

Performance Investigation of Joule–Thomson Refrigeration System with Environment Friendly Refrigerant Mixtures

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Abstract

The importance on energy conservation has resulted in new efforts to develop more efficient refrigeration systems for achieving low temperature. The Joule-Thomson (J-T) concept has been widely used for achieving low temperature. The overall performance of the refrigeration system is governed by the selection of the mixture composition and heat exchanger used. Therefore, an attempt has been made to investigate the performance of the Joule-Thomson refrigeration system with the addition of a heat exchanger in between the condenser and the expansion device. In order to study the performance of the system experimentally, a Joule-Thomson refrigeration system experimental setup has been developed with mixture of R290/R600a (40/60, 50/50, 60/40 and 70/30 by weight) are used as environmental friendly refrigerant mixtures and experiments are conducted. The effect of evaporator temperature on the coefficient of performance, exergy efficiency and exergy destruction ratio of the J-T system are investigated. The Joule-Thomson refrigeration system with compact heat exchanger yields higher coefficient of performance by 10.45% and exergy efficiency obtained by 4.25%.

Keywords: Coefficient of Performance, Exergy Destruction Ratio, Exergy Efficiency, Heat Exchanger, Joule-Thomson Refrigeration System

1. Introduction

The Ozone Depleting Potential (ODP) and Global Warming Potential (GWP) have become the important conditions in the evolution of new refrigerants. Many researchers have reported that hydrocarbon mixed refrigerants is found to be an energy efficient and environment friendly alternative option. Brodyanskii et al.¹ justified the usage of refrigerants in J-T system. In the past several decades, there has been substantial progress in closed cycle J-T system. Boiarski M. et al.² have obtained a performance evaluation among unflammable refrigerant blends and inflammable refrigerant blends. The refrigeration capacity of a Joule-Thomson system is evaluated by the enthalpy difference between low pressure and high

pressure stream at the pinch point in the heat exchanger. For optimal refrigerants the pinch point is sited at the warm end. At temperature of around 240 K it is almost 1.5 to 2 times higher than at 300 K. Subsequently the cooling capacity can be 1.5 to 2 times greater. In order to usage this result, atmosphere temperature should be moved down to around 240 K. Podtcherniaev et al.³ have described the recital of single stage and two stage Joule-Thomson system using hydrocarbon components for high temperature applications. The gas mixtures used in Joule-Thomson system have mostly lower boiling gases like helium, neon, nitrogen, and argon and light hydrocarbons ranging from methane to pentane. Walimbe et al.⁴ operated the Joule-Thomson system with usual high and low pressure limits of 2.0 Mpa - 3.0 Mpa and 0.1 Mpa - 0.4 Mpa respectively

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Parvez Iqbal A. K. M and Rahman M. M.⁵ studied the Joule-Thomson coolers operated with a sorption compressor. Linde-Hampson cooling cycle was established by thermally cycling the containers and at the same time controlling the gas flow through the JT cold stage. The cooler efficiency increased by a factor of 3 by the application of two-stage compression compared to single stage compression.

Heat exchanger is one of the most essential components of many industrial progressions, equipment's and systems covering a extensive kind of engineering applications. Aggregating consciousness for the active utilization of energy resources, diminishing working cost and care free operation have led to the increase of effective compact heat exchangers. Kays and London⁶ and Shah et al.⁷ defined compact heat exchanger as have surface area to the volume ratio greater than $700 \text{ m}^2/\text{m}^3$. Randall Barron⁸ studied a heat exchanger consists of a small inner tube in which the warm high-pressure gas flows concentric to a larger tube in which the cold low-pressure gas returns. This type of heat exchanger is called as concentric-tube heat exchanger or tube-in-tube type heat exchanger. David Reay⁹ reported in-depth information on commercially available newer compact heat exchanger designs, construction features, materials and size comparisons. Hesselgreaves¹⁰ highlighted to select a compact heat exchanger surface for a given application.

Dalkilic A.S. and Wongwises S.¹¹ studied the possible alternative replacements on a regular Vapour Compression Refrigeration (VCR) system with refrigerant blends consisting of R134a, R152a, R32, R290, R1270, R600 and R600a were compared with R12, R22, and R134a. In spite of the hydro carbon refrigerants highly inflammable features and used in many applications as other refrigerants. Theoretical outcomes revealed that the investigated alternative refrigerants in the scrutiny have a marginally lesser coefficient of performance than R12, R22, and R134a for the condensation temperature of 50°C and evaporating temperatures ranging between -30°C and 10°C . Refrigerant mixtures of R290/R600a (40/60) as an alternative of R12 and R290/R1270 (20/80) instead of R22 are initiate to be substitute refrigerants amongst new alternatives. The effects of the leading factors such as refrigerant type, degree of subcooling and superheating on the refrigerating effect, coefficient of performance and volumetric refrigeration capacity were furthermore examined for different evaporating temperatures.

Austin N. et al.¹² have conducted the exergy analysis test in a vapour compression refrigeration system to replace R134a. The effect of evaporator temperature on the coefficient of performance, exergy efficiency, exergy destruction ratio and efficiency defect for the constant condensing temperature of 52°C for the tested refrigerants are investigated.

2. Experimental Setup and Instrumentation

The schematic of the experimental set up developed in this experimental work is shown in Figure 1. The fabricated compact heat exchanger is integrated between the condenser and the expansion device. The front view (Figure 2.) and rear view (Figure 3.) of the experimental set up are exposed. The counter flow of refrigerant in the heat exchanger is preferred. The hot fluid and cold fluid flows through the inner tube and outer tube of the compact

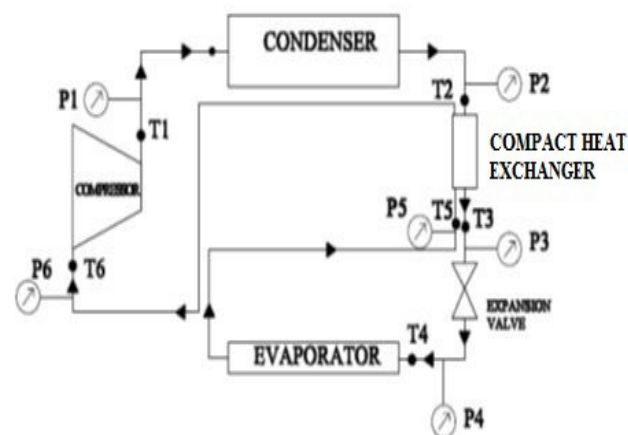


Figure 1. Schematic of the experimental setup.

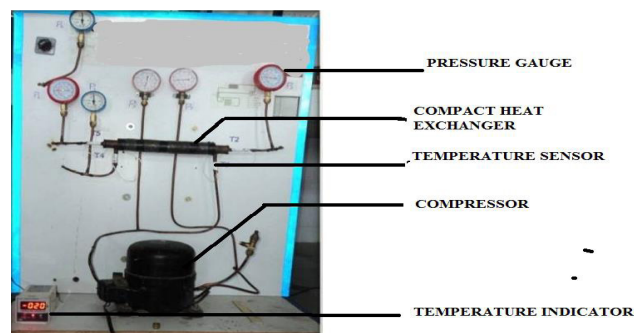


Figure 1. Schematic of the experimental setup.

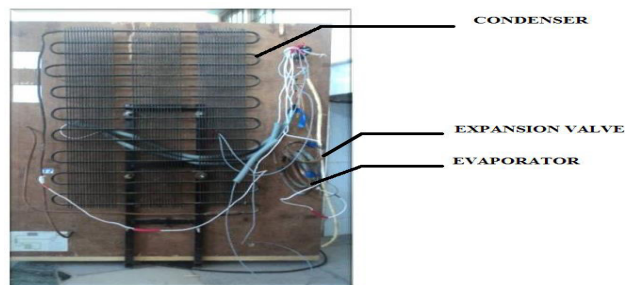


Figure 3. Rear view.

heat exchanger respectively. The outlet of the condenser is given as hot input to the heat exchanger and the output is passed as inlet to the expansion device. The outlet from the evaporator is given as cold input and output is passed as inlet to the compressor to repeat the cycle.

The heat exchanger having a surface area density on any one side is greater than $700 \text{ m}^2/\text{m}^3$ is referred to as a compact heat exchanger regardless of its structural design. Tube-in-Tube type of compact heat exchanger has been fabricated and tested in this work. The compactness

Table 1. Specification of main components

Compressor Type Hermitically Sealed Reciprocating Model Danfoss NF6.1FX.2 Code Number 105G5631 Capacity 168 Liters Clearance ratio 0.04 Stroke volume 6.13 cm^3 Speed 2900 rpm Power supply Single phase, 230 V Current 1.1 Amps Frequency 50 Hz Condenser Type Wire on tube Material Copper Length 9 m Diameter 6.35 mm Expansion Valve Type Capillary Tube Material Copper Length 5 m Diameter 0.36 m Evaporator Type - Coil type Material Copper Length 1.5 m Diameter 19 mm Parameters Measuring Instruments Temperature Type PT100 type temperature sensors Range -50°C to 100°C Accuracy $0.3/0.5^\circ\text{C}$ Resistance 100Ω at 0°C Length range from 1 m to 3 m Pressure Type Bourdon's tube pressure gauge Range 0 - 34 bar Accuracy 0.1% of the full scale Regulator and Temperature indicator Type On/Off Temperature Controller Range -50°C to 300°C Accuracy $\pm 1^\circ\text{C}$	Compact heat exchanger Outer Tube Outside Diameter 35.0 mm Outer Tube Inside Diameter 33.5 mm Outer Tube Length 315 mm Inner Tube Outside Diameter 15.8 mm Inner Tube Inside Diameter 14.8 mm Inner Tube Length 315 mm Outer Tube Inlet and Outlet Outside Diameter 7.9 mm Outer Tube Inlet and Outlet Inside Diameter 7.0 mm Outer Tube Inlet and Outlet Length 50 mm Tube Type ACR Color Code Blue Standard ASTM B280 Trade Name 30 Mesh, 0.013" P/N: 30 X 30-0-014-CU-PW Product Group Copper Wire Cloth Mesh Count 30 X 30 Material Commercially Pure Copper Standard (Material) ASTM B124M-11 Weave Type Plain Weave Standard (Weave Type) ASTM E2016-06 Wire Diameter 0.3302 mm Total Thickness 0.6604 mm Size of a mesh opening 0.871 mm X 0.871 mm Opening per sq./Inch 900 No. of disc in outertube 208 No. of disc in inner tube 230
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is $4500 \text{ m}^2/\text{m}^3$. The specification of main components of the compact heat exchanger is indicated in Table 1. Profile projector is used to measure the size of mesh opening in the disc and also the number of mesh openings in the disc is noted. By using screw gauge the wire diameter of the mesh and the thickness of the mesh are measured.

The meshes were cut by using dies to fit in the space between inner and outer tubes through which one of the fluids will pass through. After cutting it to the required size, meshes are put one by one in small wires of length just greater than 315 mm. Then these meshes are put in the space between the tubes. About 224 discs are incorporated in the annular space and 250 discs of diameter equal to the inner tube diameter are filled in the inner tube of this heat exchanger. Before doing the above said operation the copper tubes are cut according to the required length. The cutting operation is done by means of a tube cutter.

The outer tube and inner tube are cut for a length of 315 mm. In the same way the inlet and outlet tube for tube is also cut. Usually some amount of burrs is present during cutting which can be removed by using a half round file. Then the inner tube is placed inside the outer tube and is kept within by using copper plates using the brazing process. Just before the brazing process holes have drilled to let in and out the flow from the tube.

3. Experimental Procedure

Firstly, the experimental system was charged with nitrogen gas to plaid seepage, to confiscate scums, humidity and other external things inside the system, which may disturb the accuracy of the investigational effects. Then the setup was charged with R134a, the baseline tests were carried out. The refrigeration capacity and coefficient of performance, exergy flow rates of each component and piping of refrigeration system were calculated. The property are calculated using software Aspen version 11.1 using Peng–Robinson equation of state.

The detailed baseline tests procedure of the experimental setup charged with R134a are as follows,

1. The refrigerant is charged with the maximum pressure in the compressor and allowed to run. After reaching the steady state condition, the pressure gauge readings and the temperature sensors readings were taken.

2. The compressor charged pressure of the refrigerant is slightly reduced by releasing the refrigerant from the system. Then the setup is allowed to run and after achieved the steady state the readings were taken. A number of readings were taken until the pressure of the refrigerant supported to change in the evaporator inlet temperature.

The system was evacuated by using a vacuum pump. Then the system has been charged with the mixtures of hydrocarbon refrigerants such as (a) R290/R600a (40/60), (b) R290/R600a (50/50), (c) R290/R600a (60/40) and (d) R290/R600a (70/30) are charged one by one and the experiments are conducted by following the procedure 1 and 2. Then, the compact heat exchanger was installed in between the condenser and expansion device. The procedure 1 and 2 was repeated with above said refrigerants and readings are taken. The percentage of uncertainty in measured parameters were calculated and tabulated in Table 2.

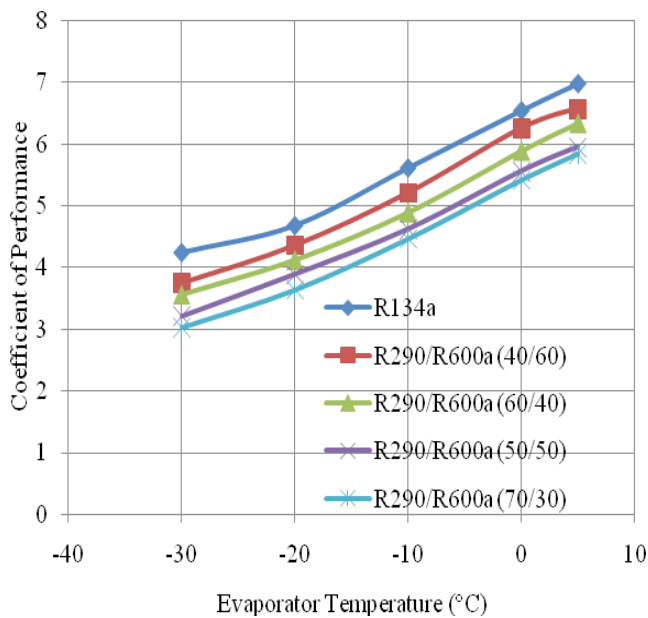
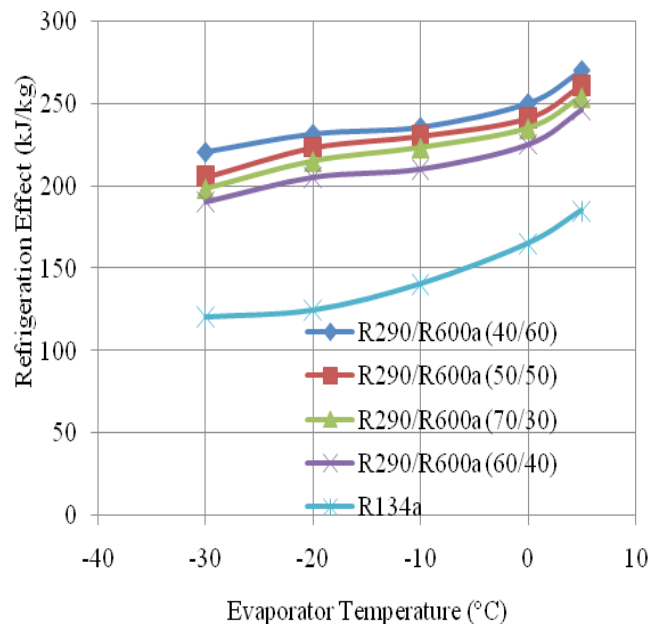
4. Results and Discussion

It is important to enhance the coefficient of performance of the Joule–Thomson refrigeration systems. This is possible only by reducing the irreversibility of the process in the cycle. One of the process which causes maximum exergy loss is the expansion device. This could be reduced by precooling the incoming refrigerant entry into the expansion device. In order to achieve this, a tube-in-tube, counter flow type Compact Heat Exchanger (CHX) is fabricated and introduced in between the condenser and the expansion device. The cold fluid (refrigerant) coming out from the evaporator is utilized to precool the refrigerant entering into the expansion device and the experiments are conducted using R134a, R290/R600a (40/60), R290/R600a (50/50), R290/R600a (60/40) and R290/R600a (70/30) and the results are discussed in this section.

The variant of coefficient of performance with evaporator temperature is revealed in Figure 4. The coefficient of performance rises as the evaporator temperature rises for the constant condensing temperature of 52°C and the evaporator temperature reaching from -30°C to 5°C . The fall in pressure ratio for the compressor also progresses the efficiency. The coefficient of performance of R134a and R290/R600a (40/60) was about 10.08% - 19.27% and 6.44% - 5.33% higher than that of R290/R600a (60/40).

Table 2. Percentage of uncertainty in measured parameters

Sl. No.	Parameter	Percentage of uncertainty (%)				
		R134a	R290/R600a (by wt. %)			
			40/60	50/50	60/40	70/30
1	Compressor outlet Temperature (°C)	±1.05	±2.15	±2.10	±1.14	±1.13
2	Condenser outlet Temperature(°C)	±1.24	±1.07	±2.89	±1.92	±2.36
3	Expansion Device inlet Temperature(°C)	±4.40	±3.06	±3.41	±3.28	±5.91
4	Evaporator inlet Temperature(°C)	±3.45	±3.16	±2.40	±3.27	±3.58
5	Evaporator outlet Temperature(°C)	±2.66	±1.24	±2.56	±1.46	±1.96
6	Compressor inlet Temperature(°C)	±3.88	±4.43	±3.06	±3.33	±3.39
7	Compressor outlet Pressure(bar)	±4.24	±4.35	±5.59	±4.27	±5.56
8	Condenser outlet Pressure(bar)	±3.54	±3.71	±4.81	±5.60	±5.05
9	Expansion Device inlet Pressure(bar)	±3.61	±4.76	±5.05	±6.97	±5.07
10	Evaporator inlet Pressure(bar)	±4.87	±4.49	±4.38	±4.62	±3.75
11	Evaporator outlet Pressure(bar)	±4.07	±3.34	±4.64	±4.91	±3.50
12	Compressor inlet Pressure(bar)	±3.94	±2.58	±3.29	±2.15	±2.65
13	Coefficient of performance	±2.26	±2.04	±2.04	±2.12	±2.11
14	Exergy Efficiency(%)	±2.54	±2.01	±1.91	±2.22	±2.41

**Figure 4.** Coefficient of performance vs evaporator temperature.**Figure 5.** Refrigerating effect vs evaporator temperature.

R290/R600a (50/50) and R290/R600a (70/30) have lesser coefficient of performance of around 6.13% - 9.82% as well 7.86% - 15.23% than that of R290/R600a (60/40).

Figure 5 displays the refrigerating effect rises with growing evaporator temperature for the continual condensing temperature of 52 °C and the evaporator temperature reaching from -30 °C to 5 °C. All the tested refrigerants have much greater refrigerating effects than R134a. The refrigerating effect of R290/R600a (40/60), R290/R600a (50/50), R290/R600a (70/30) and R290/R600a (60/40) was about 45.94% - 83.28%, 41.04% - 70.85%, 37.29% - 65.06% and 33.11% - 58.08% higher than that of R134a, respectively.

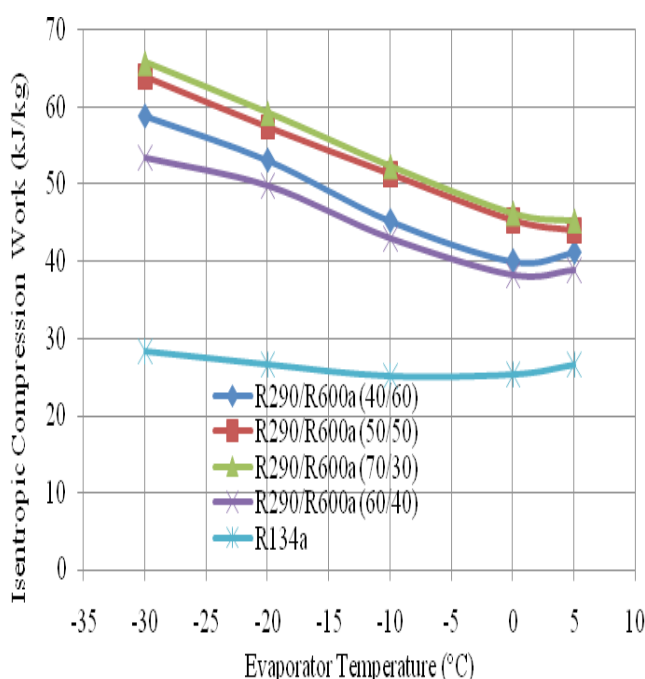


Figure 6. Isentropic compression work vs evaporator temperature.

Figure 6 displays the isentropic compression work decreases with increasing evaporator temperature for the constant condensing temperature of 52 °C and the evaporator temperature ranging from -30 °C to 5 °C. Apart from the R134a, other investigated refrigerants have much higher values of isentropic compression work. R290/R600a (70/30), R290/R600a (50/50), R290/R600a (40/60) and R290/R600a (60/40) have higher isentropic compression work than that of R134a by 70.61% - 56.94%, 65.44% - 55.74%, 50.97% - 51.81% and 46.35% - 46.96% respectively.

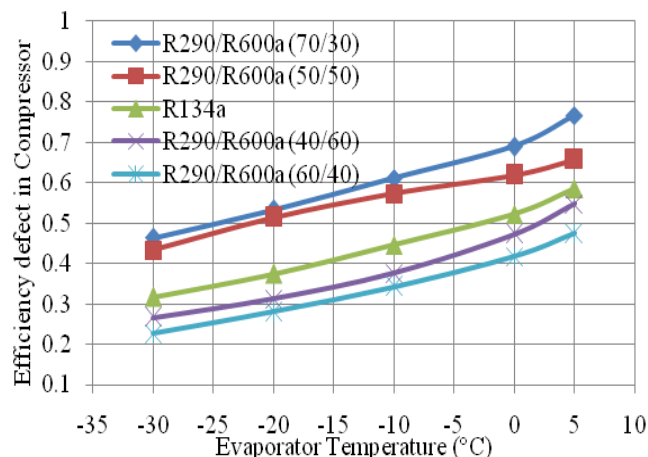


Figure 7. Efficiency defect in compressor vs evaporator temperature.

The Figure 7 shows the comparison of efficiency defect in compressor with varying evaporator temperature for the entire investigated refrigerants. The efficiency defect in compressor increases with increase in evaporator temperature. The effect revealed that efficiency defect in compressor of R290/R600a (70/30) and R290/R600a (50/50) is 30.94% - 46.51% and 12.13% - 37.02% higher than R134a and 6.66% - 16.13% and 19.14% - 28.48% lower than R134a for R290/R600a (40/60) and R290/R600a (60/40).

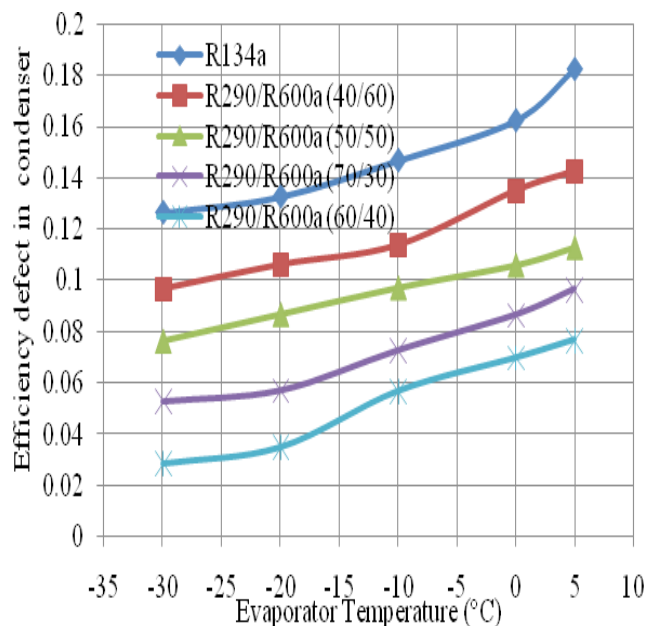


Figure 8. Efficiency defect in condenser vs evaporator temperature.

Figure 8 displays the discrepancy of efficiency defect in condenser with evaporator temperature for the tested refrigerants. The efficiency defect in condenser increases with increase in evaporator temperature. The effect revealed that efficiency defect in condenser of R134a and R290/R600a (40/60) have higher than R290/R600a (50/50) about 62.16% - 66.09% and 26.55% - 28.57% respectively. R290/R600a (70/30) and R290/R600a (60/40) have lower by 14.29% - 28.57% and 31.79% - 71.42% respectively than that of R290/R600a (50/50).

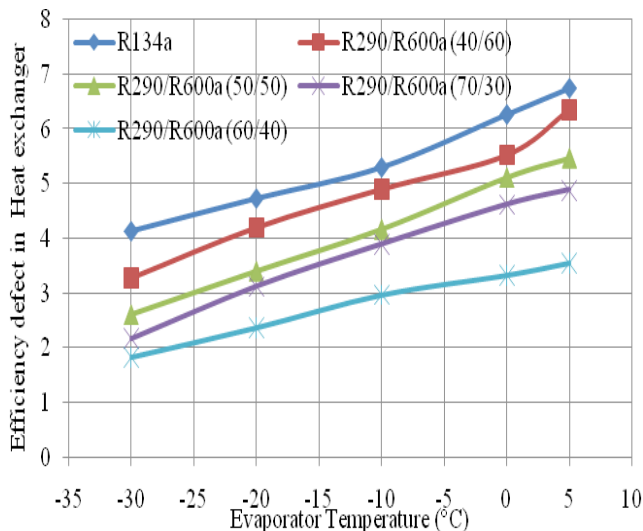


Figure 9. Efficiency defect in heat exchanger vs evaporator temperature.

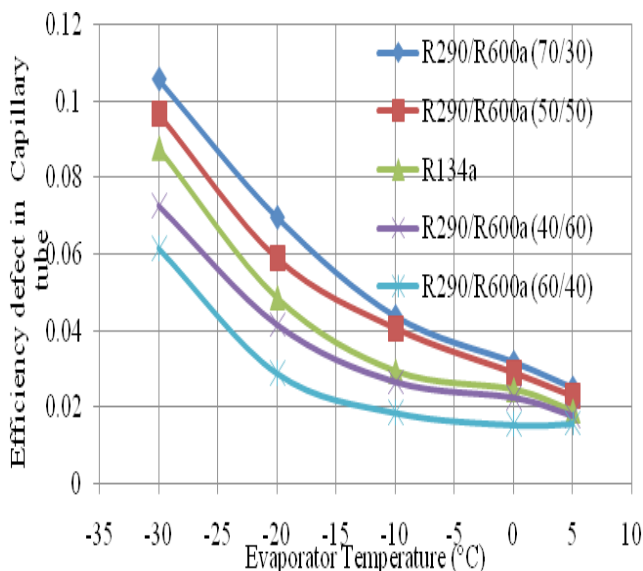


Figure 10. Efficiency defect in capillary tube vs evaporator temperature.

Figure 9 displays the evaluation of efficiency defect in heat exchanger for the tested refrigerants changing evaporator temperature. The efficiency defect in heat exchanger falls with fall in evaporator temperature. The effect gained showed that efficiency defect in heat exchanger of R134a and R290/R600a (40/60) have higher than R290/R600a (50/50) about 23.55% - 58.77% and 16.31% - 25.78% respectively. R290/R600a (70/30) and R290/R600a (60/40) have lower by 10.44% - 16.59% and 35.01% - 19.79% respectively than that of R290/R600a (50/50).

Figure 10 displays the discrepancy of efficiency defect in capillary tube with evaporator temperature for the investigated refrigerants. The result showed that the efficiency defect in capillary tube increases with decrease in evaporator temperature. The efficiency defect in capillary tube of R290/R600a (70/30) and R290/R600a (50/50) have higher than that of R134a by 31.05% - 20.68% and 20% - 10.28% respectively. R290/R600a (40/60) and R290/R600a (60/40) have the lower efficiency defect in capillary tube by 6.84% - 17.03% and 17.37% - 29.71% than that of R134a.

Figure 11 indicates the efficiency defect in evaporator variation with evaporator temperature for the tested refrigerants. The efficiency defect in evaporator increases with increase in evaporator temperature. The effect revealed that efficiency defect in evaporator of R290/R600a (70/30) and R290/R600a (50/50) have higher than that of R134a by 40.64% - 17.31% and 19.33% - 7.09%

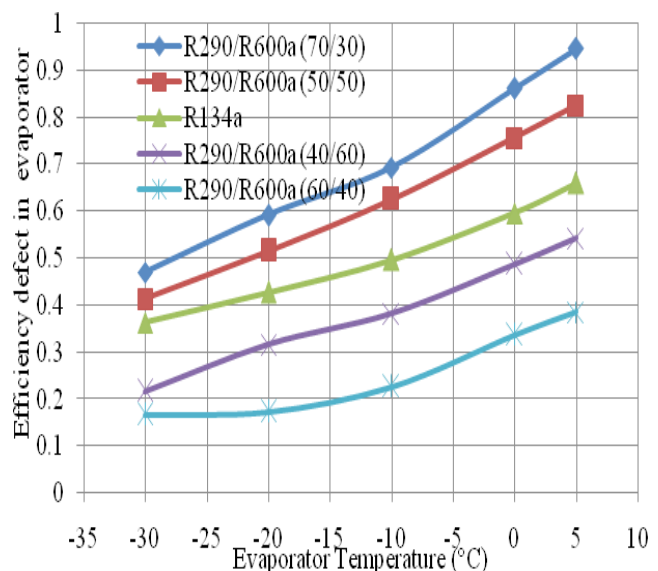


Figure 11. Efficiency defect in evaporator vs evaporator temperature.

respectively. R290/R600a (40/60) and R290/R600a (60/40) have the lower efficiency defect in capillary tube by 21.51% - 38.41% and 43.53% - 57.47% than that of R134a.

Figure 12 indicates the exergetic efficiency deviation with evaporator temperature for the investigated refrigerants. The exergetic efficiency increases with decrease in evaporator temperature. The average exergetic efficiency for R134a and R290/R600a (70/30) have 14.73% higher and 15.29% lower than that of R290/R600a (60/40) correspondingly. Exergetic efficiency of 40.26% and 33.89% are attained at evaporator temperature of -10°C for R290/R600a (40/60) and R290/R600a (50/50) respectively. R134a and R290/R600a (40/60) have higher exergetic efficiency than R290/R600a (60/40) about 24.22% - 9.54% and 9.83% - 6% respectively. The exergetic efficiency of R290/R600a (50/50) and R290/R600a (70/30) have lower by 18.87% - 6.31% and 34.31% - 9.3% than R290/R600a (60/40).

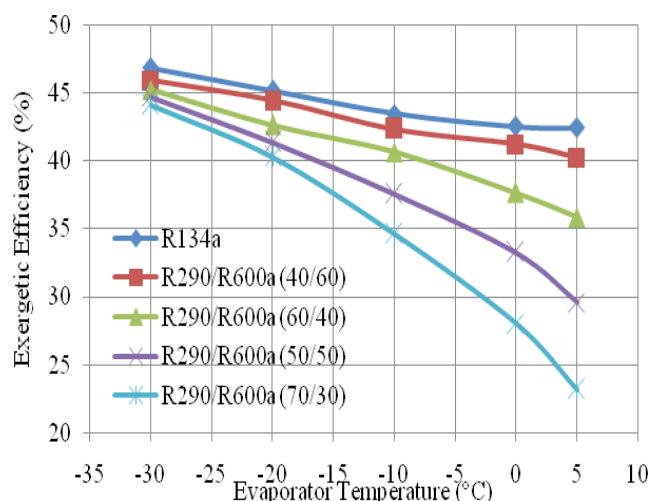


Figure 12. Exergetic efficiency vs evaporator temperature.

Figure 13 shows the comparison of exergy destruction ratio variation for the tested refrigerants with the evaporator temperature. The exergy destruction ratio falls with fall in evaporator temperature. The effect revealed that exergy destruction ratio of R290/R600a (70/30) and R290/R600a (50/50) have higher than that of R290/R600a (60/40) by 43.71% - 17.48% and 34.46% - 11.81% respectively. R290/R600a (40/60) and R134a have the lower exergy destruction ratio by 13.21% - 9.79% and 28.88% - 14.47% than that of R290/R600a (60/40).

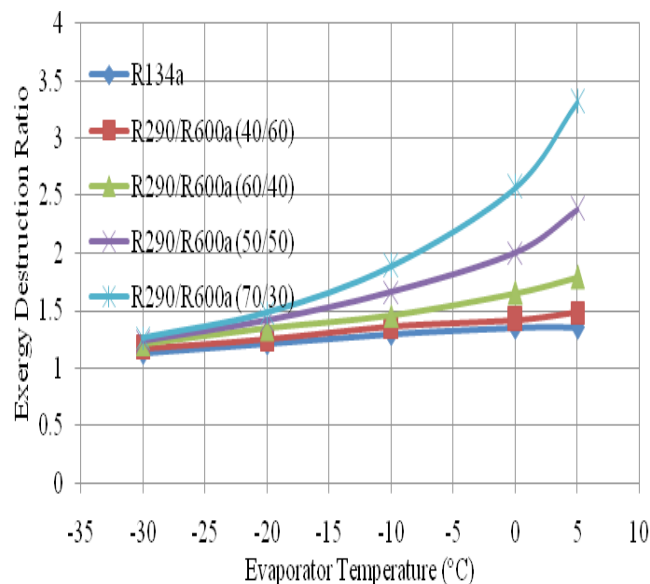


Figure 13. Exergy destruction ratio vs evaporator temperature.

5. Conclusion

- Amongst the refrigerants investigated in the system with compact heat exchanger, the system obtained the maximum coefficient of performance with R134a followed by R290/R600a (40/60).
- The system with compact heat exchanger obtained the maximum coefficient of performance of 6.985 with the refrigerant of R134a.
- The R290/R600a (40/60) mixture working with the system reaches the coefficient of performance of 6.574 with compact heat exchanger.
- The system with compact heat exchanger obtained the maximum refrigerating effect is 270.34 kJ/kg for R290/R600a (40/60).
- The minimum isentropic compression work is 26.51 kJ/kg obtained for R134a with compact heat exchanger.
- The system with compact heat exchanger consumed the minimum power consumption per ton of refrigeration with R134a is 0.501 kW/TR.
- The system with compact heat exchanger obtained the maximum exergy efficiency of 46.89% with R134a.
- The system operated with R134a obtained the minimum exergy destruction ratio of 1.132 with compact heat exchanger.

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