

Theoretical Performance Analysis of an R1234yf Refrigeration Cycle Based on the Effectiveness of Internal Heat Exchanger

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ABSTRACT

In this paper, the effects of internal heat exchanger (IHx) effectiveness on the performance parameters of the refrigeration cycle with R1234yf were theoretically investigated. For this purpose, a mathematical model was developed based on the energy balance of the cycle. The analysis were performed between -20°C and 0°C evaporation and 40°C and 50°C condensation temperatures based on the effectiveness value of IHx. The cooling capacity, coefficient of performance (COP), subcooling, superheat and compressor discharge temperature of the refrigeration cycle was examined. Finally, the performance results of the cycle with R1234yf were compared with the baseline cycle that utilizes with R134a. As a result, it was determined that the critical effectiveness to supply the same COP with R1234yf was determined 50% in comparison the baseline cycle.

Keywords:

R1234yf; R134a; Internal heat exchanger; Effectiveness.

Article History:

Received: 2016/12/02

Accepted: 2017/02/13

Online: 2017/06/30

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INTRODUCTION

According to European Commission (EC) Regulation No 517/2014, Directive No 40/2006 was rearranged so that usage of refrigerants that have GWP value higher than 150 is prohibited in mobile air conditioning systems by 2022 [1, 2]. Thus, finding a new alternative refrigerant to R134a (GWP=1430) has become a necessity and recent studies mainly have been focusing on this topic [3]. Refrigerants such as R152a, R1234yf, R1234ze and R744 (CO₂) were suggested in the literature as environment friendly refrigerants [4, 5, 6, 7]. As an alternative refrigerant, R1234yf (GWP=4) has attracted great attention due to its similar thermophysical properties in comparison with R134a [7, 8]. Therefore, various theoretical and experimental comparison studies about R1234yf can be found in the literature. Among them; Naushad et al. theoretically investigated R1234ze and R1234yf as alternatives to R134a in simple vapour compression refrigeration system. They claimed that R1234yf can be a good alternative to R134a after made some necessary modifications, such as selecting suitable compressor oil and pipe sizes. On the other hand, their study showed that R1234ze requires bigger

compressor displacement volume to supply same cooling capacity [9]. Hoşöz et al. experimentally evaluated the automotive air conditioning (AAC) system with R1234yf as a function of different ambient temperatures. Their results showed that the cooling capacity and COP of R1234yf were lower between 3.50-6.99% and 5.98-21.42% than R134a. [10]. Zilio et al. experimentally and numerically investigated R1234yf in AAC system that was originally designed to work with R134a by making simple modifications. They found that the cooling capacity and COP of R1234yf were quite lower than R134a for the same system. However, their numerical simulations showed that at the same cooling capacities, the system with R1234yf would have higher COP by enhancing the effective areas of the condenser and evaporator by 20% and 10%, respectively [11]. Daviran et al. developed a simulation program to evaluate the performance comparison of the AAC system that works with R1234yf and R134a. They observed that the COP of the system with R1234yf was 18% higher than R134a in case of constant mass flow rate and lower for the equal cooling capacity by 1.3-5% [12]. Qi investigated the potentials

to improve the system performance parameters for the operation of an AAC system using R1234yf. He found that the cooling capacity and COP values of the system was increased up to 15% by changing the subcooling degree from 1K to 10K in the condenser. Moreover, he claimed that the cooling capacity of the system can increase 72.8% by increasing the compressor volumetric efficiency from 55 to 95% [13]. Navarro et al. experimentally studied with R1234yf as a drop-in alternative for R134a in vapour compression refrigeration system. They determined that the COP of R1234yf was 5 to 30% lower than R134a. In addition, when the condensation temperature increased from 313.15 to 333.15 K, they obtained that the COP difference between R1234yf and R134a decreased up to 8%. They concluded that the IHX would cause even more reduction in the COP difference [14]. Babiloni et al. investigated the performance of an air conditioning system using R134a, R1234yf and R1234ze. The results were presented as a function of various evaporator and condenser temperatures for three refrigerants. They determined that R1234yf and R1234ze were showed lower cooling capacity values compared to R134a by 9% and 30%, respectively. Additionally, they claimed that the cooling capacity values were diminished against the increased condensation temperature. They concluded that the cooling capacity and COP of the system with R1234yf would tend to increase by adding IHX at 30% effectiveness [15]. Domanski et al. determined the influence of the liquid line/suction line heat exchanger to the refrigeration system. According to the refrigerant properties and operating conditions, they obtained that the liquid line/suction line heat exchanger can be enhanced the system performance parameters [16]. Aprea et al. demonstrated the consequences of the adapting IHX to the refrigeration system. They expressed that preventing flash gas formation at the inlet of expansion valve and liquid contained vapour which enters to compressor are the benefits of IHX [17]. Klein et al. investigated the influence of IHX on the performance of a refrigeration system for a number of alternative refrigerants. They found that when IHX adapted to the system the mass flow rate of the system was decreased due to decreasing suction density. Additionally, they implied that for some refrigerants the cooling effect was increased with increasing subcooling degree supplied with IHX [18]. Mastrullo et al. conducted experiments to show the performance of 19 ozone friendly refrigerants. They implied that installation of IHX can have positive or negative effect regarding to the refrigerant fluids. Their results showed that the IHX had the same influence on the COP and refrigerating volumetric capacity [19]. Moles et al. performed theoretical comparison analysis with R1234yf and R1234ze in different single stage vapour compression refrigeration system configurations. They obtained the best performance for the system with IHX

when ejector or expander used as an expansion device. They claimed that using IHX with effectiveness over 45% significantly enhanced the COP of the system. Consequently, they determined the relative difference between R1234yf and R134a in COP and cooling capacity were lower for R1234yf by 4-8% and 4-7%, respectively [20]. Cho et al. conducted experiments to perform the energy and exergy analysis of the AAC system with and without using IHX. They determined that COP of the system with R1234yf was 3.6 to 4.5% lower than R134a. Additionally, with presence of IHX, the COP of the system was found to be 0.3-2.9% lower for R1234yf at the compressor speed between 800-1800 rpm. Moreover, they observed that it was 0.9% higher at 2500 rpm. On the other hand, they indicated that the total exergy destruction rate was decreased by adapting IHX to the system with R1234yf [21]. Navarro et al. experimentally determined the effects of IHX in conventional vapour compression refrigeration system. They carried out 36 steady-state tests with various operating conditions and compared the results between R1234yf and R134a. They found that the relative cooling capacity and COP was reduced using IHX by 2% and 6%, respectively [22]. Pottker et al. demonstrated that the COP of the system with R1234yf was increased by increasing subcooling degree of the condensed liquid when IHX used in the system. They used 1.5 m aluminium concentric double pipe IHX with 35% effectiveness in their system. They observed that the COP of their system was improved by 16% due to increased subcooling degree with IHX [23]. Coating materials are made for increasing corrosion resistance of material, preventing discontinuity on metals (scratches, pores) and gaining functional quality [2]. In gears while coating, decreases surface roughness of gear, scoring resistance and corrosion resistance are increased.

In this study a more comprehensive performance analysis approach was followed based on variable effectiveness of IHX and operating conditions. For this purpose, a conventional refrigeration cycle was considered. The result were discussed under two different parts. In the first part, the variations of subcooling, superheat and the compressor discharge temperature of the cycle with R1234yf were examined based on effectiveness. In the second part, the influence of the effectiveness on the cycle performance parameters for R1234yf was compared with the baseline cycle that utilizes R134a. The cooling capacity and COP values

Table 1. Operating Conditions

Parameters	Value(s)	Range
Evaporation temperature (T_{evap})	-20-0 °C	5
Condensation temperature (T_{cond})	40-50 °C	2
Reference superheat	5 °C	Fixed
Reference subcooling	10 °C	Fixed
IHX effectiveness (ϵ_{IHX})	0-100%	25

were investigated as performance parameters of the refrigeration cycle. Finally, the relative performance parameters were compared under various evaporation and condensation temperatures.

MATERIALS AND METHODS

Description of the Cycle

As seen in Figure 1, the theoretical cycle consists of a compressor, condenser, evaporator, expansion valve and the IHX which is added between the evaporator and condenser outlets. Zero effectiveness represents the basic cycle and R1234yf follows the same cycle with R134a. On the other hand, in the mathematical model, the effectiveness values of IHX are considered only for R1234yf.

Thermodynamic Analysis

In order to determine the performance parameters, the mathematical model based on energy balance of the theoretical cycle were developed. For this purpose, the operating conditions represented in Table 1 were considered as basis of the calculations. The subcooling and superheat was used as constant values for the zero effectiveness. Therefore, these are selected as the reference values of the cycle. Additionally, the range of condensation and evaporation temperatures was selected according to the working conditions of real refrigeration system.

The following assumptions were made in order to carry out the calculations:

1. Steady-state operating conditions were assumed.
2. The changes in kinetic and potential energies are negligible.
3. The pressure losses within components and pipes are negligible.
4. Volumetric efficiency (η_v) of the compressor is 1.
5. Isentropic efficiency (η_{is}) of the compressor is 1.
6. Compressor, expansion valve and pipes are adiabatic.
7. Isenthalpic process is considered at the expansion valve.

The energy calculations are based on kJ per kg. Additionally, the formulas were utilized for the ideal vapour compression refrigeration cycle with IHX using the first law of thermodynamics.

The suction line temperature (T_1) can be expressed as;

$$T_1 = T_{\text{evap}} + T_{\text{sh}} \quad (1)$$

Where, T_{evap} , represents the evaporation temperature and T_{sh} is superheat.

The temperature at the condenser outlet (T_3) can be defined as;

$$T_3 = T_{\text{cond}} - T_{\text{sc}} \quad (2)$$

Where, T_{cond} represents the condensation temperature and T_{sc} is subcooling.

ΔT_{max} is the difference between condenser outlet temperature and evaporator outlet temperature:

$$\Delta T_{\text{max}} = T_3 - T_1 \quad (3)$$

The maximum heat flux (q_{max}) can be determined by Equation 3.

$$q_{\text{max}} = C_{p1} \Delta T_{\text{max}} \quad (4)$$

Where, C_{p1} is the specific heat at constant pressure for the suction line.

Accordingly, the heat flux of IHX (q_{IHX}) based on effectiveness, is calculated as follows;

$$q_{\text{IHX}} = \epsilon_{\text{IHX}} q_{\text{max}} \quad (5)$$

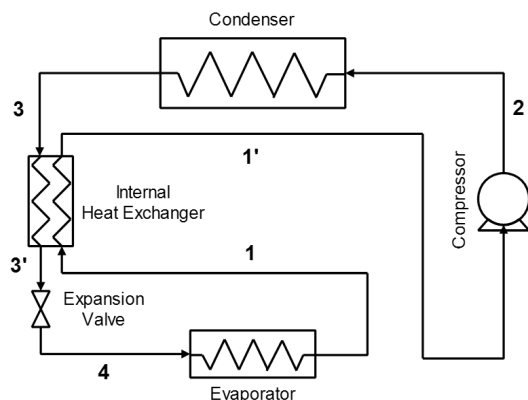


Figure 2. The changing of weight in the pinion to determine the critical scoring load

Where, ϵ is the effectiveness of IHX.

The refrigerant temperature at the IHX outlet for the evaporator side ($T_{1'}$) and for the condenser side ($T_{3'}$) can be calculated using Equations 6 and 7.

$$T_{1'} = (q_{\text{IHX}}/C_{p1}) + T_1 \quad (6)$$

$$T_{3'} = T_3 - (q_{\text{IHX}}/C_{p3}) \quad (7)$$

Where, C_{p3} is the specific heat at constant pressure for

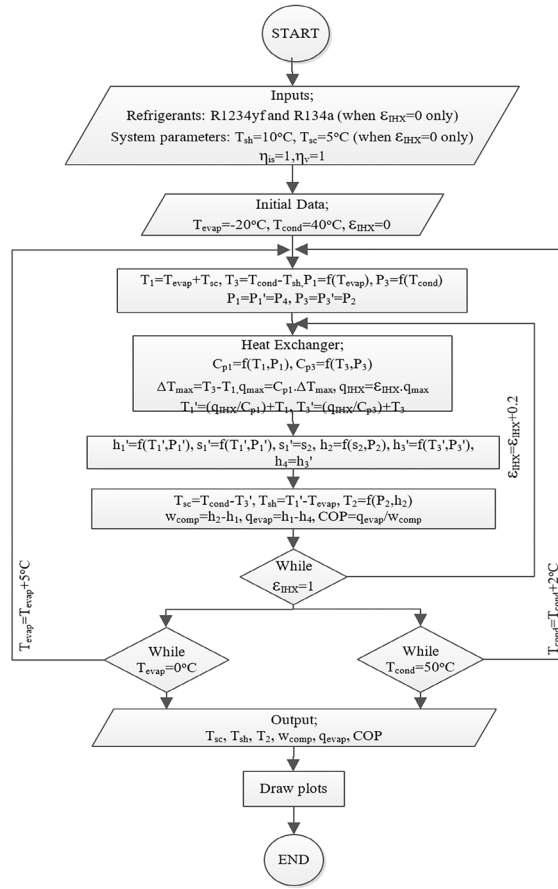


Figure 2. Flowchart for the refrigeration cycle

the subcooled refrigerant.

The compressor discharge temperature (T_2) and enthalpy value (h_2) were obtained by the isentropic compression process in the compressor. Additionally, the enthalpy value at the outlet of the expansion valve (h_4) was obtained by the isenthalpic process in the expansion valve.

The superheat and subcooling can be determined by Equations 8 and 9;

$$T_{sh} = T_{1'} - T_{evap} \quad (8)$$

$$T_{sc} = T_{cond} - T_{3'} \quad (9)$$

The cooling capacity (q_{evap}) can be expressed as;

$$q_{evap} = h_1 - h_4 \quad (10)$$

Compressor work can be calculated by Equation 11.

$$w_{comp} = h_2 - h_1 \quad (11)$$

COP can be expressed as;

$$COP = q_{evap} / w_{comp} \quad (12)$$

In order to compare the performance parameters of the cycles with R1234yf and R134a under various effectiveness values, the equations 13 and 14 was used;

$$COP\% = ((COP_{R1234yf} - COP_{R134a}) / COP_{R134a}) * 100 \quad (13)$$

$$q_{evap}\% = ((q_{evap,R1234yf} - q_{evap,R134a}) / q_{evap,R134a}) * 100 \quad (14)$$

The mathematical model was developed using Engineering Equation Solver (EES-V9.172-3D) and the enthalpy values were extracted from the EES database [24]. The program algorithm flowchart is shown in Figure 2.

RESULTS AND DISCUSSION

In the first part of this section, the variations of subcooling, superheat and the compressor discharge temperature of the cycle with R1234yf were investigated with respect to the effectiveness. In the second part, the influence of the effectiveness on the cycle performance parameters for R1234yf was compared with the baseline cycle that utilizes R134a. The changes in subcooling and superheat versus condensation temperature are shown in Figure 3a, b where the evaporation temperature is maintained

at -10°C . It is understood from the Figure 3a, that the subcooling increases with increasing effectiveness due to the raise in heat flux of the IHX against constant evaporation temperature. Additionally, the maximum

subcooling was obtained at the theoretical limits of the operating conditions where the condensation temperature and effectiveness were 50°C and 100%, respectively. On the other hand, superheat is increased

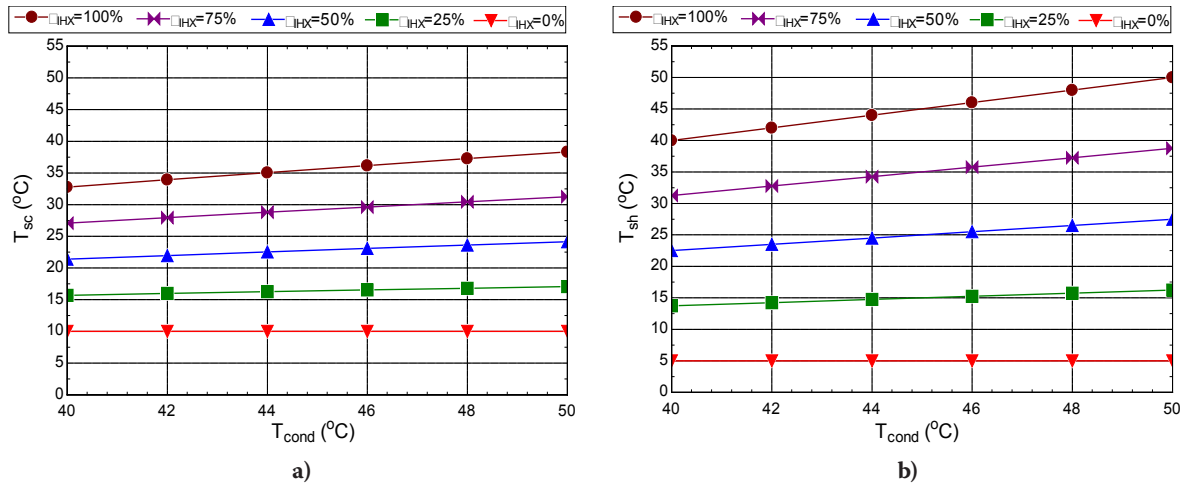


Figure 3. Variation of (a) Subcooling versus condensation temperature, (b) Superheat versus condensation temperature

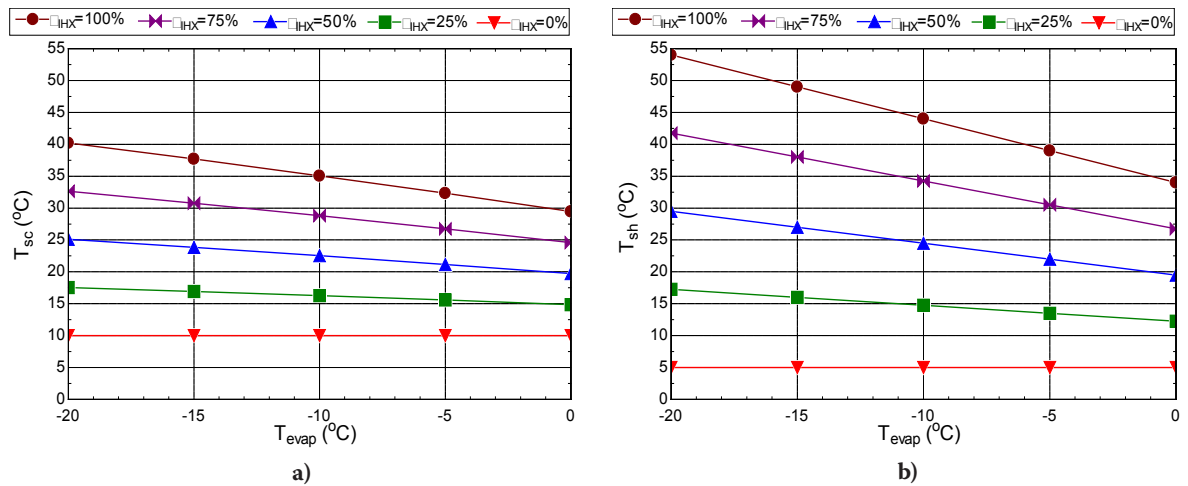


Figure 4. Variation of (a) Subcooling versus evaporation temperature, (b) Superheat versus evaporation temperature

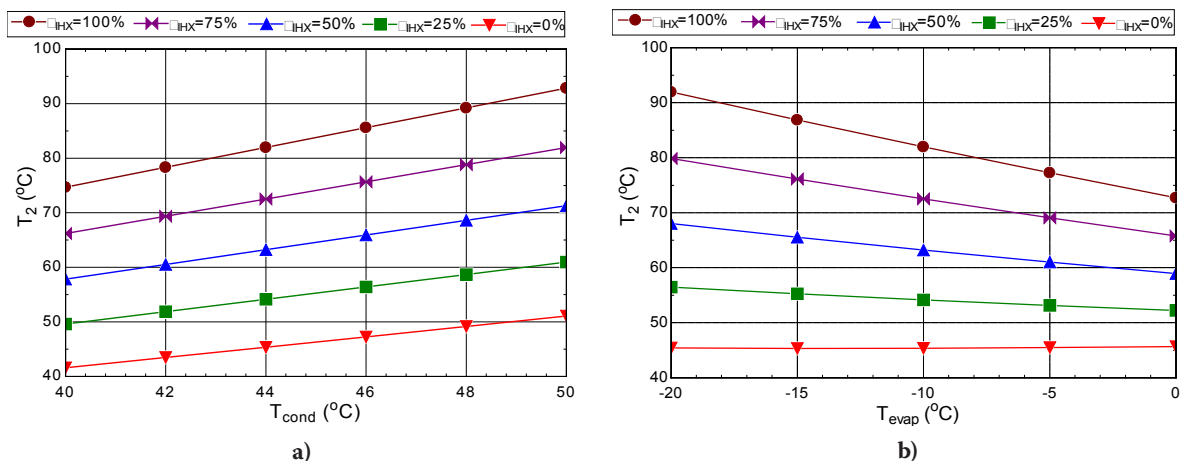


Figure 5. Variation of (a) with condensation temperature, (b) with evaporation temperature

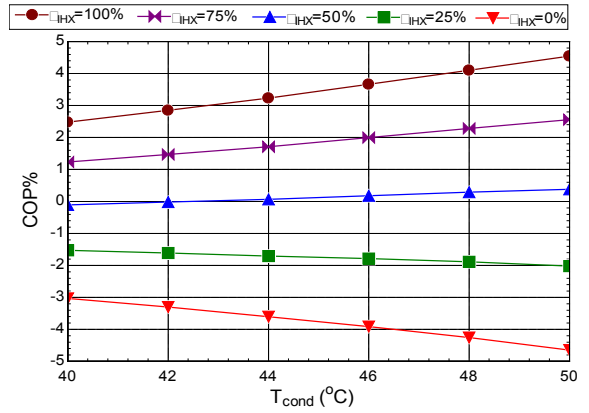
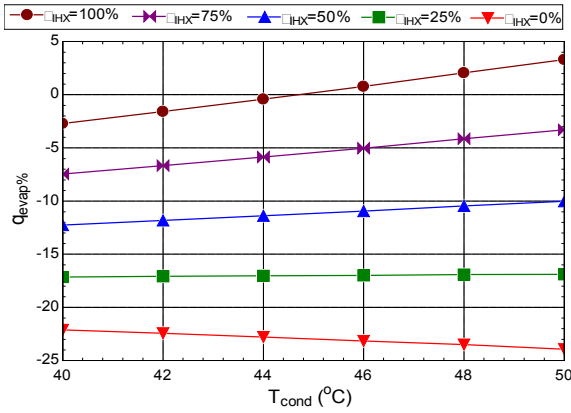


Figure 6. (a) $q_{\text{evap}}\%$ versus condensation temperatures, (b) COP% versus condensation temperatures

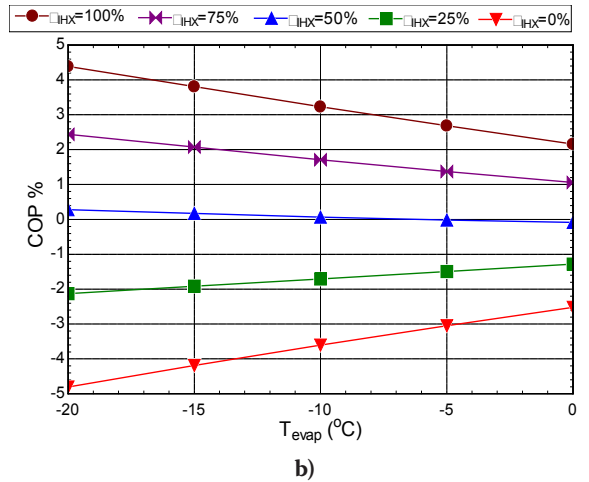
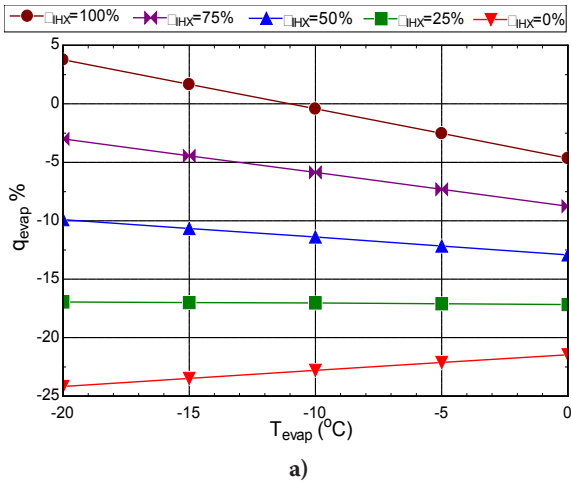


Figure 7. (a) $q_{\text{evap}}\%$ versus evaporation temperatures, (b) COP% versus evaporation temperatures

by increasing effectiveness as shown in Figure 3b. Furthermore, the rate of increase for superheat is higher than that of subcooling with increasing effectiveness.

Figure 4a, b shows the variations of subcooling and superheat versus evaporation temperature where the condensation temperature is maintained at 44°C. Subcooling and superheat tend to decrease while evaporation temperature increases due to reduction in specific heat values at the outlets of IHX for the evaporator and condenser sides. It has to be noted that at 44°C of condensation temperature and -10°C of evaporation temperature, the subcooling and superheat are the same with respect to the IHX effectiveness.

The compressor discharge temperature is an important parameter for compressor performance due to the compressor oil operating working range. Figures 5a, b indicates the variation of compressor discharge temperature based on effectiveness. It was observed that for all operating conditions, the compressor discharge tempe

rature did not exceed 93 °C. However, these values were calculated when the isentropic efficiency of the compressor assumed as 1. In reality, the actual values would be obtained higher than these amounts.

The performance parameters of the cycle such as COP and q_{evap} values are highly depend on the condensation and evaporation temperatures of the refrigerant. Therefore, in Figures 6 and 7, the influence of the effectiveness on the cycle performance parameters for R1234yf was compared with the baseline cycle that utilizes R134a under various evaporation and condensation temperatures.

Figure 6a shows that the cooling capacity was decreased with increasing condensation temperature when the effectiveness was zero for R1234yf. The main factor affecting the cooling capacity is the decreasing enthalpy difference in the evaporator associated with the increasing condensation temperature. Conversely, the cooling capacity was increased because of higher subcooling supplied with adapted IHX to the cycle. On the other hand, the

cooling capacity of R1234yf was higher than R134a after 44°C condensation temperature with 100% effectiveness. However, the relative COP difference was closed at 42°C condensation temperature with 50% effectiveness as shown in figure 6b. Above 50% effectiveness the COP of R1234yf is higher than R134a regardless to the condensation temperature. This situation is directly related with the energy consumption in the compressor. For instance, at 42°C condensation temperature with 50% effectiveness, the relative cooling capacity and COP are 12% and 0%, respectively. This is due to the smaller energy requirement of R1234yf to run the compressor for the same conditions in comparison with R134a.

Figure 7a shows that the cooling capacity increases with the increasing evaporation temperature at zero effectiveness. Besides, it was determined that the IHX was helped to improve the cooling capacity. The cooling capacity difference is nearly the same for all investigated evaporation temperature. However, when the effectiveness is above 25%, the difference is getting wider with increasing evaporation temperature. The most dramatically reduction was observed when the evaporation temperature is increased at the 100% effectiveness value. Figure 7b indicates that the relative COP difference changed similarly with the cooling capacity. However, the critical effectiveness to achieve the same COP is 50% for R1234yf.

CONCLUSION

In this theoretical study, the performance parameters of R1234yf were evaluated for various condensation and evaporation temperature by adding the IHX to the refrigeration cycle. The results were compared under varying IHX effectiveness with basic cycle of R134a. The outcomes from theoretical analysis can be summarized as follows:

- The subcooling and superheat degrees are increase with increasing condensation temperature. On the contrary, these values are declined with increasing evaporation temperature. Besides, these values increase with increasing effectiveness, regardless to the operating temperatures and the rate of increase for superheat is higher than that of subcooling.
- Similarly, increasing effectiveness causes the compressor discharge temperature increases. In reality, when the compressor discharge temperature exceeds 100°C, it should be considered for the safety reasons.
- At 25% effectiveness, the relative cooling capacity difference is not influenced by the variation of condensation and evaporation temperature.

Although, in high condensation and low evaporation temperature conditions, the relative cooling capacity difference increases with increasing IHX effectiveness.

- Moreover, relative cooling capacity and COP difference are changed similarly under various evaporation and condensation temperatures. Finally, the critical effectiveness value to achieve the same COP with R1234yf is 50%.

ACKNOWLEDGEMENTS

The authors would like to thank the University of Yalova for supporting this study through a Research Project No: 2015/BAP/123.

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