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CHARACTERIZATION OF CONDENSATION HEAT TRANSFER AND PRESSURE DROP FOR REFRIGERANT R1234YF

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ABSTRACT

Substantial efforts have been made to develop low global warming potential (GWP) refrigerants in the last decade. Many studies indicate that R1234yf has similar thermodynamic properties to R134a but much lower GWP value, suggesting the opportunity of utilizing R1234yf as a substitute of R134a due to its advantages of a very low 100-year global warming potential value of 4, zero ozone depletion potential and excellent life cycle climate performance. Nevertheless, very limited studies have been conducted in the past years related to the flow condensation heat transfer of this refrigerant. The purpose of this study is to probe the condensation heat transfer and pressure drop characteristics of R1234yf in a horizontal circular minitube with an inner diameter of 4 mm. The validation of the numerical model showed that the predictions agreed well with measured results from the literature of Yang et al., and suitably simulated the development of two-phase flow patterns along the channel. We hence extended the computational fluid dynamics (CFD) simulations to assess the heat transfer and frictional characteristics of straight and converging condensed minitubes for R1234yf refrigerant at varied operating conditions of mass flux, heat flux and vapor quality, respectively.

Keywords: Condensation heat transfer, Pressure drop, R1234yf, CFD

NONMENCLATURE

Symbols	
$\vec{F}_\sigma$	Surface tension effect
$\vec{g}$	Gravity
h	Heat transfer coefficient
h_{lv}	Latent heat
q''	Wall heat flux
T_s	Saturation temperature
$T_{avg,wall}$	Average wall temperature
ρ	Density
λ_{eff}	Effective thermal conductivity
μ_{eff}	Effective dynamic viscosity

1. INTRODUCTION

To lessen the environmental impact of refrigeration systems, one possibility as a sustainable solution is the newly developed R1234yf refrigerant, having a low 100-year GWP value of 4, zero ozone depletion potential and first-rate life service cycle climate performance [1]. The thermophysical and heat transfer characteristics of R1234yf are quite similar to those of R134a [2,3], easing the usage of R1234yf refrigerant in diverse refrigeration devices. Righetti et al. [3] investigated the boiling and condensation characteristics of hydrofluoroolefin (HFO) refrigerants from all available research papers related to in-tube two-phase heat transfer studies. Col et al. [4] explored the condensation heat transfer coefficients of two refrigerants HFO-1234yf and HFC-134a in a circular channel with an inner diameter of 0.96 mm. Later, Minor and Spatz [5] examined the performance of R1234yf as an alternative to R134a in a mobile air-conditioning system. The studied results designate that the coefficient of performance (COP) and cooling capacity of R1234yf are within 4-8% of R134a performance. In addition, R1234yf has a relatively lower compression ratio and discharge temperature relative to R134a at the same operating conditions [6-8]. The

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goal of this work is to explore the thermofluid behavior in the development of condensation heat transfer for R1234yf flowing through a minitube. We compare the predictions with the experimental measurements from the paper of Yang et al. [9] to validate the numerical model as well as to better understand the underlying physics of two-phase flow and thermal processes in the condensation heat transfer of refrigerant R1234yf. The predicted local heat transfer coefficients and pressure drops for circular tubes are also compared with broadly utilized empirical correlations. We therefore extend the computational fluid dynamics (CFD) simulations by the ANSYS/Fluent® software to evaluate the frictional and heat transfer coefficients at varied operating conditions of mass flux, heat flux, and vapor quality.

2. NUMERICAL METHODOLOGY

The numerical model treats both liquid and vapor of R1234yf refrigerant as incompressible uniform-property flows to simulate the two phase flow structure during the condensation heat transfer process in a horizontal mini-tube. The analysis is based on the time-dependent, three-dimensional (3D) conservation equations of mass, momentum and energy as below.

$$\nabla \cdot \vec{V} = 0, \tag{1}$$

$$\frac{\partial}{\partial t}\left(\rho \vec{V}\right) + \rho \vec{V} \cdot \nabla \vec{V} = -\nabla p + \nabla \cdot (\mu_{eff} \nabla \vec{V}) + \rho \vec{g} + \vec{F}_{\sigma}. \tag{2}$$

Here $\vec{V}$, p, ρ, μ_{eff} and $\vec{g}$ stand for the velocity vector, pressure, density, effective dynamic viscosity (including the laminar viscosity μ and turbulent viscosity μ_t) and gravity force, respectively. $\vec{F}_{\sigma}$ characterizes the surface tension effect at the interface, which can be modeled via the continuous surface force (CSF) model [10, 11]. We solve the volume fraction to determine the fluid distribution as below.

$$\frac{\partial \alpha}{\partial t} + \nabla \cdot \left(\vec{V} \alpha\right) = S_m. \tag{3}$$

The signs α_m and S_m are the volume fraction of the m (liquid (l) or vapor (v)) phase and mass source term due to phase change. The density and viscosity of mixture are computed by the following equations to determine the physical properties in a computational cell with the known volume fractions.

$$\rho = \alpha_l \rho_l + \alpha_v \rho_v; \quad \mu_{eff} = \alpha_l \mu_{eff\,l} + \alpha_v \mu_{eff\,v}, \tag{4}$$

here ρ_l, ρ_v, μ_{effl} and μ_{effv} are the density and effective viscosity for the liquid and vapor phase of R1234yf with the values of 1127.2/28.3 kg/m^3 and 1.739×10^{-4}/ 1.07×10^{-5} N-s/m^2 at the saturation temperature and pressure of 15 °C as the baseline case. The energy equation can be written as below.

$$\frac{\partial(\rho E)}{\partial t} + \nabla \cdot \left(\rho E \vec{V}\right) = \nabla \cdot \left(\lambda_{eff} \nabla T\right) + S_m h_{lv}. \tag{5}$$

The effective thermal conductivity λ_{eff} consists of both laminar and turbulent parts as follows.

$$\lambda_{eff} = \alpha_l \lambda_{eff\,l} + \alpha_v \lambda_{eff\,v}. \tag{6}$$

The term h_{lv} represents the latent heat of vaporization in the energy source term, indicating the heat transfer related to the condensation or evaporation occurred at the interface. The mass source term S_m involved in the phase change are expressed as [12]:

$$\text{If } T \le T_{sat} \text{ (condensation)}: S_m = r \alpha_v \rho_v \frac{T_{sat} - T}{T_{sat}};$$

$$T \ge T_{sat} \text{ (evaporation)}: S_m = -r \alpha_l \rho_l \frac{T - T_{sat}}{T_{sat}}. \tag{7}$$

Here r is an empirical coefficient called mass transfer intensity factor with the unit of s^{-1}, which is determined by comparing the model predictions with experimental data. As a two-layer model combining the *k-ω* model in the near-wall area and the *k-ε* model (with a *k-ω* form) in the outer wake area for providing reliable solutions of wall bounded flows, this study implements the shear stress transport (SST) *k-ω* model for turbulence closure. The expression for the heat transfer coefficient h can be calculated by the ratio of heat flux q'' to temperature difference between the saturation temperature T_s and average wall temperature $T_{avg,wall}$ as follows:

$$h = \frac{q''}{\left(T_s - T_{avg,wall}\right)}. \tag{8}$$

A pressure-based solver was employed with a second-order upwind scheme adopted to treat the convective and diffusion terms for spatial discretization. Moreover, a first-order implicit scheme was used for the transient term, whereas an explicit time marching technique was utilized to determine the volume fraction distributions. In the analysis, the pressure-implicit with splitting of operators (PISO) numerical algorithm is implemented for velocity-pressure coupling. With the total grids of 134363 and CFL= 0.1 (i.e., the time step of 2x10^{-5} s), the normalized residual errors of flow variables converge to 10^{-5} for calculations of each time-step with the mass balance check within 1%.

3. RESULTS AND DISCUSSION

Numerical computations are conducted by the CFD software ANSYS/Fluent® to investigate the thermo-fluid behavior of R1234yf in a condensed horizontal minitube having an inner diameter of 4 mm. Figure 1 shows the structure of the minitube and numerical grids. Figure 2 illustrates a comparison of the predicted heat transfer coefficients with the measured data for the mass flux of 200 kg/m^{2}s, heat flux of 9.9-10.5 kW/m^2 and average vapor quality of 0.13-0.84 at a saturation temperature of 15 °C [9]. Within the gravity-dominated flow regime, the local heat transfer coefficient is related to the temperature difference between the bulk refrigerant and wall. The simulated results reveal the attainment of better heat transfer outcomes as a result of a greater temperature gradient of the wall from the presence of thin liquid film thicknesses. In the entrance area of the saturated vapor through the channel, the annular flow pattern can form slim liquid film thicknesses to produce elevated thermal performance. The decreasing vapor quality also tends to coagulate liquid layer thicknesses, and thereby develops greater fluid thermal resistance with local heat transfer coefficients deteriorated for the condensation evolution along the minitube. In general, the maximum inconsistency of predicted heat transfer coefficients with the measurements is 19.2%, with the simulation results greater than those of experimental data over the entire range of vapor quality. It could be attributed to the difficulties of keeping a stable uniform heat flux condition during the experiments, leading to higher uncertainties in assessing the condensation heat transfer coefficients. On the other hand, the numerical results fall within the relative accuracy (±20%) of the experiment. Largely, the predictions are in reasonable agreement with the measured data.

Figure 3 presents a comparison of the predictions of pressure drops with the measured results for the mass flux of 200 kg/m^{2}s, heat flux of 9.9-10.5 kW/m^2 and average vapor quality of 0.13-0.84 at a saturation temperature of 15 °C [9]. Both the measurements and predicted results indicate an increase in pressure drop with respect to the vapor quality because the velocity of refrigerant escalates with vapor quality, leading to high pressure drops. As can be seen, the calculations are in reasonably good accordance with the measured results, revealing the peak discrepancy within 20%.

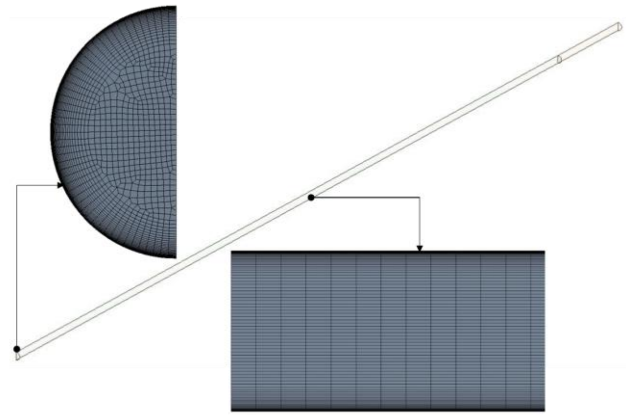

Fig 1 Structure of minitube and numerical grids

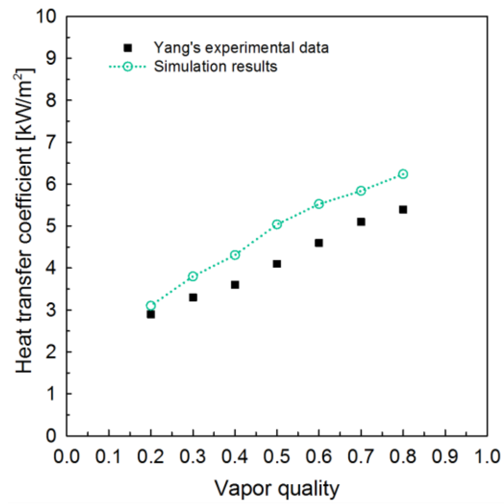

Fig 2 Comparison of predicted heat transfer coefficients with measured data for the mass flux of 200 kg/m^{2}s, heat flux of 9.9-10.5 kW/m^2 and average vapor quality of 0.13-0.84 at a saturation temperature of 15 °C

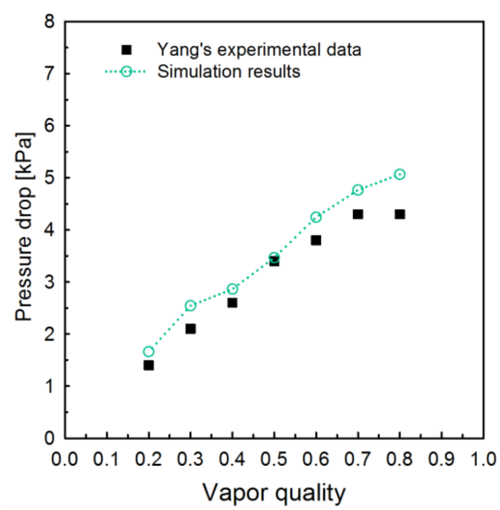

Fig 3 Comparison of predicted pressure drops with measured data for the mass flux of 200 kg/m^{2}s, heat flux of 9.9-10.5 kW/m^2 and average vapor quality of 0.13-0.84 at a saturation temperature of 15 °C

Figure 4 presents the predictions of liquid film thickness profiles around the tube for the mass flux of 200 kg/m^{2}s and the heat flux/vapor quality ranging 9.9-10.5 kW/m^2 /0.2-0.8, respectively, at a saturation temperature of 15 °C. In effect, thicker liquid films appear at relatively low vapor qualities. This trend indicates the occurrence of active condensation flow process to amalgamate thick liquid films, and thereby decline heat transfer results. In addition, relatively thinner film thicknesses stay almost constant at the angular range of 0° to 90°, and there is a clear increase in film thickness along the circumstance toward the tube bottom for all vapor qualities, revealing the condensate liquid at the top of the tube flowing to the bottom under gravity. The liquid film remains nearly invariant thickness of approximately 0.007 mm along the circumference direction at x = 0.8.

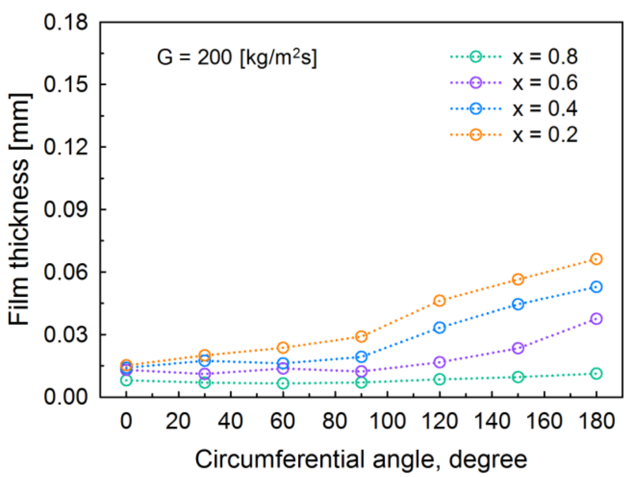

Fig 4 Predicted liquid film thickness profiles around the tube for the mass flux of 200 kg/m^{2}s, heat flux of 9.9-10.5 kW/m^2 and vapor quality of 0.2-0.8 at a saturation temperature of 15 °C

4. CONCLUSIONS

This investigation conducts the CFD simulations via the ANSYS/Fluent® software to probe the condensation heat transfer and frictional performance of R1234yf in a minichannel. To validate the computational model, the greatest differences of both the predicted heat transfer coefficients and frictional pressure drops are reasonably within 25%, comparing with the literature of Yang et al for the mass flux of 200 kg/m^{2}s, heat flux of 9.9-10.5 kW/m^2 and average vapor quality of 0.13-0.84 at a saturation temperature of 15 °C. The predicted results reveal both the condensation heat transfer coefficients and pressure drops increase with the vapor quality. The liquid films appear nearly unvarying thickness of around 0.007 mm along the circumference direction at x = 0.8. At low vapor qualities, the film at the bottom of the minitube becomes thicker due to the gravity effect, while the film thickness remains almost constant in the entire upper half of the tube.

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