

The effect of gravity on R410A condensing flow in horizontal circular tubes

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Abstract

Heat transfer characteristics of R410A condensation in horizontal tubes with inner diameter of 3.78 mm under normal and reduced gravity are investigated numerically. The mass transfer model and numerical methods are validated by comparing the numerical heat transfer coefficients under normal gravity with the experimental work and empirical correlations. The results indicate that the heat transfer coefficients increase with increasing gravitational accelerations at a lower mass flux, while the difference of these under varying gravity is insignificant at a higher mass flux. The liquid film thickness decreases with increasing gravity at the top part of the tube at vapor quality $x = 0.5$ and 0.9 , while the reverse is true for that at the tube bottom. The average liquid film thickness is nearly the same under different gravity accelerations at the same vapor quality and mass flux. The local heat transfer coefficients increase with increasing gravity at the top of the tube and decrease with increases in gravity at the bottom. The proportion of the thin liquid film region is important for the overall heat transfer coefficients for the condensing flow. A vortex with its core lying at the bottom of the tube is observed under normal gravity because of the combined effect of gravity and the mass sink at the liquid-vapor interface, while the stream traces point to the liquid-vapor interfaces under zero gravity. The mass transfer rate under zero gravity is much lower than that of normal gravity.

Keywords: Condensation; Gravity; Heat transfer; Numerical simulation; Two-phase flow

Nomenclature

A	Area, m^2	Greek letters	
c_p	Specific heat capacity, $\text{J kg}^{-1} \text{K}^{-1}$	α	Volume fraction, [–]
d_h	Hydraulic diameter, m	κ	Curvature of the interface, m^{-1}
E	Specific sensible enthalpy, J kg^{-1}	μ	Dynamic viscosity, Pa s
F	Surface tension force, N m^{-3}	μ_t	Turbulent viscosity, Pa s
g	Gravitational acceleration, m s^{-2}	ρ	Density, kg m^{-3}
G	Mass flux, $\text{kg m}^{-2} \text{s}^{-1}$	σ	Surface tension, N m^{-1}
h	Heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$	Subscripts	
h_{lv}	Latent heat of vaporization, J kg^{-1}	a	Acceleration
k	Thermal conductivity, $\text{W m}^{-1} \text{K}^{-1}$	cal	Results predicted by correlation
m	Mass source due to phase change, $\text{kg m}^{-3} \text{s}^{-1}$	sat	Saturation status
MAD	Mean average deviation, [–]	sim	Results obtained by simulation
MRD	Mean relative deviation, [–]	tp	Two phase
r	Coefficient of mass source, s^{-1}	v	Vapor phase
Pr_t	Turbulent Prandtl number, –	wall	Wall
T	Temperature, K		
v	Velocity, m s^{-1}		
x	Vapor quality, [–]		

1. INTRODUCTION

Compared with single-phase flows, two-phase flows can take advantage of the latent heat rather than the sensible heat alone and achieve heat transfer enhancement. These flows have the potential to meet the increasing demand of high heat dissipation in thermal management systems for electronic equipment, energy systems and space missions. With the growing interest in space applications and power requirements of the equipment on space vehicles, more studies are focused on the two-phase flow like pool boiling, flow boiling, and condensation under reduced and micro gravity conditions.

The gravity effect is reflected in buoyancy which is determined by gravity and the density difference between liquid and vapor. Gravity can play a crucial role in the bubble detachment and the critical heat flux in pool and flow boiling, and it also influences the flow patterns in flow boiling and condensation. Plenty of experimental and numerical investigations have been contributed on the gravity effect on nucleate and film pool boiling [1-4]. Compared with pool boiling, flow boiling could enhance critical heat flux (CHF) in microgravity because the liquid motion could take the bubbles away and replenish the wall. Konishi and Mudawar [5] made a thorough review about the flow boiling and CHF in microgravity discussing several models for predicting CHF and highlighting the need for further research emphasis on condensation flow and heat transfer characteristics under reduced gravity.

Compared with flow boiling and nucleate boiling investigations, research work on condensation under microgravity were limited. Lee et al. [6] experimentally investigated the condensing flow of FC-72 under microgravity. The reduced gravity environment was achieved by parabolic flights. At lower mass fluxes, the condensation heat transfer coefficient was determined by the thermal conduction in the liquid film. The combined effects of turbulence and interfacial waviness were found to enhance heat transfer downstream at higher mass fluxes. The empirical correlations based on the normal gravity were found to reasonably predict the microgravity data. They [7] also conducted condensation experiments under microgravity, Lunar gravity, and Martian gravity. The results showed that gravity effects were pronounced at lower mass flux. Bortolin et al. [8] reported four experiments

conducted under reduced gravity, including condensation of HFE on external surfaces, falling films condensation, in-tube condensation of n-pentane, and convective condensation of R134a in a square channel.

Other experimental works relating to the gravity effect on condensation have been mainly conducted in inclined tubes. By changing the tube inclination, the gravity component along the radial direction decreases. Lips and Meyer [9] undertook a review on two-phase condensation flow in inclined tubes, including flow pattern maps, void fractions, heat transfer coefficients, and pressure drops. The inclination angle affected the flow patterns, leading to a significant influence on heat transfer coefficients and pressure drops. An optimum inclination angle could be found for a specific configuration. Akhavan-Behabadi et al. [10] carried out an experimental work of R134a condensation in a microfin tube with inner diameter of 8.92 mm. The results indicated that the highest heat transfer coefficients were obtained when the inclination angle was 30°. The effects of inclination angles on heat transfer coefficients were more pronounced at lower mass flux, which is similar to the conclusion of Lee et al. [7]. The flow patterns of condensing flow in microfin tubes were presented in Mohseni and Akhavan-Behabadi [11]. Lips and Meyer [12, 13] investigated the effects of inclination angles on the convective condensation for R134a in a 8.38 mm smooth tube at mass fluxes ranging from 200 to 600 kg m⁻² s⁻¹ and vapor quality ranging from 0.1 to 0.9. The highest heat transfer coefficient was found at an inclination angle of -15 °, while the heat transfer was deteriorated for the upward flow. Del Col et al. [14] experimentally studied the condensation processes in mini square channels for R134a and R32 at a saturation temperature of 313.15 K and mass fluxes between 100 and 390 kg m⁻² s⁻¹. A correlation to determine the critical mass flux at which the inclination angle starts to influence the heat transfer of condensation was proposed.

Compared with the experimental studies, numerical work on condensing flow is limited. For the numerical studies relating to the multiphase flow, the VOF method and the Level-set method have been widely adopted to track the liquid-vapor interfaces. The transient and steady numerical simulations performed have been mainly focused on different aspects. The former method was adopted to study the interfacial waviness, the intermittent flow patterns like bubbly/slug flow or the transitions of the flow pattern regimes. The studies using steady

simulations have mainly focused on the steady liquid film thicknesses, the liquid-vapor interfaces, and the local heat transfer coefficients. Ganapathy et al. [15] studied the transient condensing flow in micro channels ($d_h = 100 \mu\text{m}$) using two-dimensional models. The numerical flow patterns, heat transfer coefficient, and pressure drops fitted well with other studies in the open literature. The effect of the wall heat flux on the heat transfer coefficients was less important than that of the mass flux. Lee et al. [16] conducted experimental and computational investigation of vertical down flow condensation. The computed heat transfer coefficient fitted well with the experimental results. Chen et al. [17] performed a three-dimensional simulation of FC-72 in a rectangular microchannel with a 1 mm hydraulic diameter. Though the cross-sectional meshes were coarse, the numerical work captured the transition from the annular flow to the bubbly/slug flow and presented some vectors and temperature distributions for these flow patterns. Other numerical studies were mainly performed using the steady model in which the liquid film waviness and the intermittent flow patterns could not be predicted. Wang and Rose [18-23] have made significant contributions to the condensation numerical work using the laminar flow assumption for the liquid film. The effects of surface tension, gravity, and tube shapes on condensation heat transfer coefficients were explored in detail. Compared with surface tension, gravity is unnoticeable in the non-circular channels. However, most of the numerical results showed that the heat transfer coefficient was not dependent on the mass flux. Da Riva et al. [24-26] adopted the low-Reynolds SST $k\sim\omega$ model to consider the turbulence effect in the liquid film and obtained a good agreement between the numerical results and the corresponding experimental data. Based on this approach, Bortolin et al. [27] studied the effect of channels shapes and Zhang et al. [28] investigated the effect of saturation temperatures on the condensing flow.

As most of the condensation work was carried out at normal gravity, the effect of gravity on the condensing flow has not been fully studied. Such work will aid the designer in producing more efficient and reliable thermal control systems for space vehicles or other equipment under reduced and micro gravity. Former numerical works relating to condensation under microgravity mainly used R134a as working fluid [24], while the simulations using R410A are limited. Obviously, different fluid properties will lead to different conclusions. The

present work mainly focuses on the effects of gravity on the heat transfer characteristics for R410A condensing flow inside circular smooth tubes with an inner diameter of 3.78 mm. The numerical local heat transfer coefficients are also compared with widely used correlations to validate the numerical method adopted in this work. The local heat transfer coefficients, liquid film thickness, and stream traces are presented to give a better understanding of the gravity effect on condensation.

2. NUMERICAL MODEL

The numerical simulation was conducted using ANSYS Fluent (Release 14.5) with the VOF model to capture the liquid-vapor interfaces. Detailed descriptions of the numerical model, numerical discretization methods, and boundary conditions can be found in our previous work [28]. Here, only a brief introduction of this method is given.

2.1. Governing equations

Turbulent effects in the liquid and vapor flows during the condensation process are considered as recommended by Da Riva et al. [26], using the SST $k-\omega$ turbulent model. The same approach was also adopted in Bortolin et al. [27], Kharangate et al. [29], and Lee et al. [16]. The basic equations for the continuity, momentum, energy, turbulence kinetic energy, and the specific dissipation rate are the same with those of single-phase flow as listed in Eq. (1) to Eq. (3). A volume fraction equation is coupled with these equations to distinguish the two phases. The properties used in these equations are the volume-fraction-weighted values of liquid and vapor.

A continuum surface force (CSF) model [30] is used to consider the surface tension effect which is of importance for two-phase flow in mini/micro channels. The phase-change phenomenon is modeled by adding source terms in the volume fraction equation and the energy equation using the user-defined-functions (UDF) in Ansys Fluent. As all these equations are coupled, the effect of the source terms in the volume fraction and energy equations will also influence the velocity and pressure distributions.

Continuity Equation:

$$\nabla \cdot (\rho \vec{v}) = 0 \quad (1)$$

Momentum Equation:

$$\nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla P + \nabla \cdot \left[(\mu + \mu_t) (\nabla \vec{v} + \nabla \vec{v}^T) \right] + \rho \vec{g} + \vec{F} \quad (2)$$

Energy Equation:

$$\nabla \cdot [\vec{v} (\rho E + P)] = \nabla \cdot [k (\nabla T)] + h_v m_t \quad (3)$$

Volume Fraction Equation:

$$\vec{v} \cdot \nabla \alpha_i = \frac{m_i}{\rho_i} \quad (4)$$

Turbulence Kinetic Energy Equation:

$$\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_i} (\rho k v_i) = \frac{\partial}{\partial x_j} \left(\Gamma_k \frac{\partial k}{\partial x_j} \right) + G'_k - Y_k + S_k \quad (5)$$

Specific Dissipation Rate Equation:

$$\frac{\partial}{\partial t} (\rho \omega) + \frac{\partial}{\partial x_i} (\rho \omega v_i) = \frac{\partial}{\partial x_j} \left(\Gamma_\omega \frac{\partial \omega}{\partial x_j} \right) + G_\omega - Y_\omega + S_\omega + D_\omega \quad (6)$$

where:

$$\alpha_l + \alpha_v = 1 \quad (7)$$

$$\vec{F} = \frac{2\sigma\rho\kappa_l\nabla\alpha_l}{\rho_l + \rho_v} \quad (8)$$

More details about the definitions of the parameters in these equations could be found in Zhang et al. [28] and the Ansys Fluent manuals.

The source terms for the mass transfer in Eq. 3 and Eq.4 are based on the Lee [31] model which is widely adopted for condensation[16, 17, 25-29] and evaporation[32, 33] simulations. As shown in Eq. 9, when the temperature of the vapor phase in a computational cell is lower than the saturation temperature, a negative mass source term is expected for the vapor phase corresponding to the condensation process, and vice versa for the evaporation phenomena. The volume fraction of the liquid or the vapor phase is adopted in the mass transfer model, which makes sure that the mass transfer only occurs near the liquid-vapor interface where the

volume fractions are between 0 and 1. In the cases of lower liquid temperature or higher vapor temperature than the saturation temperature, the mass term transferring through the liquid-vapor interface is zero [28].

In the Lee [31] model, a coefficient r is adopted which depends on the particular cases under study. For the evaporation and boiling simulations, a smaller r is usually applied, while a much higher r is adopted in the condensation numerical work. The values of r for the condensation simulations range from 5000 s^{-1} in Liu et al. [34] to $5 \times 10^6 \text{ s}^{-1}$ in Da Riva et al. [26] to keep the interface temperature within $T_{\text{sat}} \pm 1 \text{ K}$. In the present work, $r = 1.5 \times 10^6 \text{ s}^{-1}$ is adopted, for which the difference between the calculated liquid-vapor interface temperature and saturation temperature is lower than 0.5 K .

$$m_v = \begin{cases} r \rho_v a_v \frac{T - T_{\text{sat}}}{T_{\text{sat}}}, & \text{if } T \leq T_{\text{sat}} \\ -r \rho_l a_l \frac{T - T_{\text{sat}}}{T_{\text{sat}}}, & \text{if } T > T_{\text{sat}} \end{cases} \quad (9)$$

$$m_l = -m_v \quad (10)$$

2.2. Computational Domain

Fig. 1 shows the computational domain for a horizontal smooth tube studied in the present work. A velocity inlet boundary condition is applied at the tube inlet with a constant temperature ($T_{\text{sat}} = 320 \text{ K}$), turbulent kinetic energy, and the turbulent specific dissipation rate. A pressure outlet boundary condition is adopted at the downstream boundaries to avoid reverse flow. A symmetry boundary condition is applied on the YZ plane. The normal gradients for all solved variables are zero on this plane. A no-slip boundary condition with a constant temperature ($T_w = 310 \text{ K}$) is used as the wall boundary condition. The properties of R410A used in the present work are obtained from REFPROP 8.0. The gravity effect is considered by the gravitational acceleration along the Y axis as shown in Fig. 1.

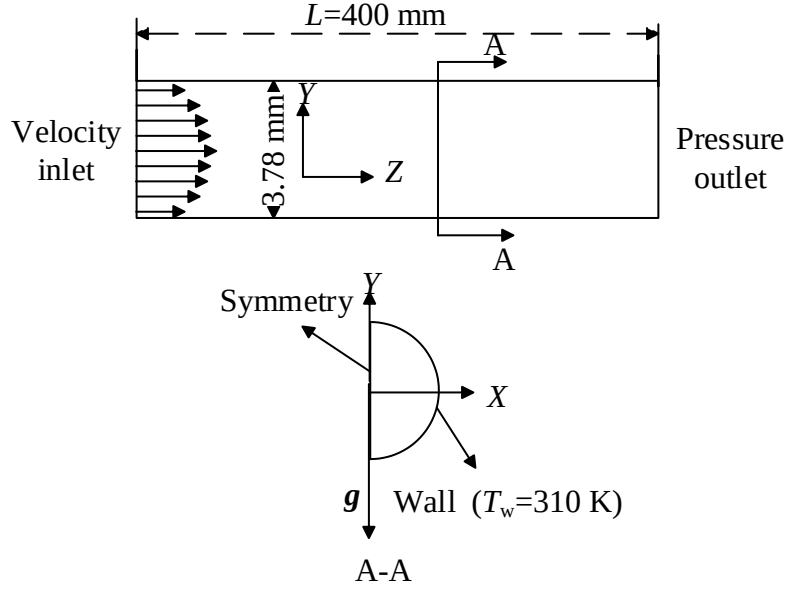


Fig. 1 computational geometry and boundary conditions.

The values for inlet velocity together with the turbulent kinetic energy and the turbulent specific dissipation rate are obtained by periodic single-phase simulations. After obtaining the distributions of the fully-developed single-phase velocity, turbulent kinetic energy, and turbulent specific dissipation rate, a profile file is used to convert these values into the condensation simulations. The length of the single-phase computational model is $10 d_h$ in length, and the cross-sectional mesh distribution is the same to that of the two-phase numerical domain. Periodic boundary conditions are applied at the tube inlet and the outlet. The velocity components repeat themselves in space with a constant pressure drop between modules. The equations of the periodic boundary conditions are shown below:

$$\vec{v}(z) = \vec{v}(z + L) = \vec{v}(z + 2L) = \dots \quad (11)$$

$$\Delta P = P(z) - P(z + L) = P(z + L) - P(z + 2L) = \dots \quad (12)$$

The hexahedral cells with a fine boundary layer mesh near the wall region is used in the present work, while much coarser meshes are adopted in the core region where the vapor phase dominates. An enhanced wall treatment, which is the default for all ω -equation based turbulent models in Ansys Fluent, is applied to solve the viscous sublayer and the logarithmic layer near the tube wall where a liquid film exists. Similar to Da Riva et al. [26], a mesh sensitivity study is conducted under earth gravity in terms of the average heat transfer

coefficients at the wall. The average heat transfer coefficients difference between the fine mesh (around 2.66 million elements) and the coarse mesh (around 1.35 million elements) is less than 0.6%. In this study, the coarse mesh is used.

3. RESULTS AND DISCUSSIONS

3.1. Flow patterns and liquid-vapor interfacial temperatures

Fig. 2 shows the flow pattern maps proposed by Dobson and Chato [35], Cavallini et al. [36], Soliman [37], and Cavallini et al. [38]. As shown in Fig. 2, annular flow occurs for the mass flux $G=307 \text{ kg m}^{-2} \text{ s}^{-1}$ and $G=720 \text{ kg m}^{-2} \text{ s}^{-1}$ at vapor quality ranging from 0.45 to 0.95, and all the cases studied belong to the ΔT -independent region. The range of the annular flow regime extends with decreasing gravity. As the steady-state simulation cannot predict the wavy and slug/bubbly flow regimes, the effect of fluctuation on the liquid-vapor interface is ignored in the present study.

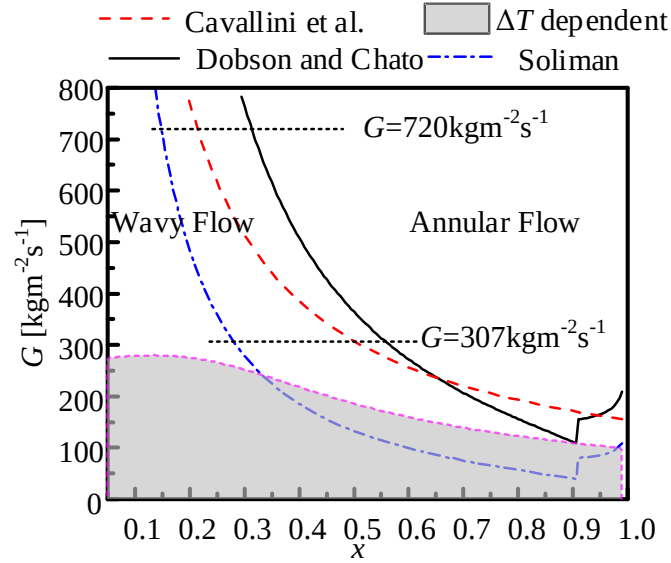


Fig. 2 Flow pattern maps proposed by Dobson and Chato [35], Cavallini et al. [36], Soliman [37], and Cavallini et al. [38].

As mentioned before, a constant temperature is assumed at the liquid-vapor interface in the Lee [31] model. However, the liquid-vapor interfacial temperature varies during the computational process. A rational r is chosen to keep the computational interfacial temperature within a small range of the refrigerant saturation temperature. Fig. 3 shows the interfacial temperature at x ranging from 0.5 to 0.9 plotted against the angular coordinate

moving from the tube top to the bottom. For the r adopted in the present work, the interfacial temperatures range from 319.5 to 319.9 K, which indicates that $r=1.5\times10^6 \text{ s}^{-1}$ is big enough to perform the Lee [31] model.

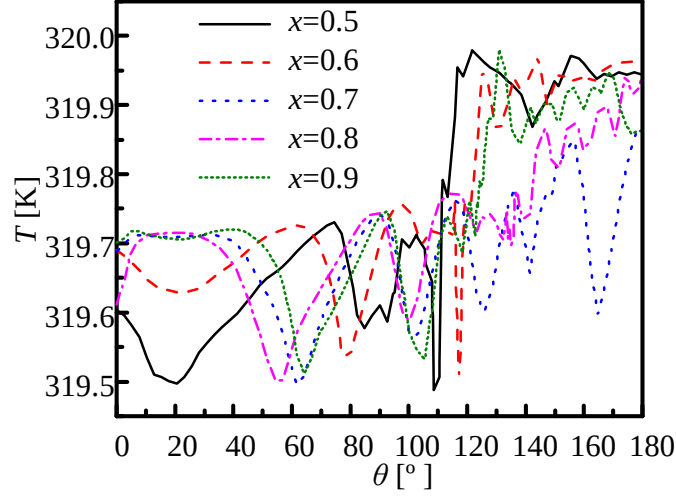


Fig. 3 Temperatures of the liquid-vapor interfaces.

3.2. Comparison of numerical data with experimental results and empirical correlations under normal gravity

The heat transfer coefficients for the condensing flow obtained using the models described before are compared with the experimental data in our previous experimental studies [39]. More details about the experimental setup and data reduction can be found in Wu et al. [39]. Fig. 4 shows the comparison between the numerical heat transfer coefficient obtained by the average local heat transfer coefficient at a local vapor quality and the experimental results. The numerical heat transfer coefficient increases with increasing mass flux and falls into the relative accuracy ($\pm 15.4\%$) of the experimental setup.

$$h = \frac{1}{A} \int \frac{q_{wall}}{T_{sat} - T_{wall}} dA \quad (13)$$

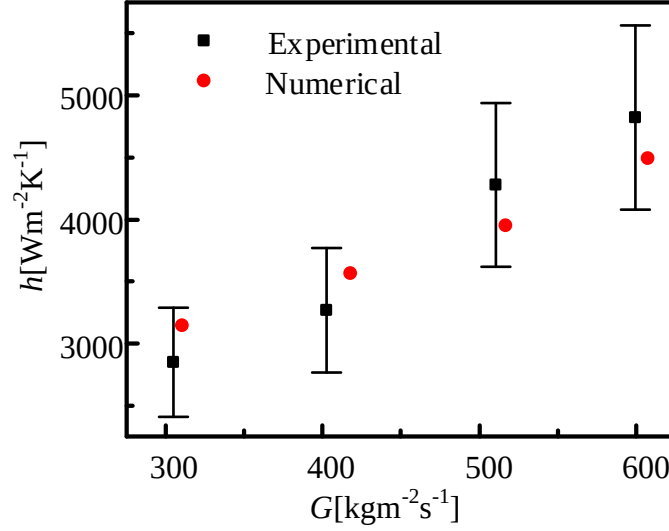


Fig. 4 Comparison of numerical data with experimental results at $x=0.45$ [40].

The local numerical heat transfer coefficients are also compared with empirical correlations developed by Cavallini et al. [38], Thome et al. [41], Shah [42]. As shown in Fig. 5, the the heat transfer correlations can predict the numerical data well. The mean absolute deviation (MAD) and the mean relative deviation (MRD) for these correlations, which are calculated in Eq. 14 and Eq. 15, are listed in Table 1. The maximum MAD for these three correlations is about 13.0 %, which validates the Lee [31] model and the numerical method used in the present work. In the following work, the same heat and mass transfer model is adopted in the numerical studies of R410A condensation under reduced and micro gravity.

$$\text{MAD} = \frac{1}{N} \sum_{i=1}^N \left| \frac{U_{pre} - U_{sim}}{U_{sim}} \right| \bullet 100\% \quad (14)$$

$$\text{MRD} = \frac{1}{N} \sum_{i=1}^N \frac{U_{pre} - U_{sim}}{U_{sim}} \bullet 100\% \quad (15)$$

where U is the value of a numerical or calculated data point.

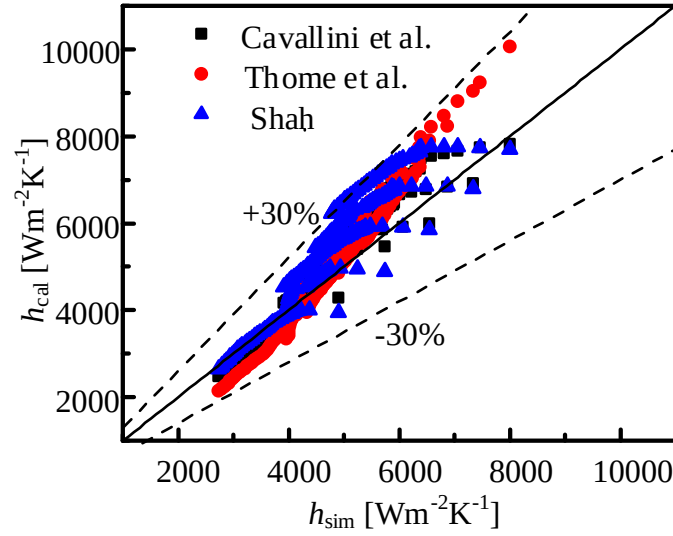


Fig. 5 Comparison with empirical correlations for heat transfer coefficients.

Table.1

MAD and MRD calculated by different correlations.

Correlations	MAD (%)	MRD (%)
Cavallini et al. [38]	8.9	5.7
Thome et al. [41]	9.3	1.1
Shah [42]	13.0	11.7

3.3. Liquid-vapor interfaces, liquid film thicknesses, and local heat transfer coefficients under reduced gravity

For a better understanding of the condensation process under various gravities, liquid-vapor interfaces are shown in Fig. 6. The gravity effect tends to drive the liquid-vapor interface from the upper part to the tube bottom, leading to a thin liquid film at the top and a thicker liquid film at the bottom. The liquid film thickness at the top of tubes remains nearly constant at each vapor quality except for the case of $0\ g$ in which the liquid film at the top increases with vapor qualities, while the liquid film thickness at the bottom tends to decrease with decreasing gravity. When the gravity effect is neglected, a total axisymmetric annular flow is observed in Fig. 6 (b) and (d). At higher mass fluxes and vapor qualities, the shear force is more pronounced, and the liquid-vapor interfaces are nearly symmetrical for all cases.

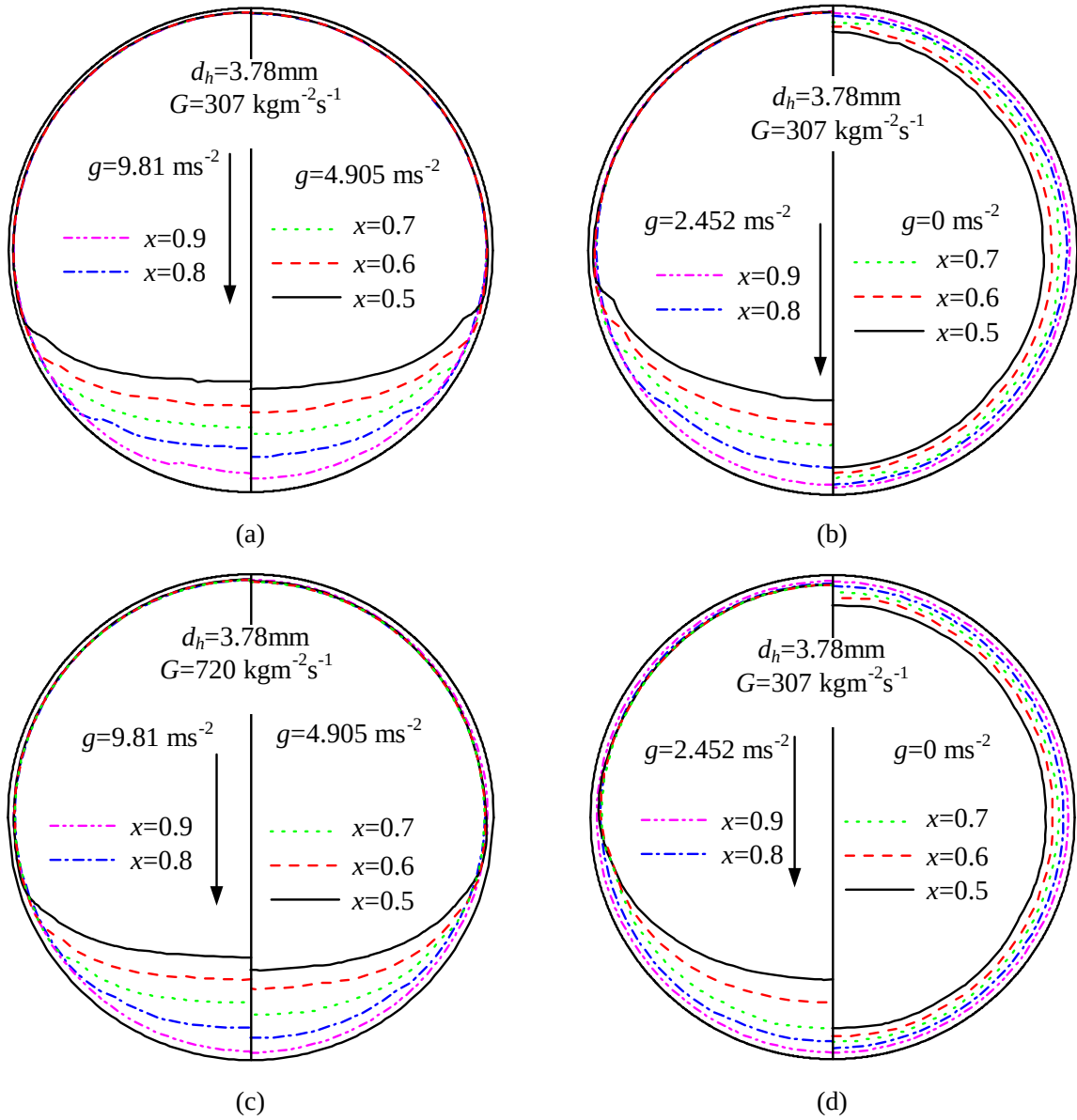


Fig. 6 Liquid-vapor interfaces inside round tubes under different gravitational accelerations at $G=307$ and $720 \text{ kg m}^{-2} \text{ s}^{-1}$.

Fig. 7 illustrates a quantitative comparison of the local liquid film thickness, defined along the local orthogonal directions, versus the angular coordinate moving from the tube top to the bottom under different gravity at $x = 0.9$ and $x = 0.5$. For the cases of zero gravity, the liquid film thickness remains constant along the angular coordinate. The liquid film thickness decreases with increasing gravity at the top part of the tube for both vapor qualities, while the reverse is true at the bottom. The difference between the maximum and the minimum liquid film thickness decreases with decreasing gravitational accelerations, which means that a more axisymmetric liquid-vapor interface will be formed at lower gravity. The liquid film thickness

remains nearly constant from the tube top ($\theta = 0^\circ$) to $\theta = 105^\circ$ under normal gravity at $x = 0.5$, and it increases dramatically at $\theta > 105^\circ$ because of the liquid accumulation due to the gravity effect. It should be noted that the minimum liquid film thickness is obtained at an angle close to 105° , while this angle decreases with decreasing gravity accelerations. The average liquid film thickness is nearly the same under different gravitational accelerations at the same vapor quality and mass flux. The average liquid film thicknesses are about $56 \mu\text{m}$ at $x=0.9$ and $230 \mu\text{m}$ at $x = 0.5$ at $G = 307 \text{ kg m}^{-2} \text{ s}^{-1}$ for various gravity, while these values decrease to $50 \mu\text{m}$ at $x=0.9$ and $215 \mu\text{m}$ at $x = 0.5$ when mass flux is $720 \text{ kg m}^{-2} \text{ s}^{-1}$.

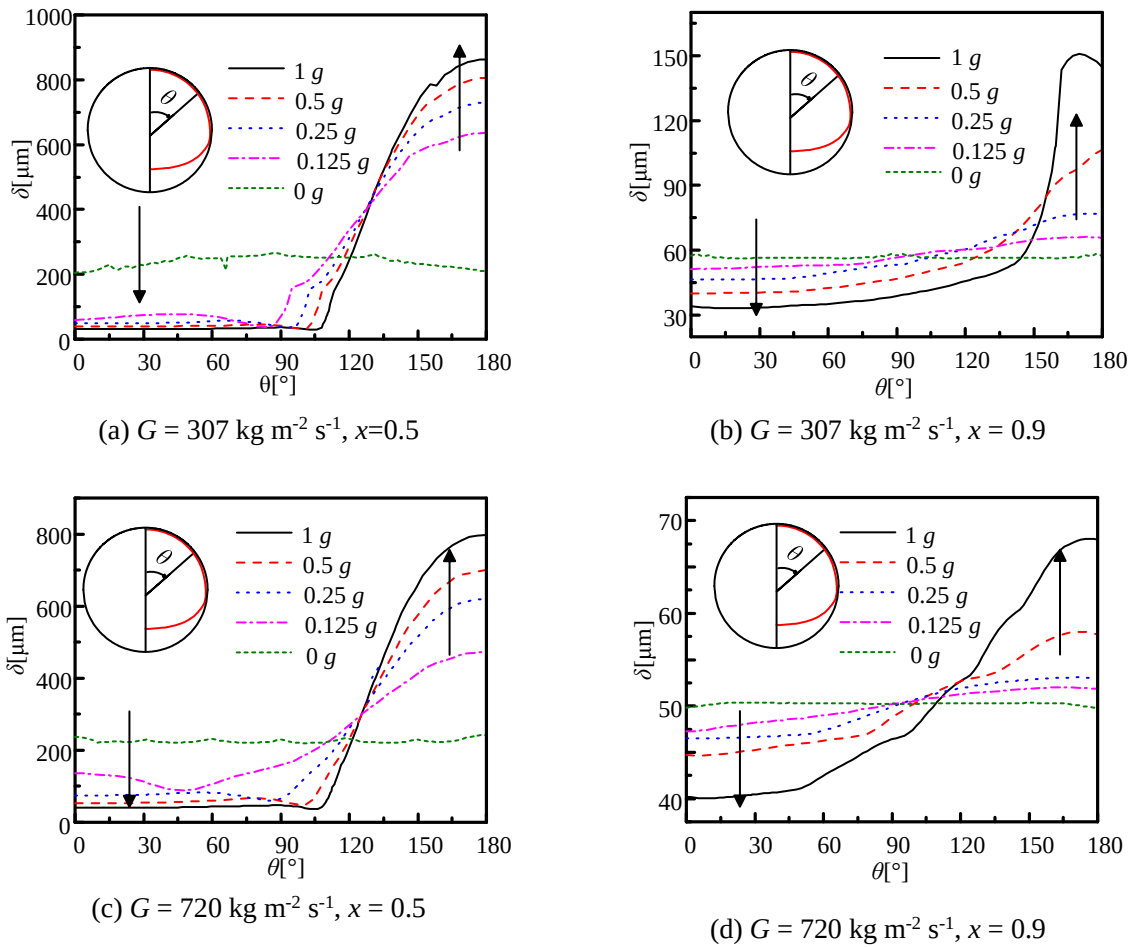
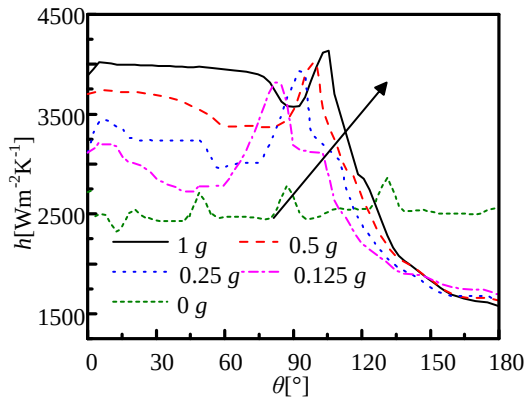


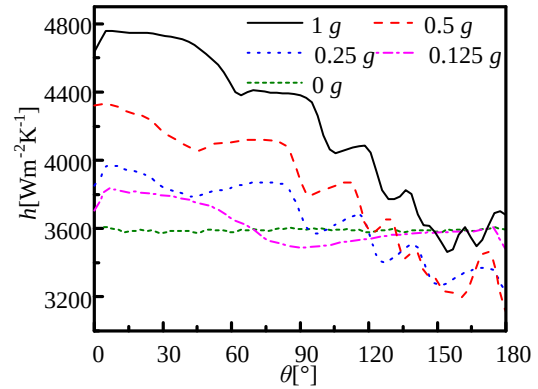
Fig. 7 The liquid film thickness inside smooth tubes under different gravity accelerations at $G = 307$ and $720 \text{ kg m}^{-2} \text{ s}^{-1}$.

A quantitative comparison of the local heat transfer coefficient versus the angular coordinate under different gravity at $x = 0.5$ and $x = 0.9$ are shown in Fig. 8. Opposite to the liquid film thickness, the local heat transfer coefficients increase with increasing gravity at the top of the tube and decrease with increases in gravity at the bottom. In the present work, a

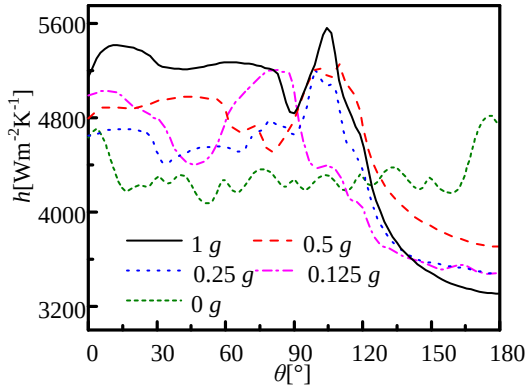
constant temperature boundary condition is applied at the tube wall, and the liquid-vapor interfacial temperature is within a small range of the saturation temperature. This means that a thin liquid film thickness will result in a larger temperature gradient, thus enhancing the heat transfer. Da Riva and Del Col [25] pointed out that the bottom part of the tube has a limited contribution to the global heat transfer coefficient for stratified flow regime. The average heat transfer coefficient increases with increasing vapor quality and mass flux at a specific gravitational acceleration, and it also increases with increasing gravity at lower vapor quality and mass flux. The average heat transfer coefficient for normal gravity is about 1.3 times of that under zero gravity at $x=0.5$ and $G=307 \text{ kg m}^{-2} \text{ s}^{-1}$. However, the difference between the average heat transfer coefficient obtained under varying gravity at $G=720 \text{ kg m}^{-2} \text{ s}^{-1}$ is unnoticeable compared with that at $G=307 \text{ kg m}^{-2} \text{ s}^{-1}$. With increasing mass flux, the shear force dominates the condensation process, while the gravity effect is unnoticeable at higher mass flux. Similar conclusions are also reported in the numerical work of Da Riva and Del Col [24] and the experimental study of Lee et al. [7]. For space missions involving multi-gravity, the influence of gravity can be negated by increasing mass flux. However, the increasing pressure drops should be considered when adopting such approaches.



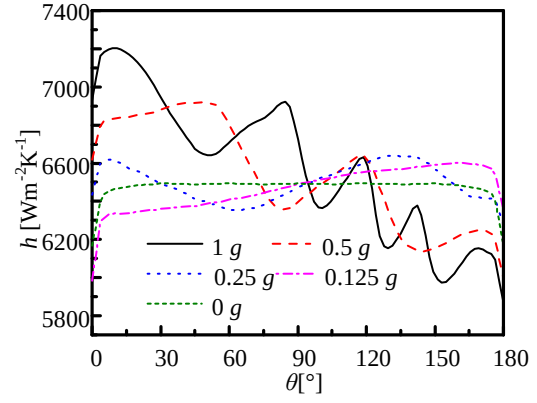
(a) $G = 307 \text{ kg m}^{-2} \text{ s}^{-1}$, $x = 0.5$



(a) $G = 307 \text{ kg m}^{-2} \text{ s}^{-1}$, $x = 0.9$



(c) $G = 720 \text{ kg m}^{-2} \text{ s}^{-1}$, $x = 0.5$



(b) $G = 720 \text{ kg m}^{-2} \text{ s}^{-1}$, $x = 0.9$

Fig. 8 The local heat transfer coefficients for R410A inside smooth tubes under different gravity accelerations.

3.3. Stream traces under normal and zero gravity

Stream traces at different vapor qualities, mass fluxes, and gravitational acceleration are shown in Figure 9. Only the X -axis and Y -axis components of velocity are considered here. The dashed red line presents $T = 319.9 \text{ K}$, and the solid red line stands for $T = 319.5 \text{ K}$. The liquid-vapor interfaces are blue solid lines. Obviously, liquid-vapor interfaces locate within 319.5 K and 319.9 K .

At $x = 0.9$ and normal gravity, a vortex with its core lying at the tube bottom is observed for both mass fluxes, while this vortex is unnoticeable at $0 g$. The stream trace starts from the Z -axis and points to the liquid-vapor interface under zero gravity. As vapor qualities decrease, the gravity effect is more pronounced. The height of vortex center decreases with increasing mass flux and decreasing vapor quality. In the liquid film region, the refrigerant flows along the tube wall from the top to the bottom like the case of falling film, while the flow pattern in this region for the case of zero gravity is very different. Some small vertexes are observed in the liquid film region at zero gravity, which will disturb the flow and thermal boundary layers and lead to some peaks of the local heat transfer coefficients as shown in Fig. 8 (a).

For the condensing flow, the vapor phase turns into the liquid phase at the liquid-vapor interface, leading to a mass sink for the vapor phase [28]. As compensation for the mass sink, the vapor phase tends to flow to the liquid-vapor interfaces as shown in Fig.9 (b) and (d). Transportation of the vapor phase from the core region to the wall region will also enhance the heat transfer. As mentioned before, the gravity effect tends to pull the vapor and liquid

phases to the bottom of the tube. The combined effect of gravity and the mass sink at the liquid-vapor interface is the reason for such stream traces for the condensation process under normal gravity.

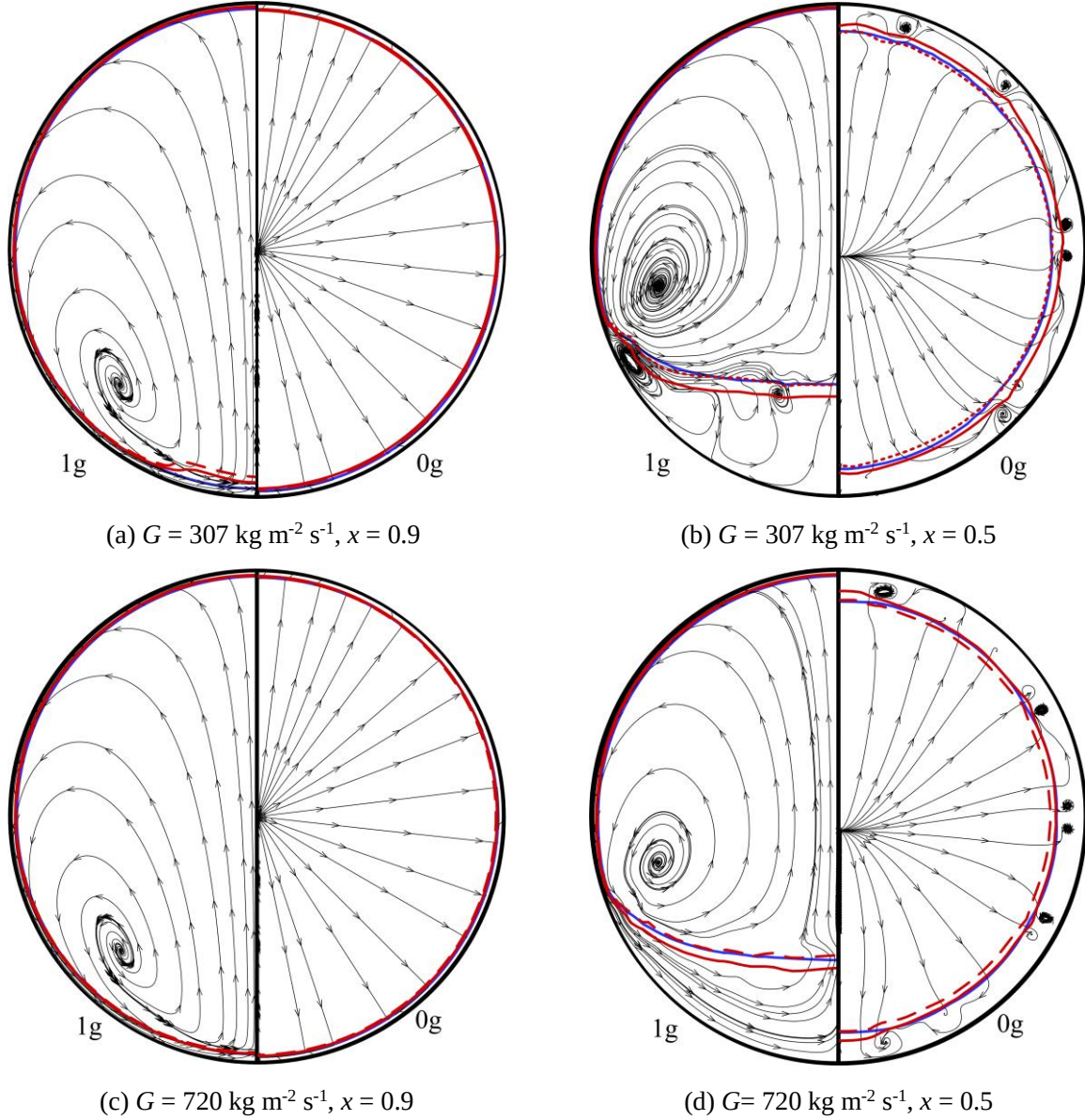


Fig. 9 The Stream traces in different cross sections under normal and zero gravity.

3.4. The mass transfer rate under normal and zero gravity

Fig. 10 shows the mass transfer rate from the vapor phase to the liquid phase at $x = 0.5$ under normal and zero gravity. The mass transfer rate at zero gravity is much lower than that of normal gravity. Under normal gravity, the mass transfer rate at the tube bottom is much smaller than that at the tube top, which means the condensation process mainly occurs at the

top of the tube. The liquid which condenses at the upper part of the tube is drained to the bottom by gravity instead of being carried in the tube axial direction by the shear stress. Under zero gravity, the mass transfer rate is nearly uniform along the perimeter direction, and the condensation process is mainly dominated by the shear force and surface tension. The mass transfer rate is directly related to the heat transfer coefficients in the condensation process under constant temperature boundary conditions through the latent heat calculated. With decreasing vapor quality, the liquid-vapor interface shrinks towards the core of the tube, resulting in a smaller heat transfer surface area, a lower mass transfer rate, and a lower heat transfer coefficient.

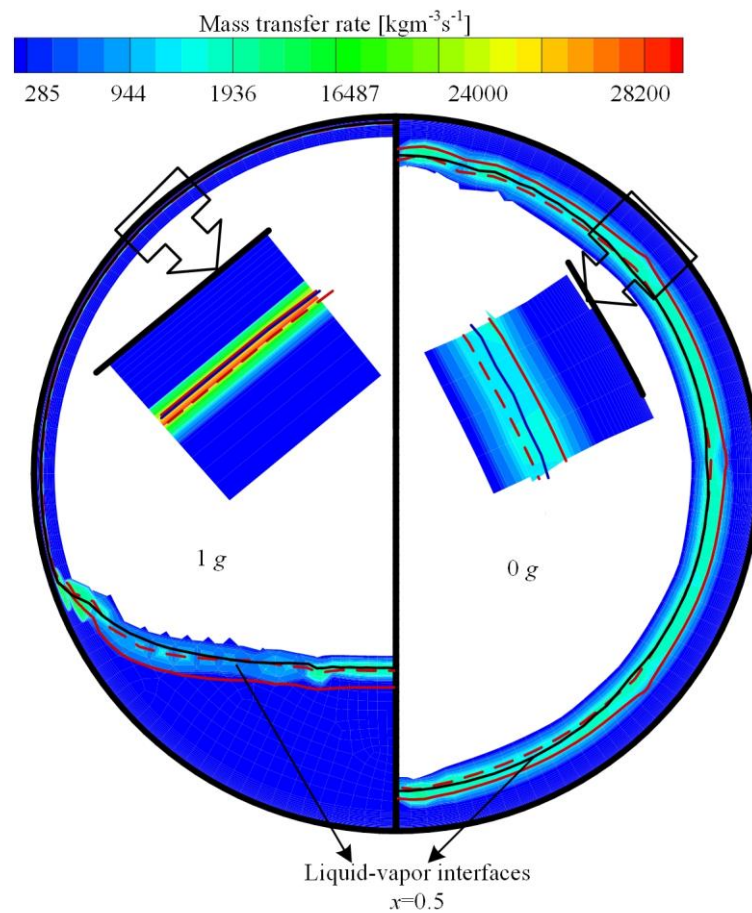


Fig. 10 The mass transfer rate from vapor to liquid for R410A inside smooth tubes under different gravity.

4. CONCLUSIONS

In this study, heat transfer characteristics of condensation inside horizontal round tubes are numerically investigated for R410A under normal and reduced gravity. The following

conclusions can be drawn from the present work.

(1) The liquid film is dragged from the tube top to the bottom by gravity, leading to a very thin liquid film at the upper part. The liquid film thickness decreases with increasing gravity at the upper part of the tube, while the reverse is true at the bottom. The average liquid film thicknesses are about 56 μm at $x = 0.9$ and 230 μm at $x = 0.5$ at $G = 307 \text{ kg m}^{-2} \text{ s}^{-1}$ for various levels of gravity, while these value decrease to 50 μm at $x = 0.9$ and 215 μm at $x = 0.5$ when the mass flux is $720 \text{ kg m}^{-2} \text{ s}^{-1}$.

(2) The local heat transfer coefficients increase with increasing gravity at the top of the tube and decrease with increases in gravity at the bottom. The bottom part of the tube has a limited contribution to the overall heat transfer coefficient when the liquid film accumulates at the bottom. The average heat transfer coefficients increase with increasing gravity accelerations at $G = 307 \text{ kg m}^{-2} \text{ s}^{-1}$, while the difference of these under various gravity levels is unnoticeable at a higher mass flux. The proportion of the thin liquid film region is important for the overall heat transfer coefficients in the condensing flow.

(3) A vortex with its core lying at the bottom of the tube is observed under normal gravity, while this vortex is unnoticeable at 0 g . In the liquid film region, the refrigerant flows along the tube wall from the top to the bottom like the case of falling film. The combined effect of gravity and the mass sink at the liquid-vapor interface is the reason for the stream traces found under normal gravity for the condensing flow. The mass transfer rate under zero gravity is much lower than that of normal gravity.

ACKNOWLEDGMENTS

This work is supported by the National Science Foundation of China (51210011) and National Science Foundation of Zhejiang Province, China (LZ13E060001).

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