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Isentropic Mixtures and Their Application in Heat Pumps in Cold Climate Region

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ABSTRACT

Organic fluids could be classified into wet, dry and isentropic types according to the slope of the saturated vapor envelope on a temperature-entropy diagram. Due to the limited availability of pure isentropic refrigerants, attempts are made to make isentropic mixtures by mixing wet and dry refrigerants with an appropriate composition in the present work. A simple principle for selecting potential component fluids for making isentropic mixtures is proposed. Based on the specified heat pump application with heat source inlet and outlet temperature being $-8\text{ }^{\circ}\text{C}$ and $-13\text{ }^{\circ}\text{C}$ respectively, and heat sink inlet and outlet temperature being $45\text{ }^{\circ}\text{C}$ and $55\text{ }^{\circ}\text{C}$, respectively, 30 combinations of component refrigerants exhibiting characteristics of an isentropic fluid are obtained. The method to reduce the effect of pressure on superheat parameter of isentropic mixtures by adding a third component and adjusting the circulating mass fractions is discussed. Finally, the influence of superheat parameter on temperature glide matching in condenser as well as cycle performance is discussed for three different types of mixtures.

1. INTRODUCTION

Organic fluids have been widely applied in energy conversion systems including vapor compression cycles (VCC) and organic Rankine cycles (ORC), and their thermodynamic properties have significant impacts on the cycle performance. Among the studies on screening criteria of working fluids for ORC, Liu *et al.* (2004) proposed that the organic fluids could be classified into three groups including dry, isentropic and wet, depending on the slope of the saturation vapor curve (dT/dS) to be positive, infinite and negative, respectively. The refrigerants applied in VCC could also be grouped based on the slope of their saturation vapor envelopes, however, the wet and dry fluids in VCC should be redefined considering the difference in the ending states of the working fluid between the vapor compression and expansion processes.

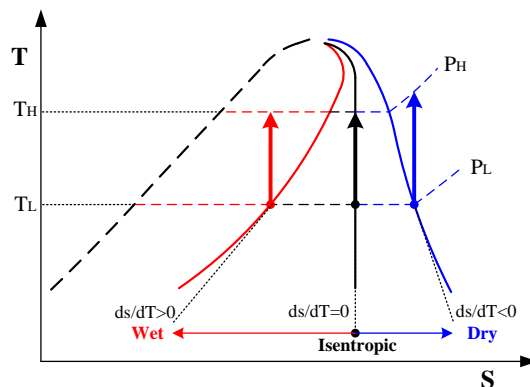


Figure 1: Classification of refrigerants in VCC

Figure 1 illustrates the saturation vapor curve and the corresponding isentropic compression process starting from the saturated vapor point at T_L for the three different types of refrigerants, respectively, in a temperature-entropy ($T-s$) diagram. As shown in the figure, the fluid exhibiting a negative saturation vapor curve slope (i.e. $dS/dT < 0$) is superheated after isentropic compression, and is defined as dry refrigerant. For the refrigerant having a positive slope (i.e. $dS/dT > 0$), the compression process ends within the liquid-vapor two-phase region, thus this kind of fluid

is regarded as wet refrigerant. The fluid showing a vertical saturation vapor curve slope (i.e. $dS/dT=0$) is defined as isentropic refrigerant. In comparison with wet refrigerants, dry and isentropic refrigerants are more suitable for the VCC because they have less chance to be in two-phase condition after isentropic compression, thereby reducing (a) the risk of liquid slugging in compression and (b) the need of high superheating degree at evaporator outlet for preventing condensation during the compression. For the same reason, wet and isentropic refrigerants are better candidates in ORC.

The isentropic fluids may be the ideal working fluid for VCC, ORC, and especially the system where the vapor expander and compressor are employed simultaneously (Zheng and Zhao, 2015). Theoretically, by using isentropic fluids, the heat exchanger area designed for superheating and desuperheating could be saved, which may lead to a better cycle performance since the compression work areas exhibited in a T-s diagram is reduced correspondingly. It is hard to find proper isentropic fluids since the slope of the saturated vapor envelope depends greatly on the operating temperatures and pressures. A few of pure fluids, such as R11 and R123 (Hung *et al.*, 1997), could be considered as isentropic fluids within a certain temperature range. However, this temperature range may not coincide with the specific working conditions of the practical application. Besides, the application of working fluids such as R11 and R123 are restricted due to their non-zero ozone depletion potential, which further limits the options.

Typically, the thermophysical properties of the binary mixture falls in between those of the two constituents, implying that an isentropic mixture could be made by mixing selected constituents with appropriate compositions. Hence, the present paper attempts to “make” isentropic fluids by mixing different types of refrigerants. A simple principle for selecting potential constituents based on specified application is proposed. Methods to reduce the impact of pressure on isentropic mixtures are provided and discussed. Finally, the effect of superheat parameter on temperature glide matching as well as cycle performance is investigated.

2. ISENTROPIC MIXTURES

2.1 Superheat parameter of mixtures

As previously mentioned, the superheat degree of a vapor could be increased, decreased or even unchanged during the isentropic compression depending on the type of the refrigerant involved. According to Bertinat (1986), the superheat degree (SHD) at compressor outlet could be expressed as:

$$SHD \approx \int_{T_{evap}}^{T_{cond}} (\zeta - 1) dT \quad (1)$$

where the dimensionless number ζ , named as superheat parameter, is defined by:

$$\zeta \equiv \beta \Delta H_{lv} V / C_p \Delta V \quad (2)$$

where β is the isobaric coefficient of thermal expansion, ΔH_{lv} the latent heat of vaporization, V specific volume of vapor, C_p specific heat at constant pressure, ΔV difference between vapor and liquid specific volumes.

Eq. 1 implies that ζ is >1 for a dry refrigerant but <1 for a wet refrigerant. The isentropic refrigerant will have $\zeta \approx 1$. This criterion could also be applicable to refrigerant mixtures. Assume two pure refrigerant R1 and R2 and their mixture RM, and their superheat parameters calculated based on the same reduced temperature (e.g. $T_r=0.8$) is ζ_{R1} , ζ_{R2} and ζ_{RM} , respectively (assume that $\zeta_{R1} > \zeta_{R2}$). At constant reduced temperature, β shows relatively little effect on ζ , and the influence of $V/\Delta V$ could also be ignored unless T_r is very closed to unit. Hence, the ratio between ΔH_{lv} and C_p will play a dominant role in determining the type of the mixture, according to Eq. 2. Then a simple linear relationship between ζ and critical pressure P_{crit} could be established while considering that ζ depends upon the intermolecular interactions and the linear dimensions of the molecules in a similar way as P_{crit} does (Hirschfelder *et al.*, 1964). In general, the critical pressure of the mixture would fall somewhere in between the critical pressures of the two constituents, hence the follow relation exists:

$$\zeta_{R1} > \zeta_{RM} > \zeta_{R2} \quad (3)$$

If $\zeta_{R2} > 1$, then ζ_{RM} is >1 , which implies that mixing two dry refrigerants makes the dry mixture; if $\zeta_{R1} < 1$, then ζ_{RM} is <1 , which implies that mixing two wet refrigerants makes wet mixture; and isentropic mixture ($\zeta_{RM}=1$) could only be made by mixing two different types of refrigerants. Although not all of the organic fluids fit the linear relationship between ζ and P_r , the simple relation expressed above reflects qualitatively at least, the variations in ζ when mixing different fluids.

Table 1: Properties of candidate Refrigerants

Fluid	M (g/mol)	P_{crit} (bar)	T_{crit} (°C)	T_{nb} (°C)	ζ (-)
R1234yf	114.04	32.57	94.70	-29.45	0.8937
R1234ze	114.04	36.36	109.37	-18.95	0.9115
R134a	102.03	40.59	101.06	-26.07	1.0694
R143a	84.04	37.61	72.71	-47.24	1.1393
R152a	66.05	45.17	113.26	-24.02	1.2286
R161	48.06	50.91	102.15	-37.55	1.2709
R218	188.02	26.4	71.87	-36.79	0.6782
R227ea	170.03	29.25	101.75	-16.34	0.7187
R32	52.02	57.82	78.11	-51.65	1.6971
R600a	58.12	36.29	134.66	-11.75	0.8027
R290	44.10	42.51	96.74	-42.11	1.0659

2.2 Screening of binary isentropic mixtures

In this section, binary isentropic mixtures are screened out based on a heat pump system applied in cold climate regions. The inlet and outlet temperatures of the heat source are set to be -8°C and -13°C , respectively, and the inlet and outlet temperatures of the heat sink 45°C and 55°C , respectively. Hence the initial selection criteria are $T_{nb} \leq -10^{\circ}\text{C}$ (the averaged evaporating temperature) to ensure above-atmospheric pressures in the system, and $T_{crit} \geq 70^{\circ}\text{C}$ to avoid approaching the critical point during condensing process. Besides, refrigerants with non-zero ODP have been excluded from consideration.

The properties of candidate refrigerants obtained from NIST REFPROP (Lemmon *et al.*, 2010) are listed in Table 1. The superheat parameter (ζ) of each refrigerant was calculated based on the same reduced temperature ($T_r=0.8$). According to the calculation results of ζ , the 11 candidate refrigerants are divided into two groups: (a) wet refrigerants including R218, R227ea, R600a, R1234yf and R1234ze, and (b) dry refrigerants including R32, R161, R152a, R143a, R134a and R290. The superheat parameters of the candidate refrigerants are plotted against their critical pressures in Figure 2. In overall, ζ increases linearly with P_{crit} . Hence, based on the simple relation proposed inequality in Eq. 3, thirty isentropic binary mixtures could be obtained by mixing the six dry candidate refrigerants with the five wet candidate refrigerants in pair. The 30 binary isentropic mixtures with their composition, expressed by the mass fraction of the dry component (x_{dry}) are shown in Table 2. The superheat parameter of each mixture is calculated based on the same mean saturation temperature (20°C). The temperature glide of each refrigerant mixture under condensing pressure is also listed in the table. According to Table 2, ten binary mixtures including R290/R600a exhibit obvious temperature glide, i.e. $\Delta T_{gl} > 5$ K, and are regarded as zeotropic mixtures. Binary mixtures like R134a/R1234yf with very small temperature glide could be considered as near-azeotropic mixtures.

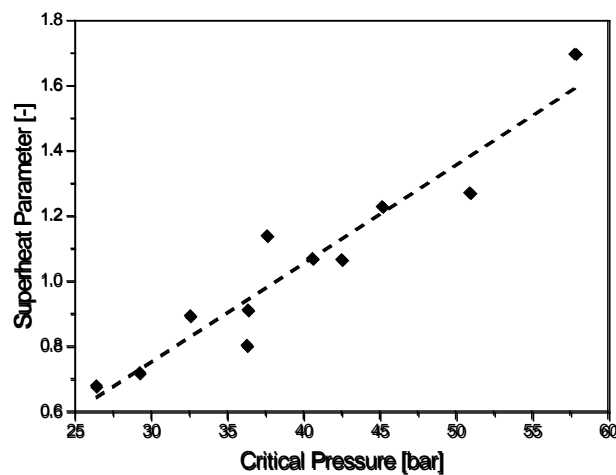


Figure 2: ζ versus P_{crit} for the candidate refrigerants

Table 2: Isentropic binary mixtures

Mixture	x_{dry} (%)	ΔT_{gl} (K)	Mixture	x_{dry} (%)	ΔT_{gl} (K)	Mixture	x_{dry} (%)	ΔT_{gl} (K)
R32/R218	46.74	0.4	R290/R218	68.18	0	R134a/R218	74.54	4.8
R32/R1234yf	7.46	7.8	R290/R1234yf	42.01	0.8	R134a/R1234yf	56.39	0.1
R32/R1234ze	3.82	6.3	R290/R1234ze	25.38	3.3	R134a/R1234ze	43.10	0.8
R32/R227ea	18.60	12.6	R290/R227ea	62.78	0.3	R134a/R227ea	74.14	0.3
R32/R600a	2.82	19.2	R290/R600a	52.63	7.4	R134a/R600a	75.47	0.8
R143a/R218	67.86	0.2	R161/R218	36.11	6.5	R152a/R218	46.12	1.9
R143a/R1234yf	41.73	2.9	R161/R1234yf	16.08	1	R152a/R1234yf	23.87	0
R143a/R1234ze	24.81	6.4	R161/R1234ze	9.01	2.3	R152a/R1234ze	15.11	0.5
R143a/R227ea	63.45	4.4	R161/R227ea	29.54	3.3	R152a/R227ea	40.51	0.5
R143a/R600a	60.04	15.4	R161/R600a	20.80	10.6	R152a/R600a	35.93	6.7

3. VARIATION OF SUPERHEAT PARAMETER

The variation tendencies of ζ with x_{dry} are illustrated for five representative binary mixtures in Figure 3(a). As expected, the value of ζ increases with x_{dry} , and the mixture changes gradually from wet mixture to dry mixture. The composition corresponding to an isentropic fluid for each mixture could be seen in Table 2. Due to the relatively large superheat parameter of R32, the isentropic mixture is formed with very low R32 mass fraction (x_{R32}), and the variation range of ζ of the mixtures containing R32 is larger than that of other three mixtures. For mixtures containing R32, the superheat parameter changes rapidly nearby the mass fraction corresponding to $\zeta=1$, which means that the prepared isentropic mixtures are more sensitive to the mass fraction change. Considering the occurrence of mass fraction shift of zeotropic mixture in a real system, R32 may not be a good choice for making isentropic mixture.

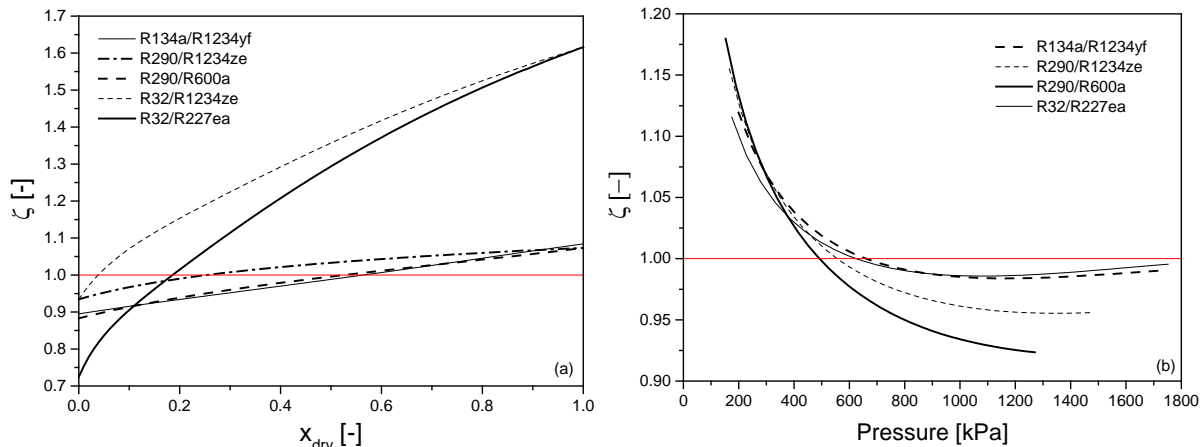


Figure 3: Variation of ζ versus: (a) x_{dry} ; and (b) pressure

The variation of superheat parameter with pressure for each representative isentropic mixture is shown in Figure 3(b). Overall, ζ decreases from above unit to below unit as the pressure increases. However, for mixture R134a/R1234yf and R32/R227ea, a slight recovery in ζ could be observed as the pressure approaches to the condensing pressure. The minimum value of ζ is about 0.984 and 0.986, respectively, for the two mixtures. For the other two mixtures containing R290, the value of ζ drops steadily with the increase of pressure. Although the decline slows down gradually as pressure increases, the values of ζ deviate from 1 obviously at condensing pressures. The mixture R134a/R1234yf and R32/R227ea show relatively steady variation of ζ as their pressures surpass 700 kPa, which means that the pressure has a weaker impact on ζ of these two mixtures under high pressure conditions.

The marked change of ζ of mixture R290/R600a versus pressure may be caused by the relatively large difference in the thermophysical properties between the two constituents. By adding an appropriate third component, the difference in properties among the constituents could be reduced, thereby reducing the effect of pressure on ζ .

According to Table 1, R134a may be a good choice as the third component since its critical temperature and critical pressure fall in between that of R290 and R600a.

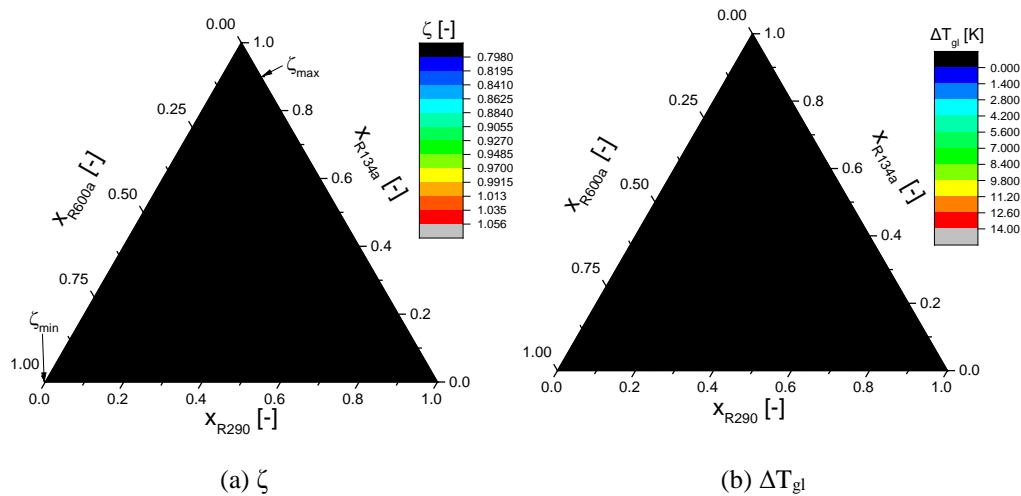


Figure 4: Result of ternary mixture R290/R134a/R600a.

Figure 4(a) shows the variation of ζ versus composition for the ternary mixture R290/R134a/R600a on a colored contour map. The superheat parameter is calculated based on the same condensing temperature for each composition. The variation range of ζ is from 0.798 to 1.056, and the compositions corresponding to the maximum and minimum values are marked on the map. A banding distribution of the superheat parameter along the mass fraction of R600a (x_{R600a}) could be observed from the map, and the value of ζ decreases from its maximum value to its minimum value as x_{R600a} increases. As x_{R600a} reaches to about 0.2, ζ of the ternary mixture is in steady around 1, regardless of the change in the mass fractions of the other two components. As shown in Table 1, R134a has a very similar ζ value with that of R600a at the same reduced temperature, which may lead to an almost constant superheat parameter for mixture R290/R134a with various compositions, according to the relation shown in inequality Eq. 3. The variation of the temperature glides versus composition is illustrated in Figure 4(b). The maximum ΔT_{gl} (12.9 K) of the ternary mixture occurs at the mass fractions of (0/0.3/0.7). Different from ζ , an annular distribution of ΔT_{gl} with the center located at the mass fractions corresponding to the maximum ΔT_{gl} appears. For the near isentropic ternary mixtures (i.e. $x_{R600a}=0.2$), obvious temperature glide could be distinguished within x_{R600a} range from 0.1 to 0.6, which means adding the third component does not impair the temperature glides matching in the condenser.

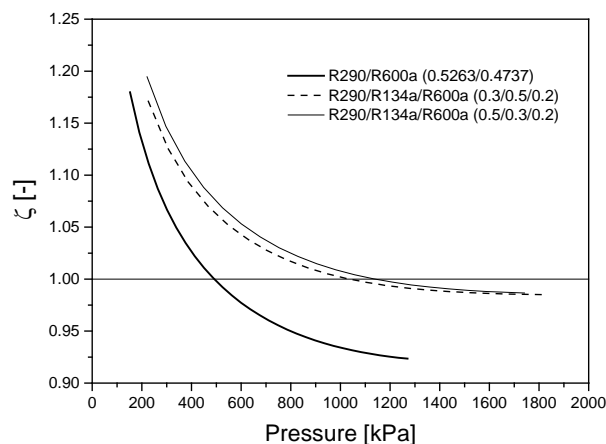


Figure 5: Effect of third component on ζ -pressure relationship

Figure 5 illustrates the ζ -pressure relationship of mixture R290/R600a after adding R134a. The two representative ternary mixtures are obtained by fixing x_{R600a} at 0.2 and changing the mass fractions of the other two constituents. The ΔT_{gl} of the two ternary mixtures is 6.1 K ($x_{R290}=0.3$) and 7.1 K ($x_{R290}=0.5$), respectively. As shown in the figure, adding R134a to the binary mixture narrows the variation range of ζ markedly, with the minimum value being

increased by over 0.06. The change of ζ versus pressure also becomes gentler, and as a result, the superheat parameter of the ternary mixture could remain quite close to $\zeta=1$.

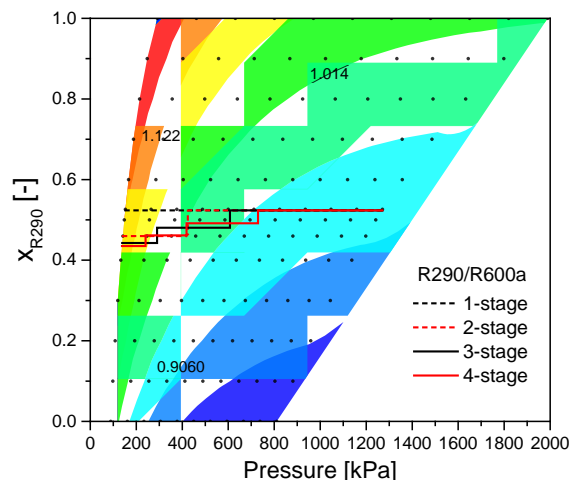


Figure 6: x - P - ζ relationship for R290/R600a plus composition change in multistage cycle

As can be seen in Figure 3(b) and Figure 5, the effect of pressure on ζ of the binary or ternary mixture could only be reduced, to some extent, within the high pressure range. However, within the low pressure range, the value of ζ still drops rapidly with an increasing pressure for any combination of the component fluids. Taking mixture R290/R600a as an example, the combined effect of composition and pressure on ζ of this mixture is illustrated in a colored contour map shown in Figure 6. Since ζ increases with x_{R290} , but decreases with pressure, the dry mixtures could be separated from the wet mixtures by the contour line of $\zeta \approx 1$ developing from the bottom left corner to the top right corner of the colored section in the figure. Hence, a potential method to weaken the effect of pressure on ζ within low pressure range is reducing the mass fraction of the dry fluid component gradually as the pressure decreases.

The circulating mass fractions of zeotropic mixtures could be changed via phase separators and T-junctions (Zheng *et al.*, 2016). In a multi-stage vapor injection cycle, for instance, the phase separators are used to supply refrigerant flow with a certain vapor quality to the compressor to decrease its discharge temperature. When zeotropic mixtures are applied in a multi-stage cycle, the mixture composition is expected to be different for each stage. The variation of circulating mass fractions versus pressure for three cycles with different number of phase separators is shown in Figure 6. As a comparison, the traditional VCC with no phase separator, named as 1-stage cycle, is also illustrated in the figure. The circulating mass fractions in the condenser is the same for the four cycles, with an R290 mass fraction of 0.5263. Every time passing through the phase separator, the mixture will lose some dry component, R290, which has a lower boiling point, and as a result, x_{R290} decreases gradually as the pressure changes from condensing pressure to evaporating pressure. The final circulating mass fraction in the evaporator does decrease as the number of phase separator in the cycle increases. However, the difference in composition between the cycles becomes smaller as the stage number increases. Increasing the number of phase separator from 1 to 2 decreases x_{R290} by 0.017, while x_{R290} is only reduced by about 0.007 by adding one more phase separator. The x_{R290} in the evaporator is expected to approach a minimum value as the number of phase separators increases, which means the variation range of mass fraction may be relatively limited when the traditional phase separation method is used. However, the variation range of composition of the mixture could be enlarged significantly by using distillation units. Although the heat pump system would become more complex by introducing multi-stage distillation units, a more desired distribution of mass fractions in the cycle could be achieved to weaken the influence of pressure on ζ of the mixture.

4. CYCLE PERFORMANCE

In this section, the effects of ζ on the temperature glide matching in the condenser and system performance are discussed based on the cycle modeling method proposed by McLinden and Radermacher (1987). In this method, counter-flow type heat exchangers are assumed with specified inlet and outlet temperatures of HTF (see Section 2.2). Refrigerant is supposed to leave the heat exchanger as a saturated fluid condition. Upon the assumption of a constant

overall heat transfer coefficient, U , which is identical for desuperheating, condensation and evaporation, the heat exchange processes could be treated by the following heat exchanger parameter:

$$\frac{UA_{total}}{Q_c} = \frac{Q_e}{Q_c} \frac{1}{\Delta T_{lm,e}} + \frac{1}{\Delta T_{lm,c}} \quad (4)$$

where A_{total} is the total heat exchange area of condenser and evaporator, and Q_c and Q_e are the condensation and evaporation heat, respectively, and $\Delta T_{lm,c}$ and $\Delta T_{lm,e}$ are the log mean temperature differences in condenser and evaporator, respectively.

The expression on the left-hand side of Eq. 4, standing for total heat transfer conductance per unit heating capacity, is specified as a constant to eliminate the influence of a varying capacity on heat exchanger loading when different mixtures are involved and compared. The irreversibility of a conventional VCC comes from four main irreversible processes, including heat transfers in the condenser and evaporator, throttling and compression. The irreversibility caused by heat transfer through a certain temperature difference in heat exchangers can be calculated as follows (Alefeld and Radermacher, 1993):

$$\delta S = \sum_{i=1}^n \left(\frac{q_i}{t_i} - \frac{q_i}{T_i} \right) \quad (5)$$

where t and T represent the temperature of refrigerant and HTF, respectively, and q_i is the heat transferred in the i^{th} segment of the heat exchanger. Symbol n stands for the number of the divided segments in heat exchangers ($n=10$ for evaporator and $n=15$ for condenser are used in this study).

To illustrate the influence of superheat parameter, the following calculations are done for three different binary mixtures including the dry zeotrope R32/R152a, the near-isentropic zeotrope R290/R600a, and the near-isentropic near-azeotropic fluid R134a/R1234yf. The heat exchanger parameter UA_{total}/Q_c is set to be 0.36 for all three mixtures.

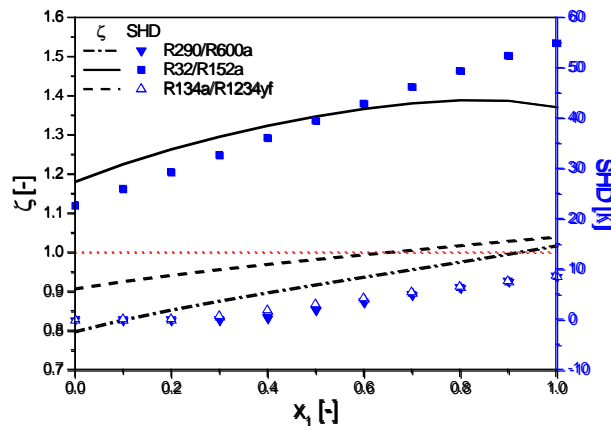


Figure 7: ζ and SHD as a function of x_1 for three mixtures

Figure 7 shows the variation tendencies of superheat parameter and superheat degree (SHD) with mass fraction, expressed by the mass fraction of the first component x_1 , for the three mixtures. For the two isentropic mixtures, ζ increases almost linearly from below unit to above unit as the x_1 increases from 0 to 1. For the dry mixture R32/R152a, the value of ζ is greater than 1 within the entire range of composition. The growth of ζ slows down gradually as x_{R32} increases and a drop in ζ occurs as x_{R32} changes from 0.9 to 1. The unique variation trend of ζ for R32/R152a may be explained by the fast increasing T_r . For mixture R32/R152a, T_r increases from 0.85 to 0.92 as x_{R32} changes from 0 to 1. An increased T_r would lead to a decrease in ζ . The SHD increases steadily as x_1 increases for all the three mixtures. Specifically, the SHD of the two isentropic mixtures increases from negative values to below 10 K. The negative values indicate two-phase refrigerant mixtures exiting the compressor. Dry mixture R32/R152a has much higher SHD with a variation range from 22.7 K to 54.9 K. It should be noted that both the superheat parameters at condensing and evaporating pressures of mixture R32/R152a are greater than 1, which takes the most responsibility for its high superheat degree.

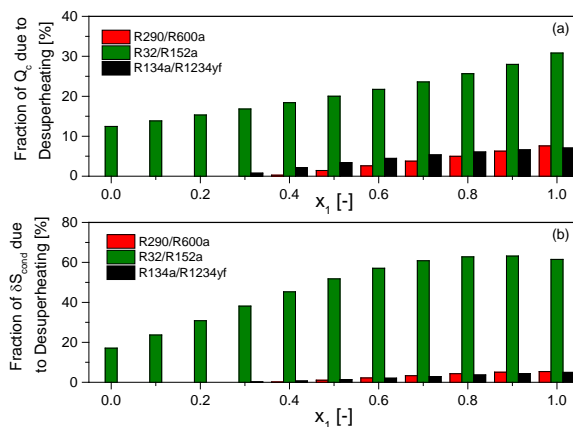


Figure 8: Contribution of desuperheating on: (a) condensing heat and (b) irreversibility

The contribution of desuperheating on the condensing heat as well as the irreversibility in the condenser (δS_{cond}) is shown in Figure 8. As expected, the fraction of heat transferred due to desuperheating of vapor refrigerant (f_{desup}) increases with the growth of the SHD for all the three mixtures. Due to the higher SHD of R32/R152a, much more proportion of its condensing heat comes from cooling down the superheated vapor, as shown in Figure 8 (a). For the isentropic mixtures with a relatively low x_{dry} , an isentropic compression process would end in the two-phase region, and hence all the condensing heat comes from condensation. The irreversibility caused by heat transfer within the desuperheating section of the condenser would increase with an increasing SHD, while its contribution to δS_{cond} would not increase steadily same time, as shown in Figure 8(b). This is because the irreversibility associated with condensation in the two-phase region also increases due to a worse temperature glides matching caused by the increasing SHD.

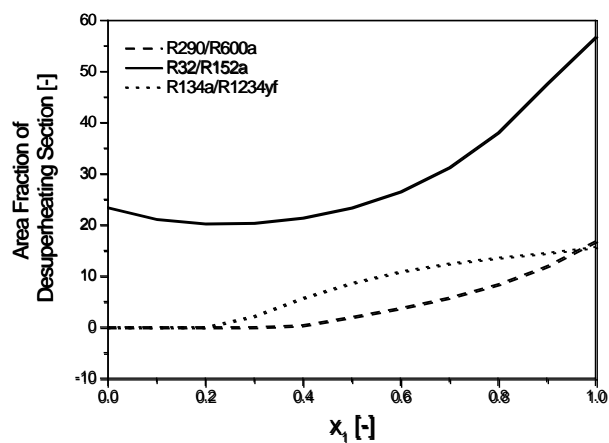


Figure 9: Variation of area fraction of desuperheating section

Based on the condensing heat and the log mean temperature difference, the proportion of condenser area used for cooling down the superheated vapor could be estimated. Since the $\Delta T_{lm,c}$ is almost identical for different mixtures due to the constant value of UA_{total}/Q_c , the area fraction of desuperheating section depends mainly on the values of f_{desup} and $\Delta T_{lm,desup}$. The variation tendency of the area fraction of desuperheating section with x_1 for each mixture is shown in Figure 9. For mixture R32/R152a, about 20-55% of the condenser area is used for desuperheating, and this proportion is much higher than that of the isentropic mixtures. As shown in the figure, the increase of the area fraction of desuperheating section becomes more rapidly as x_{R32} increases. According to Figure 8 (a), an accelerated growth of f_{desup} with x_{R32} could be observed for R32/R152a. Meanwhile, according to the calculation results, the value of $\Delta T_{lm,desup}$ first increases and then decreases with an increasing x_{R32} . As a result, the growth of the area fraction of desuperheating section is accelerated. It should be pointed out that the slight decrease of area fraction

observed within x_{R32} range from 0-0.2 is caused by the relatively fast growth of $\Delta T_{lm,desup}$. The variation tendencies of area fraction for the other two mixtures could be explained in the similar way.

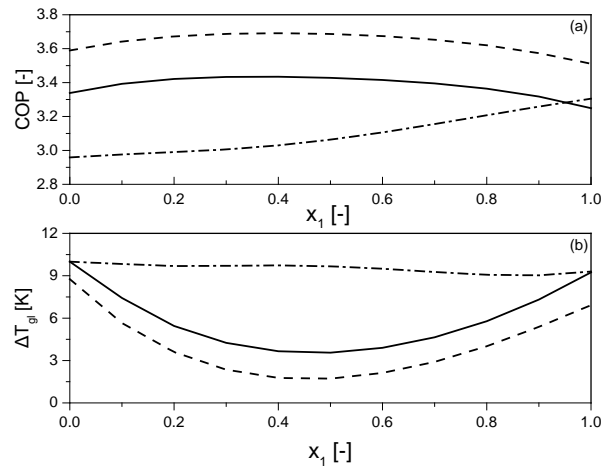


Figure 10: Calculated performance as a function of x_1 for three mixtures: (a) COP; and (b) ΔT_{gl} . (solid line: R290/R600a; dash line: R32/R152a; and dash dot line: R134a/R1234yf).

Figure 10 shows the variation of calculated COP and ΔT_{gl} with x_1 for the three mixtures. According to Figure 10 (a), the mixture R32/R152a shows the highest COP, followed by R290/R600a and then R134a/R1234yf. For zeotropes R32/R152a and R290/R600a, the maximum COP occurs at an intermediate composition, while the COP of near-azeotropic mixture R134a/R1234yf increases steadily as the R134a mass fraction (x_{R134a}) increases. The temperature glide matching in condenser affects the COP of the zeotropes significantly. As shown in Figure 10(b), the difference between the temperature glides of the refrigerant and HTF first decreases and then increases with an increasing x_{dry} . For both zeotropes, the minimum value of ΔT_{gl} occurs around x_1 of 0.5, which is slightly higher than the value of x_1 corresponding to the maximum COP (ca. 0.4). This tiny difference may be caused by the increasing SHD with x_1 . The increased SHD may counter the better match of the temperature glides at x_1 of 0.5. The mixture R290/R600a and R32/R152a have identical refrigerant-side temperature glide, however, the latter shows a better temperature glides matching, because the HTF-side temperature glide within the two-phase region is much smaller due to the higher value of f_{desup} .

5. CONCLUSIONS

The following conclusions can be drawn from this work:

- Zeotropic mixtures could be formed by mixing dry and wet fluids with appropriate mass fractions.
- 30 binary mixtures are obtained based on the specified application with heat source inlet and outlet temperature being -8 °C and -13 °C respectively, and heat sink inlet and outlet temperature being 45 °C and 55 °C respectively.
- Influence of pressure on the superheat parameter could be reduced partially by adding a third component.
- By varying the circulating mass fractions, the influence of pressure on superheat parameter could be eliminated in theory.
- The superheat degree increases with superheat parameter, leading to the larger condensing area and irreversibility due to desuperheating.
- For dry mixture R32/R152a, the better temperature glides matching during condensation could be achieved with a certain superheat degree. Due to the contribution of desuperheating to condensing heat, the COP for dry mixture R32/R152a is higher than that of an isentropic mixture such as R290/R600a.

NOMENCLATURE

A_{total}	sum of heat exchanger area	(m^2)
C_p	specific heat at constant pressure	(kJ/kg·K)
f_{desup}	fraction of condensing heat due to desuperheating	(-)
M	molecular mass	(g/mol)

ORC	organic Rankine cycle	
P	pressure	(kPa)
Q,q	condensing or evaporating heat	(W)
R1	assumed first component fluid	
R2	assumed second component fluid	
RM	assumed binary mixture	
SHD	superheat degree	(K)
T	refrigerant temperature	(°C)
U	overall heat transfer coefficient	(W/m ² ·K)
VCC	vapor compression cycle	
x	mass fraction	(-)
ΔH_{lv}	latent heat of vaporization	(kJ/kg)
ΔT_{lm}	log mean temperature difference	(°C)
ΔT_{lg}	temperature glide	(K)
β	coefficient of thermal expansion	(1/K)
ζ	superheat parameter	(-)
δS	irreversibility	(kJ/kg·K)

Subscript

1	first component of binary mixture
cond, c	condenser or condensing
crit	critical point
dry	dry component of binary mixture
evap, e	evaporator or evaporating
nb	normal boiling point
r	reduced property

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