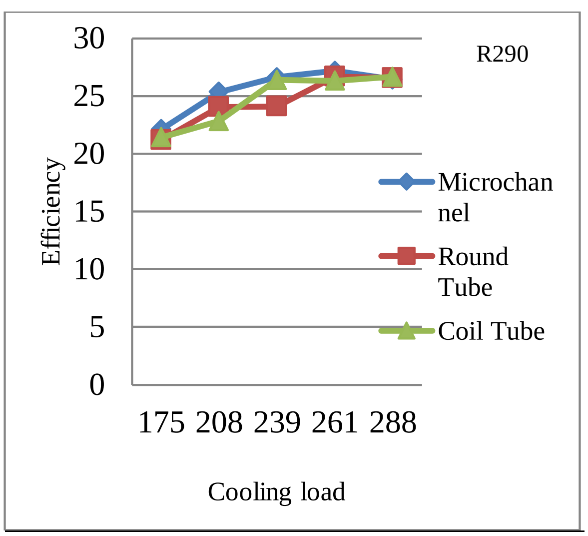

Fig. 11 Efficiency Vs Heating load for refrigerant R290

6. Effect of cooling load on theoretical coefficient of performance for R290

Figure 12 shows effect of cooling load on theoretical coefficient of performance for R290. With increase in colling load theoretical coefficient of performance increases and microchannel consenser gives higher theoretical coefficient of performance.

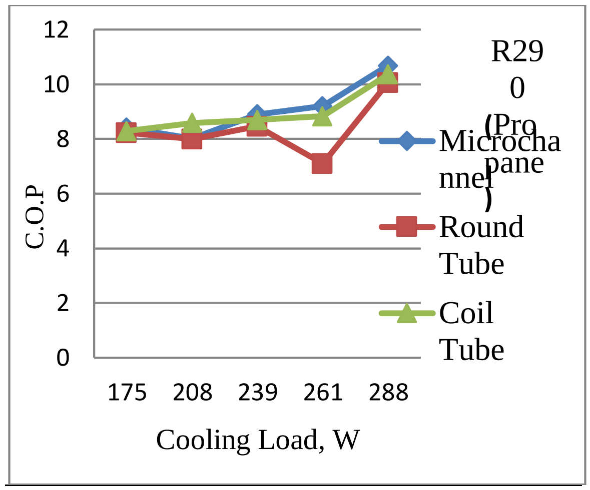

Fig.12 C.O.P theoretical Vs Heating load for refrigerant R290

7. Effect of cooling load on heat rejection ratio for R134a

Figure 13 shows effect of cooling load on heat rejection ratio for R134a. AS cooling load increases heat rejection ration decreases for all condensers. Heat rejection ratio is inversely proportional to coefficient of performance.

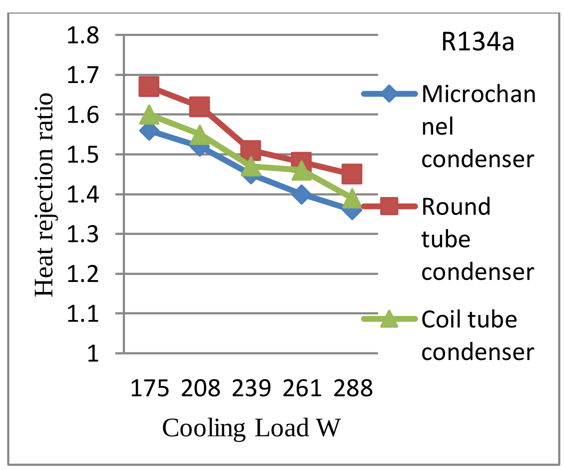

Fig 13 HRR Vs Heating load for refrigerant R134a

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8. Effect of cooling load on heat rejected from condenser for R134a

Figure 14 shows variation in heat rejected from condenser with cooling load for R134a. Heat rejected from condenser increases as cooling load increases but power consumption to drive the compressor to achieve this load increases.

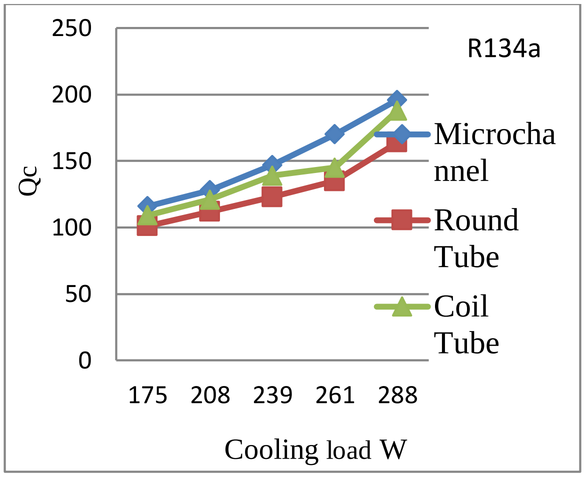

Fig.14Heat rejected from condenser Vs cooling load

9. Effect of cooling load on heat rejection ratio for R290

Figure 15 shows effect of cooling load on heat rejection ratio for R290. As cooling load increases heat rejection ration decreases for all condensers. Heat rejection ratio is inversely proportional to coefficient of performance.

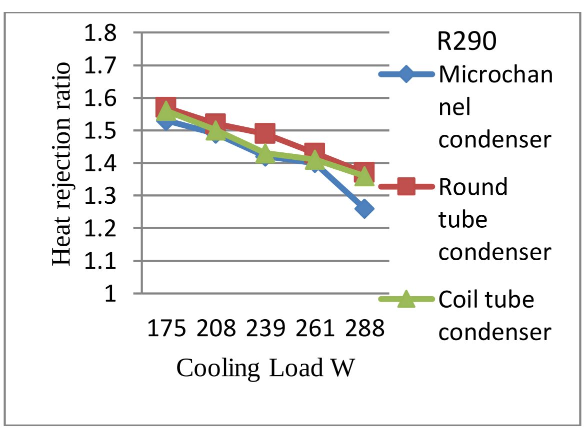

Fig.15 HRR Vs cooling load, watt

10. Effect of cooling load on heat rejected from condenser for R290

Figure 16 shows variation in heat rejected from condenser with cooling load for R290. Heat rejected from condenser increases as cooling load increases but power consumption to drive the compressor to achieve this load increases.

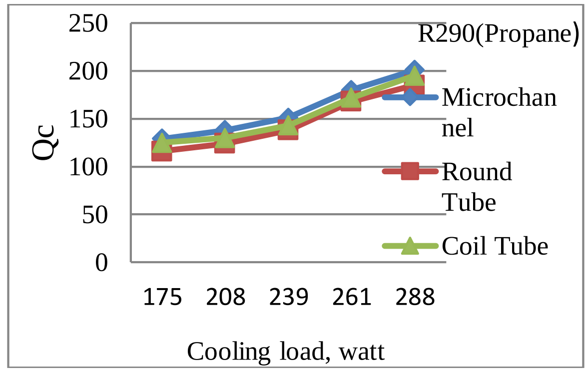

Fig.6.10 Heat rejected from condenser Vs Cooling load for refrigerant R290

11. Effect of cooling load on actual coefficient of performance for different combination of condenser and refrigerant

Figure shows effect of cooling load on actual coefficient of performance for different combination of condenser and refrigerant. For all condenser actual coefficient of performance increases with increase in cooling load. The microchannel condenser using R290 refrigerant gives highest actual coefficient of performance.

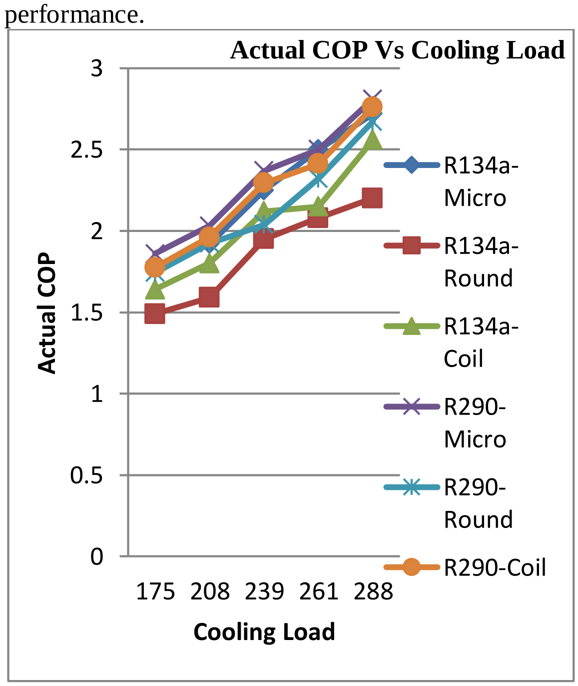

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Fig. 16 Heat rejected from condenser Vs Cooling load for R134A and R290

12. The presentation of pressure, temperature and enthalpy on P-h chart

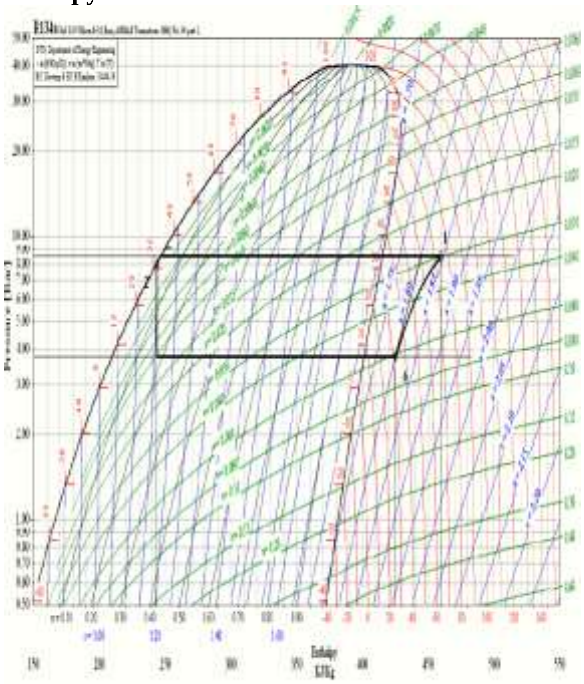

Fig. 17 P-h chart at load 175watt (Microchannel condenser, R134a)

Experimental Validation

1. Theoretical Aspects: In the condenser, . three zones, corresponding to refrigerant de-superheating, condensation and Sub-cooling are considered. In the superheating zone the surface temperature is above the saturation temperature so there is no condensation in this region. The real condensation of refrigerant occurs in the condensation zone , where two phase flow (a combination of liquid and vapor refrigerant) exists . A large number of techniques for predicting the heat-transfer coefficients during condensation inside pipes have been proposed .These range from very arbitrary correlations to highly sophisticated treatments of the mechanics of flow.

2. Shah's Correlation: The two-phase flow heat transfer model developed by Shah is a simple correlation that has been verified over a large range of experimental data. In fact, experimental data from over 20 different researchers has been used in its development. For this model, at any given quality, the two-phase heat transfer coefficient is defined as:

$Nu_l = 0.023\, Re_l^{0.8}\, Pr_l^{0.4}$

Where, $0 < x < 1$

$Re_l > 350$

$Pr_l > 0.5$

Nomenclature used in correlations

Re	Reynolds number	$\rho v D/\mu$	
Nu	Nusselt number	hD/k	
Pr	Prandtl number	$C p\mu/k$	
hr	refrigerant-side heat transfer coefficient		$W/m^2.K$
hi	enthalpy of refrigerant inter condenser		kJ/kg
Tsi	temperature of inner tube surface		ºC
Tao	temperature of air outlet		ºC
Q	rate of heat flow		Watt
Ao	outside area of tube		m2
Ai	inside area of tube		m2
ρl	saturated liquid density		kg/m3
α	heat transfer coefficient [W/m2K]		
λ	thermal conductivity [W/mK]		
D	inside diameter of tube		[m]
x	vapor quality		
Pr	Prandt number		

This correlation takes into account the pressure of the refrigerant also in addition to the quality of the mixture. This can also be used to find the local condensation heat transfer coefficient. The heat transfer coefficient is a product of heat transfer coefficient given by Dittus-Boelter equation and an additional term.

The two-phase heat transfer coefficient is defined as:

$h_{tp} = h_l * Y$

Where $Y = (1-x)^{0.8} + 3.8x^{0.76}(1-x)^{0.04}/p_r^{0.38}$

pr is the reduced pressure = condenser pressure / critical pressure

To check variation in heat transfer coefficient between experimental heat transfer coefficient and Shah's Correlation heat transfer coefficient various graphs at cooling loads are obtained.

3. Experimental aspects:

Volume flow rate of refrigerant ,inlet and outlet temperature of air across the condenser ,air velocity ,and the readings of current ,voltage ,power consumed are taken using refrigerant R134a at different volume flow rate at ambient temperature of (31ºC,28.6ºC,24.3 ºC) .The same measurements are taken for R290 at ambient temperature of (31ºC,28.6ºC,24.3 ºC) for comparison purposes. The condenser is supplied with glass tubes to show the phase of refrigerant along the condenser.

Once the temperature of refrigerant enters and leaves the condenser, condenser pressure and

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volume flow rate are measured then the heat rejected from condenser Q is calculated as,

Q= m (hi- ho) *1000

Under steady state conditions, the rate of heat transfer Q is the same from the outside surface to the inside surface of the tube and from the inside surface of the tube to the refrigerant. Although the difference between average outside tube surface temperature Tso and inner tube surface temperature Tsi is small, the inner tube surface temperature is calculated as:

Q= (kt/xt) Am (Tso – Tsi)

The average values of experimental heat transfer coefficient are calculated at average surface temperature of the condenser as:

Q= hr Ai (Tsat –Tsi)

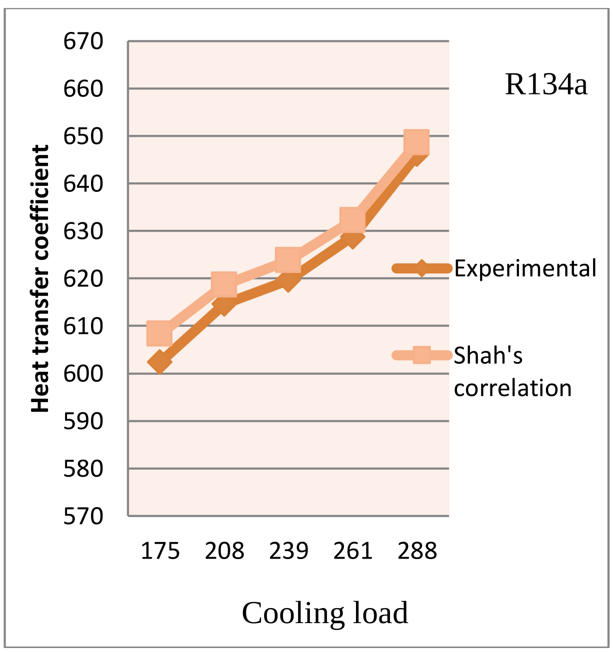

Fig. 18 Heat transfer coefficient (W/m2K) Vs cooling load (W)

Conclusion

The present work is to find performance analysis of refrigeration system using microchannel condenser, round tube condenser and coil tube condenser with two refrigerants R134a and R290. Experiments are performed to find the effects of mass flow rate, the saturation temperature, coefficient of performance, heat rejection ratio, and heat rejection rate from condenser with heat transfer coefficient.

For both refrigerants R134a and R290, coefficient of performance increases with increase in cooling load. From the three condensers, C.O.P of refrigeration system using microchannel condenser is more compared to round tube and coil tube condenser. The C.O.P of the system with the microchannel condenser is found 19.75 % higher than that with the round tube condenser and 8.65 % higher than that with the coil tube condenser using R134a. The C.O.P of the system with the microchannel condenser is found 8.21 % higher than that with the round tube condenser and 4.04 % higher than that with the coil tube condenser using R290.

For condenser parameters, heat rejection ratio with the microchannel condenser is 2.39% lower compared to coil tube condenser and 5.62% lower with round tube condenser. For a fixed condenser temperature, as the evaporator temperature decreases the COP decreases and heat rejection ratio increases. The heat rejected from microchannel condenser is 15.73% higher compared to round tube condenser and 7.136 % higher than the coil tube condenser. The overall temperature after condenser remains same for both refrigerants as well as for three condensers at various cooling load.

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