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THERMODYNAMIC PERFORMANCE OF THE TRANSCRITICAL REFRIGERATION CYCLE WITH EJECTOR EXPANSION FOR R744, R170, AND R41

Ayşe Uğurcan ATMACA*, Aytunç EREK, Orhan EKREN*** and Mustafa Turhan ÇOBAN******
*Dokuz Eylül University Faculty of Engineering Mechanical Engineering Department
Tınaztepe Yerleşkesi 35390 Buca – İzmir, ugurcan.atmaca@deu.edu.tr
**Dokuz Eylül University Faculty of Engineering Mechanical Engineering Department
Tınaztepe Yerleşkesi 35390 Buca – İzmir, aytunc.erek@deu.edu.tr
***Ege University Solar Energy Institute, EVKA-3, Bornova, 35100, İzmir, orhanekren@gmail.com
****Ege University Faculty of Engineering Mechanical Engineering Department
35040, Bornova-İzmir, turhan.coban@ege.edu.tr

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Abstract: For more than a decade, there is a great demand for finding environmentally-friendly refrigerants obeying the global warming potential value restrictions of the tough environmental legislation. Among the candidate working fluids, R744 (carbon dioxide or CO_2), R170 (ethane), and R41 (fluoromethane) are selected to be investigated parametrically in this paper. Performance comparison is made for these three working fluids individually in both transcritical (supercritical) refrigeration cycle and modification of this cycle with ejector expansion. As the first step, the effects of the gas cooler outlet temperature, evaporator temperature, and evaporator outlet superheat temperature difference on the overall performance and percentage expansion losses are investigated within a specific gas cooler pressure range. Evaporator outlet superheat temperature difference is found to be the least effective parameter on the performance; hence, secondly, the transcritical ejector expansion refrigeration cycle is analyzed considering only evaporator temperature and gas cooler outlet temperature based on the same gas cooler pressure ranges. Thermodynamic models are constructed in Matlab® and the ejector equations for the ejector expansion refrigeration cycle are established with reference to constant pressure mixing assumption. Comparisons of the performance, percentage expansion losses, and performance improvement potential through the implementation of the ejector instead of the expansion valve among these three refrigerants having low critical temperatures represent the main objective of the paper in order to make contributions to the previous researches in the literature.
Keywords: R744 (carbon dioxide), R170 (ethane), R41 (fluoromethane), Expansion losses, Ejector expansion refrigeration cycle, Constant pressure mixing (CPM) ejector.

EJEKTÖR GENLEŞTİRİCİLİ TRANSKRİTİK SOĞUTMA ÇEVRİMİNİN R744, R170 VE R41 İÇİN TERMODİNAMİK PERFORMANSI

Özet: On yıldan fazla bir süredir sıkı çevresel yönetmeliklerin küresel ısınma potansiyeli değeri kısıtlamalarına uyan çevre dostu soğutkanları bulmaya yönelik önemli bir arayış sözkonusudur. Aday akışkanlar arasından R744 (karbondioksit veya CO_2), R170 (etan) ve R41 (florometan) bu çalışmada parametrik olarak incelenmek için seçilmiştir. Performans karşılaştırmaları üç soğutkan için ayrı ayrı hem transkritik (süperkritik) soğutma çevriminde hem de bu çevrimin ejektör genleştiricili olarak geliştirildiği soğutma çevriminde yapılmıştır. Birinci adım olarak, gaz soğutucu çıkış sıcaklığının, buharlaştırıcı sıcaklığının ve buharlaştırıcı çıkışındaki kızgın buhar sıcaklığının buharlaştırıcıya göre sıcaklık farkının toplam performans ve yüzdesel genleşme kayıplarına etkileri belirli bir gaz soğutucu basıncı aralığında incelenmiştir. Buharlaştırıcı çıkışındaki kızgın buhar için sıcaklık farkı performans üstündeki en az etkili parametre olarak bulunmuştur; böylece transkritik ejektör genleştiricili soğutma çevrimi sadece buharlaştırıcı sıcaklığı ve gaz soğutucu çıkış sıcaklığı için bir önceki analizlerle aynı gaz soğutucu basıncı aralıklarında incelenmiştir. Termodinamik modeller Matlab® ortamında oluşturulmuştur ve ejektör genleştiricili soğutma çevrimi için ejektör denklemleri sabit basınçta karışım varsayımına göre elde edilmiştir. Düşük kritik sıcaklığa sahip olan bu üç soğutkan arasındaki performans, yüzdesel kısılma kayıpları ve genleşme vanasının yerine ejektörün kullanılması ile oluşan performans iyileştirme potansiyeli kıyaslamaları, literatürdeki önceki araştırmalara katkı yapabilmek adına makalenin temel hedefini oluşturmaktadır.
Anahtar Kelimeler: R744 (karbondioksit), R170 (etan), R41 (florometan), Genleşme kayıpları, Ejektör genleşmeli soğutma çevrimi, Sabit basınçta karışımlı ejektör.

NOMENCLATURE

1, 2,…,10	Operational steps of the transcritical EERC
a, b, c, d	Operational steps of the transcritical VCRC
COP	Coefficient of performance [-]
h	Enthalpy [kJ/kg]
$\dot{m}$	Mass flow rate [kg/s]
P	Pressure [kPa]
P_b	Suction chamber pressure (the pressure of the primary and the secondary fluid flows at the inlet of the mixing chamber) [kPa]
q	Heat transfer on a unit-mass basis [kJ/kg]
$\dot{Q}$	Heat transfer rate [kW]
r	The ratio of the primary fluid mass flow rate to the mixture fluid mass flow rate or quality [-]
R	Performance improvement ratio [-]
s	Entropy [kJ/kgK]
T	Temperature [K]
u	Velocity [m/s]
w	Entrainment ratio [-]
w_{in}	Work input to the compressor [kJ/kg]
w_{net}	Net work transfer to/from the system [kJ/kg]
$\dot{W}_{net}$	Net work transfer rate to/from the system [kW]
x	Exergy on a unit-mass basis [kJ/kg]
$\dot{X}$	The rate of exergy [kW]

Greek letters

η	Efficiency [-]
τ	Pressure ratio (P_{high}/P_{low})
ψ	Flow (or stream) exergy [kJ/kg]

Subscripts

0	Dead state
comp	Compressor
d	Diffuser
dest	Destruction
e	Evaporator
exp *v*	Expansion valve
gc	Gas cooler
H	High temperature environment
in	Inlet
isen	Isentropic
k	Location (boundary)
l	Liquid port of the separator
L	Low temperature environment
m	Primary (motive) nozzle/fluid
mix	Mixing section (chamber)
out (*o*)	Outlet
s	Secondary (suction) nozzle (chamber)/fluid
sep	Separator
sh	Superheating
v	Vapor port of the separator

Abbreviations

0*D*	Zero-dimensional
CPM	Constant pressure mixing
EERC	Ejector expansion refrigeration cycle
GWP	Global warming potential
HFCs	Hydrofluorocarbons
HFOs	Hydrofluoroolefins
NBP	Normal boiling point
ODP	Ozone depletion potential
VCRC	Vapor compression refrigeration cycle

INTRODUCTION

Manufacture of the environmentally-friendly and energy efficient systems are the most important target of the refrigeration industry. Refrigeration sector is in great demand for finding the perfect working fluids owing to the limitations put by the European Directive 2006/40/EC (Directive 2006/40/EC, 2006) and EU Regulation No 517/2014 which is also known as F-gas Regulation (Regulation (EU) No 517/2014, 2014) on the global warming potential (GWP) values of the refrigerants. Although there are various environmentally-friendly refrigerant alternatives, there is no perfect working fluid in all regards. Generally speaking, alternative pure fluids are hydrofluoroolefins (HFOs) such as R1234yf and R1234ze(E); some of the hydrofluorocarbons (HFCs) having low GWP values, e.g., R152a, R161, and R41; and natural refrigerants (R744, R717, 600a, R170, and R290, etc.). In this study, R744, R170, and R41 having low critical temperatures are selected to make comparisons in the transcritical refrigeration cycle and the modified cycle configuration with the ejector expansion.

First target is providing with the individual performance analyses and percentage expansion losses of these three refrigerants parametrically in the transcritical refrigeration cycle. Though there are limited number of studies regarding the comparison of these refrigerants, analyses of their blend options are drawing more interest year by year. Di Nicola et al. (2011) investigated a cascade refrigeration cycle using the low temperature working fluid as CO_2 mixture variations combined with R170, R290, R1150, R1270, and RE170 in order to propose blend alternatives that could be used at temperatures below the triple point of CO_2 (-56.56 ºC). Dai et al. (2015) proposed R41 and R32 among the ten low GWP refrigerants as the other blend component of CO_2 to be used in heat pump water heaters. Cox et al. (2008) analyzed numerically and experimentally three mixture alternatives, i.e., R1270/R161, R170/R717, and R744/R41. Wang et al. (2017) analyzed the application of the azeotropic refrigerant mixture of CO_2 and R41 in different systems, namely a refrigerated cabinet, a water-source and air-source heat pump water heaters under various operation conditions.

One of the problem of pure CO_2 is the high operation pressures; whereas the limitation of pure R41 is the flammability. Wang et al. (2017) stated that the reason for making blends of them is proposing a refrigerant mixture having lower flammability and lower operation pressures in comparison with the pure working fluids. Although there are investigations concerned with the blend options of R744, R170, and R41, the number of the comparative individual analyses among these three refrigerants is very limited in the literature. Liao and Zheng (2014) analyzed the low-temperature transcritical Rankine cycle for R41 and CO_2 and concluded that R41 performs better. A preliminary study covering comparisons of only R744 and R170 in the transcritical refrigeration cycle was conducted by the authors of this paper (Atmaca et al., 2018). In the present paper, the parametric investigation of the evaporator temperature, evaporator outlet superheat temperature difference, and gas cooler outlet temperature is conducted based on the specific gas cooler pressure ranges for R744, R170, and R41. Hence, each refrigerant performance is evaluated within a specific gas cooler pressure range that the working fluids could exhibit their maximum coefficient of performance (COP).

Lorentzen (1995) highlighted the superiority of CO_2 when compared to the other natural refrigerants such as ammonia, isobutane, etc. owing to its good environmental characteristics, non-toxicity, non-flammability, and excellent thermophysical properties. However, another hindrance of the systems utilizing CO_2 is the huge throttling losses resulting in decrease in the cycle performance (Dai et al., 2015). It is widely known from the previous researches that using ejector instead of the expansion valve is a way of minimizing these throttling losses thereby causing an increase in the overall system performance. There are many experimental and numerical studies in the literature focusing on the improvement potentials of the transcritical CO_2 cycle via ejector expansion technology (Li and Groll, 2005; Sarkar, 2008; Liu et al., 2016; Elbel and Hrnjak, 2004; Smolka et al., 2013; and etc.). Hence, the second objective of this study is displaying the performance improvement potential of these three working fluids, namely R744, R170, and R41, through the ejector expansion refrigeration cycle (EERC) with reference to the most influential parameters on the performance. The reason for comparing the percentage expansion losses is highlighting the performance improvement potentials of these high pressure refrigerants by means of the ejector expansion technology.

R41 and CO_2 are similar to each other in terms of physical properties and they have nearly the same normal boiling points (NBP) (Wang et al., 2017); whereas R170 and CO_2 have very close critical temperatures. As seen from the summarized literature, the blend alternatives of these refrigerants are commonly being studied, hence providing the individual performance behavior of this candidate group making use of thermodynamic models would be beneficial since it could give a broad idea regarding the proposed mixtures according to the operation ranges. Furthermore, the replacement of these working fluids in question should be discussed considering the performance with respect to operation conditions. The last motivation of this study from the viewpoint of the literature contribution is providing with the performance improvement values of these high pressure refrigerants experiencing very high expansion losses through the transcritical EERC with a comparative basis.

The zero-dimensional (0D) thermodynamic models of the transcritical vapor compression refrigeration cycle (VCRC) and the EERC are constructed in Matlab® applying the conservation of mass, momentum, and energy to the cycle components and the ejector sections, namely motive (primary) nozzle, suction nozzle (suction chamber or secondary nozzle), mixing section (chamber), and diffuser. Only inlet and outlet conditions are considered for each component and each section while establishing the governing equations. The thermodynamic properties of the refrigerants at each state is calculated with the help of REFPROP version 9.1 (Lemmon et al., 2013).

REFRIGERANTS AND CYCLE DEFINITIONS

The thermodynamic properties, safety classes, and environmental characteristics of the investigated refrigerants analyzed in this paper and some of the other natural refrigerants are given in Table 1. R744, R170, and R41 have very low critical temperatures that they allow heat rejection process by means of condensation only up to 31.1 ºC, 32.2 ºC, and 44.1 ºC, respectively. Focusing on the required temperature difference in the heat exchanger, the upper limit for the condensation temperature is practically at temperatures 5 to 10 K below the critical temperature (Danfoss, 2008).

Most of the refrigeration systems are designed to be able to reject heat to the atmosphere at the temperature above 25 ºC. Therefore, these working fluids are investigated in the transcritical (supercritical) refrigeration cycle in which the heat rejection process occurs above the critical temperature and the pressure as shown in Figure 1 (a). The second cycle to be investigated in this study is the modification of the base cycle with ejector expansion as shown in Figure 1 (b). The high pressure refrigerant coming from the gas cooler is expanded to a pressure lower than the evaporator pressure due to the motive nozzle of the ejector and the secondary fluid flow coming from the evaporator is sucked into the suction nozzle of the ejector. As a result of the momentum transfer between the primary and secondary fluid flows, the total flow at the exit of the ejector has an intermediate pressure higher than the evaporation pressure (Elbel and Hrnjak, 2008).

Table 1. Properties of the some of the natural refrigerants and the target group of this study (**a:** Lemmon et al., 2013; **b:** The Linde Group, 2018; **c:** Cox et al., 2008.)

Refrigerants	**R744**	**R170**	**R41**	**R717**	**R290**	**R600a**	**R1270**
Chemical name[a]	Carbon dioxide	Ethane	Fluoromethane	Ammonia	Propane	Isobutane	Propylene
Chemical formula[a]	CO_2	CH_3CH_3	CH_3F	NH_3	$CH_3CH_2CH_3$	$CH(CH_3)_3$	CH_2=CH-CH_3
Molar mass (kg/kmol)[a]	44.01	30.069	34.033	17.03	44.096	58.122	42.08
NBP (°C)[a]	-78.46	-88.58	-78.31	-33.33	-42.11	-11.75	-47.62
Critical temperature (°C)[a]	31.98	32.17	44.13	132.25	96.74	134.66	91.06
Critical pressure (MPa)[a]	7.3773	4.8722	5.897	11.333	4.2512	3.629	4.555
ASHRAE safety group	A1[b]	A3[b]	A2[c]	B2L[b]	A3[b]	A3[b]	A3[b]
ODP	0[b]	0[b]	0[c]	0[b]	0[b]	0[b]	0[b]
GWP	1[b]	6[b]	97[c]	0[b]	3[b]	3[b]	2[b]

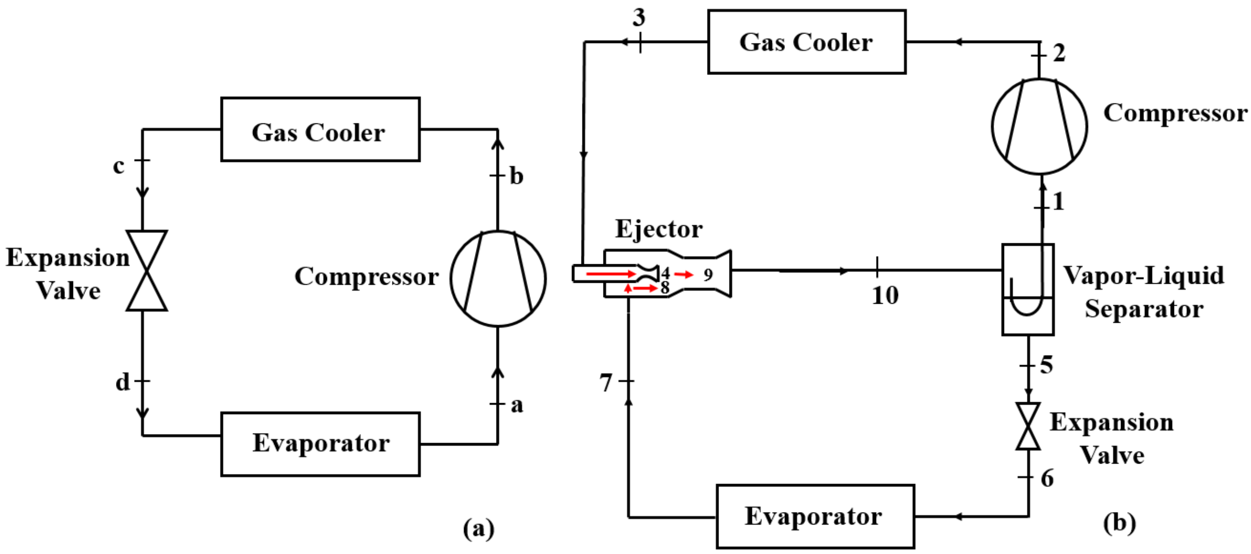


Figure 1. Schematic view of the transcritical refrigeration cycle **(a)** and the EERC **(b)**.

The benefits of the EERC could be understood from the comparison of the P-h diagrams shown in Figure 2 and obtained via Engineering Equation Solver (Klein, 2017). Thanks to the ejector expansion technology, the pressure of the refrigerant at the inlet of the compressor increases thereby decreasing the power consumption and increasing the isentropic efficiency. The evaporation capacity as well increases in comparison with the basic transcritical cycle. All diagrams are presented for a typical air conditioning application utilizing CO_2 as the refrigerant.

As it is seen from the P-h diagrams, heat rejection processes occur above the critical temperature and pressure of the CO_2. The notations used in the P-h diagrams in Figure 2 are consistent with that of the schematic views in Figure 1.

Constant pressure mixing (CPM) assumption stating that the pressure stays the same throughout mixing process (Kornhauser, 1990) is used in order to model ejector mixing section. It is seen from Figure 2 (b) that the pressure doesn't change throughout mixing process represented from number 4 (the state of the primary fluid) and number 8 (the state of the secondary fluid) to the number 9 (the state of the total fluid flow).

THERMODYNAMIC MODELLING

Basic equations for the energy and exergy analyses of the transcritical refrigeration cycle is established firstly in order to evaluate performance and to calculate the percentage exergy destruction occurring in the expansion valve; whereas the second group of the equations are given for the thermodynamic model of the ejector based on the following assumptions.

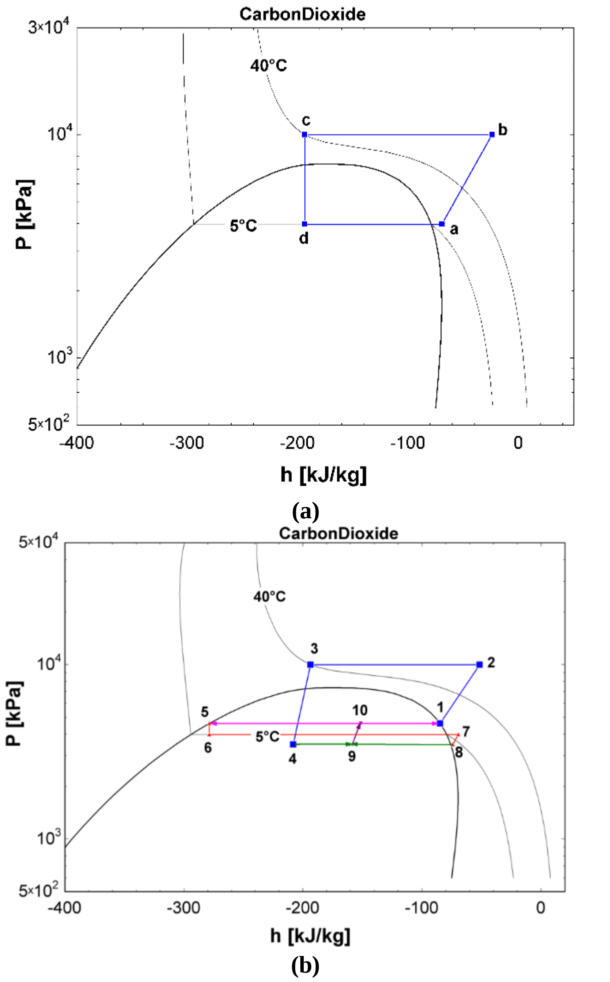


Figure 2. P-h diagrams of the transcritical refrigeration cycle **(a)** and the EERC including constant pressure mixing (CPM) ejector **(b)** for CO_2 (Klein, 2017).

(i) The refrigerants have the same pressure before mixing and the pressure doesn't change throughout mixing (Kornhauser, 1990).
(ii) The pressure drop is negligible in the pipeline, evaporator, and the gas cooler. Heat transfer to the environment is ignored except for the gas cooler and the evaporator.
(iii) The irreversibility of each ejector section and the compressor is taken into account with the properly defined efficiency values. The compressor is assumed to be adiabatic in the energy and exergy analyses. Isentropic efficiency of the compressor is defined with respect to Brunin et al. (1997) as in the studies of Li et al. (2014), Wang et al. (2016), and Nehdi et al. (2007).
(iv) The primary and the secondary flow streams and the mixed flow have negligible velocities at the inlet of the motive and suction nozzles and at the outlet of the diffuser, respectively.
(v) The separator is 100 % efficient.
(vi) The refrigerant is superheated at the evaporator outlet (Li and Groll, 2005). Its effect is added to the model with the parameter of evaporator outlet superheat temperature difference which is the difference between the temperature of the superheated refrigerant at the evaporator outlet and evaporation temperature.
(vii) The process throughout the expansion valve is isenthalpic for both of the cycles.
(viii) Homogeneous equilibrium model (HEM) stating both thermodynamic and mechanical equilibrium is assumed establishing the process in the ejector.

First and second law analyses of the transcritical VCRC

Conservation of the energy is applied to all components of the transcritical VCRC ignoring kinetic and potential energy changes to evaluate the performance. The heat loss from the gas cooler, the heat gain of the evaporator, and the work input to the compressor are calculated on a unit-mass basis as follows;

$$q_H = (h_{in} - h_{out})_{gc} \tag{1}$$

$$q_L = (h_{out} - h_{in})_e \tag{2}$$

$$w_{in} = (h_{out} - h_{in})_{comp} \tag{3}$$

With the following isentropic efficiency correlation of the compressor (Brunin et al., 1997), it is possible to account for the effects of the selected refrigerants and the operation conditions.

$$\eta_{comp} = 0.874 - 0.0135\tau \tag{4}$$

The percentage expansion losses are calculated through the second law analyses. Flow (or stream) exergy by neglecting kinetic and potential energy terms is defined as (Çengel and Boles, 2007)

$$\psi = h - h_0 - T_0(s - s_0) \tag{5}$$

25 °C and 101.325 kPa is defined as the dead state. For a steady flow process, the general exergy balance in the rate form and the exergy balance for a single stream on a unit-mass basis are given as (Çengel and Boles, 2007)

$$\dot{X}_{dest} = \sum \dot{Q}_k \left(1 - \frac{T_0}{T_k}\right) - \dot{W}_{net} + \sum \dot{m}_{in}\psi_{in} - \sum \dot{m}_{out}\psi_{out} \tag{6}$$

$$x_{dest} = \sum q_k \left(1 - \frac{T_0}{T_k}\right) - w_{net} + \psi_{in} - \psi_{out} \tag{7}$$

Exergy destruction equations on a unit-mass basis are given for each component of the transcritical refrigeration cycle as (Dinçer and Kanoğlu, 2010)

$$x_{dest,comp} = T_0 (s_{out} - s_{in})_{comp} \tag{8}$$

$$x_{dest,gc} = q_H (T_0 / T_H) + T_0 (s_{out} - s_{in})_{gc} \tag{9}$$

$$x_{dest,\exp v} = T_0 (s_{out} - s_{in})_{\exp v} \tag{10}$$

$$x_{dest,e} = T_0(s_{out} - s_{in})_e - q_L(T_0 / T_L) \quad (11)$$

Governing equations of transcritical EERC

Modelling equations of the ejector and vapor-liquid separator are established in this section according to previously defined assumptions. Inputs of the model are gas cooler pressure, gas cooler outlet temperature, evaporator temperature, and evaporator outlet superheat temperature difference, and the efficiency values of the ejector sections. One of the assumptions is concerned with the pressure of the refrigerants at the inlet of the mixing section, also known as the suction chamber pressure. The governing equations for the CPM ejector are described with reference to Kornhauser (1990) who first established the model. Entrainment ratio which is the ratio of the secondary stream mass flow rate to the mass flow rate of the motive stream is assumed at the beginning and calculated iteratively. It is expected to be as high as possible for a well-designed ejector and formulated as follows;

$$w = \frac{\dot{m}_s}{\dot{m}_m} \quad (12)$$

Actual enthalpy and the velocity at the outlet of the motive nozzle as a result of the expansion of the primary fluid is formulated as given below;

$$h_{m,out} = (1-\eta_m)h_{m,in} + \eta_m h(s_{m,in}, P_b) \quad (13)$$

$$u_{m,out} = \sqrt{2(h_{m,in} - h_{m,out})} \quad (14)$$

The same equation set is described subsequently as a result of the expansion of the secondary flow in the suction nozzle as follows;

$$h_{s,out} = (1-\eta_s)h_{s,in} + \eta_s h(s_{s,in}, P_b) \quad (15)$$

$$u_{s,out} = \sqrt{2(h_{s,in} - h_{s,out})} \quad (16)$$

The pressure of P_b given in the thermodynamic relations is the optimum suction chamber pressure to obtain the maximum performance improvement. The following equations describe the mixing process occurring at constant pressure. Thus, P_b is equal to $P_{mix,out}$ at the exit of the mixing section. Mixing section efficiency is defined according to Eames et al. (1995).

$$u_{mix,out} = \left(\frac{w}{w+1}\right)u_{s,out}\eta_{mix} + \left(\frac{1}{w+1}\right)u_{m,out}\eta_{mix} \quad (17)$$

$$h_{mix,out} = \left(\frac{w}{w+1}\right)h_{s,in} + \left(\frac{1}{w+1}\right)h_{m,in} - \frac{u_{mix,out}^2}{2} \quad (18)$$

$$s_{mix,out} = s(h_{mix,out}, P_{mix,out}) \quad (19)$$

In some of the thermodynamic models, mixing section is assumed to be 100% efficient (Li and Groll, 2005; Bilir and Ersoy, 2009; Lawrence and Elbel, 2013; and etc.). However, in only a few of the studies, it is added to the thermodynamic models of the EERC (Li et al., 2014). In this study, the mixing section efficiency is added to the thermodynamic model as a parameter for more realistic conclusions to be derived.

The governing equations of the diffuser to recover the rest of the kinetic energy still available at the outlet of the mixing section are provided below

$$h_{d,out} = h_{mix,out} + \frac{u_{mix,out}^2}{2} \quad (20)$$

$$h_{d,out,isen} = h_{mix,out} + \eta_d \frac{u_{mix,out}^2}{2} \quad (21)$$

The outcomes to be calculated are given as

$$P_{d,out} = P(s_{mix,out}, h_{d,out,isen}) \quad (22)$$

$$r_{d,out} = r(P_{d,out}, h_{d,out}) \quad (23)$$

For a separator having efficiency equal to unity, the quality at the diffuser outlet and the entrainment ratio have the following relationship;

$$r_{d,out} = \frac{1}{w+1} \quad (24)$$

In the transcritical EERC, a vapor-liquid separator is used after the diffuser; hence saturated vapor is sent to the compressor and saturated liquid is sent to a small scale expansion valve before the evaporator. Thus, the enthalpies of the refrigerants are as follows;

$$h_{sep,l,out} = h(P_{d,out}, r_{sep,l,out} = 0) \quad (25)$$

$$h_{sep,v,out} = h(P_{d,out}, r_{sep,v,out} = 1) \quad (26)$$

COP of the transcritical VCRC and the EERC is defined as follows;

$$COP_{VCRC} = \frac{(h_{e,out} - h_{e,in})_{VCRC}}{(h_{comp,out} - h_{comp,in})_{VCRC}} \quad (27)$$

$$COP_{EERC} = w\frac{(h_{e,out} - h_{e,in})_{EERC}}{(h_{comp,out} - h_{comp,in})_{EERC}} \quad (28)$$

The performance improvement ratio is the ratio of the COP of the transcritical EERC to the COP of the base cycle and expressed as

$$R = \frac{COP_{EERC}}{COP_{VCRC}} \quad (29)$$

Model validation

Experimental validation is made with reference to two kinds of refrigerants, namely, R134a and R744, thereby verifying the model for both subcritical and transcritical EERC concepts, respectively. Experimental data of Ersoy and Sag (2014) is used for the model validation of the subcritical EERC. The difference between the numerical and experimental data of the diffuser outlet pressure and the entrainment ratio is within 10% error band as given in Table 2 and the agreement is found satisfactory. Section efficiency values of the ejector are added to the validation calculations with respect to thermodynamic analyses from the literature (Li and Groll, 2005; Lawrence, 2012).

Study of Liu et al. (2012) is utilized to obtain the experimental operation conditions of the transcritical EERC using CO_2 as the working fluid. This study is chosen intentionally since they provide with the experimental section efficiencies except for the diffuser. Two options are analyzed for the diffuser efficiency as 0.80 and 0.75 with reference to Li and Groll (2005) and Lawrence (2012), respectively as given in Table 3. Under these conditions, the maximum percentage error for the global parameters of the transcritical EERC utilizing CO_2 is around 10% and the agreement is satisfactory as in the previous case. Hence the thermodynamic model constructed with respect to CPM approach could be used in the numerical analyses of the transcritical EERC.

RESULTS AND DISCUSSION

Results are divided into two sections. First group is focusing on the parametric comparison of the refrigerants in the transcritical VCRC; whereas the second group is giving the comparisons of the modified cycle with the ejector expansion. Operation conditions are defined for a typical air conditioning application and given in Table 4.

As for the gas cooler outlet temperature, it is varied between 40-46 ºC for R744 and R170 and the range is defined between 46-49 ºC for R41. Since the critical temperature for R41 is around 44.1 ºC, it is necessary to select a higher temperature in order to be assured of staying in the transcritical refrigeration cycle region. That's why, the operation conditions are given in Table 4 for each refrigerant individually. For the rest of the parametric analyses, 40 ºC for R744 and R170 and 46 ºC for R41 are used as the reasonable gas cooler outlet temperatures for staying within the transcritical cycle region and calculating COP values making sense.

Table 2. Declaration of the percentage error between the experimental data provided by Ersoy and Sag (2014) and the numerical data for the diffuser outlet pressure and the entrainment ratio**.**

Validation Parameters	**Experimental Data**	**Numerical-1 ($\eta_m = \eta_s$=0.9 &η_d=0.8)**	**Error 1 (%)**	**Numerical-2 ($\eta_m = \eta_s$=0.8 &η_d=0.75)**	**Error 2 (%)**
w	0.6364	0.6617	**3.9793**	0.6511	**2.3176**
$P_{d,out}$ (kPa)	421.34	460.1837	**9.2191**	439.0198	**4.1961**

Table 3. Display of the percentage error between the numerical results and the experimental data of Liu et al. (2012) under two different diffuser efficiency assumptions.

Validation Parameters	**Experimental Data**	**Numerical-1 (η_d=0.8)**	**Error 1 (%)**	**Numerical-2 (η_d=0.75)**	**Error 2 (%)**
w	0.3889	0.3922	**0.8**	0.3932	**1.1**
$P_{d,out}$ (kPa)	4499	4998.9	**11.1**	4864.2	**8.1**

Table 4. Parametric operation ranges of the refrigerants according to their critical temperatures and pressures for a typical air-conditioning application.

Parameters	**Values**		
	R744	**R170**	**R41**
Gas cooler pressure (P_{gc})	8.5-10 MPa	5.5-12 MPa	6-12 MPa
Gas cooler outlet temperature ($T_{gc,o}$)	40 ˚C	40 ˚C	46 ˚C
Evaporator temperature (T_e)	5 ˚C	5 ˚C	5 ˚C
Evaporator outlet superheat temperature difference (ΔT_{sh})	5 ˚C	5 ˚C	5 ˚C
Primary nozzle efficiency (η_m)	0.9	0.9	0.9
Suction nozzle efficiency (η_s)	0.9	0.9	0.9
Mixing section efficiency (η_{mix})	0.9	0.9	0.9
Diffusor efficiency (η_d)	0.8	0.8	0.8

Parametric analyses of transcritical refrigeration cycle

All parameters are investigated with reference to gas cooler pressure since its value yielding the highest performance is dependent not only on the refrigerant, but also operation conditions. Each solid line in the rest of the following figures represents one configuration of the investigated parameter, i.e., variations in the gas cooler outlet temperature, evaporator temperature, evaporator outlet superheat temperature difference, based on the gas cooler pressure and the same color dashed line belongs to the counterpart of the same aspect. Figure 3 shows the effects of the gas cooler outlet temperature based on the gas cooler pressure.

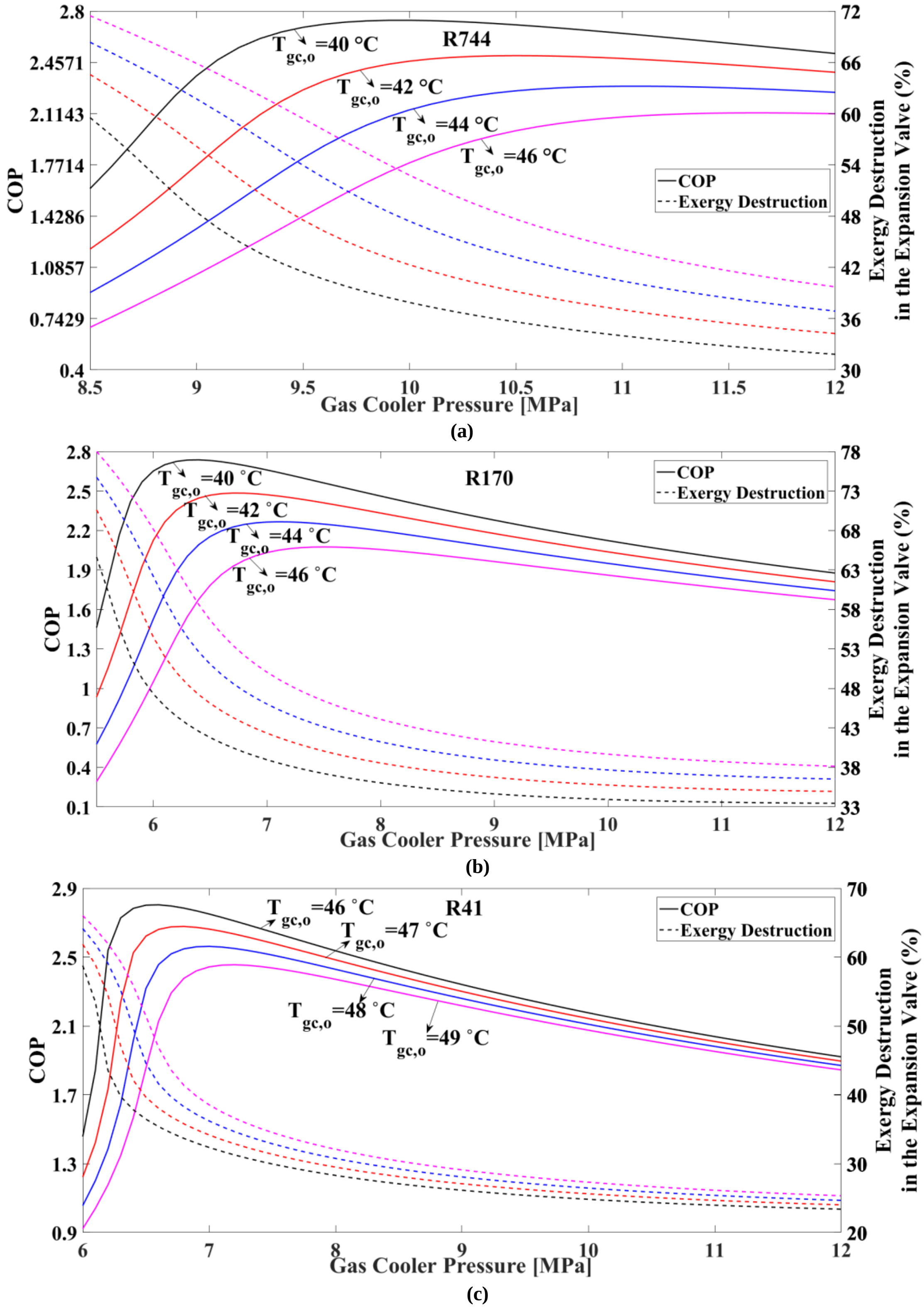


Figure 3. Effects of the gas cooler outlet temperature according to the gas cooler pressure on the performance and percentage expansion losses for R744 **(a)**, R170 **(b)**, and R41 **(c)**.

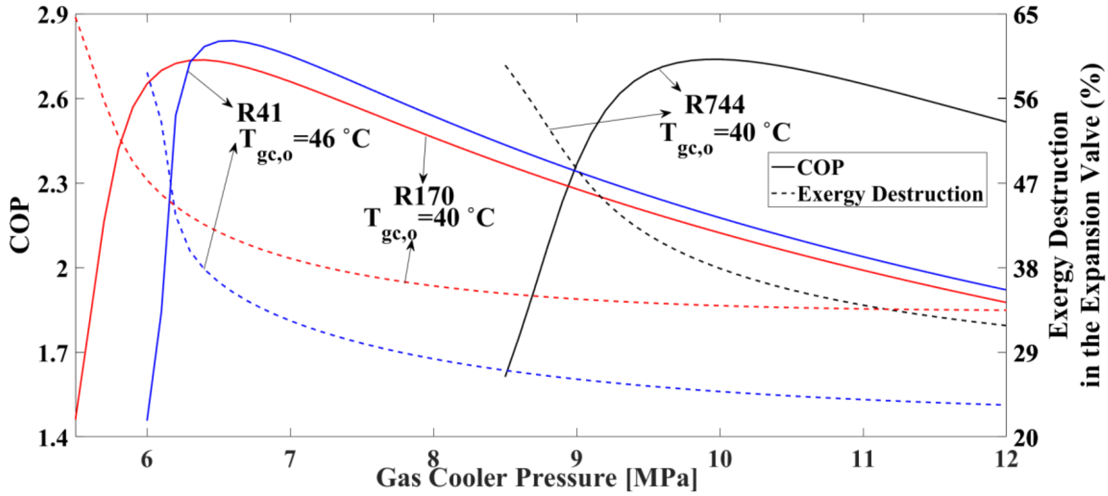


Figure 4. Comparisons of the COP, percentage expansion losses, and gas cooler pressure of the highest performance point for R744, R170, and R41 with respect to different gas cooler outlet temperatures at which they perform their highest performance.

Maximum performance curve is obtained at gas cooler outlet temperature of 40 ºC for R744 and R170 as shown in Figure 3 (a) and (b); whereas it is at 46 ºC for R41 as given in Figure 3 (c). Generally speaking, when gas cooler outlet temperature increases the performance of the refrigerants decrease and the percentage exergy destruction in the expansion valve increases.

As shown in Figure 3, the highest gas cooler pressure range at the refrigerants' best performance belongs to R744. When gas cooler outlet temperature increases, the gas cooler pressure yielding the highest performance increases as well for each refrigerant. Variation of change in the performance profile around this pressure point is more steep for R41 and R170 when compared to R744 with reference to investigated gas cooler outlet temperatures. Another interpretation from Figure 3 is that the performance variation around the gas cooler pressure yielding the highest performance becomes more gradual for all refrigerants as the gas cooler outlet temperature increases. Furthermore, the value of the gas cooler pressure resulting in the highest performance varies more as reaction to the changes in the gas cooler outlet temperature for R744 than R170 and R41.

As for the expansion losses, they decrease dramatically around the gas cooler pressure of the highest performance and starts stabilizing after this pressure point is achieved. In other words, rate of change of expansion losses have a different tendency before and after the operation pressure yielding the maximum performance for a specific gas cooler outlet temperature. When the gas cooler outlet temperature increases, the percentage exergy destruction in the expansion valve as well increases. The gas cooler outlet temperature affects the expansion losses mostly for R744 and R170 rather than R41. Within the investigated ranges of gas cooler pressure and outlet temperature, R744 and R170 experience very high expansion losses around 30-40%; whereas it stabilizes around 23-25% for R41. Although the improvement potentials of these three refrigerants are different in the EERC due to the variations of the percentage expansion losses based on the investigated operation ranges, they are all promising candidates to be investigated in the transcritical cycle with the ejector expansion.

Figure 4 presents the general overview related to these three refrigerants for the operation points at which they display their highest performance. Operation pressure of R41 and R170 resulting in the highest performance are very close to each other at the gas cooler outlet temperatures yielding their own maximum performance. Moreover, within the investigated operation conditions, maximum performance is calculated for R41.

The effect of the evaporator temperature on the cycle performance and percentage expansion losses are displayed in Figure 5 (a), (b), and (c) for R744, R170, and R41, respectively based on the gas cooler pressure as in the same way of gas cooler outlet temperature investigation. Standard comments regarding the general trends of the percentage exergy destruction in the expansion valve and the cycle COP could be interpreted from these plots.

Gas cooler pressure yielding the highest performance with respect to evaporation temperature nearly stays the same for each refrigerant in contrast to the variations observed in the gas cooler outlet temperature analyses. When evaporator temperature increases, the refrigerants display better performance while they experience less expansion losses as shown in Figure 5. The percentage exergy destruction of each refrigerant in the expansion valve is calculated very similar to each other and stabilized around a constant value for the investigated evaporation temperatures. The stabilized percentages could be expressed roughly as 32 %, 33%, and 23% for R744, R170, and R41, respectively. Hence, it is obvious that R744 and R170 has the highest improvement potentials in the transcritical EERC. The effect of the last operational parameter, evaporator outlet superheat temperature difference, is shown in Figure 6 (a), (b), and (c) for R744, R170, and R41, respectively.

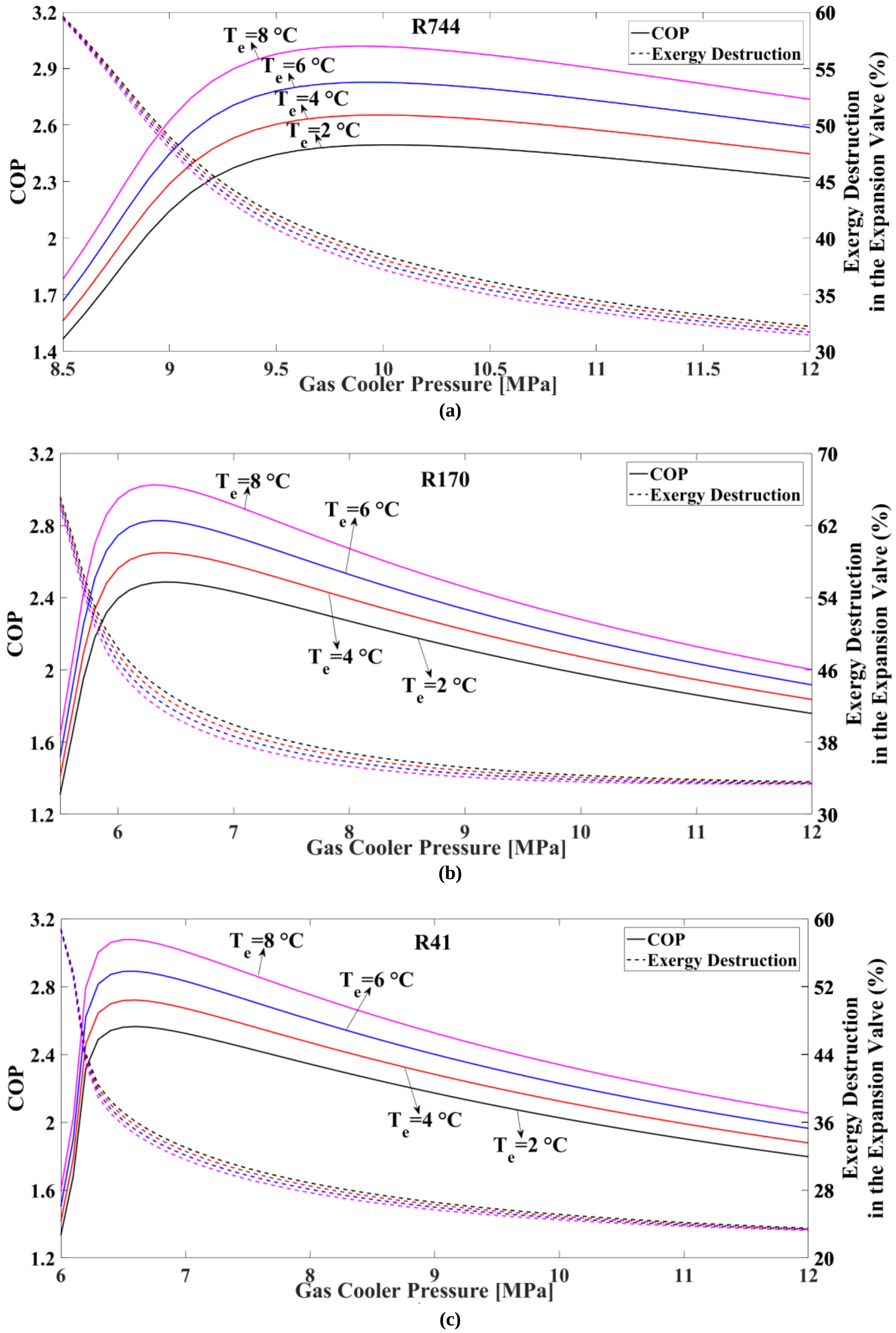
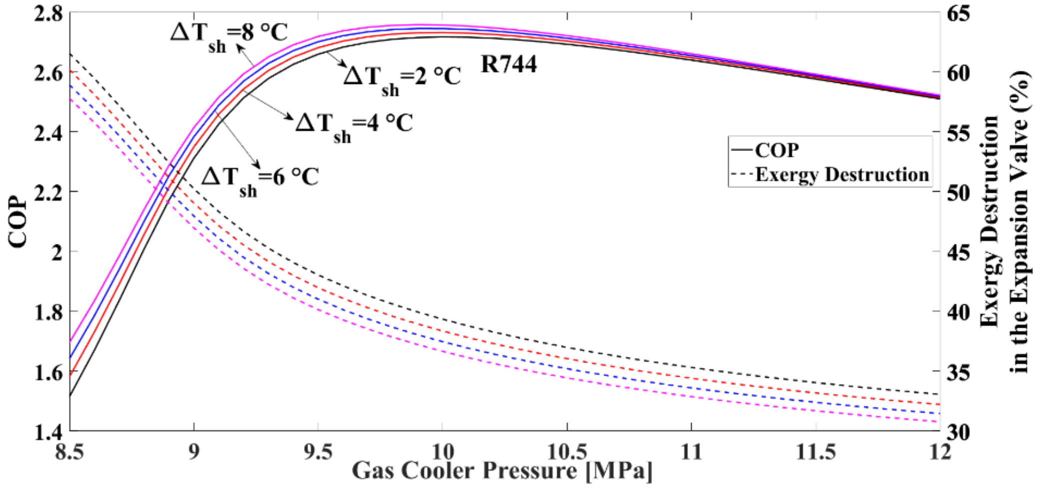
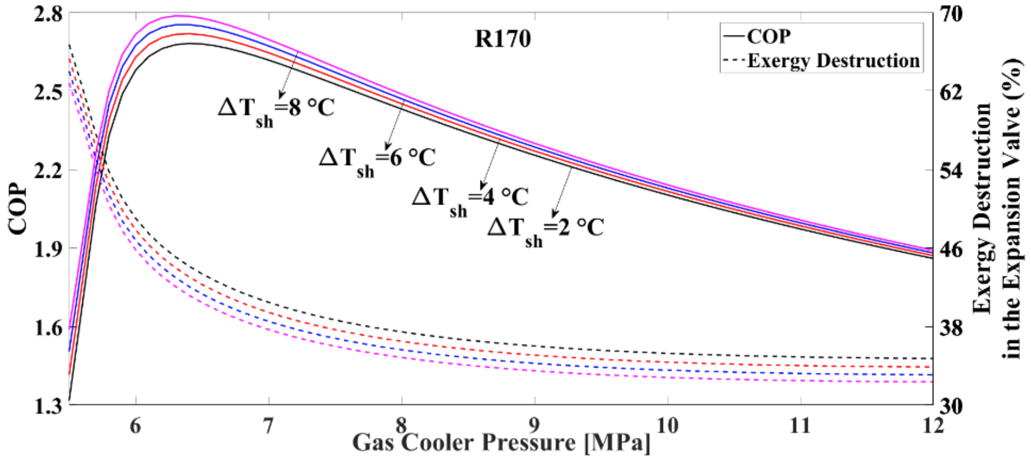


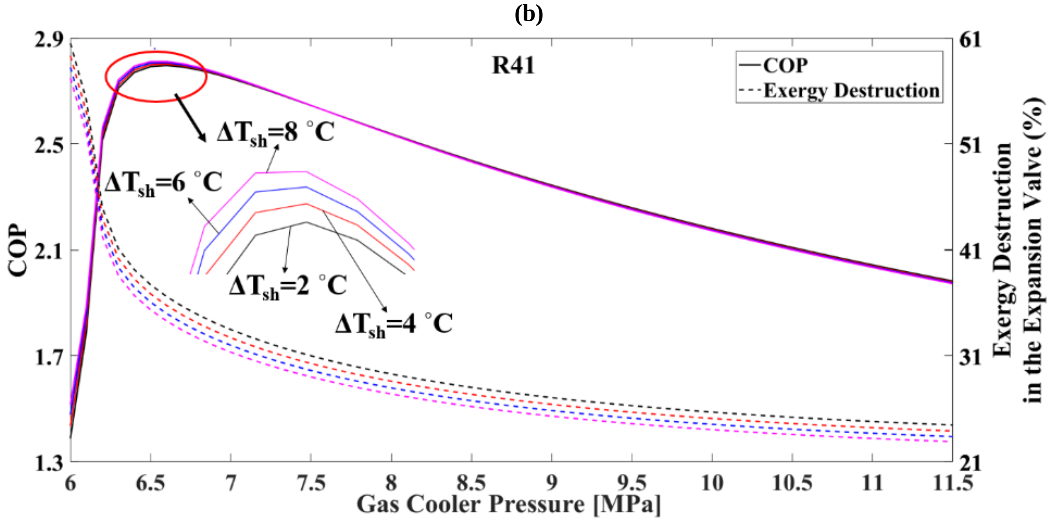
Figure 5. Effects of the evaporator temperature on the performance and percentage expansion losses for R744 **(a)**, R170 **(b)**, and R41 **(c)** with respect to the gas cooler pressure.



(a)



(b)



(c)

Figure 6. Effects of the evaporator outlet superheat temperature difference on the performance and percentage expansion losses based on the gas cooler pressure for R744 **(a)**, R170 **(b)**, and R41 **(c)**.

When evaporator outlet superheat temperature difference increases, the performance increases and the percentage exergy destruction in the expansion valve decreases. When compared to R744 and R170, R41 is the least sensitive refrigerant to the changes in this parameter and has less expansion losses. As shown in Figure 5 and Figure 6, the least effective parameter is the evaporator outlet superheat temperature difference in terms of the performance; whereas it is the evaporator temperature for the percentage expansion losses. All in all, the most important parameters are the gas cooler pressure and outlet temperature on the performance variations and expansion losses.

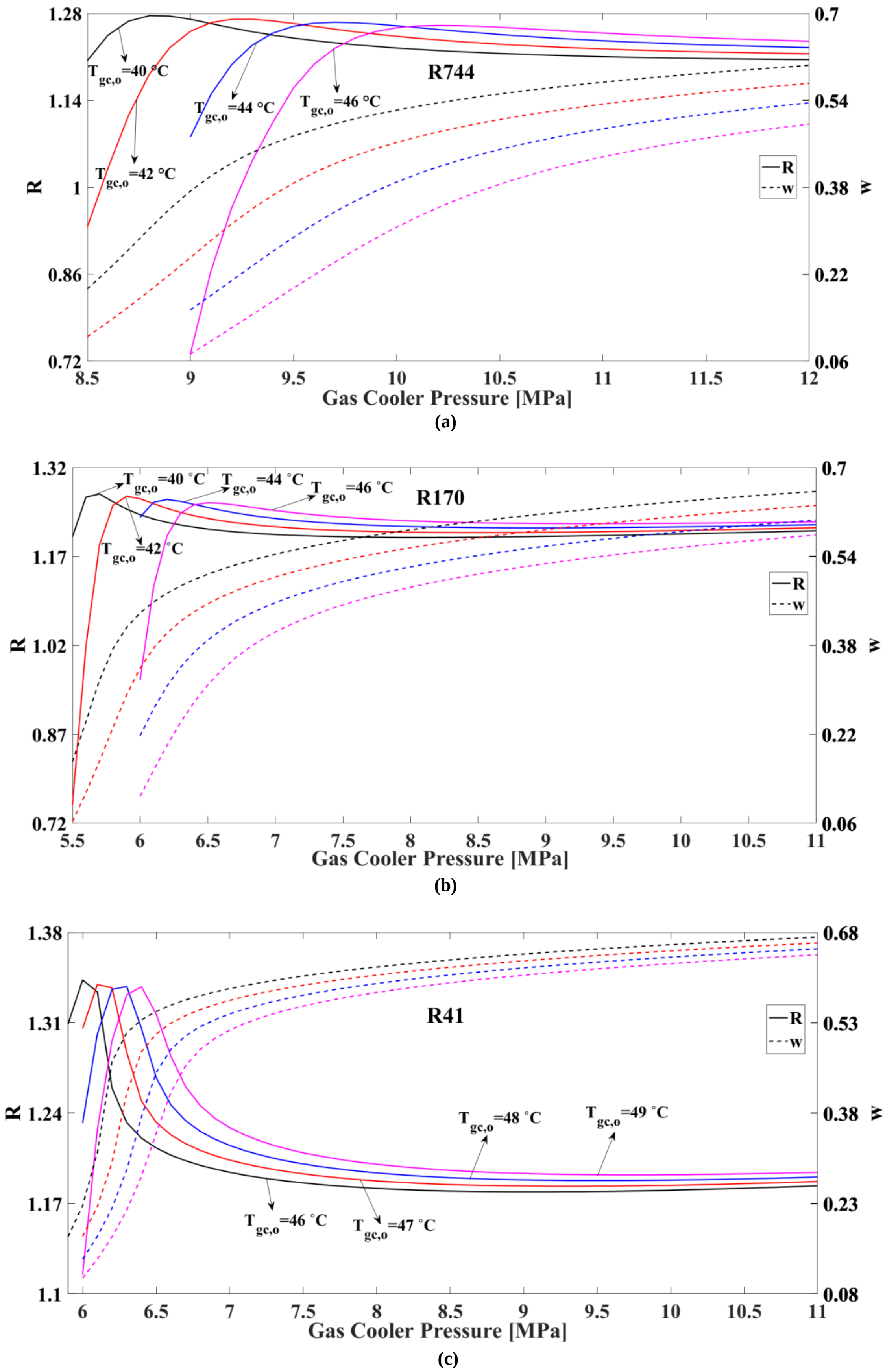


Figure 7. Effects of the gas cooler outlet temperature based on the gas cooler pressure on the optimum performance improvement ratio and entrainment ratio for R744 **(a)**, R170 **(b)**, and R41 **(c)**.

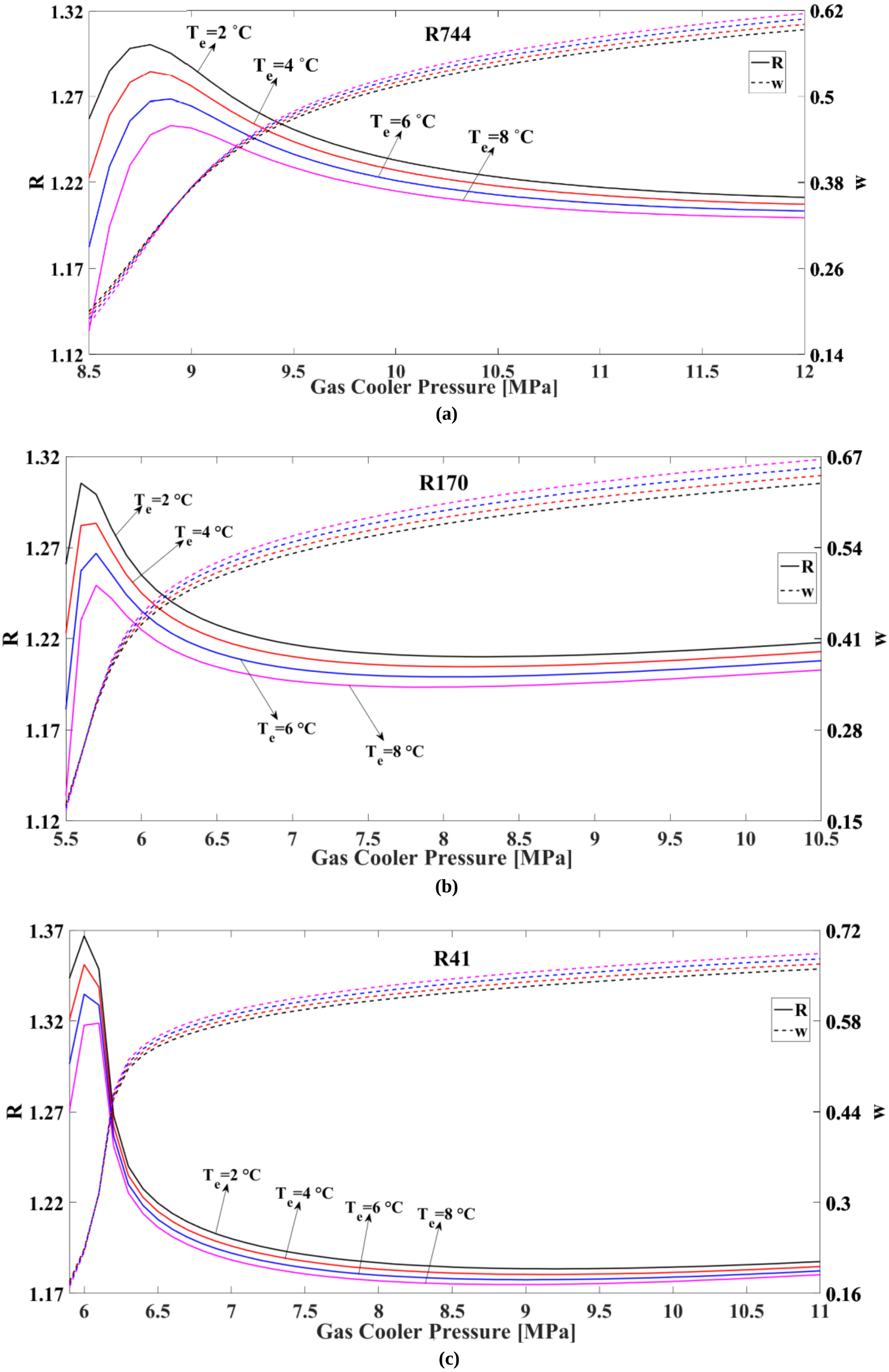


Figure 8. Effects of the evaporator temperature based on the gas cooler pressure on the optimum performance improvement ratio and entrainment ratio for R744 **(a)**, R170 **(b)**, and R41 **(c)**.

Performance evaluation of the transcritical EERC

In terms of the expansion losses, the most effective parameters are the gas cooler outlet temperature and the evaporator outlet superheat temperature with reference to analyses based on the gas cooler pressure. However, the similar amounts of the expansion losses are calculated according to evaporator temperature and evaporator outlet superheat temperature difference for each refrigerant. Rest of the analyses concerned with the performance improvement potentials in the transcritical EERC are conducted according to the most important critical parameters of the performance based on the gas cooler pressure as in the same way of the previous analyses. As stated before, mixing section efficiency is added to the model for more realistic performance improvement estimations. Optimum performance improvement ratio and entrainment ratio is calculated iteratively based on the optimum suction chamber pressure (P_b) yielding the maximum performance.

Figure 7 displays the performance improvement potential of the refrigerants in the transcritical EERC when gas cooler outlet temperature is changed with respect to gas cooler pressure. Higher gas cooler outlet temperatures result in higher performance improvement ratios due to the higher expansion losses at those ranges for all refrigerants.

After the gas cooler pressure yielding the maximum performance improvement ratio is achieved for each gas cooler outlet temperature, a decrease is observed in the performance improvement ratio profile for R744 and R170. On the other hand, the decrease is more dramatic for R41 after the gas cooler pressure of the maximum performance point. The performance improvement ratio is stabilized more or less around a specific interval which are 20%-24%, 21%-23%, and 18%-19% for R744, R170, and R41, respectively for a wide range of the gas cooler pressures.

R170 and R744 display better performance improvement potential in the transcritical EERC within the stabilized region of the performance improvement ratio. Moreover, when gas cooler outlet temperature increases, the entrainment ratio decreases for each refrigerant within the same investigated gas cooler pressure range. Trend of the entrainment ratio for each gas cooler outlet temperature curve displays an increasing profile with respect to gas cooler pressure increments. R744 and R170 exhibit more gradual increase in the entrainment ratio in contrast to the dramatic increase of the entrainment ratio calculated for R41 around their own gas cooler pressures resulting in the highest performance improvement ratio.

Figure 8 is for the investigation of the evaporator temperature effects on the optimum performance improvement ratio and the entrainment ratio in the transcritical EERC utilizing R744, R170, and R41. Performance improvement ratio increases as the evaporator temperature is decreased. However, entrainment ratio decreases as a result of the decrease in the evaporator temperature for a specific gas cooler pressure range. For each evaporator temperature analysis, the entrainment ratio increases as reaction to increases in the gas cooler pressures. After the operation pressure of the highest performance point for each evaporator temperature configuration, there is a decrease in the performance improvement ratios of all refrigerants.

The gas cooler pressures yielding the best performance improvement ratios according to variations in the evaporator temperatures nearly stay the same for each refrigerant. The rate of decrease in the performance improvement ratio profile for each evaporator temperature is more sudden in comparison with the corresponding profile of the gas cooler outlet temperatures after the the best performance pressure point is achieved at the analyzed conditions. Among these three refrigerants, R41 performs the most sudden decrease in the performance improvement ratio and the most dramatic increase in the entrainment ratio after the gas cooler pressure of the maximum performance improvement ratio for each evaporator temperature. The performance improvement potential stabilizes around 22-20% for R744 and R170; whereas it is about 18% for R41 at various evaporator temperature configurations.

As it is seen from Figure 7 and 8, around the operation pressure of the highest perfromance for each refrigerant, there is a region that the entrainment ratio is very low, and the transcritical EERC is not applicable actually. These kinds of performance lines for various operation conditions and refrigerants could be beneficial to display the actual operation ranges of the refrigerants in the EERC.

CONCLUSION

In conclusion, the objectives of this study are divided into two branches, i.e., comparing the environmentally-friendly refrigerants having low critical temperatures, namely R744, R170, and R41, parametrically in the transcritical refrigeration cycle and calculating the improvement potential of these high-pressure refrigerants in the modified cycle with ejector expansion. Firstly, the effects of the gas cooler outlet temperature, evaporator temperature, evaporator outlet superheat temperature difference on the performance and percentage expansion losses with respect to the specific gas cooler pressure ranges are investigated for these three refrigerants in the transcritical refrigeration cycle.

Gas cooler pressure yielding the highest performance is dependent sensitively on the gas cooler outlet temperature rather than the evaporator and evaporator outlet superheat temperature difference in the transcritical cycle. The most effective parameters are gas cooler outlet temperature and the evaporator temperature in terms of the performance; whereas the gas cooler outlet temperature and evaporator outlet superheat temperature difference according to the percentage exergy destruction in the expansion valve as a result of the analyses based on a specific gas cooler pressure range for each refrigerant. When all these refrigerants are compared according to their performance values at the gas cooler outlet temperatures yielding their own highest COP, it is seen that the best performance is calculated for R41 at a low gas cooler pressure.

The rest of the analyses relative to EERC configuration is conducted with reference to two critical parameters which are effective mostly on the performance. The performance improvement potential of the refrigerants is different from each other and depends on the operation conditions. To sum up, thermodynamic analyses show that the improvement potential in the EERC is higher for R744 and R170 than R41 with respect to the investigated parameters.

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Ayşe Uğurcan ATMACA

Ayşe Uğurcan Atmaca is a research assistant and Ph.D. student at Dokuz Eylül University, Mechanical Engineering Department, Thermodynamics division since 2014. She graduated from İzmir Institute of Technology, Mechanical Engineering Department in 2011. She received her M.Sc. degree from Dokuz Eylül University, Mechanical Engineering Department, Thermodynamics division in 2013. Between 2011-2014, she worked as a development engineer at Bosch Termoteknik Isıtma ve Klima Sanayi Ticaret A.Ş. She is a member of Turkish Society of Thermal Sciences and Technology and the Chamber of Mechanical Engineers. Her special area of interest includes combi boiler type heating appliances, flow visualization, ejector expansion refrigeration cycle, and natural convection.

Aytunç EREK

Dr. Aytunç Erek is a professor in the Department of Mechanical Engineering at Dokuz Eylül University in İzmir, Turkey. He received his Ph.D. in the area of thermodynamics from Dokuz Eylül University in 1999. His main research interest include CFD analysis of heat exchangers and thermal energy storage systems. He is a member of the Turkish Association of HVAC Engineers and the Chamber of Mechanical Engineers. He has published several refereed journal and conference papers in the area of thermodynamics and heat transfer.

Orhan EKREN

Dr. Orhan Ekren is Assoc.Prof. in the Solar Energy Institute at Ege University**.** He received the M.Sc. degree from the Department of Energy Engineering, Izmir Institute of Technology, Turkey, and Ph.D. degree from the Department of Mechanical Engineering (Thermodynamics), Turkey. He has been working at Ege University with the 15 years of work experience. He studied as a Postdoc researcher in Southern Illinois University, USA as well. His main research interests are energy efficiency, energy saving, capacity modulation on refrigeration and HVAC&R systems, renewable heating and cooling, magnetic cooling, integration of renewable energy systems into buildings, optimum sizing of hybrid renewable

energy sources (wind/solar etc.), also sustainability in energy and buildings. Dr. Ekren has authored and co-authored 25 international journal papers, 25 international conference papers, 5 books and book chapters and also 2 utility models on efficient operation of HVAC&R systems. Therefore, system design and experimental tests and demo studies are in his research scope. He has currently involved several research project funded by national research council (TUBITAK) and EC-FP7 project (ECOSHOPPING).

Mustafa Turhan ÇOBAN

Mustafa Turhan Çoban, borned in Bolu, Seben, Turkey in 1957. He received his BSc degree from Ege University, faculty of mechanical engineering, mechanical engineering department in 1978; his M.Sc. degree in Mechanical Engineering from Michigan Technological University (USA) in 1982. He graduated from University of Utah (USA) in 1986 with a Ph.D. in Mechanical Engineering. In 1995, he received postgraduate degree in computer science from Victoria Technological University (Australia). He worked as a lecturer and researcher at University of Nebraska, Lincoln (USA), Victoria Technological University (Australia), Ballarat University (Australia), Dokuz Eylül University, Gebze Institute of Technology, Ege University. He worked at Aras compressors (Turkey), Mineral Research and Exploration Institute of Turkey, TUBITAK (Turkish Technological and Scientific Research Foundation) Institute of Energy, Imperial Chemical Industries (Australia), Ceramic Fuel Cells Limited (Australia) as an engineer/senior engineer/researcher; TUBITAK UME (National Metrology Institute of Turkey) as technical vice-chairman.