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PERFORMANCE IMPROVEMENT POTENTIALS OF LOW GLOBAL WARMING POTENTIAL REFRIGERANTS FOR INTERCITY BUS AIR CONDITIONING SYSTEM

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In this study, theoretical investigation of two evaporator ejector refrigeration system was carried out based on energy and exergy analysis using R134a and low global warming potential refrigerants (namely R1234yf and R1234ze(E)). In order to perform the analyses, a simulation model was developed and then the influence of different parameters on the COP, COP increase rate and exergy destructions were discussed for each refrigerant. The model was validated with experimental data for R134a and later used to predict the behavior with R1234yf and R1234ze(E). It was found that the highest exergy destruction occurs in the R1234yf system and the increase rate in the COP with respect to the conventional system by using ejector as an expansion device are 15% for R134a, about 17% for R1234yf and about 15% for R1234ze(E), respectively.

Key words: *ejector, two-evaporator, COP, R1234yf, exergy, low GWP*

Introduction

Air-conditioning is important for the transportation industry. The need is huge, especially for intercity buses. It takes about 10 kW of mechanical energy from the engine. Nearly all buses and minibuses are using air-conditioning systems to ensure passengers' comfort recently. Standard bus refrigeration systems have a significant effect on fuel consumption and exhaust gas emissions to the environment because refrigeration system's compressor is connected to the engine shaft by a pulley. Standard bus refrigeration systems generally work with R134a and it is known that mobile refrigeration system gas leakage plays a big role in global warming. According to the Kyoto protocol [1] and the Directive of European Parliament [2], mobile refrigeration system studies have been focused on alternative refrigerants which have lower global warming potential.

In order to meet global ecological goals, conventional refrigerants should be replaced by more environmentally friendly ones. Researches have focused on developing new refrigerants [3].The first candidate of the new refrigerants R1234yf has been proposed as a replacement for R134a in mobile air conditioning systems [4]. It has comparable thermodynamic properties to R134a, make it ideal as a replacement, as it requires for few or no alterations in order to replace R134a with R1234yf in a pre-designed system [5]. Researches on R1234yf concluded that the evaporation heat transfer coefficient is nearly same as R134a [6] and condensation heat transfer coefficient of R1234yf was lower than R134a by 12-35% depending on the test conditions [7].

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Another candidate refrigerant R1234ze(E) has been proposed in new systems of medium temperature applications [8] because the vapor pressure of R1234ze(E) is between –28% and –24% compared with R134a. Yataganbaba *et al*. [9] presented exergy analysis of R1234yf and R1234ze(E) in a two-evaporator refrigeration system. Lastly, Qi [10] investigated the performance improvement potentials of R1234yf in mobile air conditioning system under various operation conditions.

While environment-friendly refrigerant research is progressing, several studies have been focused on the recover lost energy in expansion valve by using two-phase ejector instead of the expansion valve in refrigeration systems. Thus the performance of the conventional refrigeration cycle can be improved.

Various ejector based refrigeration systems have been studied with different working substances. The ejector-expansion technology of replacing the throttling valve was summarized by Sarkar [11], Besagni *et al*. [12], and Chen *et al*. [13]. A number of theoretical and experimental studies are given in the review papers. Besagni *et al*. [14] investigated working fluids which used differently in models and compared to these models with a set of experimental data available in the literature, they show that the working fluids had a great impact on the ejector performance and each refrigerant had its own range of operating conditions. On the other hand, it should be noted that the ejectors in the mentioned studies operated with only one evaporation temperature, which was known as the single-evaporator refrigeration system. In fact, introducing the ejector into the multi-evaporator refrigeration system has been an attractive option since the ejector could be used to maintain the required pressure differences between the high temperature and low-temperature evaporators and obviously lower the power consumption by its pressure recovery effect [15]. Li *et al*. [16] experimentally investigated entrainment and pressure recovery performances of the variable area-ratio ejector applied in the multi-evaporator refrigeration system in detail. Kutlu *et al*. [17] studied bi-evaporator transcritical ejector refrigeration system and they reported 21% COP improvement can be obtained with using ejector in the R744 transcritical cycle. Unal and Yılmaz [18] performed the thermodynamic analysis of a bus air-conditioning system enhanced with a two-phase ejector. The analysis showed that the system COP improvement could reach 15% for proper design.

A survey of the literature shows that the ejector cycle becomes complicated due to the phase separator from its mass balance point of view. Practically this may give a lower performance during separation. So there is a clear need for further investigations of the bi-evaporator system in which instead of the phase separator, an auxiliary evaporator is applied for the case of low GWP refrigerants as a working fluid. In this paper, thermodynamic and exergetic analysis of two evaporator ejector refrigeration systems working with low GWP refrigerants in buses is performed. In calculations, refrigerant velocities at the ejector inlets are taken into consideration in energy and momentum equations. Change of COP of the conventional cycle and ejector refrigeration cycle is compared and shown graphically. Exergy destruction rates are also investigated.

Two evaporator ejector bus refrigeration systems

In buses, air conditioning systems are vapor compression cycles driven by the engine through the clutch. Bus refrigeration systems components are a compressor, condenser, liquid tank, evaporator and expansion valve. The layouts of the ejector bus refrigeration system's components are settled on the bus roof, except the compressor. Figure 1 gives a schematic view of the system. Two-phase ejector system is one of the alternate cycles for expansion work recovery. The liquid at the outlet of the condenser is split into two streams. One of these streams
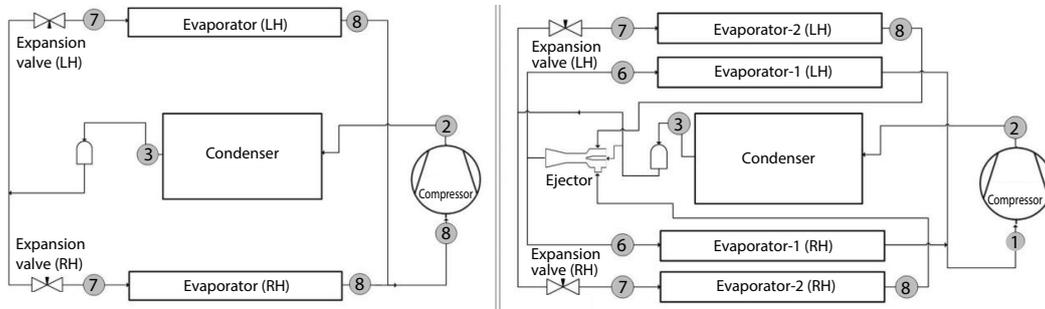
Figure 1. Schematic view of the conventional and ejector refrigeration systems

is isentropically expanded through the ejector, and the other stream is isenthalpically throttled, sent through an evaporator, and then sent to the suction nozzle of the ejector. The two streams are combined in the ejector mixing section, and the two-phase fluid at the ejector outlet is sent through a secondary evaporator before entering the compressor. Main advantages of this system that because the two-phase fluid exiting the ejector are evaporated before entering the compressor, this cycle does not require a liquid-vapor separator, and because of the increase in saturation pressure and temperature provided by the ejector, this cycle yields the possibility for two different evaporation temperatures.

Pressure-enthalpy (P-h) diagram of the system is shown in fig. 2. The refrigerant which comes from the first evaporator enters the compressor. Then the fluid is compressed to the desired level. After the refrigerant is cooled in condenser then it is divided into to be sent to the ejector and expansion valve at the condenser exit. The fluid, which leaves the expansion valve, goes the second evaporator then the refrigerant flows into the ejector as the secondary flow. Primary and secondary flows are mixed in the mixing section. After the process, the refrigerant enters diffuser section of the ejector. Then it enters to the first evaporator. The refrigerant which comes from the first evaporator goes to the compressor, thus the cycle is completed.
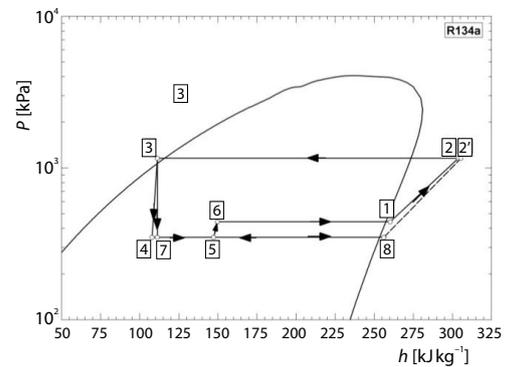
Figure 2. The P-h diagram of the conventional and ejector refrigeration systems

In order to conduct thermodynamic analyses of the cycle, we accepted the following assumptions:
– condensing and evaporation temperatures were constant,
– pressure losses of the whole system are neglected,
– nozzle and diffuser isentropic efficiencies are known, and
– the process in the mixing section takes place at constant pressure and constant cross-sectional area and efficiency of the mixing section of the ejector are known.

Thermodynamic properties of point (1) can be determined if the first evaporator temperature is known. For calculating the thermodynamic properties at the compressor exit, compressor isentropic efficiency expression can be used:

$$\eta_{comp} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1}$$

Isentropic efficiency of compressor changes with compression ratio and compressor rpm value. However, as the general assumption, can be used empirical formulation given in reference [18]. The h_{2s} is determined by using eq. (2) as a function of the P_{2s} and s_{2s}. P_{2s} and s_{2s} indicate pressure and entropy after isentropic compression, respectively. As known, s_{2s} is equal to s_1:

$$h_{2s} = f(s_{2s}, P_{2s}) \tag{2}$$

The enthalpy of the refrigerant at the compressor discharge can be found from eq. (1) by using the compressor isentropic efficiency. Since the condenser temperature is known, thermodynamic properties at the condenser exit are calculated.

The ejector consists of three main sections that are nozzle, mixing, and diffuser. Points (3), (4), (5) and (6) indicate the nozzle inlet, nozzle exit, diffuser inlet and diffuser exit, respectively.

Thermodynamic properties at the point (4) can be calculated by using the energy equation between points (3) and (4) given in Eq. (3) and eq. (4). Due to conservation of mass principle, it can be considered that and velocity of the refrigerant at the nozzle inlet is neglected in eq. (3).

$$h_3 = h_4 + \frac{V_4^2}{2} \tag{3}$$

$$\eta_n = \frac{h_3 - h_4}{h_3 - h_{4s}} \tag{4}$$

In the mixing section, constant pressure-constant cross-sectional area model is used. Pressure difference produced by the entrainment process of the ejector is neglected. However, inlet velocity of the secondary flow is taken into consideration in mixing section energy equation. Here, conservation of mass principle is applicable as given in eq. (5) and the thermodynamic properties of the refrigerant at the point (5) can be calculated by using energy and momentum equations which are given in eqs. (6) and (7).

$$\dot{m}_4 + \dot{m}_8 = \dot{m}_5 \tag{5}$$

$$\left(h_4 + \frac{V_4^2}{2}\right) + \omega\left(h_8 + \frac{V_8^2}{2}\right) = (1+\omega)\left(h_5 + \frac{V_5^2}{2}\right) \tag{6}$$

$$\eta_m = \frac{(1+\omega)V_5^2}{V_4^2 + \omega V_8^2} \tag{7}$$

ω is defined as the entrainment ratio, which shows the ratio of mass flow rates of primary and secondary fluids that enter the ejector, given in eq. (8):

$$\omega = \frac{\dot{m}_8}{\dot{m}_3} \tag{8}$$

In order to determine thermodynamic properties at the diffuser exit, diffuser isentropic efficiency, and energy equation can be used as given eqs. (9) and (10), respectively.

$$h_5 + \frac{V_5^2}{2} = h_6 + \frac{V_6^2}{2} \tag{9}$$

$$\eta_d = \frac{h_{6s} - h_5}{h_6 - h_5} \tag{10}$$

In order to provide proper oil return, the minimum refrigerant velocity is recommended as 5-7 m/s in the compressor suction line [19]. However, there is an evaporator between the compressor and diffuser outlet in this work. So, velocity of the fluid at the ejector diffuser exit is considered as $V_6 = 15$ m/s for the sake of safe oil return.

The refrigerant goes to the expansion valve at the point (3) and exits from there at point (7). The expansion process in the expansion valve is constant enthalpy process.

Primary mass flow rate ($\dot{m}_3$) can be calculated from the cooling capacity of the system which is given in eq. (11):

$$\dot{Q} = \dot{m}\left[\omega(h_8 - h_7) + (1+\omega)(h_1 - h_6)\right] \tag{11}$$

The coefficient of performance of the system can be calculated by the equation below:

$$\text{COP} = \frac{\dot{Q}_{e1} + \dot{Q}_{e2}}{\dot{W}} = \frac{\omega(h_8 - h_7) + (1+\omega)(h_2 - h_1)}{(1+\omega)(h_2 - h_1)} \tag{12}$$

COP of the ejector refrigeration system given in eq. (12) is compared to the COP of the conventional refrigeration system (COP_{conv}), and the COP increase rate (COP^*) is determined:

$$\text{COP}^* = \left(\frac{\text{COP} - \text{COP}_{\text{conv}}}{\text{COP}_{\text{conv}}}\right)100 \tag{13}$$

The secondary evaporation temperature is used for the conventional cycle computation. Therefore, COP of the conventional refrigeration system can be calculated by eq (14):

$$COP_{\text{conv}} = \frac{h_8 - h_7}{h_{2'} - h_8} \tag{14}$$

Exergy analysis

Exergy analysis can overcome many limitations of energy analysis. It is useful to identify the locations, magnitudes, and causes of process inefficiencies [20]. Exergy analysis is a powerful tool for designing, optimization, and performance evaluation of energy systems and it is aimed to determine the maximum performance of the system with overcoming exergy destructions [21, 22].

The system's exergy models are established as follows [23-25]. According to the definition of exergy and exergy balance at steady operation, the exergy at any point and exergy destruction in a component can be expressed as follows:

$$Ex = \dot{m}_i\left[(h_i - h_0) - T_0(s_i - s_0)\right] \tag{15}$$

$$\dot{Ex}_d = \dot{Ex}_{\text{in}} - \dot{Ex}_{\text{out}} + \sum\left[\dot{Q}\left(1 - \frac{T_0}{T}\right)\right]_{\text{in}} - \sum\left[\dot{Q}\left(1 - \frac{T_0}{T}\right)\right]_{\text{out}} + \sum\dot{W}_{\text{in}} - \sum\dot{W}_{\text{out}} \tag{16}$$

Where T_0 is as reference temperature maintained at 27 °C and the reference pressure sets at 101.32 kPa throughout this study for reference enthalpy and entropy. Subscript *i* denotes to each point shown in fig. 2. The temperature *T* was assumed to be 5 °C greater than the evaporation temperature in the evaporators and 5 °C less than the condensation temperature in the condenser.

Results and discussion

In this study, under different operating conditions, a variation of COP, entrainment ratio and exergy analysis of a bus air conditioning system that has two separate evaporators using two-phase ejector are investigated. To evaluate thermodynamic properties and solve the equations Klein [26] is used. Air conditioning systems with the cooling capacity of 14 kW are widely used on midi-buses. The ambient temperature can be assumed to be 35 °C for the design condition of the bus air conditioning system for the Mediterranean climate zone [27]. For practical applications, evaporator temperatures were taken as 12 and 5 as the primary and secondary evaporators, respectively [18]. In the calculations, nozzle, mixing, and diffuser section efficiencies were taken as 0.9, 0.8 and, 0.9, respectively, [28].

The model to be used in this system was validated by Unal and Yılmaz [18]. In addition, R134a refrigerant system, Unal [29] prepared an experimental set-up for the bus ejector air conditioning system which is depicted in fig. 3. An air conditioner currently used on the midibuses was turned into an ejector air conditioning system. It has 14 kW cooling capacity, and experimental studies were conducted on this system. The range and accuracy of the pressure transmitter are 0-25 bar and ±1.0%, respectively. K-type thermocouples are calibrated to provide an accuracy of ±0.3. According to thermodynamic analysis, the pressure difference between the refrigerant at diffuser outlet and the secondary evaporator outlet value was 93.4 kPa and temperature difference was 4 °C. According to the experimental results, pressure difference was measure around 80 kPa and temperature difference was 3.8 °C after the system reaches steady state. COP increase rates were also calculated 15% and measured 8%. According to these results, it can be said that differences between the theoretical and experimental results were within acceptable limits and therefore, the calculation method for the ejector refrigeration system can be considered as validated.

Figure 3. Ejector refrigeration system for bus

The effect of condensing and evaporation temperatures

Mobile air conditioning systems can operate under wide range ambient conditions. To conclude that reason, the performance of air conditional system should be investigated with a variation of condensing temperature.

Figure 4 shows a variation of COP, COP*, and ω with condensing temperature for three refrigerants. Similar to standard systems, in ejector refrigeration system also behave similar trend. As shown in fig. 4(a), increasing condenser temperature results with decrease in performance. The significant result of the figure that the system which is working with R1234yf, has lower performance than others about 5%. In stark contrast, however, COP increase rate for R1234yf is better. So, potential of the system performance of the R1234yf system can be improved more by using two-phase ejector. As seen in fig. 4(b), entrainment ratio is inversely proportional to COP.

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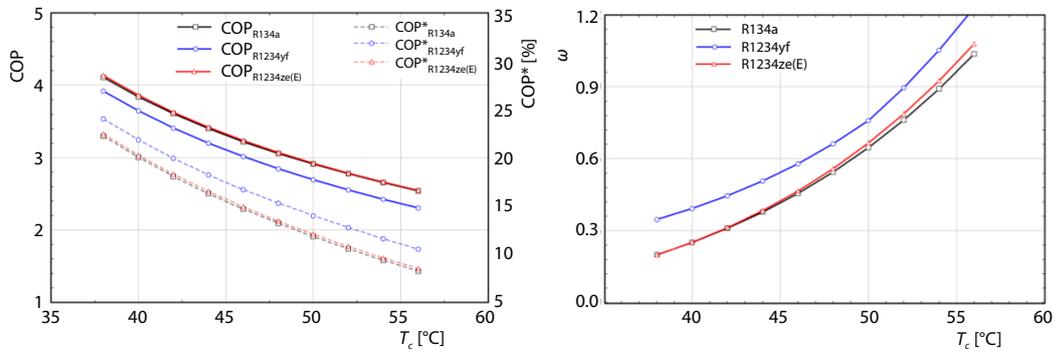
Figure 4. (a) effect of condensing temperature on COP and COP*, (b) effect of condensing temperature on entrainment ratio
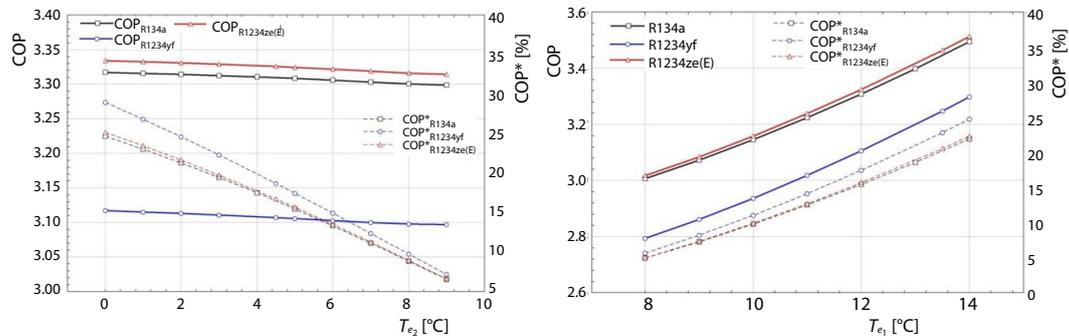
Figure 5. (a) effect of second evaporator temperature on COP and COP*, (b) effect of first evaporator temperature on COP and COP*

The effect of the second evaporator temperature on COP is given in fig. 5(a). COP of the standard system increases with second evaporator temperature, however, the ejector system COP decreases slightly. Since cooling load is kept as constant, COP depend on the compressor work only. When second evaporator temperature gets higher, its saturation pressure increases. In conventional systems, as the second evaporator pressure increases, compression ratio decreases, and therefore COP increases. Second evaporator temperature does not affect compression ratio in ejector refrigeration system. Because of the total refrigerant mass flow rate rises, COP decreases slightly. The COP* decreases because of the conventional system performance increase but ejector system performance is nearly stable. Although R1234yf has lower ejector refrigeration system performance, it has higher COP increase rate potential. The effect of the first evaporator temperature on COP and COP* is given in fig. 5(b). In a conventional refrigeration system, there is no first evaporator, so it does not effect on conventional COP. In ejector refrigeration system, COP depends on the first evaporator pressure and temperature because the compressor is placed on the first evaporator line. So, COP of the ejector refrigeration system increases with first evaporator temperature.

The effect of compressor efficiency

The compressor plays a key role in vapor compression refrigeration system performance. In addition to this, mobile air conditioning system such as bus refrigeration systems can operate under a wide range of ambient conditions. As mentioned above, the compressor driven

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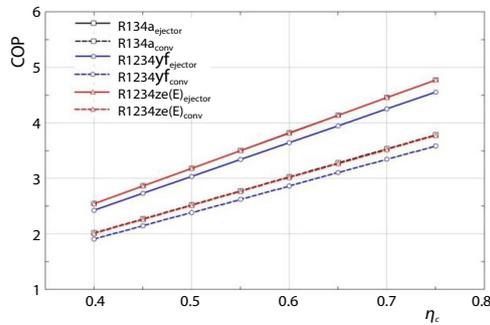
Figure 6. Effect of compressor efficiency on COP

by the engine and engine speed is changed according to road conditions (stopping at stations or traffic lights, shifting gear, *etc.*) as a result of these situations, the speed of air conditioning compressor is changed, the isentropic efficiency of the compressor does not remain constant. Thus, COP of the air conditioning system is affected. In fig. 6, the benefit from compressor isentropic efficiency improvement was the system COP increasing, which resulted from compressor power consumption reduction.

Exergy destruction rates

Exergy calculations are carried out for both conventional and ejector based cycles to have a clear view of losses. Exergy destruction rates are estimated at condensing temperature 45ºC, cooling load is 14 kW, first evaporator and second evaporator temperatures are 12 ºC and 5 ºC, respectively. For the compressor, isentropic efficiency empiric equation in reference [18] is used.

Figure 7(a) shows a comparison of exergy destruction rates in components for the conventional refrigeration system and ejector refrigeration system as an R1234yf refrigerant. Exergy destruction rate in the evaporator and condenser of both the cycles are almost the same for the given operating conditions and cooling capacity. As seen in the figure that the compressor has the biggest exergy destruction for both systems. Since compressor isentropic efficiency is low in bus compressors due to the transmission components, a large amount of energy is lost in the compressor. And ejector system has lower exergy destruction than the conventional system as 3.132 kW and 3.652 kW, respectively. When the cycle is turned into ejector refrigeration system, destruction in compressor and expansion valve decreases because of the falling in compression ratio. So the improvement in COP is achieved with this reduction. Figure 7(b) shows the exergy destruction rates for all components in ejector refrigeration system with substances such as R134a, R1234yf, and R1234ze(E). The biggest exergy destruction occurs in the R1234yf system, it explains why the performance is lower than other refrigerant systems. Total exergy destructions are 2.86 kW, 2.88 kW, and 3.13 kW for R1234ze(E), R134a, and R1234yf, respectively. The important difference of the destruction rate occurs in the compressor, other components cause a similar amount of exergy destruction for all refrigerants.
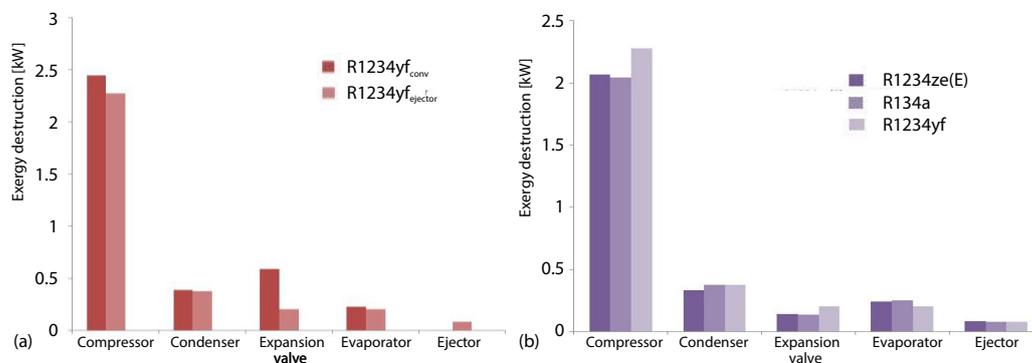
Figure 7. (a) Comparison of exergy destruction rates for conventional and ejector system, (b) exergy destruction rates for all components

Conclusions

In this paper, the thermodynamic and exergetic analysis of low GWP refrigerants R1234yf and R1234ze(E) intercity bus air condition system were analyzed under variable operating conditions. The performance improvement by using ejector was mainly focused on new refrigerants. The following conclusions for theoretical study can be drawn.

- The analysis results revealed that although R1234yf ejector refrigeration system COP was lower by 6% than the R134a ejector system, R1234yf more perfectly meets environmental concerns. That makes it ideal as a replacement, as it may be possible that few or no alterations are required in order to replace R134a with R1234yf in a pre-designed system.
- The highest exergy destruction occurs in the R1234yf system.
- The highest exergy destruction occurs in the compressor for each refrigerant.
- The increase rates in the COP with respect to the conventional system by using ejector as an expansion device are 15% for R134a, about 17% for R1234yf, and about 15% for R1234ze(E), respectively.
- Entrainment ratio is inversely proportional to COP values.

It should be noted that the aim of this study is to investigate the performance improvement potential. Future studies should be performed in order to validate experimentally for alternative refrigerants on real road conditions.

Nomenclature

COP – coefficient of performance, [–]
COP* – increase rate, [%]
$\dot{Ex}$ – exergy, [W]
h – enthalpy [Jkg^{-1}]
$\dot{m}$ – mass flow rate, [kgs^{-1}]
P – pressure, [Nm^{-2}]
$\dot{Q}$ – cooling capacity, [W]
s – entropy, [Jkg^{-1}K^{-1}]
T – temperature, [ºC]
V – refrigerant velocity, [ms^{-1}]
$\dot{W}$ – work [W]

Greek symbols

η – efficiency, [–]
ω – entrainment ratio, [–]

Subscripts

c – condenser
comp – compressor
conv – conventional
d – diffuser
e_1 – first evaporator
e_2 – second evaporator
m – mixing section
n – nozzle
s – isentropic state

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