

Numerical Study of low-GWP Mixture Refrigerants Used to Replace R410A

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Abstract

In this paper, M1(R32/R290, 67/33) and M2(R32/R1270, 63/37) were proposed to replace R410A refrigerant. The thermodynamic parameters of R410A, M1 and M2 were compared, the effects of mass flux, vapor quality and physical properties on heat transfer performance were discussed. The results showed that the thermodynamic performances of M1 and M2 are similar to R410A, which GWP was 76.3% and 77.8% lower than R410A, respectively. At the same condition, the heat transfer coefficients and pressure gradients increase with increasing mass flux and vapor quality, the heat transfer coefficients of M1 and M2 were on average 13% and 17% higher than that of R410A, respectively. However, the pressure gradients of the three are not much different. M1 and M2 can be used as low GWP alternative refrigerants of R410A. The obtained results of the present study are of great significance to the industrial application of alternative refrigerants.

Keywords

Mixture Refrigerants; Replaces R410A; Thermodynamic Performance; Condensation; Heat Transfer.

1. Introduction

"Carbon peak" and "carbon neutralization" have become a hot topic in the world today. The large-scale use of high GWP refrigerants is one of the main problems facing the refrigeration industry [1]. R410A has been used in the refrigeration industry for many years as an alternative refrigerant to R22. However, the GWP of R410A is relatively high and faces the problem of replacement in the future. Therefore, new refrigerants need to be developed. The purpose of this paper is to propose new mixed refrigerant M1 and M2, and to study and compare the thermodynamic properties of R410A, M1 and M2 as well as the condensation heat transfer characteristics and two-phase pressure drop characteristics in microtubes.

[2] summarized the alternative refrigerants of R410A and compared the thermodynamic properties of various alternative refrigerants. [3] studied the convective boiling heat transfer coefficient and two-phase pressure drop of R410A in rectangular microchannels with hydraulic diameters of 1.36 and 1.44 mm. The heat transfer correlation of the microchannel is established, which yielded satisfactory predictions with a mean deviation of 18%. [4] experimentally studied the condensation heat transfer and pressure drop of R22, R410A and R407C in two single round stainless steel tubes with inner diameter of 1.088 mm and 1.289 mm. R410A has more advantages than R407C. [5] experimentally and theoretically investigated the condensation heat transfer performance of R32 instead of R410A in microchannel tubes. The heat transfer coefficient and frictional pressure gradient of R32 were higher than those of R410A. [6] presents the experimental results of the condensation heat transfer characteristics of R410A, R22 and R32 flowing in horizontal micro-finned tubes, and a new condensation heat transfer coefficient correlation is proposed.

Although natural refrigerants have a very low GWP, efficiency and safety are issues that need to be overcome. In addition, HFOs refrigerants have a price disadvantage. Therefore, the author believes that it is necessary to study new transitional alternative refrigerants. In this study, thermodynamic performances of M1, M2 and R410A are compared, and the heat transfer coefficient and pressure drop of R410A M1 and M2 in microchannel were studied by numerical simulation.

2. Numerical Model

2.1. Thermodynamic Model

The main composition and characteristics of refrigerants R410A, M1 and M2 are listed in Table 1. In order to facilitate the comparison of the thermodynamic properties of R410A and alternative refrigerants, the vapor compression refrigeration cycle (VCRC) will be used as the basis for comparison. Mathematical calculations involved in the performance evaluation of VCRC system are given below.

The compression and expansion processes follow the model assumptions. The isentropic efficiency expressions of compressor is expressed in the following equations [7]:

$$\eta_c = 0.8981 - 0.09238CR + 0.00476CR^2 \quad \text{Eq. (1)}$$

Non-zeotropic have non-isothermal evaporation and condensation. During the evaporation and condensation, the mixture refrigerants temperature glides due to the changing component concentrations in each phase of the mixture, it is regarded as temperature glide. The definition is as follows:

$$T_{\text{glide}} = T(P_e, x=1) - T(P_e, x=0) \quad \text{Eq. (2)}$$

COP is defined as:

$$COP = \frac{\dot{W}_c}{\dot{Q}_e} \quad \text{Eq. (3)}$$

In this study, NIST database REFPROP 9.1 [8] is called by MATLAB R2017b for programming to evaluate the thermodynamic performance characteristics of various considered R410A alternatives.

Table 1. Physical properties of R410A and alternative refrigerants.

Refrigerants	R410A	M1	M2
Components	R32/R125	R32/R290	R32/R1270
Composition by mass (%)	(50/50)	(67/33)	(63/37)
Molecular mass (g/mol)	72.59	49.11	47.84
Critical temperature (°C)	71.34	62.10	83.56
Critical pressure (MPa)	4.9	4.74	7.29
Temperature glide (K)	0	0.009	0.001
Safety group	A1	A2*	A2*
ODP	0	0	0
GWP _{AR5}	1920	455	427

2.2. Numerical Model of Heat Transfer

2.2.1. Geometric Model and Boundary Conditions

The horizontal smooth circular tube is adopted as the simulated geometric model in this study. Figure 1 shows a schematic diagram of the geometric model. The steady-state evaporation process of R410A and M1 in a single horizontal circular tube were simulated. Hexahedral mesh is generated, which has a higher quality, and it's densified near the wall to set a finer mesh. The first layer of the mesh in the fluid zone is also adjusted to make sure $y^+ < 1$. The refrigerant liquid flows into the tube under the condition of $T_{sat}=313.15K$. The wall surface is considered to be a smooth, non-thickness surface, and the heat exchange temperature difference with the wall surface is set to 10K. The geometric parameters and boundary conditions of the entire model are listed in Table 2, it is chosen according to previous studies within common ranges.

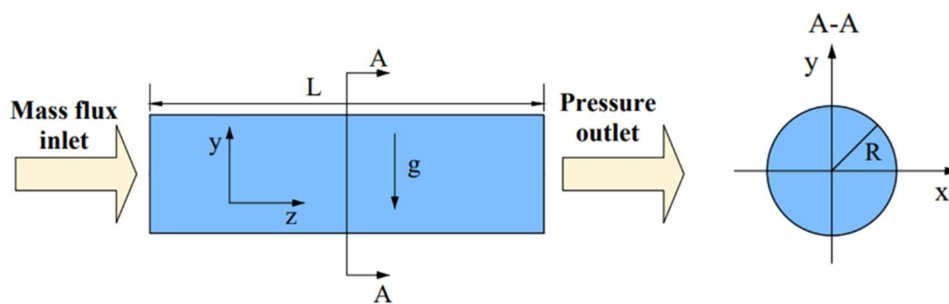


Figure 1. Geometry and boundary conditions for the computational units.

Table 2. Geometric parameters and research conditions.

Item	value
Working fluid	R410A, M1, M2
Tube length (L), mm	300
Tube diameter (D), mm	1
Saturation temperature (T_{sat}), K	313.15
Mass flux (G), kg/m ² ·s	400, 600, 800

2.2.2. VOF Model

The Volume of Fluid (VOF) method [9] is chosen in this study. As it can track the motion of interface between liquid and vapor through the use of a single momentum equation shared by gas and liquid. For each control volume, the volume fractions of each phase sum to unity. Void regions in which no fluid of any type is occupied were disallowed. The fluid properties and flow variables were shared by the phases and calculated based on the volume fraction of each phase. In the VOF model, the sum of volume fraction in the liquid and gas phase equals 1. Volume fraction equation:

$$\alpha_L + \alpha_G = 1 \quad \text{Eq. (4)}$$

$$\nabla \cdot (\bar{v} \alpha_G) = \frac{\dot{m}_G}{\dot{\rho}_G} \quad \text{Eq. (5)}$$

$$\nabla \cdot (\bar{v} \alpha_L) = \frac{\dot{m}_L}{\dot{\rho}_L} \quad \text{Eq. (6)}$$

Fluid properties and flow variables are calculated based on the volume fraction of each phase. The density ρ , viscosity λ and thermal conductivity μ for vapor–liquid mixture are defined as follows:

$$\rho = \rho_L \alpha_L + \rho_G \alpha_G \quad \text{Eq. (7)}$$

$$\mu = \mu_L \alpha_L + \mu_G \alpha_G \quad \text{Eq. (8)}$$

$$\lambda = \lambda_L \alpha_L + \lambda_G \alpha_G \quad \text{Eq. (9)}$$

The velocity, pressure and temperature field are solved based on the above control equations, and the above volume fraction α will be used to calculate properties of each phase of the refrigerant. The numerical simulation is based on the mass, momentum, and energy balance. The governing equations are shown below. Continuity equation, Momentum equation:

$$\nabla \cdot (\rho \vec{v}) = 0 \quad \text{Eq. (10)}$$

$$\nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot \left[\mu (\nabla \vec{v} + \nabla \vec{v}^T) \right] + \rho \vec{g} + \vec{F} \quad \text{Eq. (11)}$$

The continuum surface force (CFS) model [10] is used to evaluate the role of surface tension. The surface tension was expressed as a volume force $\vec{F}$. It is expressed as:

$$\vec{F} = \sigma \frac{2\rho \kappa_L \nabla \alpha_L}{\rho_L + \rho_G} \quad \text{Eq. (12)}$$

where κ_L is interface curvature defined as below.

$$\kappa_L = \nabla \cdot \frac{\nabla \alpha_L}{|\nabla \alpha_L|} \quad \text{Eq. (13)}$$

Energy equation:

$$\nabla \cdot [\vec{v} (\rho E + p)] = \nabla \cdot (\lambda_{eff} \nabla T) + h_{lv} \dot{m}_L \quad \text{Eq. (14)}$$

where λ_{eff} is the effective thermal conductivity.

$$\lambda_{eff} = \lambda + \frac{C_p \mu_t}{Pr_t} \quad \text{Eq. (15)}$$

$$E = \frac{\rho_L \alpha_L E_L + \rho_G \alpha_G E_G}{\rho_L \alpha_L + \rho_G \alpha_G} \quad \text{Eq. (16)}$$

In the present simulation, the implicit scheme was adopted for the time discretization for numerical simulation. The Green-Gauss Node-Based method was applied to compute the gradients; the PRESTO! scheme was used to interpolate the pressure values at the faces; The third order MUSCL scheme was used to discretize the momentum equation, energy equation and turbulence equations; The pressure-velocity coupling algorithms adopted in this paper was SIMPLE; the compressive scheme is selected for volume fraction discretization.

2.2.3. Phase-change Model

The phase-change model is indispensable in the simulation of the condensation process, [11] model is adopted in this study.

$$\dot{m}_G = -\dot{m}_L = \gamma \alpha_G \rho_G \frac{T - T_{sat}}{T_{sat}}, \text{ if } T \leq T_{sat} \quad \text{Eq. (17)}$$

$$\dot{m}_G = -\dot{m}_L = \gamma \alpha_L \rho_L \frac{T - T_{sat}}{T_{sat}}, \text{ if } T > T_{sat} \quad \text{Eq. (18)}$$

In this study, the value of γ is selected from $3 \times 10^5 \text{ s}^{-1}$ to $5 \times 10^6 \text{ s}^{-1}$ ensure the temperature difference between the saturation temperature and interface temperature was always limited to 1.0 K.

2.2.4. Turbulence Model

The Shear Stress Transport (SST) k - ω model [12] was selected as the turbulence model. The transport equations for the turbulent kinetic energy and its specific dissipation rate were as follows:

$$\nabla \cdot (\rho k \bar{v}) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right] + \mu_t S^2 + \rho \beta^* k \omega \quad \text{Eq. (19)}$$

$$\nabla \cdot (\rho \omega \bar{v}) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_\omega} \right) \nabla \omega \right] + \frac{a a^*}{v_t} \mu_t S^2 + \rho \beta \omega^2 + 2(1 - F_1) \rho \frac{1}{\omega \sigma_{\omega,2}} \nabla k \nabla \omega + S_\omega \quad \text{Eq. (20)}$$

$$\mu_t = \frac{\rho k}{\omega} \frac{1}{\max \left[\frac{1}{a^*}, \frac{SF_2}{a_1 \omega} \right]} \quad \text{Eq. (21)}$$

$$a = \frac{a_\infty}{a^*} \left(\frac{a_0 + Re_t / R_\omega}{1 + Re_t / R_\omega} \right) \quad \text{Eq. (22)}$$

$$a^* = a_\infty \left(\frac{a_0^* + Re_t / R_k}{1 + Re_t / R_k} \right) \quad \text{Eq. (23)}$$

$$Re_t = \frac{\rho k}{\mu \omega} \quad \text{Eq. (24)}$$

The detailed information of F_1 , F_2 , β , β^* and a_∞ can be found in the reference of [12]. The empirical constants for the SST k- ω model were set to the default value.

3. Model Validation

In order to ensure the accuracy of the results and avoid excessive meshes leading to higher calculation costs, the mesh number is finally set as about 1.26 million. As the lack of experimental studies on R410A microtubule heat transfer, the experimental data for R404A were used as experimental validation. Figure 2 presents the simulated heat transfer coefficient and pressure gradient of condensation model compared to that of the experimental data [13]. Mean Absolute Deviation (MAD) is used to measure the deviation between simulated data and experimental data, and the MAD was defined as follows:

$$MAD = \frac{1}{N} \sum_{i=1}^N \left| \frac{y(i)_{sim} - y(i)_{exp}}{y(i)_{exp}} \right| \quad \text{Eq. (25)}$$

It can be seen from Figure 2(a) and (b) that the MAD of the heat transfer coefficient is 12.04%, and the MAD of the pressure gradient is 12.8%, and the error was in the acceptable range. Indicating the reliability of the heat transfer coefficient and pressure gradient results simulated by the condensation model. Wall temperature is one of the causes of deviation, the wall temperature cannot be completely guaranteed to be constant during the experiment. In addition, the roughness of the tube surface and the thickness of the tube are also causes of deviation.

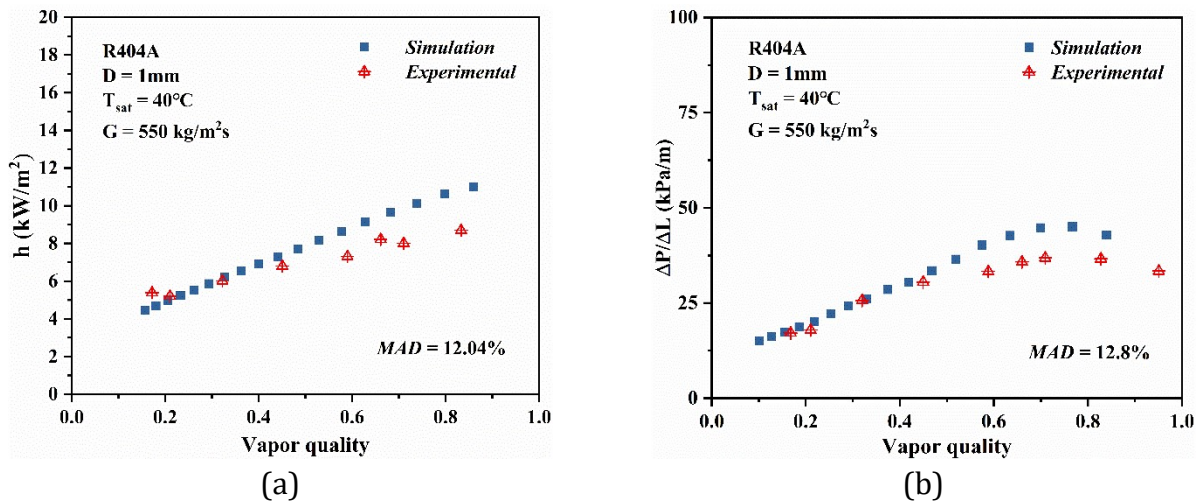


Figure 2. Comparison of simulated value and experimental value [13] of condensation model. (a) comparison of heat transfer coefficient. (b) comparison of pressure gradients.

4. Results and Discussions

4.1. Thermodynamic Performance of R410A, M1 and M2

Same conditions ($T_c=303.15K$) are simulated to discuss the thermodynamic performance of R410A and alternative refrigerants M1 and M2. It can be seen from Figure 3 that the COPs of the three refrigerants are similar; the power consumption W_c and cooling capacity Q_e of M1 and M2 are higher than those of R410A, because the vapor enthalpy and latent heat of vaporization of refrigerants M1 and M2 are higher than R410A, which leads to higher power consumption of

the compressor and cooling capacity; the isentropic efficiency of the compressor can be improved because the compression ratio PR of M1 and M2 is smaller; the discharge temperature of the three refrigerants is close; the VCC of M1 and M2 is slightly higher than that of R410A, which means that the same R410A compressor may be used for M1, M2. In addition, the GWP of M1 and M2 is 76.3% and 77.8% lower than R410A, respectively.

In conclusion, thermodynamic performance of newly proposed refrigerant mixture M1 and M2 was better than that of R410A from the stand point of PR, VCC, GWP and power savings, and the COPs of the three are similar. Indicating that M1 and M2 refrigerants can be used as an environmentally friendly substitute for R410A from the perspective of thermodynamic theory analysis.

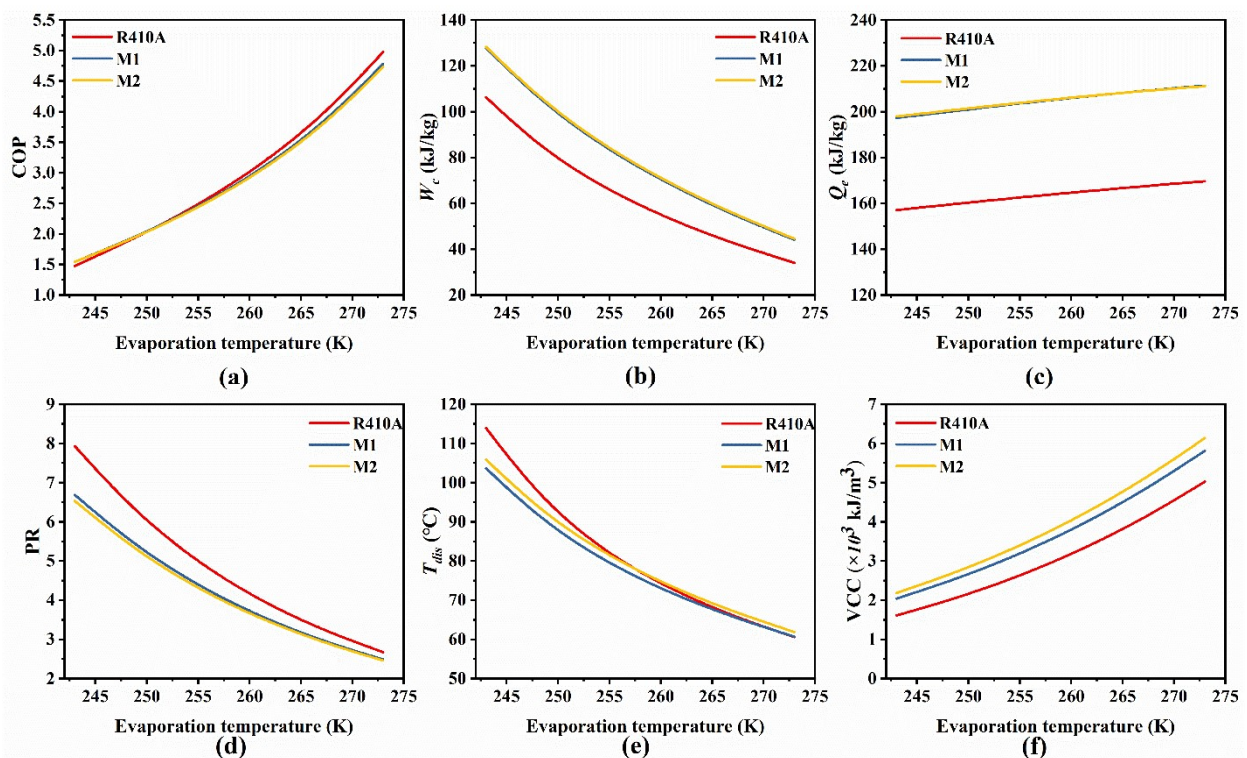


Figure 3. The thermodynamic performance of R410A, M1 and M2. (a) comparison of COP; (b) comparison of W_c ; (c) comparison of Q_e ; (d) comparison of PR; (e) comparison of T_{dis} ; (f) comparison of VCC.

4.2. Heat Transfer Characteristics of R410A, M1 and M2

4.2.1. Turbulence Model

Figure 4 depicts a direct comparison of the heat transfer coefficients between R410A, M1 and M2 when mass fluxes varied from 400 to 800 kg/m²·s. The results show that the heat transfer coefficient goes up as the vapor quality rises, the liquid film surrounding the wall is thinner when the vapor quality is large, the thermal resistance is small at this time, so the heat transfer coefficient is large. In addition, it can be seen that M1 and M2 have better heat transfer performance than R410A. Within the mass flux range discussed, the heat transfer coefficient of M1 is on average 11%-15% higher than R410A; the heat transfer coefficient of M2 is average 15%-19% higher than R410A, and under high vapor quality, the difference in heat transfer coefficient is more obvious. This is due to the different physical properties of the refrigerant. The density of M1 and M2 is lower than R410A, so under the same mass flux conditions, the flow velocity in the tube will be larger, and the thermal conductivity of M1 and M2 is higher than R410A, which will increase the heat transfer coefficient.

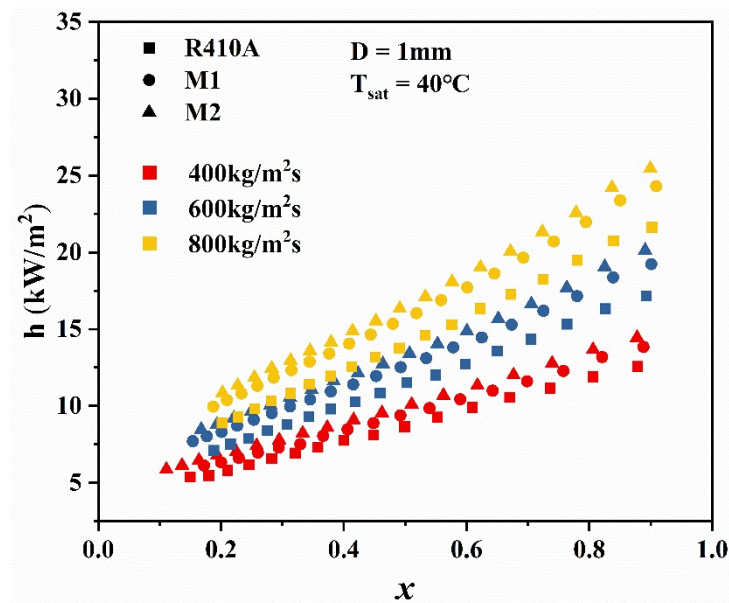


Figure 4. Comparison of condensation heat transfer coefficients between R410A, M1 and M2.

4.2.2. Pressure Gradient

The comparison of the pressure gradient of R410A, M1 and M2 when the mass flux changes from 400 to 800 kg/m²·s is shown in Figure 5. Obviously, for R410A, M1 and M2, the pressure gradient increases as the vapor quality increases. The larger velocity magnitude at higher vapor quality, and the greater the frictional resistance, which is the cause of this phenomenon. Overall, the pressure gradient of M1 was the maximum and then the R410A and the M2. On average, the pressure gradient of M1 was 8% larger than that of R410A; the pressure gradient of M2 was 3% lower than that of R410A. The results show that M1 and M2 do not differ much from R410A in terms of pressure drop because of their similar gaseous densities.

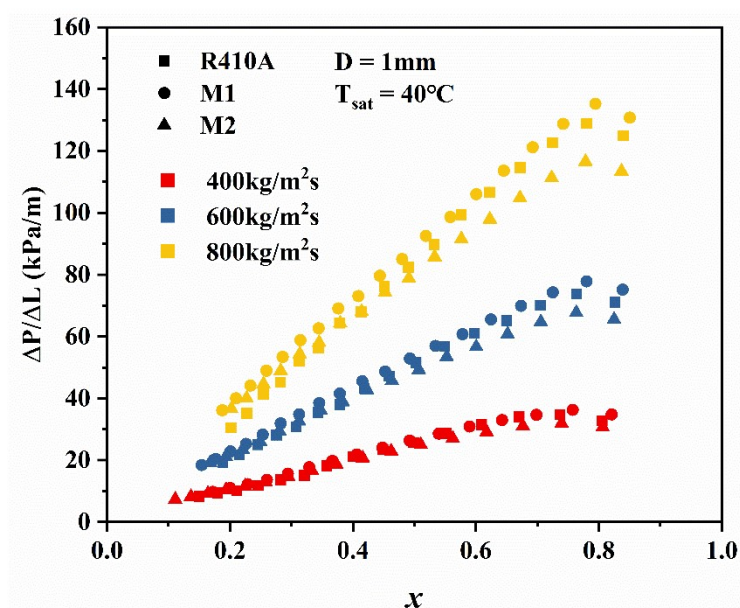


Figure 5. Comparison of condensation pressure gradients between R410A, M1 and M2.

4.2.3. Liquid Film Thickness

Figure 6 depicts change of the liquid film thickness of R410A, M1 and M2 with the increase of angular coordinates. It can be seen that the liquid film distribution of the three refrigerants is relatively uniform when the vapor mass is large, and the liquid film thickness is similar. The growth rate of R410A liquid film is similar to that of M1 and M2. The result of Figure 7 shows that as the mass flux increases, the shear stress increases, so the thickness of the liquid film decreases.

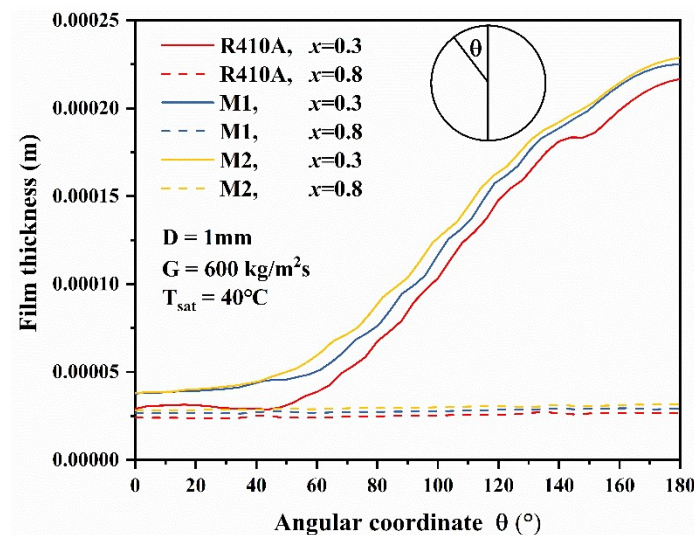


Figure 6. Comparison of condensation heat transfer coefficients between R410A, M1 and M2.

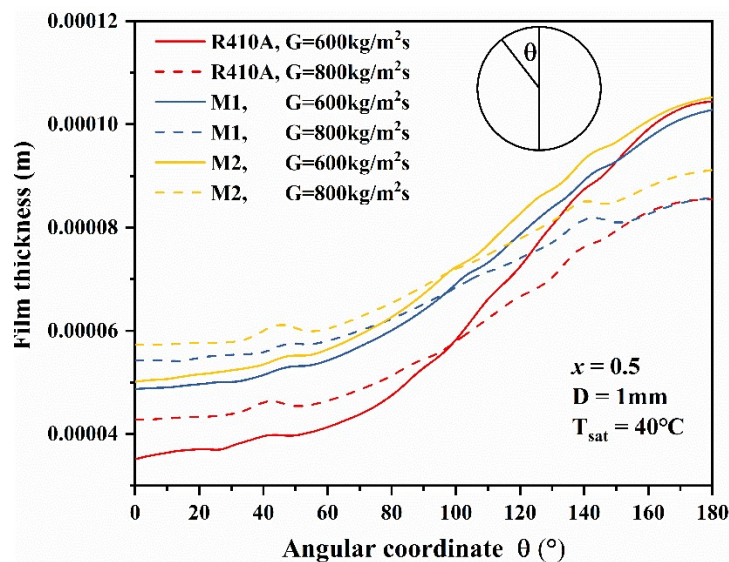


Figure 7. Comparison of condensation pressure gradients between R410A, M1 and M2.

5. Conclusion

In this study, two mixed refrigerants (M1(R32/R290,67/33), M2(R32/R1270, 63/37)) are proposed to replace R410A refrigerant. The thermodynamic performances and heat transfer characteristics of R410A, M1 and M2 were compared. In the low temperature range (243.15K-273.15K), the thermodynamic properties of M1 and M2 are similar to R410A, the GWP of M1 and M2 is 76.3% and 77.8% lower than R410A, respectively. At the same condition, the

condensation heat performance of the M1 and M2 were the best and then the R410A. When the mass flux is 400-800 kg/m²·s, the heat transfer coefficient of M1 is on average 13% higher than R410A; the heat transfer coefficient of M2 is average 17% higher than R410A. The friction pressure gradients of R410A, M1 and M2 increase with increasing in vapor quality and mass flux. R410A, M1 and M2 have little difference in pressure gradient. Based on the above conclusions, M1 and M2 are considered to be low GWP transitional refrigerants that can replace R410A.

Acknowledgments

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