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Theoretical Study on A Modified Subcooling Vapor-compression Refrigeration Cycle Using Hydrocarbon Mixture R290/R600a

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ABSTRACT

In this study, a modified subcooling vapor-compression refrigeration cycle (MSVRC) using refrigerant mixture R290/R600a was proposed for applications in refrigerator-freezers. In the MSVRC, a phase separator is utilized to split the refrigerant mixture and obtain refrigerants at two different mass fractions after partial condensation in the condenser. Moreover, an internal heat exchanger is introduced to enhance the overall system performance. Energetic and exergetic analysis methods were used to theoretically evaluate the system operating performance of MSVRC and compared with the conventional vapor-compression refrigeration cycle (CVRC). The simulation results show that the MSVRC outperforms CVRC in terms of coefficient of performance (COP) and exergy efficiency. Under the given operating condition, the COP and exergy efficiency of the MSVRC can be improved up to 5.27% and 11.4%, respectively, as compared to those of CVRC. The system performance characteristics of the proposed cycle demonstrate its potential advantages for application in domestic refrigerator-freezers.

Keywords: Refrigeration cycle; Hydrocarbon mixture; Internal heat exchanger; Performance improvement

1. INTRODUCTION

Due to environmental concerns, there has been an increasing universal interest to apply environment-friendly refrigerants to various vapor-compression refrigeration systems over the past years (Bolaji and Huan, 2013; Sarbu, 2014). Hydrocarbons (HCs) and their mixtures are considered as a replacement for HCFCs and HFCs since they have zero ozone depletion potential (ODP) and relatively low global warming potential (GWP) (Granryd, 2001; Harby, 2017). Some hydrocarbons, such as R600a and R290, have been approved for actual use in some countries in small refrigeration systems like refrigerators and freezers (Palm, 2008). The potential applications of these pure hydrocarbons are numerous when the refrigerant charge in the refrigeration systems is limited in the range of allowable safety amount. On the other hand, the use of hydrocarbon mixtures as refrigerants has also attracted wide interest. Hydrocarbon mixtures are the zeotropic refrigerants which have greater potential for improvements in energy efficiency and capacity modulation of refrigeration systems (Mohanraj *et al.*, 2009). Thus, many relevant investigations on hydrocarbon mixtures applied in different small refrigeration systems were reported in recent years.

Dalkilic and Wongwises (2010) conducted a theoretical study on a conventional vapor-compression refrigeration system with several binary refrigerant mixtures including R290, R600, and R600a, and investigated the effects of the main parameters on performance. Rasti *et al.* (2012) experimentally investigated the effect of using different charges R436A (a mixture of R290/R600a) as a refrigerant in a domestic refrigerator and showed that R436A appears to be a suitable replacement for R134a. Yu and Teng (2014) analyzed the use of mixture R290/R600a with different mixed mass ratios in a small refrigerator and indicated that replacing R134a with HC mixtures is feasible and can obtain higher energy factor. Yoon *et al.* (2012) carried out a thermodynamic analysis on a Lorenze-Meutzner cycle with hydrocarbon mixtures for a domestic refrigerator-freezer and confirmed that the energy consumption of the LM cycle using R290/R600 (40:60%) was reduced in comparison with that of the bypass two-circuit cycle using R600a. He *et al. (2014)* have reported applications of R290/R600a in a large capacity chest freezer and showed that the power consumption can be lowered. In addition, d'Angelo *et al.* (2016) presented a performance evaluation of a vapor injection refrigeration system using a mixture refrigerant R290/R600a and claimed that COP of the vapor injection refrigeration cycle is 16–32% greater than the one of a vapor compression cycle. Overall, the interest in the use of hydrocarbon mixtures in refrigerator-freezers and other vapor-compression systems has grown continuously. Principally, energy savings of actual refrigeration systems are directly related to the performance of the relevant refrigeration cycle. The conventional vapor-compression refrigeration cycle (CVRC) is already well known in literature and in industry. However, in order to improve the cycle performance, some modifications to the basic CVRC are employed (Domanski *et al.*, 2014). Typically, one of these modifications is the use of a suction line heat exchanger (SLHX) or internal heat exchanger (IHX), i.e. so-called subcooling cycles (Mota-Babiloni *et al.*, 2015; Park *et al.*, 2015). In the subcooling cycles, the refrigerant at the condenser outlet is subcooled by an additional heat exchanger, which can potentially improve the coefficient of performance. Therefore, using subcooling cycles to increase performances of vapor-compression refrigeration systems is an effective method to save energy and increase efficiency (Hermes, 2013; Qureshi and Zubair, 2013; She *et al.*, 2014).

In this study, a modified subcooling vapor-compression refrigeration cycle (MSVRC) using zeotropic mixture R290/R600a was proposed. As compared to the CVRC, the MSVRC adopts a condenser unit with phase separation to obtain the components separation for the mixture R290/R600a and fabricates two circuits with R290-rich mixture and R600a-rich mixture. In the MSVRC, this portion of the R290-rich mixture is used to realize refrigeration effect, whereas the portion of the R600a-rich mixture is used to cool down the R290-rich mixture through an IHX and resulting in the increased subcooling degree. This case could be more useful for improving the cycle performance. The objective of the present study is to theoretically evaluate the performance characteristics of the MSVRC. The effects of the main parameters on the performance have been studied. In addition, the performance of the MSVRC is also compared with that of the CVRC.

2. CYCLE DESCRIPTION AND MODELING

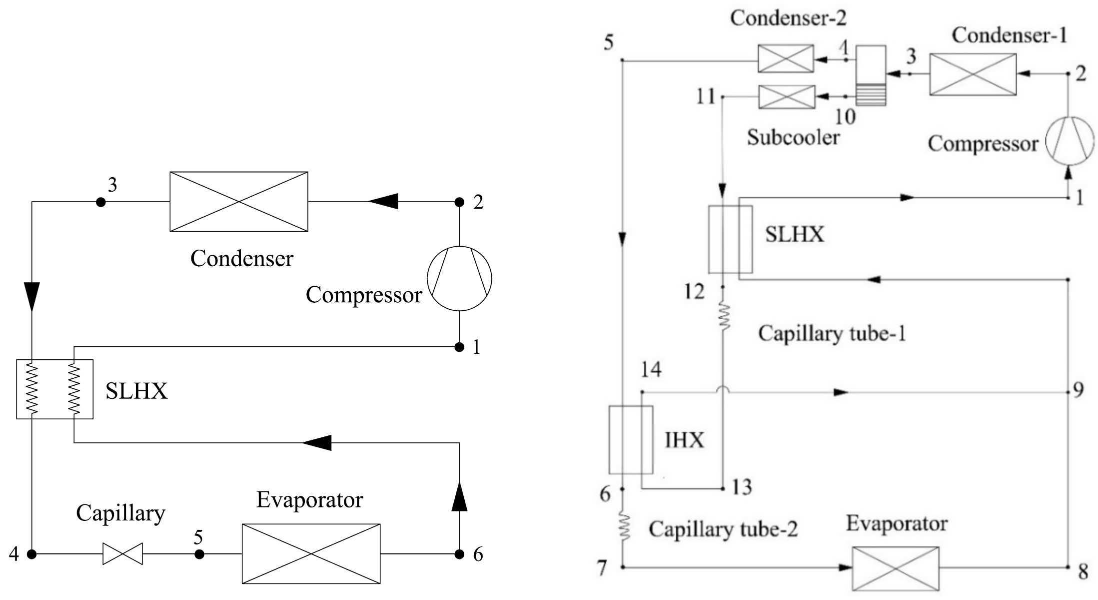
Fig. 1 shows the schematic diagrams of two cycle systems, i.e. CVRC and MSVRC systems. Fig. 2 shows the working process of MSVRC on pressure-enthalpy diagram. The CVRC system consists of a compressor, a condenser, a suction line heat exchanger (SLHX), an expansion valve and an evaporator. The MSVRC are based on CVRC with the addition of an internal heat exchanger (IHX), a phase separator and an expansion valve as well as the condenser comprising three portions (condenser-1, condenser-2, subcooler). The refrigeration process of MSVRC is described as follows: the vapor refrigerant mixture R290/R600a (state 1) is compressed to superheated refrigerant vapor (state 2) and then enters the condenser-1 to obtain partial condensation. The two-phase fluid (state 3) leaving condenser-1 is split into two different composition steams by the phase separator. The R600a-rich saturated liquid refrigerant (state10) is further subcooled in the subcooler (state 11) and flows into the IHX through SLHX, and capillary tube-1 (process 11-12-13). Then, this refrigerant is evaporated to the superheated vapor refrigerant (state 14). On the other hand, the R290-rich saturated vapor refrigerant (state 4) is totally condensed to saturated liquid (state 5) in the air-cooled condenser-2 and is further cooled in the IHX to subcooled liquid (state 6). Then, the refrigerant from IHX enters the evaporator after a throttled process (state 6-7) in the capillary tube-2 and achieves the useful refrigeration effect during the vaporization process. The saturated vapor refrigerant (state 8) at the evaporator outlet is mixed with the refrigerant from IHX (state 14). Finally, the mixing refrigerant (state 9) flows into the SLHX to become the superheated vapor (state 9) and returns to the compressor. In this way, the entire refrigeration cycle of MSVRC is completed. The mass fraction of working fluid changes with the system's operating condition. Table 1 shows the mass fraction of working fluid under the fixed operating condition: the evaporation temperature -32 ºC, R290 mass fraction at the compressor suction port 0.5, the condensation temperature 40 ºC, and the subcooling degree in SLHX outlet is 20 K. Generally, the use of the phase separator and IHX can bring large subcooling of refrigerant entering the evaporator and take good advantage of the zeotropic refrigerant's temperature glide attribute, leading to higher evaporation pressure and enhancement of the system performance.

Table 1: Compositions of state points under the given operation condition

State point	**State 1-3**	**State 4**	**State 5-8**	**State 9**	**State 10-14**
R290/R600a mass fraction	0.5/0.5	0.37/0.63	0.54/0.46	0.5/0.5	0.37/0.63

The performances of MSVRC are theoretically evaluated based on the energetic and exergetic methods. The following assumptions are made to simplify the analysis:

(1) All components are assumed to be a steady-state and steady-flow process;
(2) The compression process in the compressor is irreversible and has variable isentropic efficiency;
(3) The throttling processes in expansion valve are isenthalpic;
(4) The vapor and liquid from the phase separator are completed separated and saturated;
(5) The evaporator outlet vapor and condenser outlet liquid are both saturated; and
(6) Refrigerant pressure drops and heat losses in the cycle are neglected.



(a) CVRC (b) MSVRC

Figure 1: The schematic diagrams of CVRC and MSVRC

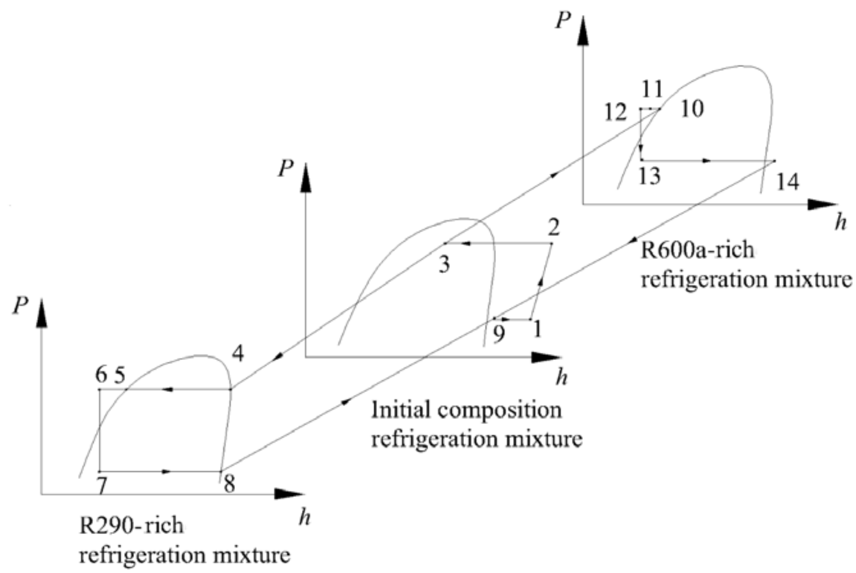


Figure 2: The pressure-enthalpy diagram of MSVRC

Based on the above assumptions, the following equations for main components can be obtained in terms of the mass and energy conservation. The system refrigeration capacity can be obtained by Equation (1).

$$\dot{Q}_e = \dot{m}(1-x_3)(h_8-h_7) \tag{1}$$

Volumetric refrigeration capacity is given by Equation (2).

$$q_{\mathrm{ev}} = \frac{\dot{Q}_e}{\dot{m}\ v_1} \tag{2}$$

where v_1 is specific volume of the compressor suction gas.

The compressor consumption power is expressed as Equation (3).

$$W_{\mathrm{c}} = \dot{m}(h_2 - h_1) = \dot{m}\frac{h_{2\mathrm{s}} - h_1}{\eta_{\mathrm{s}}} \tag{3}$$

where $h_{2\mathrm{s}}$ is the refrigerant specific enthalpy at the compressor outlet under the isentropic compression process; η_{s} is the compressor isentropic efficiency, given as Equation (4) (Elakdhar *et al.*, 2007).

$$\eta_{\mathrm{s}} = 0.874 - 0.0135\frac{P_2}{P_1} \tag{4}$$

where P_1 and P_2 is the refrigerant pressure at the inlet and outlet of the compressor, respectively.

The coefficient of performance (COP) of the cycle can be obtained by Equation (5).

$$\mathrm{COP} = \frac{\dot{Q}_e}{W_{\mathrm{c}}} \tag{5}$$

The energy balance in the SLHX and IHX can be expressed as Equations (6) and (7), respectively.

$$\dot{m}(h_1 - h_9) = \dot{m}(1 - x_3)(h_{11} - h_{12}) \tag{6}$$

$$\dot{m}(1 - q_3)(h_{14} - h_{13}) = \dot{m}x_3(h_5 - h_6) \tag{7}$$

The energy balance equation in two different composition refrigerants mixing process at state point 9 is calculated by Equation (8).

$$\dot{m}(1 - x_3)h_{14} + \dot{m}x_3h_8 = \dot{m}h_9 \tag{8}$$

The exergy analysis method is presented to reveal potential thermodynamic improvements of MSVRC. Based on the assumption that kinetic and potential energy changes are negligible, the exergy destruction for MVRC system components can be obtained as follows:

For the compressor, the exergy destruction is given by Equation (9).

$$\dot{E}x_{\mathrm{d,com}} = \dot{m}(h_1 - h_2) - T_0\left(s_1 - s_2\right) + W_{\mathrm{c}} = \dot{m}T_0\left(s_2 - s_1\right) \tag{9}$$

where the reference temperature T_0 and the reference pressure P_0 are 298.15K, 101.325kPa, respectively.

For the condensers and subcooler, the exergy destruction is expressed as Equations (10-12), respectively.

$$\dot{E}x_{\mathrm{d,con\text{-}1}} = \dot{m}[(h_2 - h_3) - T_0\left(s_2 - s_3\right)] \tag{10}$$

$$\dot{E}x_{\mathrm{d,con\text{-}2}} = \dot{m}[(h_4 - h_5) - T_0\left(s_4 - s_5\right)] \tag{11}$$

$$\dot{E}x_{\mathrm{d,sub}} = \dot{m}(1 - x_3)[(h_{10} - h_{11}) - T_0\left(s_{10} - s_{11}\right)] \tag{12}$$

For the SLHX and IHX, the exergy destruction is given as Equations (13) and (14), respectively.

$$\dot{E}x_{\mathrm{d,SLHX}} = \dot{m}(1 - x_3)[(h_{11} - h_{12}) - T_0\left(s_{11} - s_{12}\right)] + \dot{m}[(h_9 - h_1) - T_0\left(s_9 - s_1\right)] \tag{13}$$

$$\dot{E}x_{\mathrm{d,IHX}} = \dot{m}(1 - x_3)[(h_{13} - h_{14}) - T_0\left(s_{13} - s_{14}\right)] + x_3\dot{m}[(h_5 - h_6) - T_0\left(s_5 - s_6\right)] \tag{14}$$

For the two capillary tubes, the exergy destruction is calculated as Equations (15-16), respectively.

$$Ex_{\mathrm{d,cap\text{-}1}} = \dot{m}(1-x_3)[(h_{12}-h_{13})-T_0\left(s_{12}-s_{13}\right)] = \dot{m}(1-x_3)T_0\left(s_{13}-s_{12}\right) \quad (15)$$

$$\dot{E}x_{\mathrm{d,cap\text{-}2}} = x_3\dot{m}[(h_6-h_7)-T_0\left(s_6-s_7\right)] = x_3\dot{m}T_0\left(s_7-s_6\right) \quad (16)$$

For the evaporator, the exergy destruction is calculated according to the literatures in Equation (17) (Dopazo *et al.*, 2009; Sarkar *et al.*, 2004).

$$\dot{E}x_{\mathrm{d,eva}} = \dot{m}(1-x_3)[(h_7-h_8)-T_0\left(s_7-s_8\right)] + Q_{\mathrm{e}}(1-\frac{T_0}{\bar{T}_{\mathrm{e}}}) = \dot{m}(1-x_3)[T_0\left(s_8-s_7\right)-(h_8-h_7)\frac{T_0}{\bar{T}_{\mathrm{e}}}] \quad (17)$$

where $\bar{T}_{\mathrm{e}}$ is the refrigerant thermodynamic average temperature.

The total exergy destruction of the MVRC system can be obtained by Equation (18).

$$\dot{E}x_{\mathrm{d,tot}} = \dot{E}x_{\mathrm{d,com}} + \dot{E}x_{\mathrm{d,con\text{-}1}} + \dot{E}x_{\mathrm{d,con\text{-}2}} + \dot{E}x_{\mathrm{d,SLHX}} + \dot{E}x_{\mathrm{d,IHX}} + \dot{E}x_{\mathrm{d,sub}} + \dot{E}x_{\mathrm{d,cap\text{-}1}} + \dot{E}x_{\mathrm{d,cap\text{-}2}} + \dot{E}x_{\mathrm{d,eva}} \quad (18)$$

The exergy efficiency of the MSVRC system is given by Equation (19).

$$\eta_{\mathrm{ex}} = 1-\frac{\dot{E}x_{\mathrm{d,tot}}}{W_{\mathrm{c}}} \quad (19)$$

As the detail CVRC model is well known, it is not presented here to keep this paper concise. However, the relevant simulations for CVRC are performed for a comparison with MSVRC. The simulation program is written in Fortran Language, and the required refrigerant properties are calculated by using the property subroutines of REFPROP 8.0 (Lemmon *et al.*, 2007).

3. RESULTS AND DISSCUSION

The performances of the CVRC and MSVRC were evaluated by energy and exergy method under the following operation conditions: the evaporation temperature T_{e} (i.e., the temperature at the evaporator inlet T_7) ranges from -40 ºC to -24 ºC; the R290/R600a zeotropic mixture is in the composition range of 0.3-0.7; the condensation temperature T_{c} (i.e., the temperature at the condensation-2 outlet T_5) ranges from 50 ºC to 30 ºC; the subcooling degree in SLHX outlet ΔT_{sc} is set at 20 K; the refrigerant quality of the phase separator inlet x_3 is fixed at 0.75; the refrigerant mass flow rate at the compressor inlet is 1 g/s and the refrigerant at the evaporator outlet is saturated.

Fig. 3 shows the variation tendencies of COP and compressor power with the evaporator inlet temperature ranging from -40 ºC to -24 ºC at the given operating condition: T_{c} =40 ºC, Z_{R290}=0.5, x_3=0.75, ΔT_{sc} =20 K. It can be founded that COP increases significantly and compressor work decreases sharply with the rise of T_{e} for both CVRC and MSVRC. Moreover, MSVRC exhibits better system performance than CVRC in terms of COP, especially for low evaporator inlet temperature. As T_{e} varies from-40 ºC to -24 ºC, MSVRC yields 5.2-3.1% higher COP and 8.2-8.5% lower compressor work than CVRC. It attributes that the use of IHX can further increase the refrigerant subcooling, which leads to a higher evaporation pressure at the same evaporator inlet temperature due to the temperature glide characteristics of zeotropic mixtures. This tends to produce 7.15% lower compression ratio, as a result, the compressor work of MSVRC is 8.36% lower than that of CVRC. Consequently, the system performance of MSVRC is superior to that of CVRC. Fig. 4 displays the variations of the total exergy destruction $Ex_{\mathrm{d,tot}}$ and exergy efficiency η_{ex} with the evaporator inlet temperature T_{e} under the above operation condition. It can be observed that for both cycles $Ex_{\mathrm{d,tot}}$ decreases markedly with T_{e}, whereas η_{ex} shows the opposite trend. Compared with CVRC, $Ex_{\mathrm{d,tot}}$ of MSVRC is decreased by 11.84% and the corresponding η_{ex} is improved by 8.76% on average while T_{e} ranges from -40 ºC to -24 ºC. From these results, the better system performance of MSVRC is demonstrated.

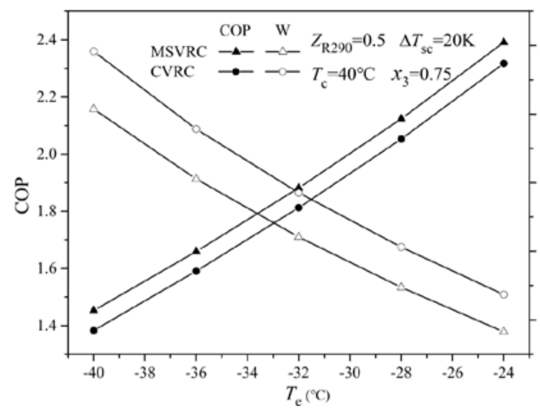


Figure 3: The effect of T_e on system COP and W_c

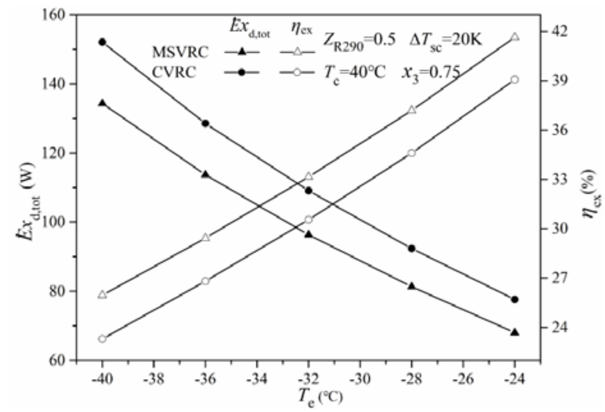


Figure 4: The effect of T_e on $Ex_{d,tot}$ and η_{ex}

Fig. 5 illustrates the various values of COP and compressor power against the condensation temperature T_c under the selected working condition: T_e =-32 ºC, Z_{R290}=0.5, x_3=0.75, ΔT_{sc} =20 K. It can be seen that COP for both cycles decrease with the rise of T_c as expected, whereas the compressor power increases accordingly. MSVRC exhibits 8.3% lower compressor power than CVRC on average. Meanwhile, the discharge temperature of MSVRC is 12ºC lower compared with that of CVRC. Although the cooling capacity of MSVRC is 4% lower than that of CVRC due to temperature glide and higher average evaporation temperature, MSVRC still outperforms CVRC in the aspect of COP by 3.87% on average. Fig. 6 presents the variation tendencies of the total exergy destruction $Ex_{d,tot}$ and exergy efficiency η_{ex} with the T_c for the two cycles. As expected, the $Ex_{d,tot}$ of both cycles increases with T_c, whereas η_{ex} the decreases, which is similar to the COP tendency. As T_c ranges from 30 ºC to 50 ºC, $Ex_{d,tot}$ of MSVRC is reduced by 11.8% on average and η_{ex} is correspondingly improved by from 6.92-11.0% as compared with CVRC. This fact indicates that MSVRC gives more improvement in exergy efficiency for higher condensation temperature.

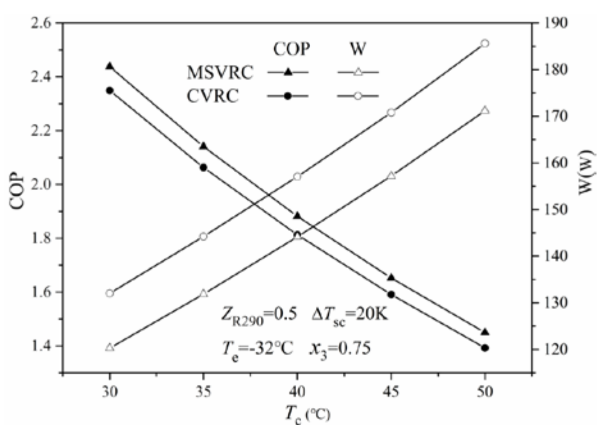


Figure 5: The effect of T_c on system COP and W_c

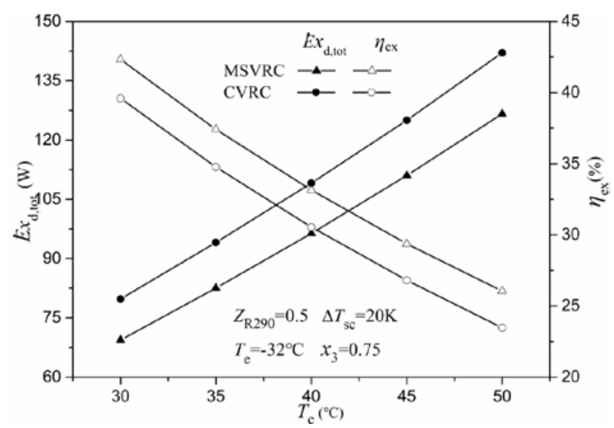


Figure. 6: The effect of T_c on $Ex_{d,tot}$ and η_{ex}

It is well known that the pressure-temperature corresponding relationship of zeotropic mixture in the evaporator and condenser is strongly dependent on the mass fraction of refrigerant mixture, which could impact the refrigeration cycle performance. Considering this point, the influence of propane mass fraction Z_{R290} on the system performance should be evaluated. Fig. 7 indicates the variation tendencies of COP and compressor power versus the propane mass fraction Z_{R290} under the given operation condition: T_c =40 ºC, T_e =-32 ºC, x_3=0.75, ΔT_{sc} =20 K. It should be noted that Z_{R290} at the inlet of the phase separator ranges from 0.3 to 0.7 as Z_{R290} at the compressor inlet ranges from 0.3 to 0.7. Meanwhile, Z_{R290} at the liquid outlet of the phase separator varies from 0.2 to 0.58 and Z_{R290} at the vapor outlet of the phase separator ranges from 0.33 to 0.74. As shown in Fig. 7, COP of MSVRC increases first and then decreases, thus, there exists an optimum composition Z_{R290}=0.4 for MSVRC to reach the maximum COP at 1.884. In addition, the COP of MSVRC is superior to that of CVRC in the entire Z_{R290} range.

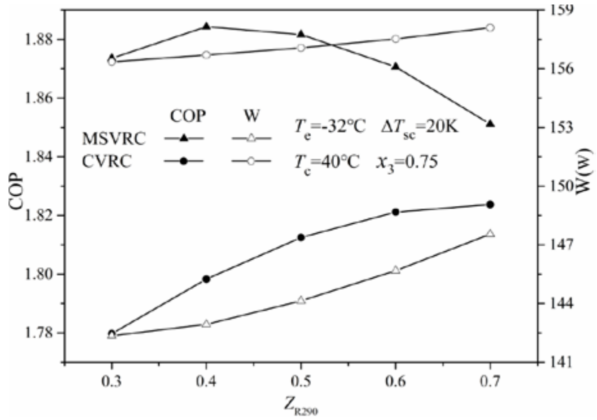


Figure 7: The effect of Z_{R290} on system COP and W_c

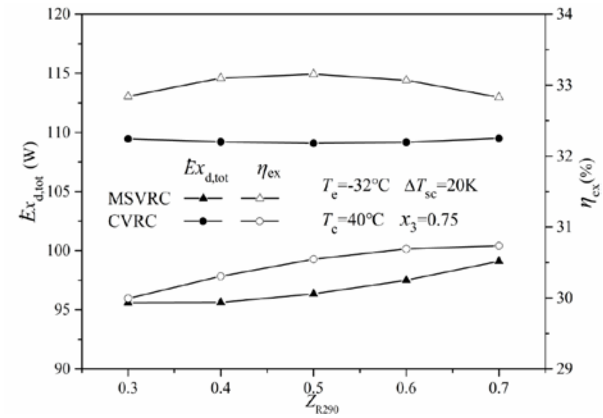


Figure 8: The effect of Z_{R290} on $Ex_{d,tot}$ and η_{ex}

As compared with CVRC, MSVRC yields 3.63% higher COP on average. The reason is due to the fact that the use of the IHX causes further subcooling of refrigerant entering the evaporator, leading to higher evaporation pressure at the same evaporator inlet temperature, lower compression ratio, higher compressor efficiency and better system performance. The compression ratio is reduced by 6.85% as compared with that of CVRC. Furthermore, MSVRC shows a remarkable decrease in compressor work by 8% on average than CVRC. Fig. 8 indicates the variation trends of the total exergy destruction $Ex_{d,tot}$ and exergy efficiency η_{ex} with the propane mass fraction Z_{R290} under the same operating condition. Similar to the variation trend of the COP, the MSVRC is always superior to the CVRC in the exergy efficiency and there is an optimum Z_{R290} for the maximum exergy efficiency η_{ex}. MSVRC reduces the total exergy destruction $Ex_{d,tot}$ by 11.4% as compared to CVRC. As described above, the compressor work of MSVRC is also decreased by 8%, as compared with that of CVRC. Consequently, the exergy efficiency η_{ex} of MSVRC shows 8.4% increase in comparison with CVRC. Thus, it can be concluded that the system performance of MSVRC outperforms that of CVRC.

4. CONCLUSIONS

This paper proposes a modified subcooling vapor-compression refrigeration cycle (MSVRC) using refrigerant mixture R290/R600a. The IHX and phase separator are utilized to make good use of the zeotropic mixture's temperature glide characteristics and achieve further subcooling of refrigerant. The mathematical model based on energetic and exergetic analysis method was developed to theoretically investigate the overall system performance and compare with CVRC under different operating conditions. The effects of key operational parameters such as evaporation temperature, condensation temperature, mass fraction of refrigerant mixture on the system performance are evaluated. The simulation results show that MSVRC is superior to CVRC in terms of COP, compressor work and exergy efficiency. Compared with CVRC, MSVRC can improve the COP and exergy efficiency by up to 5.27% and 11.4%, respectively. Moreover, the compressor work and total exergy destruction of MSVRC can be reduced by 8.36% and 11.8% on average, respectively, as compared to that of CVRC. In general, MSVRC can provide apparent advantages over CVRC by employing the phase separator and IHX. Although the theoretical study demonstrates the potential performance improvement of the modified cycle, further theoretical and experimental works will be necessary in next step.

NOMENCLATURE

CVRC	conventional vapor-compression refrigeration cycle	(–)
COP	coefficient of performance	(–)
Ex_d	exergy destruction	(W)
h	specific enthalpy	(J kg^{-1})
$\dot{m}$	mass flow rate	(kg s^{-1})
MSVRC	modified subcooling vapor-compression refrigeration cycle	(–)

P	pressure	(Pa)
$\dot{Q}$	cooling capacity	(W)
q_{ev}	volumetric cooling capacity	($J\ m^{-3}$)
s	specific entropy	($J\ kg^{-1}\ K^{-1}$)
T	temperature	(°C)
v	specific volume	($m^3 \cdot kg^{-1}$)
W_c	input work of compressor	(W)
x_3	refrigerant quality	(–)
Z	mass composition	(–)

Greeks symbol

η_{ex}	exergy efficiency
η_s	isentropic efficiency of compressor

Subscript

0	reference state
1-14	state points of refrigerant
com	compressor
con-1	condenser-1
con-2	condenser-2
cap-1	capillary tube-1
cap-2	capillary tube-2
eva	evaporator
IHX	internal heat exchanger
SLHX	suction line heat exchanger
sub	subcooler
tot	total

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