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A COMPARATIVE ENERGETIC ANALYSIS FOR SOME LOW-GWP REFRIGERANTS AS R134a REPLACEMENTS IN VARIOUS VAPOR COMPRESSION REFRIGERATION SYSTEMS

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Abstract: In this study, the energy parameters (i.e., cooling capacity and COP) were theoretically investigated for three different vapour compression refrigeration system (basic cycle, basic cycle with liquid-to-suction heat exchanger and two-stage cascade cycle) for which R1234yf, R1234ze(E), R513A, R445A and R450A alternative refrigerants with low GWP value were used instead of R134a. The studied refrigerants demonstrated similar thermodynamic behaviour. The exergetic efficiency of the systems was also compared. The comparison of the energy parameters was carried out for two different evaporation temperatures (-30 and 0°C) and two condensing temperatures (40 and 55°C). According to the calculation results, R450A which almost has the same COP values as R134a comes into prominence with 58% lower GWP value compared to R134a. Among the studied refrigeration cycles, system with LSHEX can be suggested for providing a better effect in terms of COP for the considered refrigerants and temperature cases as well as assumed system parameters. It was seen for the investigated cycles that the highest exergetic efficiency could be obtained in the case of R445A.
Keywords: R450A; R513A; R1234yf; R134a; COP; Exergy

FARKLI BUHAR SIKIŞTIRMALI SOĞUTMA SİSTEMLERINDE R134a YERİNE DÜŞÜK GWP DEĞERLİ SOĞUTKANLARIN ENERJI PARAMATERELERİNİN TEORİK MUKAYESESİ

Özet: Bu çalışmada, R134a yerine kullanılan düşük GWP değerine sahip R1234yf, R1234ze(E), R513A, R445A ve R450A soğutucu akışkanlarının, enerji parametreleri (soğutma kapasitesi, COP vb) üç farklı buhar sıkıştırmalı soğutma sistemi (basit çevrim, iç ısı değiştiricili basit çevrim ve iki kademeli kaskad çevrim) için teorik olarak mukayese edilmiştir. Soğutucu akışkanların termodinamik özellikleri benzer davranışlar göstermektedir. Ayrıca sistemlerin ekserji verimleri de karşılaştırılmıştır. Yapılan çalışmada enerji parametreleri, iki farklı evaporatör sıcaklığı (-30 ve 0°C) ve iki farklı kondenser sıcaklığı (40 ve 55°C) için mukayese edilmiştir. Hesaplanan sonuçlara göre R450A, R134a ile hemen hemen aynı COP değerindedir. R450A'nın GWP değeri R134a ile karşılaştırıldığında %58 daha düşüktür. Çalışılan soğutma çevrimleri arasında LSHEX'li çevrim, varsayılan sistem parametreleri, sıcaklıklar ve tüm soğutucu akışkanlar için yüksek COP değerleri açısından önerilebilir. İncelenen çevrimler için en yüksek ekserji verimliliğinin R445A durumunda elde edilebildiği görülmüştür.
Anahtar kelimeler: R450A; R513A; R1234yf; R134a; COP; Ekserji

Nomenclature

COP	coefficient of performance
GWP	global warming potential
h	enthalpy, kJ·kg^{-1}
$\dot{m}$	mass flow rate, kg·s^{-1}
P	pressure, kPa
Q	heat transfer, kW
T	temperature, °C or K
W_{el}	electrical power, kW
η_s	isentropic efficiency
η_{ex}	exergetic efficiency

Subscripts

comp	compressor
con	condenser
ev	evaporator
exp	expansion valve
HP	high pressure
in	inlet
LP	low pressure
out	outlet
Carnot	Carnot cycle

INTRODUCTION

R134a with Ozone Depletion Potential (ODP) of zero value is widely used in many of the refrigeration applications as a substitute for R12 which has a high ODP value [Riffat and Shankland, 1993]. However, it has been noticed in the past time that only ODP is not enough for the selection of refrigerant type. A necessity about using gas types reducing CO_2 emission that contributes the global warming has developed. Using refrigerants with smaller Global Warming Potential (GWP) value as well as zero ODP value has an importance in terms of environmental effect. R134a has a high GWP value such as 1430. According to EU F-Gas regulation, the refrigerants with a GWP value greater than 150 shall not be used since 2011 in the mobile air conditioning (MAC) devices [European Parliament and of the Council Official, 2006]. Instead of R134a, more environmentally sensitive refrigerants will be used by this regulation in the near future [Lee, and Jung, 2012]. In addition, the lower GWP values of the all investigated alternative refrigerants would provide an important contribution for decreasing the amount of CO_2 releasing by 20% with respect to the level of 1990's which is an objective of EU for the year 2020.

Among the refrigeration fluids, the proper alternatives may be R1234yf, R1234ze(E), R513A, R445A or R450A as a substitute for R134a. There is a lack of investigations related to refrigeration units operating with R513A, R445A and R450A which are recently commercialized. The studies on the alternative refrigerants (especially R1234ze(E) and R1234yf) are increasing in recent years [Fukuda et al., 2014; Minor, and Spatz, 2008; Navarro-Esbri et al., 2013; Mota-Babiloni et al., 2015a]. R1234yf and R1234ze(E) which have GWP of only 4 and 1, respectively satisfies EU criteria by this value [European Parliament and the Council, 2014]. Another important characteristic of R1234yf and R1234ze(E) are that they could be directly charged into any refrigeration system constructed for using R134a without making any modification on the system. A restrictive feature of R1234yf and R1234ze(E) are the flammability rate identified as A2L by ASHRAE. R513A and R450A may also be directly used without carrying out some changes on the system, but their possible safety code would be A1 (non-flammable). Another alternative refrigerant which is R445A has an acceptable low GWP and it is expected that its flammable rating would be classified as A2L. Hence, the aforementioned refrigerants can be preferred to use for the suitable places and applications.

Mota-Babiloni et al. [Mota-Babiloni, 2016] reviewed previous studies about systems working with R1234ze(E). The basic result from their research pointed out that some modifications (for example, using internal heat exchanger) in the system should be necessary in order to improve the energy parameters. Nevertheless, CO_2 emission would be reduced because of lower GWP of R1234ze(E) as they emphasized.

R134a and R450A were experimentally compared for a single-stage vapour compression refrigeration system with an internal heat exchanger, IHX [Mota-Babiloni et al., 2015a]. It was determined for various evaporation and condenser temperatures that cooling capacity was seen to decrease by 4 to 8% while COP was increased up to 2%. Furthermore, Mota-Babiloni [Mota-Babiloni et al., 2015b] noted for R1234ze(E), R450A and R134a refrigerants that cooling capacity amounts were enhanced regardless evaporation temperature value by considering the system with IHX compared to that without IHX.

The COP of R445A was higher than that for R134a and R1234yf as determined theoretically by Lee at al for a mobile air conditioning system. [Lee et al., 2015]. Kondou and Koyoma suggested four different heat recovery systems including triple tandem cycle, two-stage extraction cycle, three-stage extraction cycle, and cascade cycle from waste heat [Kondou and Koyama, 2015]. They theoretically examined refrigerants with low GWP which may be alternatives to R134a. The energy parameters were evaluated and discussed considering R1234ze(E). For both refrigeration and air-conditioning systems, Mota-Babiloni et al. [Mota-Babiloni et al., 2015c] theoretically compared low-GWP refrigerants which can be used as alternatives to refrigerants those are restricted with respect to EU regulation 517/2014.

Mota-Babiloni et al. compared the cooling capacity and the COP values of R1234yf and R1234ze(E) refrigerants which are the alternatives for R134a [Mota-Babiloni et al., 2014]. The investigation was carried out for three different evaporation temperatures (260, 270 and 280 K) and condenser temperatures (310, 320 and 330 K). The cooling capacity and COP values of both alternative refrigerants were found to be lower about 9 to 30% and 6%, respectively than those of R134a. Bolaji and Huan theoretically investigated refrigerants with low GWP as substitutes for R134a [Bolaji and Huan, 2014]. The compressor energy consumption of R1234yf was determined to be smaller compared to R134a. Additionally, it has been expressed that the cooling effect of R1234yf was smaller than that of R134a for different evaporation temperatures. Zilio et al. directly used R1234yf (i.e., a drop-in application) as an alternative to R134a and changed expansion valve adjustments. The results were presented for different compressor speed and ambient temperatures [Zilio et al., 2011]. Novarro-Esbri et al. performed an experimental investigation on the direct use of R1234yf in a system operating with R134a [Navarro-Esbri et al., 2013]. They stated that reduction in the cooling capacity of 6% to 13% approximately was noted using R1234yf instead of R134a. Rinne et al. has determined that COP of R513A was smaller than that of R134a by 1.0% [Rinne et al., 2011].

Nawaz et al. modelled R1234yf and R1234ze(E) refrigerants for a domestic type heat pump and compared the results with R134a [Nawaz et al., 2017]. They expressed that HFO based refrigerants can be used as alternatives to R134a and suggested that capacity of compressor should be increased for a low amount of heat load. Devecioğlu and Oruç [Devecioğlu and Oruç, 2017] theoretically compared R1234yf, R445A and R444A for mobile air-conditioning systems [Devecioğlu and Oruç, 2017]. Although cooling capacity of R445A was greater, its COP value was determined to be lower in comparison with R1234yf. Mota-Babiloni et al. [Mota-Babiloni et al., 2017] experimentally studied R513A as a substitute for R134a. They found that both cooling capacity and COP values of R513A were better than that of R134a at different evaporation and condenser temperatures.

Many alternative refrigerants as R134a replacements have been developed and offered to the markets due to GWP, flammability as well as COP considerations. The further investigations with the new refrigerants are strongly necessary for various refrigeration systems and extensive operating conditions. The purpose of the present study is thereby to determine the suitability of low-GWP refrigerants of R1234yf, R1234ze(E), R513A, R445A and R450A by theoretically obtaining parameters such as COP, power consumption and compression ratio in a refrigeration system designed for operating with R134a. The investigation was based on evaporation temperatures of -30 and 0°C as well as condensing temperatures of 40 and 55°C were considered for the computations. The estimated results of energy parameters with a variety of refrigerants for three vapour compression systems as basic cycle, basic cycle with liquid-to-suction heat exchanger, and two-stage cascade cycle systems were then compared and discussed. The theoretical data presents alternative refrigerants (R1234yf, R1234ze(E), R445A, R450A and R513A), which do not deplete the ozone, can directly be preferred as the refrigerants in any refrigeration plant originally constructed to operate with R134a. Theoretical results are probably the most powerful guiding prior to the experimental studies. Hence, the extensive information presented herein may lead to proper selection of cycle type and temperature conditions for alternative refrigerants which could be used as substitutes for R134a in refrigeration systems.

STUDIED REFRIGERANTS

The alternative gases of R1234yf, R1234ze(E), R445A, R450A and R513A may be taken into account as the potential refrigerants those can be used as alternatives to R134a in the direct expansion vapor compression refrigeration systems. They can be utilized for all commercial or domestic applications where R134a is used. Some significant properties of R134a, R1234yf, R450A, R1234ze(E), R445A and R513A considered in the present investigation are listed in Table 1 [Chemours, 2016; Lemmon et al., 2013; Honeywell, 2015a; Honeywell, 2015b; Honeywell, 2014; Chemours, 2018]. Both R445A and R450A are zeotropic refrigerants composed of natural blend of HFC/HFO, however R513A is an azeotropic refrigerant with HFC/HFO mixture.

The information about the composition of mixtures obtained from manufacturer companies is given in Table 2. Note that R134a, R1234yf, and R1234ze(E) are the pure substances. The saturation pressure distributions for the considered refrigerants are depicted in Fig. 1. It can be inferred that although thermodynamic properties of R134a, R1234yf, R450A and R513A refrigerants are similar, the saturation pressure values of both R1234yf and R513A are somewhat greater while that of R450A seems to be smaller compared to R134a for a given temperature. Furthermore R1234ze(E) and R445A demonstrate different behaviour among the considered refrigerants.

Table 2. Mass composition of mixtures

Refrigerant	Composition	Mass concentration (%)
R450A	R134a/R1234ze(E)	42/58
R513A	R134a/R1234yf	44/56
R445A	R134a/R1234ze(E)/R744	9/85/6

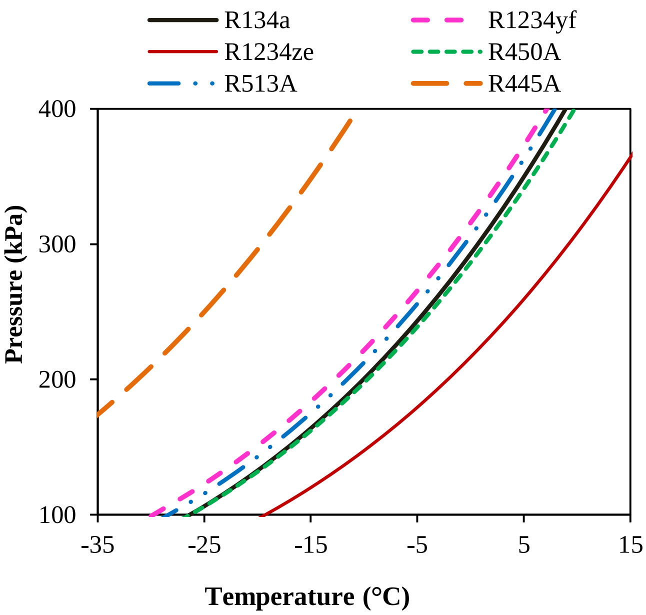


Fig. 1. Variation of saturation pressure with temperature for the studied refrigerants

Table 1. Characteristics of the investigated refrigerants

	Boiling point temperature, (100 kPa) (°C)	Critical temperature (°C)	Critical pressure (kPa)	Temperature glide (°C)	Flammability (ASHRAE classification)	GWP
R134a	-26.07	101	4059	0	A1	1430
R1234yf	-29.45	94.7	3381	0	A2L	4
R1234ze(E)	-18.97	109.4	3635	0	A2L	1
R450A	-25.6	103.1	3971	0.1	A1	547
R513A	-27.9	97.7	3681	0	A1	631
R445A	-23.4	104.7	4497	0	A2L	130

THEORETICAL ANALYSIS

The theoretical study has been carried out using R1234yf, R1234ze(E), R445A, R513A or R450A as alternatives to R134a for three different systems as (i) simple vapour compression refrigeration system, (ii) simple refrigeration with liquid to suction heat exchanger (LSHEX) system, and (iii) two-stage cascade system. The systems with the basic components (compressor, condenser, expansion valve, and evaporator) are schematically shown in Fig. 2 as well the corresponding pressure-enthalpy diagrams. Note that the cascade cycle uses the same refrigerant in both low pressure and high pressure sides.

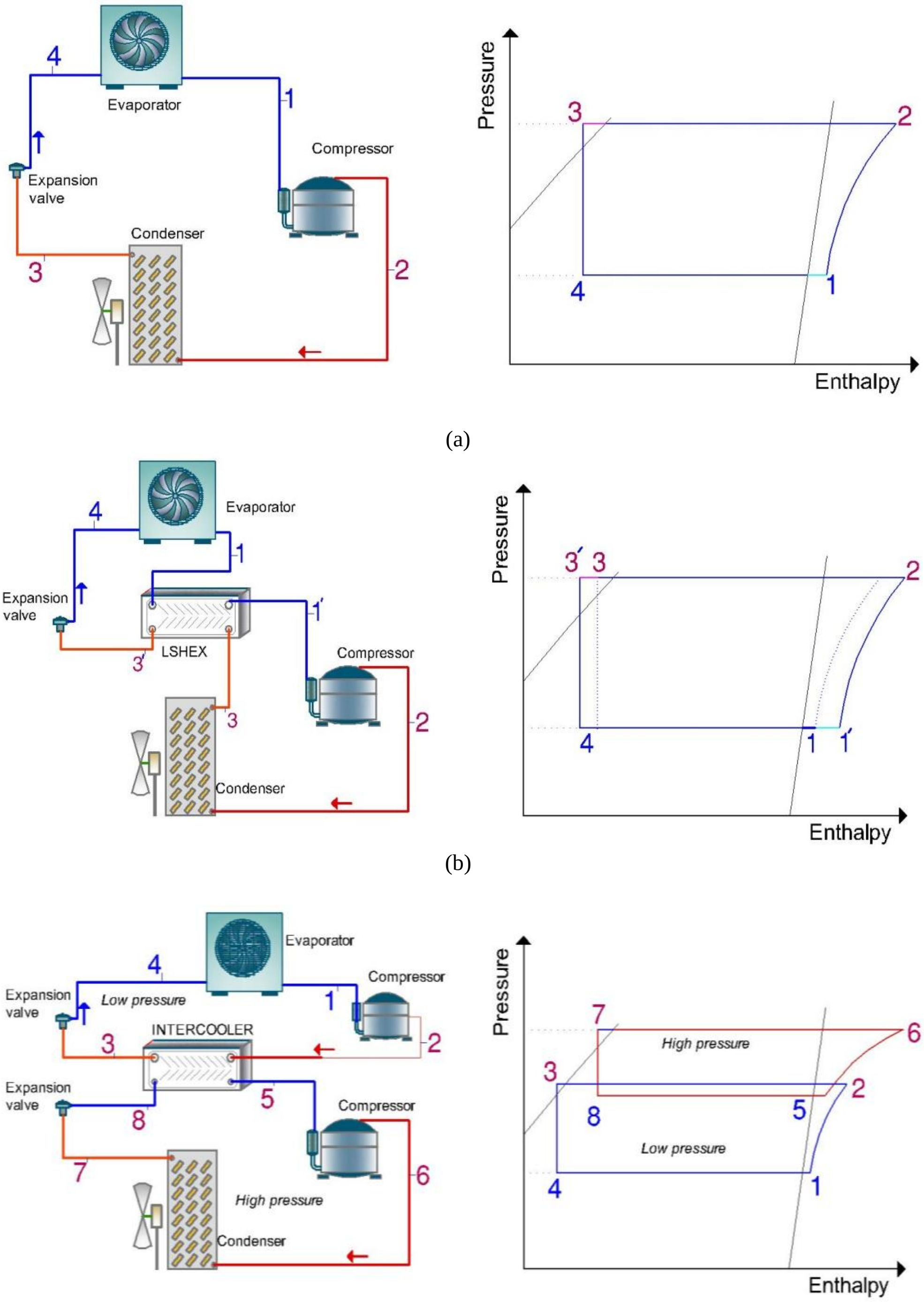


Fig. 2. A schematic representation for the vapour-compression refrigeration cycles a) Basic cycle b) Basic cycle with liquid-to-suction heat exchanger c) Two-stage cascade cycle.

The assumed system parameters for the theoretical analysis are specified in Table 3. Furthermore, it has been supposed that the system operates in a steady-state regime; the heat and pressure loss in the refrigeration system components and pipes is neglected; kinetic and potential energy changes are neglected. The efficiency of LSHEX device in Fig. 2b, $\varepsilon_{\mathrm{LSHEX}}$ is the ratio of actual heat transfer amount to the maximum heat transfer

$$\varepsilon_{\mathrm{LSHEX}} = \frac{T_{\mathrm{suc}} - T_{\mathrm{out,ev}}}{T_{\mathrm{out,con}} - T_{\mathrm{out,ev}}} \quad (1)$$

where T_{suc} is temperature at the suction of compressor, $T_{out,ev}$ and $T_{out,con}$ are the temperature of fluid leaving evaporator and condenser, respectively.

Table 3. Assumed system parameters in the analysis

Evaporation temperature, T_{ev} (°C)	-30, 0
Condensing temperature, T_{con} (°C)	40, 55
Superheat (K)	5
Sub-cooling (K)	3
Isentropic efficiency, η_s (%)	70
Liquid to suction heat exchanger efficiency ε_{LSHEX} (%)	35
Intercooler efficiency (%)	65
Electromechanical and volumetric efficiencies (%)	95

The preferred evaporation temperatures in the study (-30 and 0°C) are commonly used values in practice. These temperature values were investigated since they cover a wide range of application from milk cooler to deep freeze, for example. The ambient temperature is generally 30 to 45°C which led to select condenser temperatures of 40 and 55°C (assuming the difference between the ambient temperature and condenser temperature is 5 to 10°C). It should be remarked that similar evaporation and condenser temperatures were considered in the previous studies [Mota-Babiloni et al., 2014; Moles et al., 2014].

Determination of thermodynamic properties in the cycle

First of all, pressure and temperature values should be fixed in order to find the thermodynamic properties of the points in the considered cycle. A sample analysis is discussed about single-stage vapour compression refrigeration system. The enthalpy across the expansion device is constant such that $h_3 = h_4$. The suction temperature and the liquid temperature values can be obtained through assumed superheat and sub-cooling values. Condensation and evaporation occur at constant pressure. The mass flow rate can be evaluated by taking cooling capacity as a constant value. Liquid temperatures can be regarded as almost the same value on the constant pressure line crossing saturated liquid curve. The points in the cycle can be determined as follows:

Point 1: Corresponds to suction temperature and low pressure values.

Point 2: Enthalpy at the exit of compressor is determined from isentropic efficiency, than point 2 corresponds to the obtained enthalpy and high pressure values.

Point 3: Liquid temperature can be found by subtracting sub-cooling value from the selected condenser temperature. The intersection point of liquid temperature, high-pressure and saturated liquid curve is determined, and then thermodynamic properties at this state refer to point 3.

Point 4: It is the state corresponding to intersection of constant enthalpy ($h_3 = h_4$) and low pressure values.

According to the above information, the energy parameters can be obtained through data corresponding to points of the cycle in Fig. 2. Hence, the cooling capacity, Q_{ev}, is computed as

$$Q_{\mathrm{ev}} = \dot{m}(h_{\mathrm{out,ev}} - h_{\mathrm{in,ev}}) \quad (2)$$

where $\dot{m}$ is the mass flow rate in kg/s, $h_{out,ev}$ and $h_{in,ev}$ are the enthalpy values of the refrigerant at outlet and inlet of the evaporator, respectively in kJ/kg. Energy consumed by the compressor is evaluated as

$$W_{\mathrm{el}} = \dot{m}(h_{\mathrm{out,comp}} - h_{\mathrm{in,comp}}) \quad (3)$$

where W_{el} is the electrical power supplied to the system in kW. Isentropic efficiency, η_s, of the system is determined as

$$\eta_s = \frac{(h'_{\mathrm{out,comp}} - h_{\mathrm{in,comp}})}{(h_{\mathrm{out,comp}} - h_{\mathrm{in,comp}})} \quad (4)$$

where h' corresponds to the enthalpy for the case where entropy remains constant at exit and inlet states of the compressor ($s_2 = s_1$).

The energy performance of any refrigeration or air conditioning system is defined through coefficient of performance (COP) which can be obtained by the ratio of cooling capacity to the supplied electrical power [29]:

$$\mathrm{COP} = \frac{Q_{\mathrm{ev}}}{W_{\mathrm{el}}} \quad (5)$$

In the cascade systems (Fig. 2c), W_{el} is the total power consumed by both compressors which can be determined as

$$W_{\mathrm{el}} = \dot{m}_{\mathrm{LP}} \Delta h_{\mathrm{comp,LP}} + \dot{m}_{\mathrm{HP}} \Delta h_{\mathrm{comp,HP}} \quad (6)$$

where LP and HP represents low pressure and high pressure, respectively while Δh_{comp} is the enthalpy difference between discharge and suction of the compressor. Therefore COP values related to Fig. 2 investigated in this study can be evaluated using Eq.(5) as

basic cycle:

$$\mathrm{COP} = \frac{(h_1 - h_4)}{(h_2 - h_1)} \quad (7)$$

basic cycle with LSHEX:

$$\mathrm{COP} = \frac{(h_1 - h_4)}{(h_2 - h_{1'})} \quad (8)$$

two-stage cascade cycle:

$$\mathrm{COP} = \frac{\dot{m}_{\mathrm{LP}}(h_1 - h_4)}{\dot{m}_{\mathrm{LP}}(h_2 - h_1) + \dot{m}_{\mathrm{HP}}(h_6 - h_5)} \quad (9)$$

Note that the numerical subscripts in Eqs.(7 to 9) refer to the states shown in Fig. 2.
The exergy information is a useful parameter for total irreversibility distribution in a refrigeration cycle components. The exergetic efficiency, η_{ex} of a system is defined as [Dinçer and Kanoğlu, 2010]

$$\eta_{\mathrm{ex}} = \frac{\mathrm{COP}}{\mathrm{COP}_{\mathrm{Carnot}}} \quad (10)$$

where $\mathrm{COP}_{\mathrm{Carnot}}$ is defined as follows:

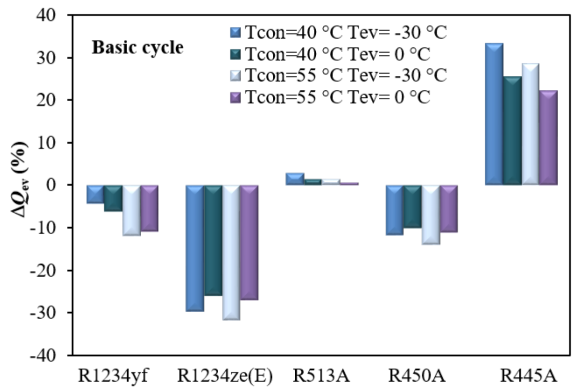
$$\mathrm{COP}_{\mathrm{Carnot}} = \frac{T_L}{T_H - T_L} \quad (11)$$

where T_L is temperature of refrigerated space, T_H is ambient temperature where heat rejection occurs. In the calculations, T_L temperatures are –20°C and 10°C for T_{ev} values of –30°C and 0°C, respectively. Similarly, T_H temperatures are 40°C and 25°C for T_{con} values of 55°C and 40°C, respectively.

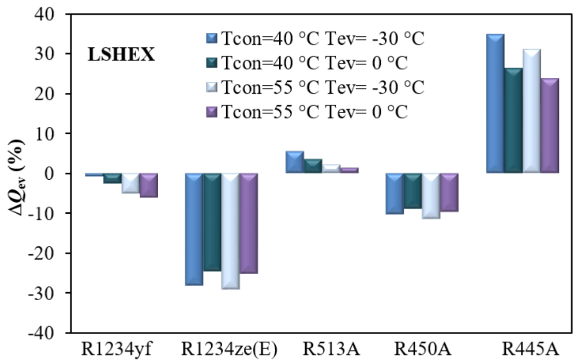
RESULTS AND DISCUSSION

The cooling capacity amounts for the investigated cases were calculated using Eq.(2). The change in cooling capacity (ΔQ_{ev}) values for alternative refrigerants under different T_{ev} and T_{con} cases were then determined compared to R134a. The obtained results are plotted as in Fig. 3. First of all, basic cycle is considered in Fig. 3a which shows that there is a reduction in ΔQ_{ev} for R1234yf, R1234ze(E) and R450A. Likewise, Moles et al. [15], theoretically found for basic cycle that Q_{ev} magnitudes of R1234yf and R1234ze(E) were smaller than that R134a as detected in Fig. 3. The decrease rate in ΔQ_{ev} for R1234yf and R450A is about in the same order which is –10%, approximately regardless of temperature. The higher decrease in ΔQ_{ev} for R1234ze(E) can be easily noted up to -30%. However there is an enhancement in ΔQ_{ev} for R513A and especially R445A which leads to an improvement with almost 30%. Furthermore, it seems that T_{ev}=-30°C and T_{con}=55°C case causes the greatest reduction in ΔQ_{ev} independent of refrigerant type while T_{ev}=-30°C and T_{con}=40°C case should be better in terms of improvement for ΔQ_{ev} for the basic cycle (R513A and R445A).

The system with LSHEX and cascade system as in Figs. 3(b,c) presented similar behaviour to that expressed for Fig. 3a. Namely, the reduction (for R1234yf, R1234ze(E) and R450A) or increase (for R513A and R445A) in ΔQ_{ev} develops. However, LSHEX and cascade systems give better results compared with basic cycle, in other words smaller decrease in ΔQ_{ev} is developed. For instance, reductions in ΔQ_{ev} for R450A with T_{ev}=-30°C and T_{con}=40°C are -11.7%, -10.4% and -9.5% for basic cycle, LSHEX system and cascade cycle, respectively as seen in Fig. 3. It can also be noted from Fig. 3 that Q_{ev} is improved using LSHEX system and cascade cycle instead of basic cycle such that ΔQ_{ev} amounts for R513A at T_{ev}=-30°C and T_{con}=40°C are 2.9%, 5.4%, and 8.4% considering basic cycle, LSHEX system and cascade cycle, respectively.



(a)



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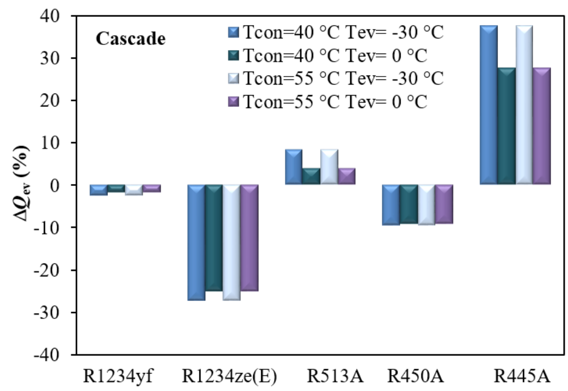


(c)

Fig. 3. Comparison on the change in cooling capacity for the alternative refrigerants with respect to R134a (a) Basic cycle (b) System with LSHEX (c) Cascade cycle.

Although Q_{ev} is a significant parameter in refrigeration applications, it does not present overall physical behaviour alone. The more meaningful descriptions should be expressed in terms of COP determined by Eq.

(5) for general refrigeration systems or through Eqs. (7 to 9) for special cases as discussed herein. As a result the variations of COP with studied temperature cases are provided in Fig. 4 for each cycle type. First of all the best case regarding the effect of temperature occurs clearly for T_{ev}=0°C and T_{con}=40°C whatever the refrigerant kind. It is useful to note that the influence of temperature was not obvious in Fig. 3 since it created different behaviour as refrigerant type varied. However, Fig. 4 points out evidently that the highest COP could be obtained for T_{ev}=0°C and T_{con}=40°C while the lowest COP is related to T_{ev}=-30°C and T_{con}=55°C independent of refrigerant type. In fact, this may already be evaluated as a reasonable behaviour due to the fact that both higher T_{ev} and the smaller T_{con} give rise to an expected enhancement in COP.

The other interesting issue observed in Fig. 4 is the effect of cycle type on COP such that the best situation is the cycle with LSHEX for all studied refrigerants and all covered T_{ev} and the smaller T_{con} cases. For example, if R1234yf is used COP values can be obtained as 2.06, 2.27, 2.46 for cascade system, basic cycle and LSHEX system, respectively at T_{ev}=0°C and T_{con}=55°C. On the other hand, basic cycle seems to give higher COP with respect to cascade cycle for all refrigerants and studied temperature cases except for T_{ev}=-30°C and T_{con}=55°C in which COP is slightly greater for cascade cycle, for instance, COP values can be determined due to basic cycle, cascade cycle and system with LSHEX as 1.01, 1.10, 1.13, respectively for the mentioned temperature and R1234yf case. The cascade cycle could give better results if operated with a larger temperature differential between evaporation and condensation. This is the great advantage of the cascade.

The proper refrigerant type can be determined by carefully observing Fig. 4. Since the highest COP case was found for T_{ev}=0°C and T_{con}=40°C, this case with LSHEX system was focused for the comparison among the refrigerants. Accordingly, COP values are determined from Fig. 4 for the discussed conditions of R134a, R1234yf, R1234ze(E), R513A, R450A and R445A as 3.93, 3.86, 3.90, 3.89, 3.92, and 4.19, respectively. Similar behaviours can evidently be figured out for the remaining temperature cases and cycle types in Fig. 4. Hence, R445A with LSHEX system obviously presents the highest COP case among the investigated refrigerants. Note also that COP values of R1234ze(E) and R450A are very close to that of R134a. These may therefore be considered as good alternative refrigerants for the refrigeration systems.

The cooling capacity amounts for R450A were smaller than that for R134a as observed in Fig. 3. This result agrees with the behaviour obtained by Mota-Babiloni et al. [Mota-Babiloni et al., 2015a]. On the other hand, they determined higher COP about 2% for R450A. Note that COP values of R450A for basic cycle and LSHEX system are mildly lower than that of R134a contrarily to the experimental measurements [Mota-Babiloni et al., 2015a]. The mentioned difference between present theoretical investigation and experimental study [Mota-Babiloni et al., 2015a] may be acceptable due to loss of energy which cannot be eliminated in the real conditions. Similar to distribution plotted in Fig. 4, Lee et al. [Lee et al., 2015] also determined greater COP of R445A compared with R134a. Furthermore, Mota-Babiloni et al. [Mota-Babiloni et al., 2015c] theoretically studied R450A, R513A, and R445A refrigerants as alternatives for R134a and their results were compatible with the present investigation.

The distribution of the change in mass flow rate, $\Delta\dot{m}$ for alternative refrigerants compared to R134a (T_{ev}=-30°C and T_{con}=40°C) is presented in Fig. 5. Not only refrigerant type but also cycle type clearly affects $\Delta\dot{m}$. Furthermore, mass flow rate decreases as a result of using all alternative refrigerants for low pressure side of cascade system. Additionally, the greatest reduction in $\dot{m}$ occurs for R1234ze(E) with high-pressure side of cascade system in which $\Delta\dot{m}$ is about -70% while the biggest increase in mass flow rate with respect to R134a occurs as about $\Delta\dot{m}$ =125% for R1234yf with the same system type. Another point in Fig. 5 is related to the similar behaviour of basic cycle and LSHEX system, because they have almost same $\Delta\dot{m}$ amount regardless of kind of alternative refrigerant. Finally, R1234ze(E) and R450A always creates a reduction in $\Delta\dot{m}$ whatever the cycle type, but the higher decrease in $\Delta\dot{m}$ is noted for cascade system of these refrigerants compared to basic cycle and system with LSHEX.

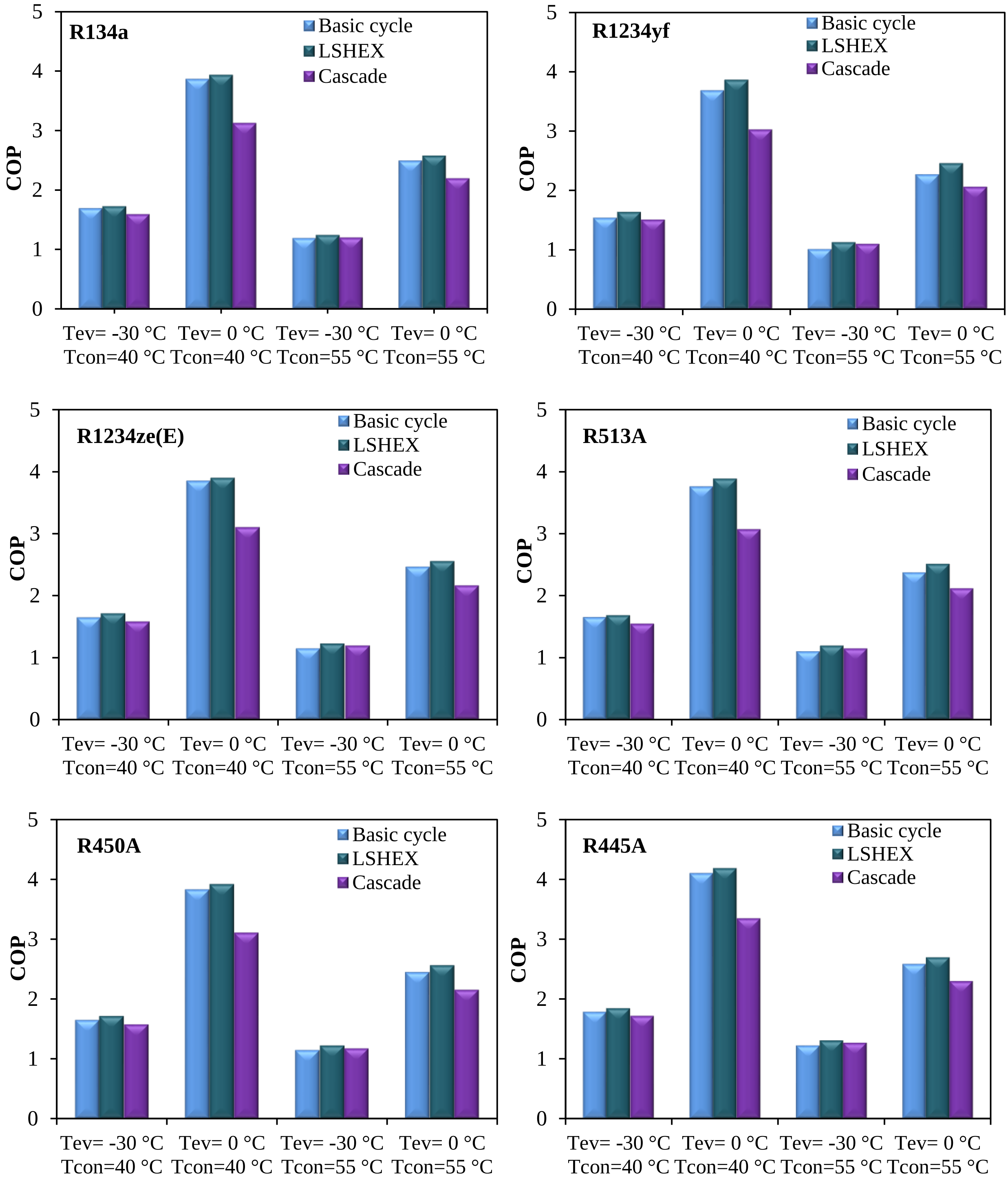


Fig. 4. The effect of cycle type on COP for the investigated refrigerants with different T_{ev} and T_{con} cases.

Investigating only energy parameters for the cycles may not be enough in order to select suitable refrigerant type. Therefore, exergetic efficiency information should be additional parameter for this aim. COP_{Carnot} is depended only on source temperatures as defined in Eq.(10), it remains constant for all covered cases depending on T_{con} and T_{ev} values. The second-law efficiency of refrigerants for each cycle is presented in Fig. 6. At T_{ev}= –30°C exergetic efficiency, η_{ex} is reduced with increasing T_{con}. On the other hand, η_{ex} is augmented as T_{con} increases for T_{ev}= 0°C. This distribution develops due to COP_{Carnot} values such that increase in COP_{Carnot} and COP are 50% and 35%, respectively at T_{ev}= 0°C. However, enhancement in COP_{Carnot} and COP are 25% and 30%, respectively at T_{ev}= –30°C. This condition is same for all three cycles. The highest η_{ex} was found for R445A while the lowest η_{ex} was obtained for R1234yf case. Additionally, for a given refrigerant type, the maximum η_{ex} has been determined for a cycle with LSHEX.

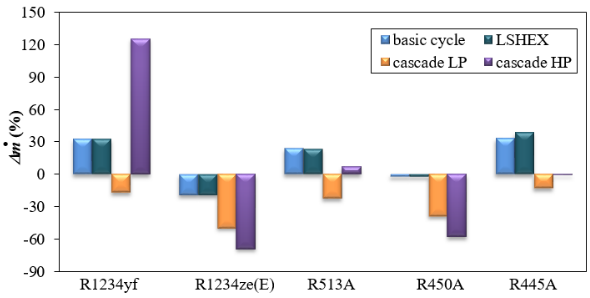
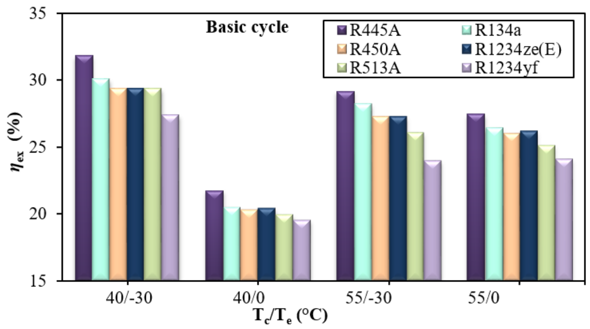
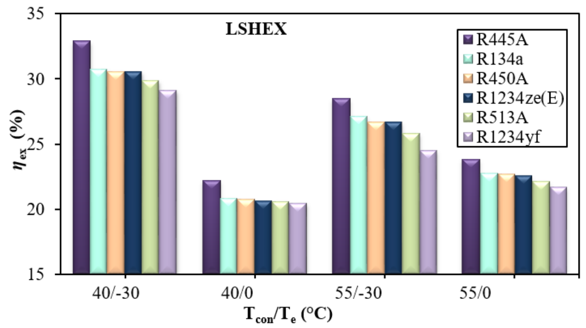


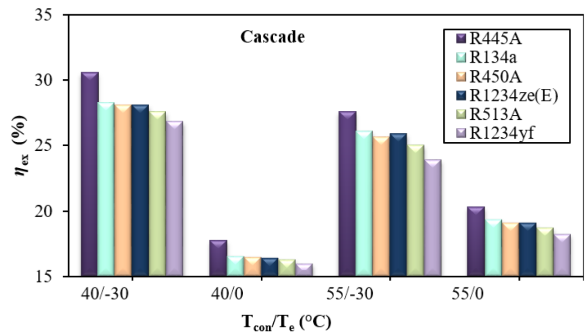
Fig.5. The change in mass flow rate for the refrigerants compared to R134a with studied refrigeration systems at T_{ev}= −30°C and T_{con}=40°



(a)



(b)



(c)

Fig. 6. The exergetic efficiency versus T_{con} /T_{ev} for refrigerants (a) Basic cycle (b) LSHEX (c) Cascade

Generally, the researchers focused on basic cycle systems in previous theoretical investigations. In the conducted studies, different refrigerants and temperature cases were investigated according to principle aim of refrigerant utilization. The refrigerants with low-GWP and providing suitable energy performance should be identified as alternatives to refrigerants those will be phased-out shortly. This study point out both suitable refrigerant type and temperature range for a cascade cycle with LSHEX used to improve energy performances of the system. The present theoretical study has significance since it may be a useful guide prior to experimental investigations. R1234yf, R1234ze(E), R513A, R445A and R450A,which may be used as alternatives to R134a, were compared for different refrigeration cycles and several cases of T_{con} and T_{ev}. The most proper refrigerant and cycle type can be effectively determined through second-law efficiency of the refrigerants.

CONCLUSION

The finding that COP values among the alternative gases for R450A is closer to that of R134a verifying the suitability of using R450A in the refrigeration systems. It was seen, for example, regarding the basic cycle with T_{ev}=0°C and T_{con}=40°C case that COP values have been obtained as 3.86, 3.68, 3.85, 3.76, 3.83, and 4.10 for R134a, R1234yf, R1234ze(E), R513A, R450A and R445A, respectively. Namely there is an improvement in COP about 6.2% when R445A is used as a substitute for R134a. Alternatively R450A, which has only a lower COP by 0.8% compared to R134a, may be more safely preferred. Since R450A and R513A exist in the non-flammable gas class, they could be preferred rather than R1234ze(E), R1234yf, and R445A for some restrictive applications. Nevertheless, R445 is a best alternative refrigerant to R134a for any system as long as flammability is not a major risk for a specific application. Furthermore, it would be suitable in the medium term that the refrigerants with higher GWP should be replaced by the alternative refrigerants with lower GWP values and better energy parameters without making so much constructional modification on the refrigeration system. The present study should be useful in terms of providing original results on using new generation low-GWP refrigerants, including particularly R450A, R445A, and R513A in the refrigeration plants as alternatives to R134a. Additionally, LSHEX system should be selected instead of basic cycle for any refrigerant to improve the efficiency of refrigeration units. However, due to lower magnitudes of COP, cascade cycle may not be a good alternative to basic cycle with respect to the assumptions and cases covered herein. The results of this paper should be useful since includes a wider range of T_{ev} and T_{con} as well as five alternative refrigerants to R134a and three different cycle types. The experimental investigations on the subject should be carried out by testing proper expansion valve type as well to determine whether COP values could be enhanced. The highest η_{ex} was noted for a system with LSHEX. Similarly, R445A provided maximum amount of η_{ex} among the studied refrigerants. If the difference between T_{con} and T_{ev} was high, it was noted also that exergetic efficiency of the system was enhanced using cascade cycle.

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