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THEORETICAL ANALYSIS ON THE PREDICTION OF PERFORMANCE COEFFICIENT OF TWO-STAGE CASCADE REFRIGERATION SYSTEM USING VARIOUS ALTERNATIVE REFRIGERANTS

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Abstract: This paper presents a comparison of the refrigerants CFC-12, HCFC-22, CFC-502, and their alternatives, such as HFC-134a, HFC-152a, HFC-404A, HFC-407C, HC-290, HC-600a, R717 (ammonia), and three mixtures composed of HFC-134a, HFC-152a, HC-600a, and HC-290. A theoretical performance study on a cascade refrigeration system was performed using two refrigeration cycles connected through a heat exchanger in the middle working as an evaporator for the high-pressure cycle and a condenser for the low-pressure cycle. The condensing temperatures varied between 40 and 60 °C, the evaporating temperatures varied between -50 and -10 °C, and the heat exchanger temperature was kept constant at 1 °C. The refrigerants in both cycles are assumed to be same at first. The alternative refrigerants of HFC-152a and R717 had higher coefficients of performance (COPs) than other refrigerants and also low refrigerant charge rates for all operating conditions in the case study, according to the results of the theoretical analysis. The effects of the investigated refrigerants on the depletion of the ozone layer, increase in global warming, and flammable and toxic characteristics were considered and alternative refrigerant mixtures that may be suggested were determined. It was found that refrigerant blends of HC290/HC600a (55/45 by wt%), as a non-azeotropic mixture, and HFC-152a/HFC-134a (14/86 by wt%) and HFC-134a/HC600a (82/18 by wt%), as azeotropic mixtures, gave lower performance coefficients (COPs) and required lower refrigerant charge rates than their base pure refrigerants in the analysis. The separate usage of the refrigerants R717 and HFC-152a for the high- and low-pressure sections of the cascade system, respectively, were found to be the best combination in terms of the analysis for the determination of coefficients of performance (COP) and refrigerant charge rates. The results of this study are of technological importance for efficient design when the industrial systems are assigned to utilize various alternative refrigerants.
Keywords: Refrigeration, Cascade system, R290, R600a, R152a, R717.

ÇEŞİTLİ ALTERNATİF SOĞUTKANLAR KULLANILARAK İKİ KADEMELİ KASKAD SOĞUTMA SİSTEMİNİN PERFORMANS KATSAYISININ TAHMİNİ HAKKINDA TEORİK ÇALIŞMA

Özet: Bu çalışmada CFC-12, HCFC-22, CFC-502 soğutkanları ve HFC-134a, HFC-152a, HFC-404A, HFC-407C, HC-290, HC-600a, R717 (amonyak) gibi alternatifleri ile üç adet HFC-134a, HFC-152a, HC-600a, HC-290'dan oluşan karışım soğutkanların karşılaştırılması sunulmuştur.Bir kaskad soğutma sistemi hakkındaki teorik çalışma bir adet ısı değiştirici ile bağlantılı iki adet soğutma çevrimi kullanılarak yapılmıştır. Isı değiştirici yüksek basınç çevriminde buharlaştırıcı olarak, düşük basınç çevriminde yoğuşturucu olarak çalışmaktadır. Yoğuşma sıcaklıkları 40°C ve 60 °C, buharlaştırıcı sıcaklıkları -50 °C ve -10 °C arasında değişmekte olup ısı değiştirici sıcaklığı 1 °C sıcaklıkta sabit tutulmuştur. Başlangıçta her iki çevrimdeki akışkanlar aynı kabul edilmiştir. Teorik analizin sonucuna gore HFC-152a ve R717 alternatif soğutkanları diğer soğutkanlardan daha yüksek performans katsayısına (COP) sahip oldukları ve daha düşük şarj miktarları gerektirdikleri örnek çalışmada incelenen tüm çalışma şartlarında bulunmuştur. Bazi soğutkanların ozon tabakasının bozulmasına olan etkileri, küresel ısınmayı artıcı etkileri, alevlenme ve zehirleme karakteristikleri göz önüne alınarak alternatif soğutkan karışımları önerilmiştir. Analizlerde azeotrop olmayan HC290/HC600a (55/45 %) karşımının, azeotrop olan HFC-152a/HFC-134a (14/86 %) ve HFC-134a/HC600a (82/18 %) karışımlarının kendi baz soğutkanlarından daha düşük performans katsayıları verdikleri ve daha düşük şarj miktarlarına gereksinim duydukları bulunmuştur. Kaskad sisteminin yüksek basınç bölgesinde R717 ve alçak basınç bölgesinde HFC-152a soğutkanlarının ayrı kullanımlarının performans katsayısı ve soğutkan şarj miktarının bulunmasıyla ilgili olarak en iyi kombinasyon olduğu bulunmuştur. Endüstriyel sistemlerin alternatif soğutkanlardan yararlanarak çalıştırılması durumunda çalışmanın sonuçları teknolojik öneme sahiptir.
Anahtar Kelimeler: Soğutma, Kaskad sistem, R290, R600a, R152a, R717.

INTRODUCTION

Recently, the ozone-depleting potential (ODP) and global-warming potential (GWP) have become the most important criteria in the development of new refrigerants apart from the refrigerant CFCs and HCFCs, both of which have high ODP and GWP due to the their contribution to ozone layer depletion and global warming. In spite of their high GWP, alternatives to CFCs and HCFCs, such as hydro fluorocarbon (HFC) refrigerants, with their zero ODP, have been preferred for use in many industrial and domestic applications intensively for a decade. HFC refrigerants also have suitable specifications such as non-flammability, stability, and similar vapor pressure to CFCs and HCFCs. The problems of the depletion of the ozone layer and the increase in global warming have caused scientists to investigate more environmentally friendly refrigerants than HFC refrigerants for the protection of the environment, such as the hydrocarbon (HC) refrigerants of propane, isobutene, n-butane, or hydrocarbon mixtures as working fluids in refrigeration and air-conditioning systems. Although HC refrigerants have highly flammable characteristics (A3), according to the standards of ASHRAE, as a negative specification, they not only have several preferable specifications such as zero ozone depletion potential, very low global warming, non-toxicity, and higher performance than other refrigerant types but also high miscibility with mineral oil and good compatibility with existing refrigerating systems. Ammonia (R717) has been used for all new land-based systems, often in combination with secondary refrigerants such as water-based mixtures of propylene glycol or ethylene glycol and CO2 in cascade systems. It is a working fluid in the high-temperature sides of the systems, and it has toxicity, zero ODP, and zero GWP. Moreover, the toxicity specification of R717 makes it unsuitable for domestic use. These kinds of refrigerants are used in as many applications, with attention being paid to safety of leakage from the system, as other refrigerants in recent years.

Many investigations have been conducted regarding substitutes for CFC12 and CFC22. Chlorine atoms must not be contained in the chemical structure of candidate refrigerants due to their harmful effect of ozone layer depletion and global warming. Since the discovery of the depletion of the earth's ozone layer and as a result of the 1992 United Nations Environment Program meeting, the phase-out of CFC-11 and CFC-12, used mainly in conventional refrigeration and air-conditioning equipment, was expected by 1996, and that of HCFCs is expected after 2030 (Lee and Su, 2002). HFC134a and HFC152a were proposed instead of CFC12 by the American Household Appliances Manufacturers (AHAM) (Devotta et al., 1993) as alternative refrigerants in domestic refrigeration due to their specifications of non-toxicity, flammability classes of A1 for R134a and A2 for R152a, and better ODP and GWP. Commercial refrigerant propane/isobutane blends have begun to be used in small systems as substitutes for CFC12 in recent years. Moreover, R407C is accepted as a short-term replacement for R22 due to the fact that it has similar properties to R22.

Wongwises et al. (2006) presented an experimental study on the application of hydrocarbon mixtures to replace HFC-134a in automotive air conditioners. The hydrocarbons investigated are propane (R290), butane (R600), and isobutane (R600a). The measured data are obtained from an automotive air-conditioning test facility utilizing HFC-134a as the refrigerant. Wongwises and Chimres (2005) presented an experimental study on the application of a mixture of propane, butane, and isobutene to replace HFC-134a in a domestic refrigerator. The results showed that a 60%/40% propane/butane mixture was the most appropriate alternative refrigerant. Alsaad and Hammad (1999) investigated the performance of a domestic refrigerator using LPG (24.4% propane, 56.4% butane, and 17.2% isobutane), which is available locally in many countries, is cheap, and possesses an environmentally friendly nature with no ozone depletion potential (ODP), as an alternative refrigerant to CFC12. Jung et al. (1996) used a propane/isobutane (R290/R600a) mixture to determine its performance in domestic refrigerators. According to their thermodynamic cycle analysis, the propane/isobutane blend in the composition range of 0.2 to 0.6 mass fraction of propane yields an increase in the coefficient of performance (COP) of up to 2.3% compared to CFC12. Granryd (2001) mentioned the possibilities and problems of using hydrocarbons as working fluids in refrigeration equipment. In spite of their flammability specifications, it was shown in his paper that hydrocarbons propose interesting refrigerant alternatives for energy-efficient and environmentally friendly refrigerating equipment and heat pumps as a result of his study. Han et al. (2007) experimentally studied a new hydrocarbon refrigerant mixture instead of R407C for vapor-compression refrigeration systems. As a result of the experimental and theoretical analysis, their new ternary non-azeotropic mixture of R32/R125/R161, whose ODP and GWP is zero and lower than R407C, respectively, showed better refrigerating capacity and coefficient of performance (COP) than R407C. Park et al. (2007) tested two pure hydrocarbons and seven mixtures composed of propylene, propane, HFC152a, and dimethylether as alternatives to HCFC22 in residential air conditioners and heat pumps. Their experimental results showed that the coefficients of performances (COPs) of these mixtures are up to 5.7% higher than that of HCFC22. Mani and Selladurai (2008) performed experiments using a vapor compression refrigeration system with the new R290/R600a refrigerant mixture as a substitute refrigerant for CFC12 and HFC134a. According to the results of their experiments, the refrigerant R290/R600a has a refrigerating capacity that is 19.9% to 50.1%

higher than that of R12 and 28.6% to 87.2% higher than that of R134a. The R290/R600a blend's performance coefficient (COP) was improved by 3.9-25.1% compared to that of R12 at lower evaporating temperatures and by 11.8-17.6% at higher evaporating temperatures. The refrigerant R134a had a slightly lower coefficient of performance (COP) than R12. Chen and Yu (2008) presented a new refrigeration cycle and introduced an alternative refrigeration cycle applied in residential air conditioners, using the binary non-azeotropic refrigerant mixture R32/R134a. The comparison between the conventional cycle configuration and the new one showed that the coefficient of performance (COP) increased by 8% to 9% compared to the conventional cycle configuration, and the volumetric refrigerating capacity increased by approximately 9.5%.

Dopazo et al. (2009) investigated a cascade refrigeration system with CO2 and NH3 as working fluids in the low- and high-temperature stages, respectively. Determinations of COP and exergetic efficiency and optimization studies based on the optimum CO2 condensing temperature have been performed. Getu and Bansal (2008) presented a thermodynamic analysis of a carbon dioxide-ammonia (R744-R717) cascade refrigeration system to optimize the design and operating parameters of the system. They showed that the best coefficient of performance (COP) order of some alternative refrigerants, such as ethanol, R717, R290, R1270, and R404a, changes with the alteration of the subcooling or superheating degree. Kilicarslan (2004) performed an experimental and theoretical study of a different type of a two-stage vapor compression cascade refrigeration system using R-134. As a result of the analysis, the coefficient of performance (COP) was found to be mainly a function of evaporator temperature and pressure. Lee et al. (2006) used carbon dioxide and ammonia in a cascade refrigeration system to determine the optimal condensing temperature of the cascade condenser given various design parameters to maximize the COP and minimize the exergy destruction of the system. In their study, it was found that the maximum coefficient of performance (COP) increased with evaporating temperature but decreased as condensing temperature or temperature difference increased. Hoşöz (2005) presented an experimental comparison of single-stage and cascade vapor-compression refrigeration systems using R134a as the refrigerant. According to his results, the cascade system provides a lower evaporating temperature, lower compressor discharge temperature, lower ratio of discharge to suction pressures, and higher compressor volumetric efficiency at the expense of a lower coefficient performance (COP) for a given refrigeration capacity. Kanoğlu (2002) performed an exergy analysis of the multistage cascade refrigeration cycle used for natural gas liquefaction. The equations of exergy destruction and exergetic efficiency for the main cycle components, such as evaporators, condensers, compressors, and expansion valves, are developed in his study.

Table 1. Some physical properties of blend refrigerants used in the analysis (REFPROP, 2001).

Ref.	**Rate % by wt.**	**P (MPa)**	**T_l (°C)**	**T_g (°C)**	**ρ_l (kg m^{-3})**	**ρ_g (kg m^{-3})**	**h_l (kJ kg^{-1})**	**h_g kJ kg^{-1})**
R134a/R600a	82/18	0.8	29.19	29.36	949.67	34.83	246.23	431.49
R134a/R600a	82/18	1	37.46	37.69	922.14	43.97	260.07	436.20
R134a/R600a	82/18	1.2	44.58	44.85	896.90	53.48	272.31	439.91
R134a/R600a	82/18	1.4	50.86	51.16	873.16	63.46	283.41	442.85
R134a/R600a	82/18	1.6	56.5	56.83	850.37	73.96	293.66	445.16
R152a/R134a	14/86	0.8	32.30	32.39	1122.3	36.13	246.77	431.87
R152a/R134a	14/86	1	40.42	40.50	1091.2	45.57	259.19	435.63
R152a/R134a	14/86	1.2	47.39	47.47	1062.8	55.36	270.14	438.54
R152a/R134a	14/86	1.4	53.54	53.62	1036.1	65.57	280.05	440.80
R152a/R134a	14/86	1.6	59.06	59.14	1010.5	76.28	289.18	442.54
R290/R600a	55/45	0.8	30.43	37.53	510.16	18.2	276.69	615.33
R290/R600a	55/45	1	39.65	46.45	496.28	22.84	301.38	625.32
R290/R600a	55/45	1.2	47.61	54.14	483.63	27.65	323.34	633.47
R290/R600a	55/45	1.4	54.67	60.92	471.81	32.63	343.34	640.22
R290/R600a	55/45	1.6	61.03	67.02	460.56	37.84	361.87	645.83

In this paper, a two-stage cascade refrigeration cycle for some well-known pure refrigerants, binary non-azeotropic and azeotropic blend refrigerants, is analyzed to reach better coefficients of performance (COP) and to reduce the refrigerant charge rate. Theoretical investigations were performed to reach this aim using the REFPROP program (2001). The coefficients of performance (COP) of nine pure refrigerants and seven commercially available/custom-made blend refrigerants, as shown in Table 1, containing R290, R600a, R152a, and R134a in various concentrations were calculated to examine the possibility of substitution for the conventional refrigerants of R12, R22, and R502. The specifications of zero ozone-depleting potential (ODP) and quite low global-warming potential (GWP) makes most of these refrigerants suitable candidates. Comparisons are made to identify high-performance refrigerants for various evaporating temperatures ranging from -50 to -10 °C, condensing temperatures ranging from 40 to 60 °C, and a constant heat exchange temperature of 1 °C in a cascade system. In addition to these, sample case studies for the specified operating conditions are given in Tables 2 and 3 using the high-performance refrigerants and blend refrigerants identified in the analysis. The effect of the refrigerant alteration in the low- and high-temperature cycles of the cascade system and changes in evaporating and condensing temperatures with their pressures for all investigated refrigerants and their blends are also discussed in the paper.

THEORETICAL COMPUTATION MODEL

Cycle performance determination is performed to ease the theoretical calculations by means of some assumptions as follows: steady-state operating conditions, negligible alterations of kinetic and potential, adiabatic (well-insulated) heat exchanger, saturated liquid at the outlet of the condenser, saturated vapor at the inlet of the compressor, neglect of the pressure drops and the heat losses to the environment from the evaporators and condensers (see Figure 1), the isenthalpic flow across the expansion valves, and the considered isentropic efficiency for the compressor.

The related physical properties of the refrigerant cycle states, in accordance with the various refrigerants and their mixtures (see Table 1) are determined using REFPROP 7 (2001). When the thermodynamic properties of each state of the cycle are determined, the equations for the cycle analysis can be obtained by means of mass and energy conservation.

A large temperature range between the evaporating and condensing temperatures causes the usage of stages in the refrigeration system. This aim can be achieved by cascade refrigeration systems. A two-stage cascade refrigeration system, shown in Figure 1, works between the varied condensing and evaporating temperature limits and at a constant heat exchanger temperature. The high- and low-pressure parts of the cycle are indicated as “A” and “B” in Figure 1a, respectively. The refrigerants in the “A” and “B” cycles may be different from each other and cannot be mixed in the heat exchanger. It should be noted that the working fluid of the lower cycle is at a higher pressure and temperature in the heat exchanger for effective heat transfer in practice. Moreover, there are two different refrigerants operating in two individual cycles in the system (see Figs. 1b and 1c). They are thermally coupled in the cascade condenser. The selected refrigerants are supposed to have suitable pressure-temperature characteristics. The data reduction of the cascade refrigeration system can be analyzed as follows:

$$PR = \frac{P_{cond}}{P_{evap}} \quad (1)$$

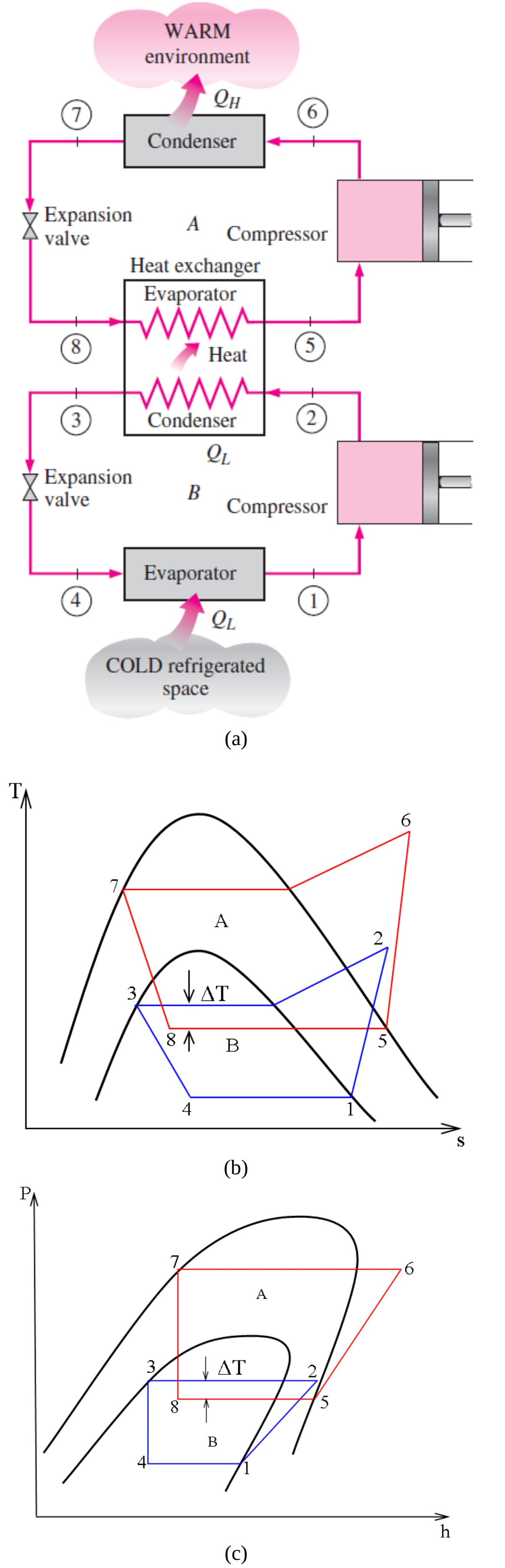


Figure 1. Two-stage cascade refrigeration system (Cengel and Boles, 2010) with T-s and P-h diagrams for the performance analysis of alternative refrigerants.

where the pressure ratio (PR) of the cycle is calculated by the division of condensing pressure (P_{cond}) into evaporating pressure (P_{evap}).

$$\dot{m}_A h_5 + \dot{m}_B h_3 = \dot{m}_A h_8 + \dot{m}_B h_2 \quad (2)$$

where the refrigerant mass flow rate flowing through the low-temperature cycle ($\dot{m}_B$) is calculated from the energy balance ($\dot{E}_{out}$ = $\dot{E}_{in}$) in the heat exchanger.

$$\dot{Q}_L = \dot{m}_B (h_1 - h_4) \quad (3)$$

where the rate of heat removal by a cascade cycle ($\dot{Q}_L$) is determined by means of the rate of heat absorption in the evaporator of the low-temperature stage.

$$\dot{W}_{in} = \dot{W}_{compI,in} + \dot{W}_{compII,in}$$

$$\dot{W}_{in} = \dot{m}_A (h_6 - h_5) + \dot{m}_B (h_2 - h_1) \quad (4)$$

where the power input to the cascade cycle ($\dot{W}_{in}$) is expressed as the sum of the power inputs to the compressors.

$$COP = \frac{\dot{Q}_L}{\dot{W}_{in}} \quad (5)$$

where the coefficient of performance (COP) of a refrigeration system is defined as the ratio of the refrigeration rate ($\dot{Q}_L$) to the net power input ($\dot{W}_{in}$).

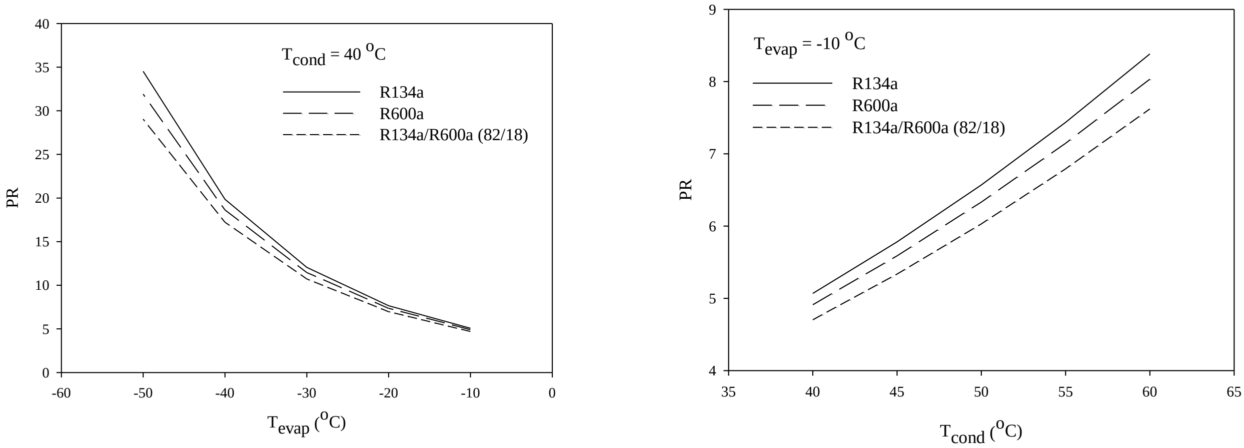
RESULTS AND DISCUSSION

The substances of the refrigerants used in refrigeration systems should have stable characteristics at the ambient temperature chemically and short atmospheric lifetimes in the case of leakage from the system, in consideration of protection of the ozone layer and global warming. The selection of refrigerant working fluids is decided according to the necessary refrigeration temperature and the type of equipment to be used. In addition, it should be noted that there are other variable factors for this selection in terms of the usage area of the refrigeration such as industrial and domestic units. Another important factor is the refrigerant's coefficient of performance (COP) for the cycle. The refrigerant with a high performance coefficient (COP) is preferable for use in the cycle.

The dependency on the composition of refrigerant mixtures makes the pressure-temperature relationship in the cycle significant for condensers and evaporators. Cycle performance is affected by the pressure alterations, while the ratios of the blend vary at the given temperature. For this reason, the proportion of the components in the mixture is one of the significant factors in the cycle. For comparison of the theoretical data, R12, R22, and R502 are chosen in this paper as reference fluids due to their common usage in cooling systems and prohibition by the Montreal Protocol.

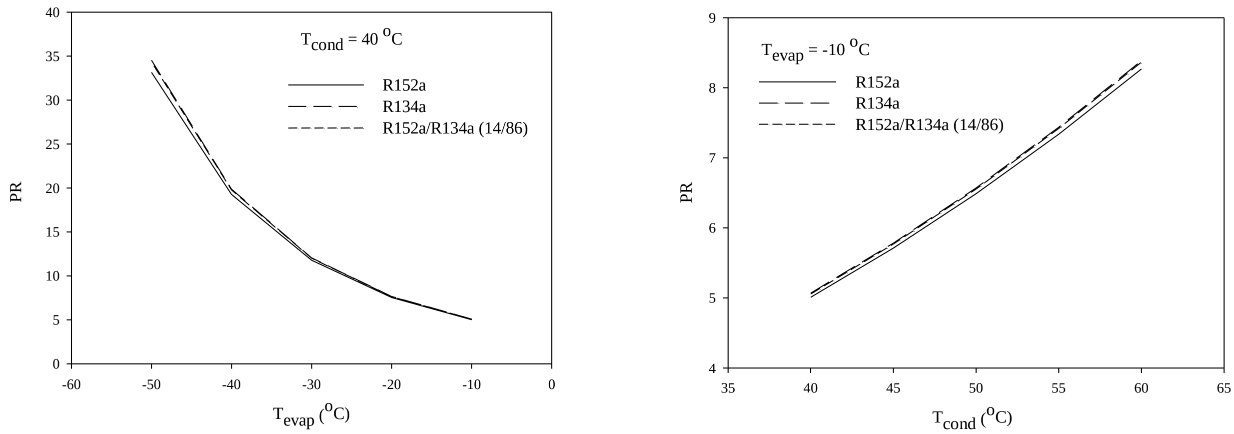
The alterations in pressure ratio (PR) with the temperatures of the condenser (T_{cond}) and evaporator (T_{evap}) are shown in Figs. 2 to 4 for the refrigerant mixtures listed in Table 1 and their base refrigerants as follows: R-134a/R-600a (82/18 by wt%) vs. R-134a vs. R-600a (Figure 2), R-134a/R-152a (86/14 by wt%) vs. R-134a vs. R-152a (Figure 3), and R290/R600a (55/45 by wt%) vs. R290 vs. R600a (Figure 4). The nearest pressure ratio (PR) of the substitute refrigerant to the base one is that of R-134a/R-152a (86/14 by wt%), whose pressure ratio (PR) is 0.89% higher than that of R152a and 0.26% lower than that of R134a for the constant condensing and evaporating temperatures of 40 °C and -10 °C, respectively. The pressure ratio (PR) of the R-134a/R-600a mixture (82/18 by wt%) is 7.7% lower than that of R134a and 4.43% lower than R600a for the constant condenser and evaporator temperatures of 40 °C and -10 °C, respectively. In addition to this, the R290/R600a mixture (55/45 by wt%), whose pressure ratio (PR) is 42.55% lower than that of R600a and 15.13% lower than that of R290, gives the highest ratios as substitutes for R600a and R290 for the constant condensing and evaporating temperatures of 40 °C and -10 °C, respectively.

In Tables 2 and 3, the significant parameters of cycle performance such as refrigerant charge rate ($\dot{m}$), heat transfer rate ($\dot{Q}$), power input rate ($\dot{W}$), coefficient of performance (COP), and change in refrigerant charge rate ($\dot{m}\%$) under different theoretical conditions are calculated as case studies of high-performance refrigerants and blend refrigerants determined as a result of the analysis shown in Figs. 5 to 7. It should be noted that the same refrigerant couples were used in the low- and high-pressure zones of the cascade system for the analyses in shown in Figs. 5 to 9. As shown in Figure 1a, the operating conditions of R152a for the high-pressure zone (A) of the cascade system and R717 for the low-pressure zone (B) of the cascade system needs the lowest refrigerant charge rate for the low-temperature zone of the cycle. The highest coefficient of performance (COP) is obtained for the condition of R717 for the high-pressure zone (A) of the cascade system and R152a for the low-pressure zone (B) of the cascade system among all refrigerants and their mixtures used in the paper. It should be noted that alteration of coefficient of performance (COP) can be seen in these tables for the constant evaporating temperature of the low-pressure zone and various condensing temperatures of the high-pressure zone.



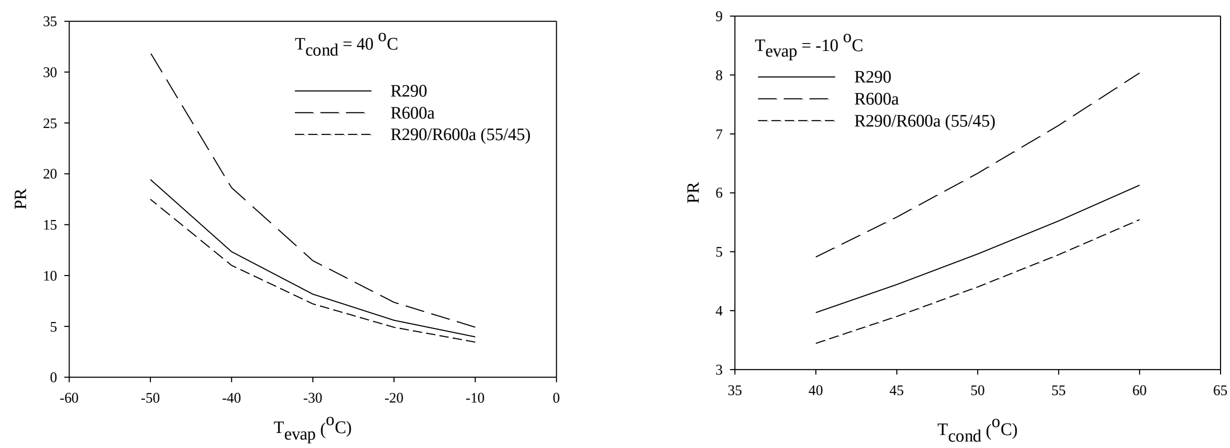
(a) (b)

Figure 2. Pressure ratio vs. evaporating (a)/condensing (b) temperatures for R134a-R600a.



(a) (b)

Figure 3. Pressure ratio vs. evaporating (a)/condensing (b) temperatures for R134a-R152a.



(a) (b)

Figure 4. Pressure ratio vs. evaporating (a)/condensing (b) temperatures for R290-R600a.

Table 2. Alteration of refrigerant charge rates at specific conditions for the high-performance refrigerants used in the high-(A) and low-(B) pressure zones of the system in Figure 1.

Ref.	**T_{evap}-B (^oC)**	**T_{cond}-A (^oC)**	**T_{HE} (^oC)**	$\dot{m}_A$ **(kg s^{-1})**	$\dot{m}_B$ **(kg s^{-1})**	$\dot{Q}_L$ **(kW)**	$\dot{W}_{in}$ **(kW)**	**COP**	$\dot{m}$ **(%)**
R12	-30	40	1	1	0.72	99.77	33.9	2.94	27.7
R12	-30	45	1	1	0.69	95.22	35.46	2.68	30.99
R12	-30	50	1	1	0.65	90.59	36.93	2.45	34.35
R12	-30	55	1	1	0.62	85.87	38.33	2.24	37.76
R12	-30	60	1	1	0.58	81.08	39.63	2.04	41.24
R22	-30	40	1	1	0.71	136.16	46.74	2.91	2.91
R22	-30	45	1	1	0.68	130.3	49.04	2.65	2.65
R22	-30	50	1	1	0.64	124.27	51.25	2.42	2.42
R22	-30	55	1	1	0.61	118.09	53.36	2.21	2.21
R22	-30	60	1	1	0.58	111.72	55.37	2.01	2.01
R290	-30	40	1	1	0.69	234.07	81.93	2.85	2.85
R290	-30	45	1	1	0.65	221.23	85.43	2.58	2.58
R290	-30	50	1	1	0.61	208.07	88.58	2.34	2.34
R290	-30	55	1	1	0.57	194.55	91.55	2.12	2.12
R290	-30	60	1	1	0.53	180.61	94.29	1.91	1.91
R152a	-30	40	1	1	0.72	207.01	69.37	2.98	27.06
R152a	-30	45	1	1	0.7	198.7	72.74	2.73	29.99
R152a	-30	50	1	1	0.67	190.25	75.95	2.5	32.97
R152a	-30	55	1	1	0.63	181.63	79.03	2.29	36
R152a	-30	60	1	1	0.6	172.84	81.96	2.1	39.1
R717	-30	40	1	1	0.76	936.47	315.53	2.96	23.15
R717	-30	45	1	1	0.75	914.78	335.08	2.73	24.93
R717	-30	50	1	1	0.73	892.83	354.89	2.51	26.73
R717	-30	55	1	1	0.71	870.58	373.65	2.32	28.56
R717	-30	60	1	1	0.69	848.01	392.66	2.15	30.41

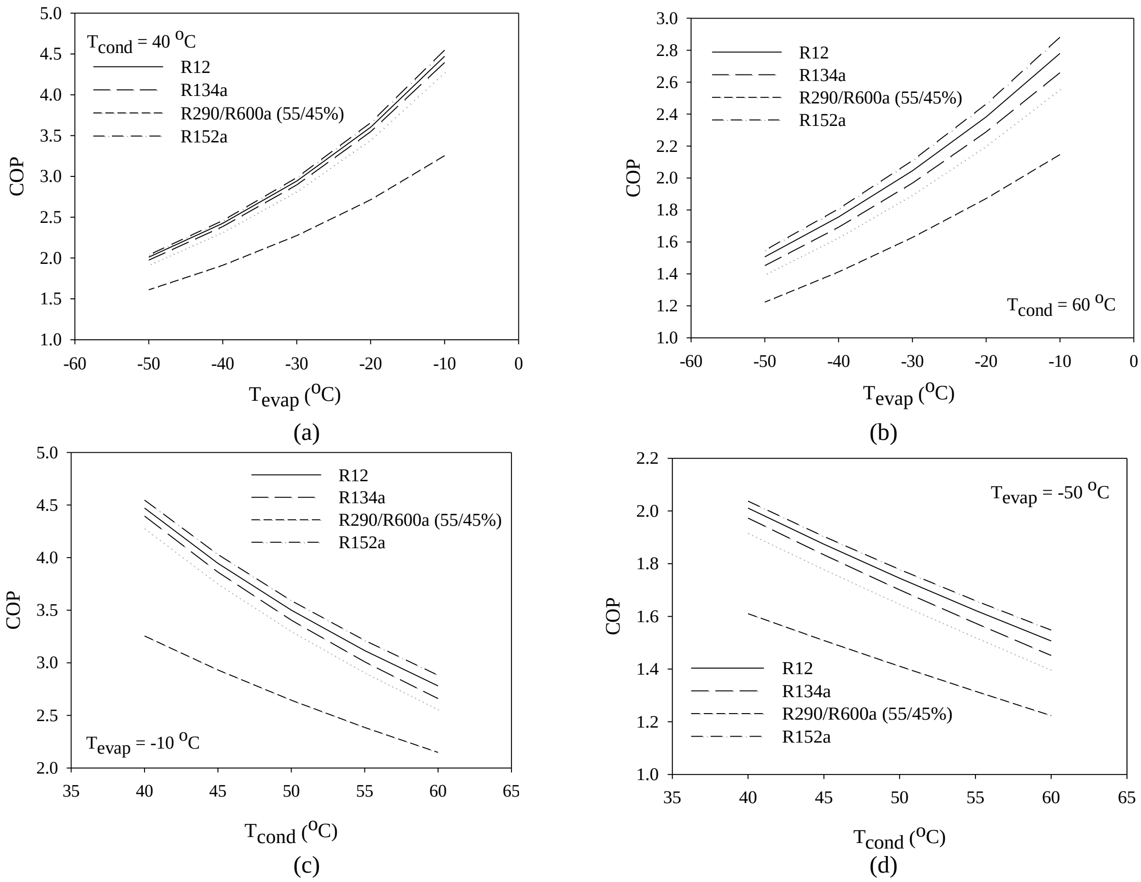


Figure 5. Comparison of performance coefficients (COP) of R12 and its alternatives using the system in Figure 1 at different condensing temperatures of 40 ^oC (a) and 60 ^oC (b) and evaporating temperatures of -10 ^oC (c) and -50 ^oC (d).

Table 3. Alteration of refrigerant charge rates under specific conditions for the blended refrigerants and high-performance refrigerants used in the high-(A) and low-(B) pressure zones of the system in Figure 1.

Ref.	**Rate % by wt.**	**T_{evap}-B (^oC)**	**T_{cond}-A (^oC)**	**T_{HE} (^oC)**	$\dot{m}_A$ **(kg s^{-1})**	$\dot{m}_B$ **(kg s^{-1})**	$\dot{Q}_L$ **(kW)**	$\dot{W}_{in}$ **(kW)**	**COP**	$\dot{m}$ **(%)**
R290/R600a	55/45	-30	40	1	1	0.68	226.43	99.57	2.27	31.25
R290/R600a	55/45	-30	45	1	1	0.65	214.86	102.48	2.09	34.76
R290/R600a	55/45	-30	50	1	1	0.61	203.09	105.16	1.93	38.33
R290/R600a	55/45	-30	55	1	1	0.58	191.08	107.62	1.77	41.98
R290/R600a	55/45	-30	60	1	1	0.54	178.82	109.85	1.62	45.7
R152a/R134a	14/86	-30	40	1	1	0.7	136.53	46.92	2.9	29.61
R152a/R134a	14/86	-30	45	1	1	0.66	129.7	48.95	2.64	33.13
R152a/R134a	14/86	-30	50	1	1	0.63	122.74	50.85	2.41	36.71
R152a/R134a	14/86	-30	55	1	1	0.59	115.64	52.63	2.19	40.37
R152a/R134a	14/86	-30	60	1	1	0.55	108.37	54.29	1.99	44.12
R134a/R600a	82/18	-30	40	1	1	0.68	130	45.72	2.84	31.49
R134a/R600a	82/18	-30	45	1	1	0.64	122.46	47.53	2.57	35.46
R134a/R600a	82/18	-30	50	1	1	0.6	114.75	49.2	2.33	39.52
R134a/R600a	82/18	-30	55	1	1	0.56	106.87	50.72	2.1	43.68
R134a/R600a	82/18	-30	60	1	1	0.52	98.77	52.12	1.89	47.94
R152a+R717	A/B	-30	40	1	1	0.16	206.41	69.97	2.94	83.06
R152a+R717	A/B	-30	45	1	1	0.16	198.12	73.32	2.7	83.74
R152a+R717	A/B	-30	50	1	1	0.15	189.69	76.51	2.47	84.43
R152a+R717	A/B	-30	55	1	1	0.14	181.1	79.56	2.27	85.13
R152a+R717	A/B	-30	60	1	1	0.14	172.33	82.47	2.08	85.85
R717+R152a	A/B	-30	40	1	1	3.3	939.22	312.78	3	-230.88
R717+R152a	A/B	-30	45	1	1	3.23	917.48	332.38	2.76	-223.22
R717+R152a	A/B	-30	50	1	1	3.15	895.46	352.26	2.54	-215.47
R717+R152a	A/B	-30	55	1	1	3.07	873.15	371.08	2.35	-207.61
R717+R152a	A/B	-30	60	1	1	2.99	850.5	390.17	2.17	-199.63

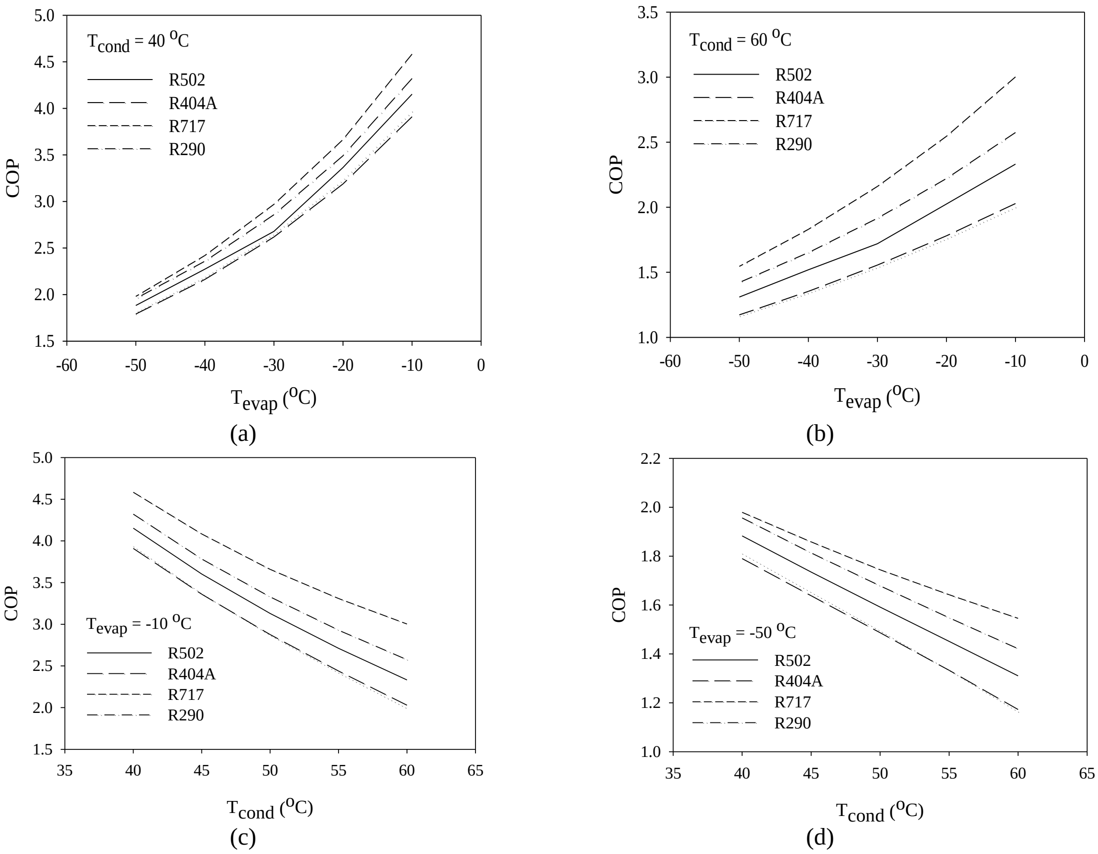


Figure 6. Comparison of performance coefficients (COP) of R502 and its alternatives using the system in Figure 1 at different condensing temperatures of 40 ^oC (a) and 60 ^oC (b) and evaporating temperatures of -10 ^oC (c) and -50 ^oC (d).

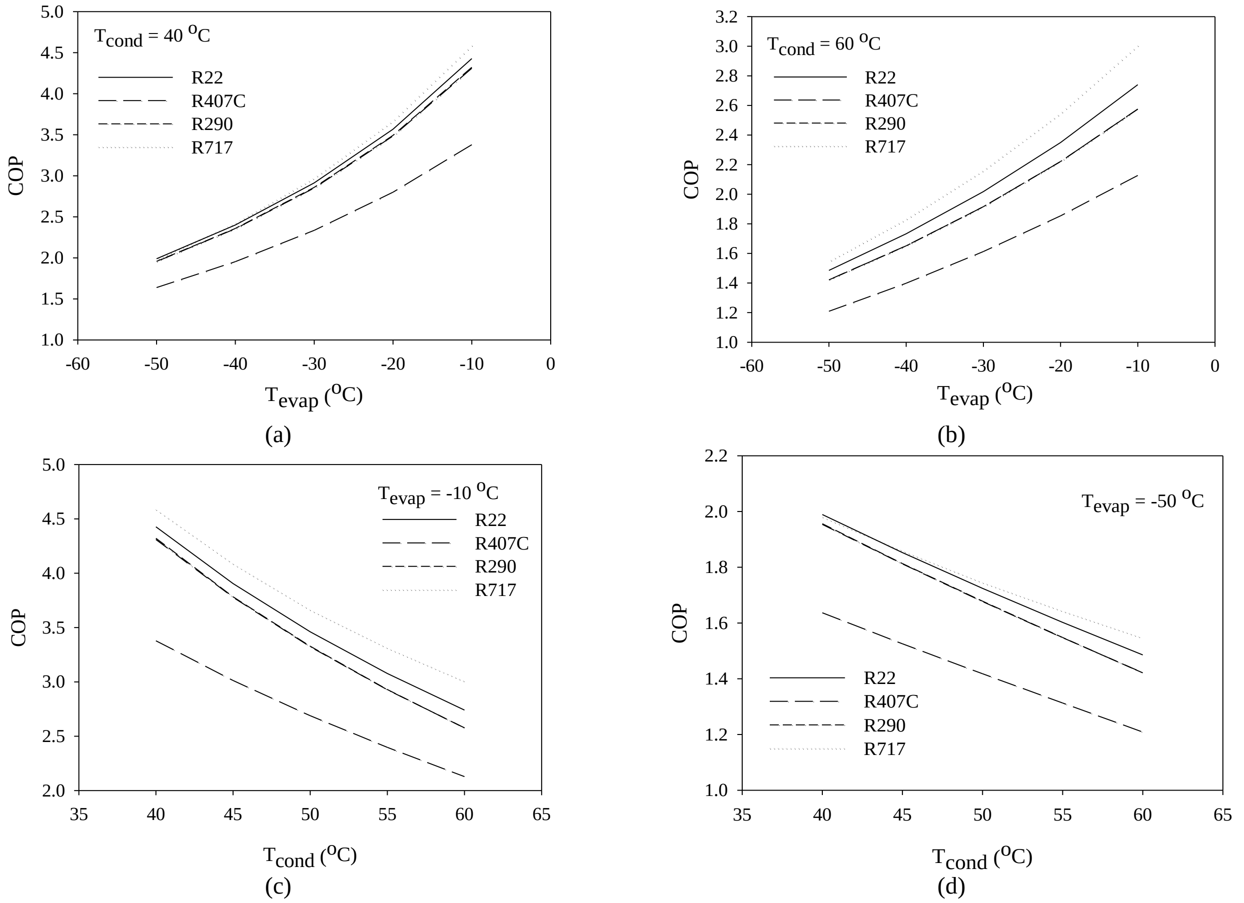


Figure 7. Comparison of performance coefficients (COP) of R22 and its alternatives using the system in Figure 1 at different condensing temperatures of 40 ^{o}C (a) and 60 ^{o}C (b) and evaporating temperatures of -10 ^{o}C (c) and -50 ^{o}C (d).

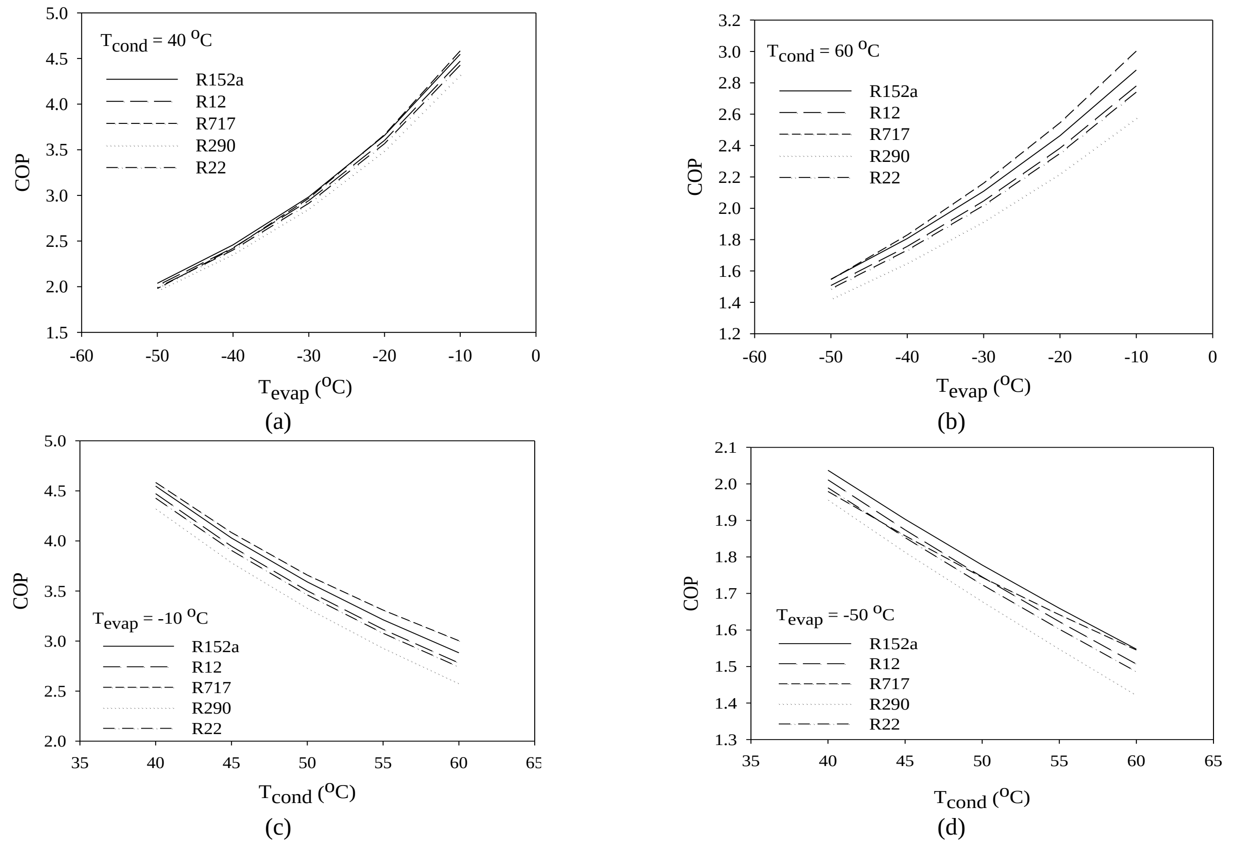


Figure 8. The comparison of performance coefficients (COP) of high-performance refrigerants determined for the system model in Figure 1 at different condensing temperatures of 40 **^{o}C (a) and 60 ^{o}C (b) and evaporating temperatures of -10 ^{o}C (c) and -50 ^{o}C (d).**

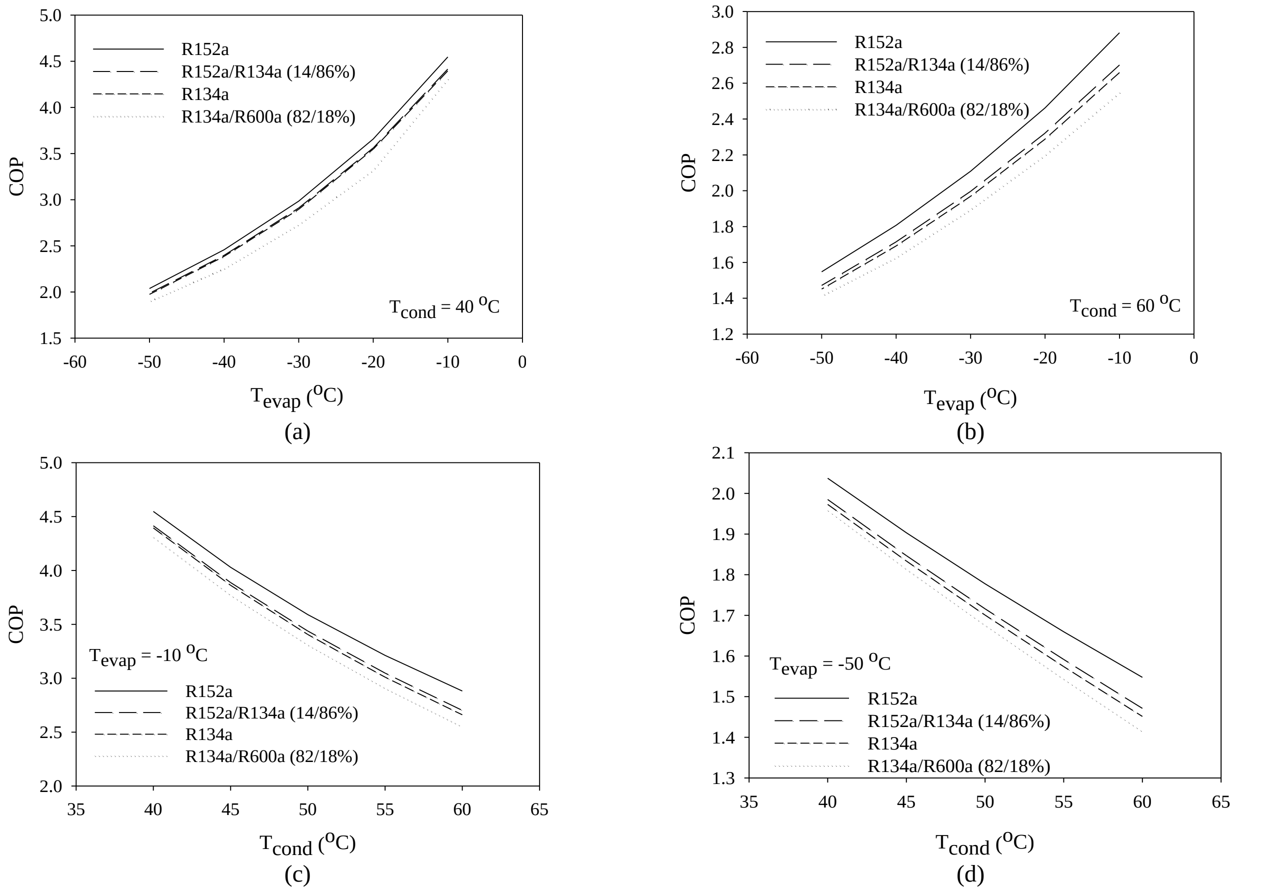


Figure 9. Comparison of performance coefficients (COP) of mixture refrigerants developed for R152a and R134a using the system in Figure 1 at different condensing temperatures of 40 ^{o}C (a) and 60 ^{o}C (b) and evaporating temperatures of -10 ^{o}C (c) and -50 ^{o}C (d).

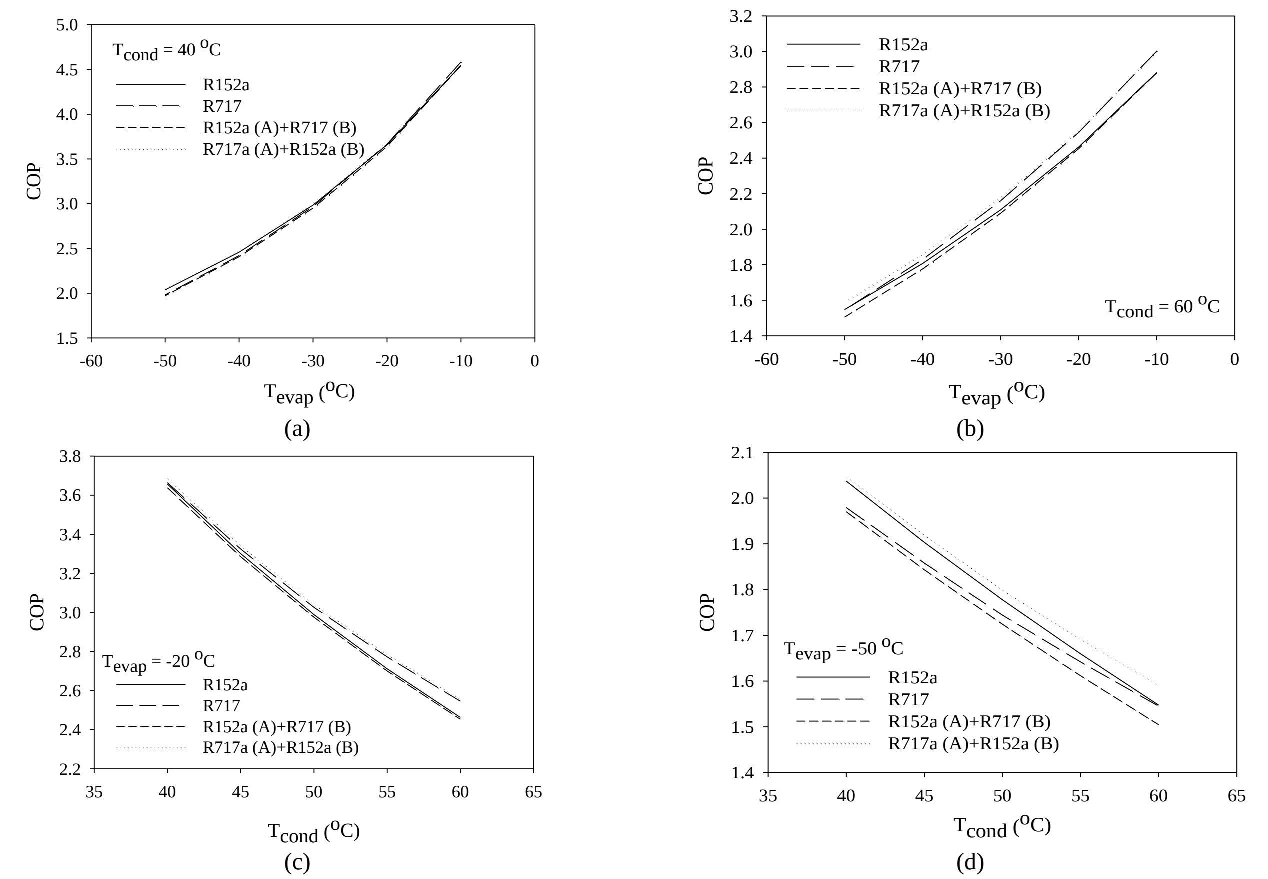


Figure 10. Comparison of performance coefficients (COP) of R152a and R717 using the high- (A) and low- (B) pressure zones of Figure 1 at different condensing temperatures of 40 ^{o}C (a) and 60 ^{o}C (b) and evaporating temperatures of -10 ^{o}C (c) and -50 ^{o}C (d).

The variation in the performance coefficient (COP) with evaporating (T_{evap}) and condensing temperatures (T_{cond}) is illustrated in Figs. 5 to 7. According to these figures, the coefficient of performance (COP) increases as the evaporating temperature (T_{evap}) increases for the constant condensing temperatures of 40 and 60 °C and evaporating temperatures ranging from -50 °C to -10 °C. The coefficient of performance (COP) increases as the condensing temperature (T_{cond}) decreases for the constant evaporating temperatures of -10 and -50 °C and condensing temperatures ranging from 40 °C to 60 °C. The decrease in pressure ratio (PR) for the compressor also improves the efficiency. All the performance coefficients (COP) of the alternative refrigerants tested are found to be lower than those of their base refrigerants under the simulation conditions. In addition to this, determination of alternative high-performance refrigerants to R12, R502, and R22 is performed for the studied cycle by means of these figures (shown in Figure 8). It can be seen in Figure 8 that the R152a and R717 refrigerants have higher coefficients of performance (COP) values than others. According to the comparison of the blended R-134a/R-152a refrigerant (86/14 by wt%) with R152a and R134a in Figure 9, the blend refrigerant has a higher coefficient of performance (COP) value than R-134a .

Figure 10 illustrates the variation of the high-performance R717 and R152a refrigerants for the studied cascade system's high-(A) and low-(B) temperature zones. When the R717 refrigerant is used in the high-(A) temperature cycle, and the R152a refrigerant is used in the low-(B) temperature cycle, the highest coefficient of performance (COP) is obtained for this operating condition and validated by the test of wide-ranging evaporator and condenser temperatures. It should be noted that these operating conditions have the largest rates of refrigerant charge for the low-temperature cycle, while the high-temperature's refrigerant charge rate is taken as 1 kg s^{-1} for all analyses in the paper. Moreover, the maximum coefficient of performance (COP) for these operating conditions is 5.37% higher than the operating conditions of R152a (A)+R717 (B), 2.82% higher than R717, and 2.67% higher than R152a, as shown in Figure 10d for the constant evaporating temperature of -50 °C and condensing temperatures ranging from 40 °C to 60 °C.

A large temperature range necessitates the use of cascade refrigeration systems. It can be seen from the T-s diagram in Figure 1 that the compressor work decreases and the amount of heat absorbed from the refrigerated space increases as a result of cascading. For this reason, the coefficients of performance (COP) values of cascade systems are higher than those of the ideal vapor compression system, as indicated by the literature. It should be noted that the initial cost of the cascade system is about double, as indicated by the simple vapor compression system, and, for this reason, it is necessary to estimate after how many years of usage the additional money invested in a cascade system can be saved.

CONCLUSION

Through a theoretical analysis of a comparison of coefficients of performance (COPs) belonging mainly to a two-stage cascade refrigeration cycle, the following results are obtained:

1. The thermo-physical properties (performance, efficiency), limitations, and restrictions related to safety, environmental impact, and associated legislation are the most significant factors in choosing a new refrigerant.

2. Low fluid viscosities and vapor phases, high liquid specific heat, high thermal conductivities of liquid and vapor phases, high latent heat, and small temperature glide are the desired thermo-physical properties of refrigerant mixtures.

3. The performance coefficient (COP) of the system increases with increasing evaporating temperature and decreasing condensing temperature for constant condensing and evaporating temperatures, respectively, in the analysis.

4. The comparison of performance coefficients (COP) and pressure ratios (PR) of the tested refrigerants showed that HFC-152a/HFC-134a (14/86 by wt%) and HFC-134a/HC600a (82/18 by wt%), as azeotropic mixtures, have higher performance coefficients (COP) than the non-azeotropic mixture of HC290/HC600a (55/45 by wt%).

5. The coefficient of performance (COP) values of the CFC-12, HCFC-22, CFC-502 refrigerants and their alternatives, such as HFC-134a, HFC-152a, HFC-404A, HFC-407C, HC-290, HC-600a, R717 (ammonia), and three mixtures composed of HFC-134a, HFC-152a, HC-600a, and HC-290 are compared to each other using a wide range of theoretical data. R152a and R717 are determined to be the best alternative refrigerants for the case study.

6. The effect of the usage of different refrigerants in the low- (B) and high- (A) stages of the cascade system is shown in the paper. The best refrigerants for the high- (A) and low- (B) temperature stages are determined to be R717 (A) and R152a (B) among all of the tested refrigerants in terms of their high coefficient of performance (COP) values.

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NOMENCLATURE

CFCs	chlorofluorocarbons
COP	coefficient of performance
$\dot{E}$	energy, kW
GWP	global warming potential
h	enthalpy, kJ kg^{-1}
HCFCs	hydro chlorofluorocarbons
HCs	hydrocarbons
HFCs	hydro fluorocarbons
$\dot{m}$	mass charge rate of refrigerant, kg s^{-1}
ODP	ozone depletion potential
P	pressure, MPa
PR	pressure ratio
Ref	refrigeration
s	entropy, kJ kg^{-1}K^{-1}
T	temperature, °C
ΔT	temperature difference, K
$\dot{Q}$	heat transfer rate, kW
$\dot{W}$	power, kW
Greek symbols	
ρ	density, kg m^{-3}
Subscripts	
A	high temperature zone
B	low temperature zone
cond	condensing/condenser
comp	compressor
evap	evaporating/evaporator
g	vapor
H	heat
in	input
l	liquid
L	low
th	thermal
1,5	evaporator superheat
2, 6	compressor superheat
3, 7	condenser saturated liquid
4, 8	evaporator saturated mixture

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