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Numerical Study on Condensation Flow and Heat Transfer of Hydrocarbon Mixtures in Inclined Tubes under Static and Swaying Conditions

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ABSTRACT: To investigate the complex phase change behavior in two-phase condensation flow of hydrocarbon mixtures in inclined tubes, a numerical model was developed in Fluent using the Volume of Fluid method combined with the Lee phase change model. A mixing effect correction was incorporated to enhance the simulation accuracy, and its impact on the flow and heat transfer characteristics was systematically evaluated. Numerical simulations were performed and subsequently corrected for mixing effects; the final results show good agreement with classical experimental data. The average deviation of the heat transfer coefficient is -0.76% , while that of the frictional pressure drop is 4.5% . Furthermore, by introducing the swaying-motion equation into the model, the effects of different swaying periods and amplitudes on the heat transfer under swaying conditions are investigated. The results show that shorter swaying periods and larger swaying amplitudes lead to more pronounced fluctuations in the heat transfer coefficient. Within the parameter range considered in this study, the influence on the time averaged heat transfer coefficient does not exceed 25% . This study provides a solid theoretical and data foundation for the design of relevant condensation heat exchangers, thereby supporting the development of the offshore LNG industry and enhancing energy security.

KEYWORDS: Hydrocarbon refrigerants; condensation; heat transfer coefficient; frictional pressure drop; numerical simulation

1 Introduction

With the global energy mix shifting toward low-carbon development, natural gas has become a widely used cleaner energy carrier because it combines high combustion efficiency with relatively low pollutant emissions. It is applied in multiple sectors, including industrial production, urban heating, and transportation [1]. However, at ambient conditions, natural gas is gaseous, and its large specific volume makes storage and long-distance transport difficult. To overcome this limitation, the gas is commonly liquefied after extraction and processing. Cooling it to approximately -162°C condenses it to a liquid and reduces the volume to about $1/600$ of the original, which markedly relaxes the requirements for storage capacity and transport infrastructure and enables large-scale interregional distribution of natural gas [2–4].

During natural gas liquefaction, the condensation of non-azeotropic mixtures is particularly important and can restrict the overall liquefaction rate. Natural gas is not a single component fluid; it is a non-azeotropic mixture mainly composed of hydrocarbons such as methane, ethane, and propane, with small amounts of other gases. Owing to differences in their physical properties, particularly their boiling points, these

components do not undergo condensation at the same temperature or time. Instead, condensation proceeds in stages: components with higher boiling points first reach local saturation and begin to liquefy, whereas components with lower boiling points condense only at lower temperatures [5,6]. This staged condensation tightly couples heat transfer and mass transfer. The evolving flow regime, the axial and radial temperature fields, and the component-wise phase change behavior directly influence liquefaction efficiency and overall energy consumption. A clear description of the condensation flow characteristics of non-azeotropic mixtures is therefore essential for designing and optimizing efficient heat exchangers.

As key equipment in liquefaction systems, heat exchangers must be designed with structural layouts, heat transfer area distributions, and flow channel configurations that reflect an accurate understanding of condensation phenomena. A reliable description of the component behavior during phase change allows engineers to select appropriate design parameters, enhance heat transfer performance, reduce energy losses, and ensure stable and efficient operation of the liquefaction train. These improvements, in turn, support the wider deployment of natural gas as a cleaner energy source. Numerous studies have reported experimental and numerical investigations on the condensation of pure and mixed working fluids. For pure fluids, Morrow and Derby [7] measured the heat transfer coefficients of R134a, R513A, and R450A during condensation in a 0.95 mm diameter channel. They reported that at low mass fluxes, the coefficients for R513A and R134a differ significantly; when the mass flux approaches 500 kg/(m²·s), this difference becomes small, and the average heat transfer performance of R450A is about 5.5% lower than that of R134a. Dai et al. [8] analyzed condensation of R1234ze(E) and R152a in a 4 mm horizontal circular tube and proposed a correlation for the heat transfer coefficient with an absolute deviation of 13.08%. Li et al. [9] examined flow and heat transfer for R152a and R1234ze in a 4 mm horizontal circular tube and showed that heat flux strongly affects heat transfer, but has only a minor influence on frictional pressure drop. Fazelnia et al. [10] reviewed condensation of R1234yf in circular tubes and identified four main flow patterns: slug, stratified, intermittent, and annular. Moghadam et al. [11] investigated R1234yf condensation in an inclined tube with an inner diameter of 8.3 mm and observed that, at high mass flux or high vapor quality, the inclination angle has little effect on heat transfer performance.

For mixed working fluids, Mazumder et al. [12] studied condensation of the mixed refrigerant R32/R1234ze in a vertical circular tube and found that when the vapor quality is below 0.5, the overall thermal resistance is large and becomes the dominant factor limiting heat transfer. Yu et al. [13] conducted experiments on a methane–propane mixture in a spiral tube and analyzed how operating parameters, such as inlet mass flow rate, affect flow behavior and heat transfer performance. Cai et al. [14,15] carried out numerical simulations of propane and ethane–propane mixtures condensing in spiral tubes using a two-fluid formulation with a k – ϵ turbulence model; they reported that the applied heat flux has only a limited influence on the overall flow and heat transfer response. Neeraas [16] performed experiments on condensation of propane, R22, an ethane–propane mixture and a methane–propane mixture in spiral tubes, providing heat transfer coefficients and frictional pressure drop data that are useful for model validation and engineering design.

Large reserves of recoverable natural gas also occur in deep-sea regions, where floating liquefied natural gas (FLNG) vessels enable offshore extraction, processing and storage. In such marine environments, platform motion becomes a major factor. Under rolling conditions, the flow and heat transfer behavior of mixed hydrocarbon working fluids inside heat exchangers is complex and time dependent. Most existing studies focus on single phase flow under rolling motion. For example, Yu et al. [17] showed that for single phase flow under rolling conditions, the dominant parameter is the rolling acceleration; when this acceleration is held constant, the time-averaged flow rate remains nearly unchanged even as rolling frequency and amplitude vary. Wang et al. [18] examined single phase flow and heat transfer in tubes under rolling

and demonstrated that both the heat transfer coefficient and frictional pressure drop vary periodically, whereas their time-averaged values differ little from those under stationary conditions. For two phase flow under rolling motion, Chen et al. [19] investigated boiling of water and found that, as mass flux increases, the instantaneous heat transfer coefficient exhibits stronger fluctuations, while the instantaneous pressure drop shows smaller fluctuations. Tian et al. [20] experimentally investigated the condensation behavior of a methane/ethane/propane/isobutane mixture under oscillatory (swaying) motion, with a maximum displacement of 1500 mm. The flow was predominantly annular, and the liquid phase oscillated laterally under the imposed motion. Their results indicated that the oscillation can enhance heat transfer. Han et al. [21] experimentally investigated the condensation of n-pentane under oscillatory motion. Their results showed that, at low Reynolds numbers, increasing the oscillation displacement can reduce heat transfer.

Although many working fluids have been examined, differences remain between the fluids reported in the literature and the multicomponent mixtures used in industrial natural gas liquefaction. In particular, numerical studies on non-azeotropic hydrocarbon mixtures with three or more components are still limited. Understanding of how sloshing-induced motion alters condensation flow and heat transfer behavior is also incomplete. To address these gaps, the present work combines numerical simulation with published experimental data to develop a detailed model for two-phase condensation flow of mixed hydrocarbon working fluids. On this basis, we simulate condensation of a representative gas-field composition in an inclined tube under both stationary and swaying conditions. We then systematically analyze the effects of sloshing-related parameters on the mixture, with the aim of providing practical design references and engineering guidance for efficient heat exchangers used in natural gas liquefaction.

2 Numerical Model

2.1 Physical Model

The structural configuration of the inclined tube physical model used in this study is shown in Fig. 1. The model consists of three consecutive sections: a fully developed section with a length of 0.8 m, an experimental measurement and reference section with a length of 0.4 m, and a pressure stabilizing section with a length of 0.2 m. The diameter of the pipe is 10 mm. The gravity direction is along the Z-axis, vertically downward. The assembly is installed at an inclination angle of 10° . Although secondary-flow characteristics differ between an inclined straight tube and a helical coil, for practical heat exchangers the discrepancy is expected to be limited. Previous studies indicate that [22] the exchanger height is typically 10–50 m, the coil (winding) diameter is generally 0.8–1.0 m, the helix angle is about 4° – 10° , and the tube diameter is usually 8–14 mm. When the tube diameter is much smaller than the winding diameter, the curvature effect exists but is relatively weak. Therefore, the swaying-induced influence can be theoretically captured using a simplified straight-tube model as a first-order approximation.

This arrangement is consistent with that reported by Yu et al. [23], which enables direct comparison with the literature and represents a moderate tube inclination relevant to offshore applications. The fully developed section conditions the flow before it enters the measurement region. As the fluid passes through this upstream length, inlet effects such as non-uniform velocity profiles and small perturbations in phase distribution are attenuated, and the flow evolves toward a stable pattern suitable for subsequent evaluation. In the simulations, this section also acts as a numerical buffer, allowing the imposed inlet boundary condition to relax to an internally consistent profile and thereby reducing the sensitivity of the results to the specific inlet formulation. The experimental measurement and reference section serves as the primary zone for data extraction. Monitoring planes located within this 0.4 m segment are used to sample area averaged quantities, including pressure, wall shear stress, vapor quality or void fraction, and the local heat transfer coefficient at fixed axial positions. The use of a dedicated measurement section, distinct from the inlet and outlet buffer

regions, helps to isolate entrance and exit effects, enhances repeatability, and facilitates direct comparison with the reference data in Yu et al. [23]. The pressure stabilizing section, located at the outlet end of the model, is designed to prevent back flow and to maintain a well posed outlet boundary condition. Back flow would disturb the stable flow field in the measurement region, introduce deviations in the computed fields, such as fluctuations in velocity, temperature, and phase distribution, and could invalidate the extracted metrics. Due to its layout and function, the pressure stabilizing section redistributes the internal pressure field, suppresses reverse flow, and provides a numerical buffer when a pressure outlet condition is applied. Consequently, it helps ensure the accuracy and reliability of the results obtained in the measurement section and reduces the influence of outlet boundary choices on the reported quantities.

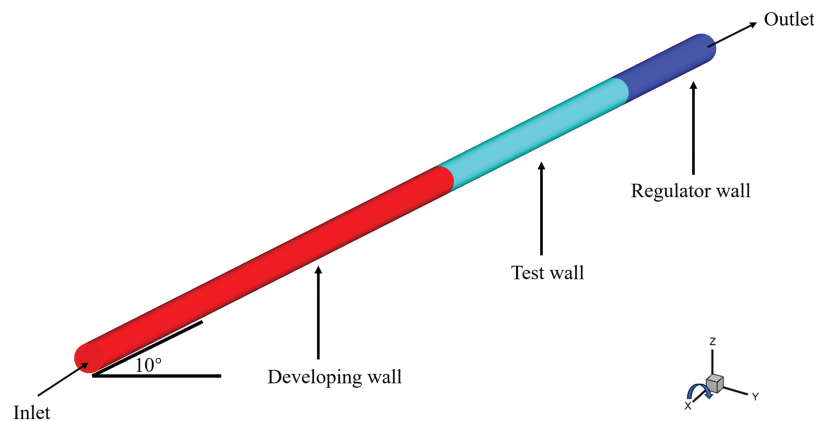


Figure 1: Physical model.

2.2 Governing Equations

The Volume of Fluid (VOF) model was employed to capture the gas liquid interface during condensation and to resolve the evolution of flow patterns and wall bounded liquid films. This model was selected because it conserves phase volume, accurately represents large interfacial deformations, and is suitable for thin film formation along inclined walls. Turbulence was described using the Reynolds Stress Model (RSM) in order to account for anisotropy and streamline curvature under inclination and possible swaying, conditions in which eddy viscosity models such as the k epsilon model may underpredict normal stress effects and interfacial shear. At high vapor velocities, turbulent inertial forces tend to equalize the liquid film thickness on the upper and lower walls. As the vapor velocity increases, the internal tube flow becomes more symmetric, which improves prediction fidelity [24]. Interfacial mass transfer was modeled using the Lee phase change model, with the mass transfer coefficient set to 10^{-4} in accordance with common practice for condensable mixtures [25]. The latent heat source term and the vapor liquid mass source term were coupled to the energy and volume fraction equations to ensure conservation of energy during phase change. Interface tracking employed the Modified High Resolution Interface Capturing (Modified HRIC) scheme to reduce numerical diffusion at the interface front while maintaining boundedness of the volume fraction. Consistent with the characteristics of the Modified HRIC method, entrainment effects were represented through user defined functions that introduced an additional mass exchange term in the near interface control volumes [26]. Spatial discretization of momentum and energy used second order schemes. Near wall treatment adopted enhanced wall functions, and grid spacing was selected to keep the near wall y^+ within the logarithmic layer range in the vapor core and within the viscous affected range in the liquid film. The Courant number in interfacial cells was maintained below unity, and relaxation factors were adjusted to avoid overshoot of the volume fraction. Mesh and time step sensitivity analyses were carried out to verify that the key outputs,

namely the heat transfer coefficient and the frictional pressure drop, are insensitive to further refinement within the reported ranges.

Volume Fraction Equation:

$$\frac{\partial \alpha_l}{\partial t} + \nabla \cdot (\vec{u} \alpha_l) = \frac{S_{\alpha_l}}{\rho_l} \quad (1)$$

$$\frac{\partial \alpha_g}{\partial t} + \nabla \cdot (\vec{u} \alpha_g) = -\frac{S_{\alpha_l}}{\rho_g} \quad (2)$$

The gas and liquid phase volume fractions satisfy the following condition:

$$\alpha_l + \alpha_g = 1 \quad (3)$$

In the VOF model, the gas and liquid phases within each computational cell are treated as a single phase. The flow is considered to share a common velocity field for both phases, and the momentum equation takes the same form as that for single-phase flow. The momentum conservation equation is expressed as follows:

$$\frac{\partial (\rho \vec{u})}{\partial t} + \nabla (\rho \vec{u} \vec{u}) = -\nabla p + \nabla \left[\mu \left(\nabla \vec{u} + \nabla \vec{u}^T \right) - \frac{2}{3} \mu \cdot \nabla \vec{u} \right] + \rho g + F_\sigma \quad (4)$$

Energy equation:

$$\frac{\partial}{\partial t} (\rho h) + \nabla (\vec{u} \rho h) = \nabla \cdot (\lambda_{eff} \nabla T) + S_h \quad (5)$$

Phase transition Lee model:

$$S_{\alpha_l} = -r \cdot \alpha_l \rho_l \frac{T - T_s}{T_s} \quad T \geq T_s \quad (6)$$

$$S_{\alpha_l} = r \cdot \alpha_g \rho_g \frac{T_s - T}{T_s} \quad T < T_s \quad (7)$$

where S_{α_l} represents the mass transfer rate associated with phase change per unit volume and per unit time; α_l represent liquid phase volume fraction; α_g represent gas phase volume fraction; $\vec{u}$ represents the shared velocity of the two-phases m/s; ρ is the mixture density obtained by volume-fraction-weighted averaging kg/m³; μ denotes the dynamic viscosity of the mixture Pa·s; F_σ represents the surface tension; h is the average enthalpy of the gas and liquid phases J/kg; λ_{eff} is the effective thermal conductivity between the gas and liquid phases W/(m·K); r is the time relaxation factor 1/s; T_s is saturation temperature. The behavior of a mixture of working fluids during condensation differs from that of pure working fluids, primarily due to the volatility of the components. Components with higher volatility tend to accumulate at the interface, resulting in a poorer condensation effect for components with lower volatility, thereby increasing the thermal resistance to heat transfer. In existing methods, the Silver method is often used to correct for the effects of mixture composition, thereby improving the accuracy of the simulation. When a mixed refrigerant undergoes condensation, the component with higher volatility tends to accumulate at the interface, thereby hindering the condensation of the less volatile component and increasing the thermal resistance to heat transfer. The widely used calculation method, originally proposed by Silver [27] and later modified, has shown high accuracy and broad applicability. It is important to note that the mixing effect correction factor defined in Eq. (8) is applied as an offline post-processing method. In this study, this method is employed to

calculate the heat transfer coefficient of the mixed refrigerant. Following Silver's assumptions, this method neglects the influence of mass transfer on sensible heat transfer in the gas phase as well as the enhancement due to interfacial roughness, while assuming thermodynamic equilibrium between the phases. Despite these simplifications, the method provides reliable predictions. It is widely adopted due to its simplicity, efficiency, and the fact that it eliminates the need for diffusion coefficients. As expressed by the following equation:

$$h_{mix} = \frac{h_{film}}{1 + h_{film} \cdot \Phi} \quad (8)$$

$$\Phi = \frac{Z}{h_g} \quad (9)$$

where h_{film} is the condensate film heat transfer coefficient $W/(m^2 \cdot K)$; Φ is the heat and mass transfer resistance in vapor core for mixture $(m^2 \cdot K)/W$; h_g is the heat transfer coefficient of gas-core $W/(m^2 \cdot K)$; Z is the sensible heat ratio, derived from the temperature-enthalpy diagram of the fluid mixture.

2.3 Validation of Boundary Conditions and Grid Independence

The inlet was prescribed as a mass flow boundary condition, the outlet as a pressure outlet, and all walls as constant heat flux boundaries. The inlet is set as a mass flow rate inlet, and the impact of entrainment on the simulation results has been considered. The method proposed by Ishii et al. is used to calculate the inlet liquid phase mass flow rate. The inlet is specified as a mass flow inlet. The influence of entrainment on the simulation is taken into account by determining the inlet liquid-phase mass flow rate using the method proposed by Ishii and Mishima [26]. The PRESTO! scheme was adopted for pressure discretization, and the Least Squares Cell Based method was employed for gradient evaluation. Thermophysical properties were obtained from REFPROP [28].

In practical operation, the oscillatory motion of the system is relatively complex and may involve several degrees of freedom. In the present study, attention is focused on lateral oscillation, which has been identified as the dominant mode influencing the flow and heat transfer characteristics [29]. In existing investigations, lateral oscillation is commonly approximated as a regular sinusoidal motion [30]. By superimposing the oscillation equation on a stationary reference case and implementing it through a user defined function, the moving coordinate framework can be used to represent the oscillating condition. The resulting motion is expressed as shown in Eq. (10).

$$X = X_{max} \sin\left(\frac{2\pi t}{T_c}\right) \quad (10)$$

In the equation, T_c represents the swaying period, and X denotes displacement generated by swaying.

To better resolve the condensate film, six mesh sets with different cell counts were examined to verify grid independence. The verification operating condition is a mixture of ethane/propane as the working fluid, with a molar composition ratio of 1:1, a pressure of 3.2 MPa, and a mass flow rate of 300 $kg/(m^2 \cdot s)$. As shown in Fig. 2, once the total number of cells exceeds approximately 1.4 million, further mesh refinement results in negligible variations in the monitored quantities. This indicates that the mesh resolution is independent of the simulation results beyond this threshold. Given the balance between computational cost and accuracy, this mesh resolution is considered adequate for the current simulations and has been adopted for all subsequent calculations. This grid resolution has been verified to be suitable for both stationary and swaying conditions.

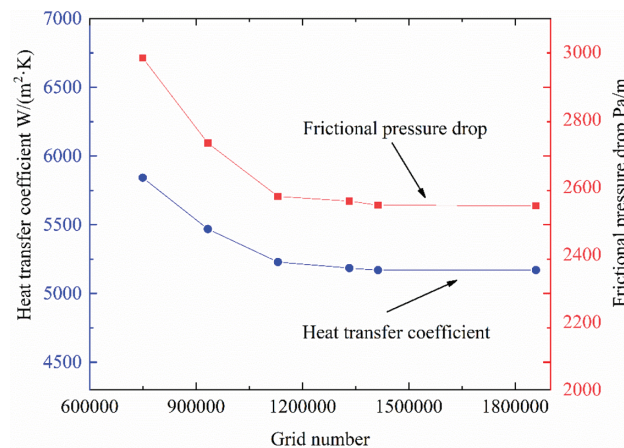


Figure 2: Results of grid independence.

In terms of time step independence, four time steps of 0.01, 0.005, 0.001, and 0.0005 s were selected for repeated calculations under the same conditions to compare the heat transfer coefficient and frictional pressure drop of the measurement section. When the time step was reduced from 0.005 to 0.001 s, the differences in both the heat transfer coefficient and frictional pressure drop were less than 1%. Similarly, when the time step was further reduced from 0.001 to 0.0005 s, the differences remained under 1%. Therefore, the time step of 0.001 s chosen in this study meets the required criteria.

2.4 Numerical Model Validation

The numerical model was validated against experimental data for a binary mixture of ethane and propane [31,32]. The experimental conditions corresponded to a pressure of 3.2 MPa, a mass flux of $300 \text{ kg}/(\text{m}^2 \cdot \text{s})$, a vapor quality in the range from 0.1 to 0.9, and a mass ratio of ethane to propane of 1:1. Fig. 3a compares the simulated heat transfer coefficients with the experimental data reported by Neerass et al. [31,32], while Fig. 3b presents the corresponding comparison for the frictional pressure drop. As shown in Fig. 3, both the heat transfer coefficient and the frictional pressure drop increase monotonically with vapor quality from 0.1 to 0.9. Physically, an increase in vapor fraction leads to thinning of the condensate film and an enhancement of the interfacial area and interfacial shear. At the same time, the mixture density decreases and the mean flow velocity increases, which jointly promote higher heat transfer rates and larger pressure losses. The maximum deviation between simulation and experiment occurs at a vapor quality close to 0.9. Within the simulated vapor quality range, the relative deviation of the heat transfer coefficient remains below 20%, with an average deviation of -0.76% . For the frictional pressure drop, the relative deviation is below 15%, with an average deviation of 4.5% .

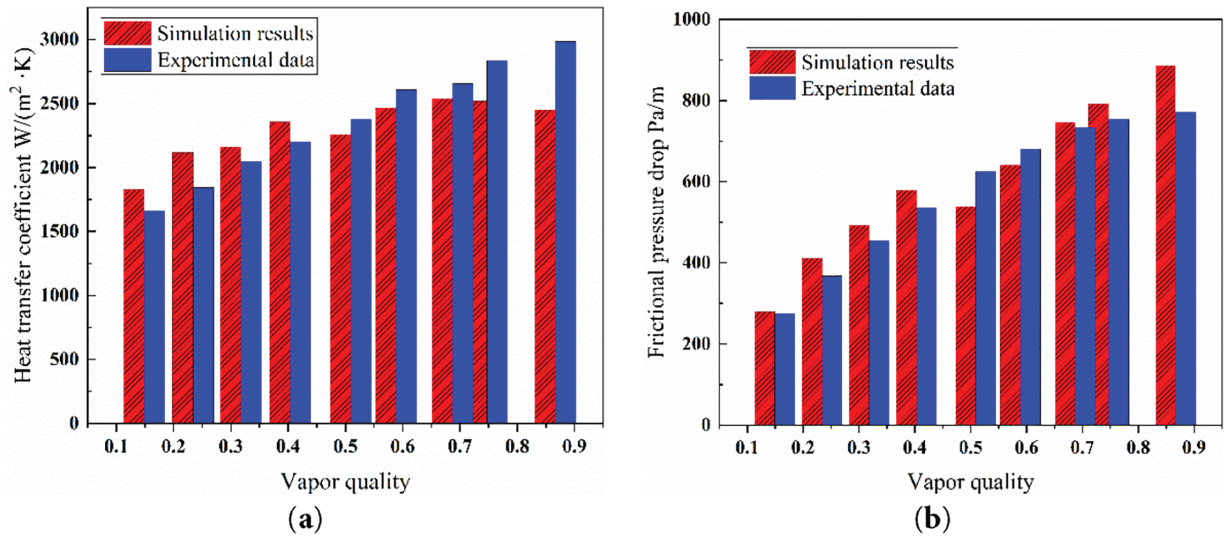


Figure 3: (a) Comparison of heat transfer coefficients obtained from Neerass experimental data and numerical simulations; (b) Comparison of frictional pressure drop obtained from Neerass experimental data and numerical simulations.

3 Analysis and Discussion of Numerical Simulation Results

3.1 Simulated Heat Transfer Coefficients under Stationary Conditions

We followed the same numerical simulation approach for multi-component mixtures as proposed by Li et al. [15]. The data used herein were obtained from our preliminary research and surveys [33]. The working fluid is a nitrogen, methane, ethylene, propane, and isopentane mixture with mole fractions of 20, 54, 24, 1.5, and 0.5 percent, respectively. The operating pressure is 3 MPa, the mass flux ranges from 350 to 500 $kg/(m^2 \cdot s)$, and the vapor quality varies from 0.1 to 0.9. The time interval for the heat transfer coefficient under stationary conditions in this study is 5–10 s.

The simulated heat transfer coefficients obtained without considering the mixing effect are presented in Fig. 4a, whereas the corresponding results including the mixing effect are shown in Fig. 4b. The absence of the mixing effect causes a significant overprediction of the heat transfer coefficient. In general, the heat transfer coefficient increases with vapor quality. When the mixing effect is taken into account, the heat transfer coefficient rises rapidly at vapor qualities below 0.5 and then increases more gradually at vapor qualities above 0.5. This behavior is primarily attributed to the large differences in boiling point among the components of the mixed refrigerant, which give rise to a pronounced interfacial thermal resistance. Consequently, the mixing effect should be included when simulating the heat transfer coefficient. At a fixed vapor quality, an increase in mass flux increases the film Reynolds number and thereby enhances the heat transfer coefficient. At a fixed mass flux, an increase in vapor quality reduces the mixture density, which strengthens convective heat transfer within the liquid film and leads to a higher heat transfer coefficient. The Boyko correlation [34] provides good predictive performance for the heat transfer coefficient. The Boyko correlation is given by Eq. (11). In the section Re_l represents the Reynolds number of liquid and Pr_l represents the Prandtl number of liquid.

$$h = 0.021 Re_l^{0.8} Pr_l^{0.43} \sqrt{1 + \frac{\rho_l - \rho_g}{\rho_l} x \frac{\lambda_l}{d}} \quad (11)$$

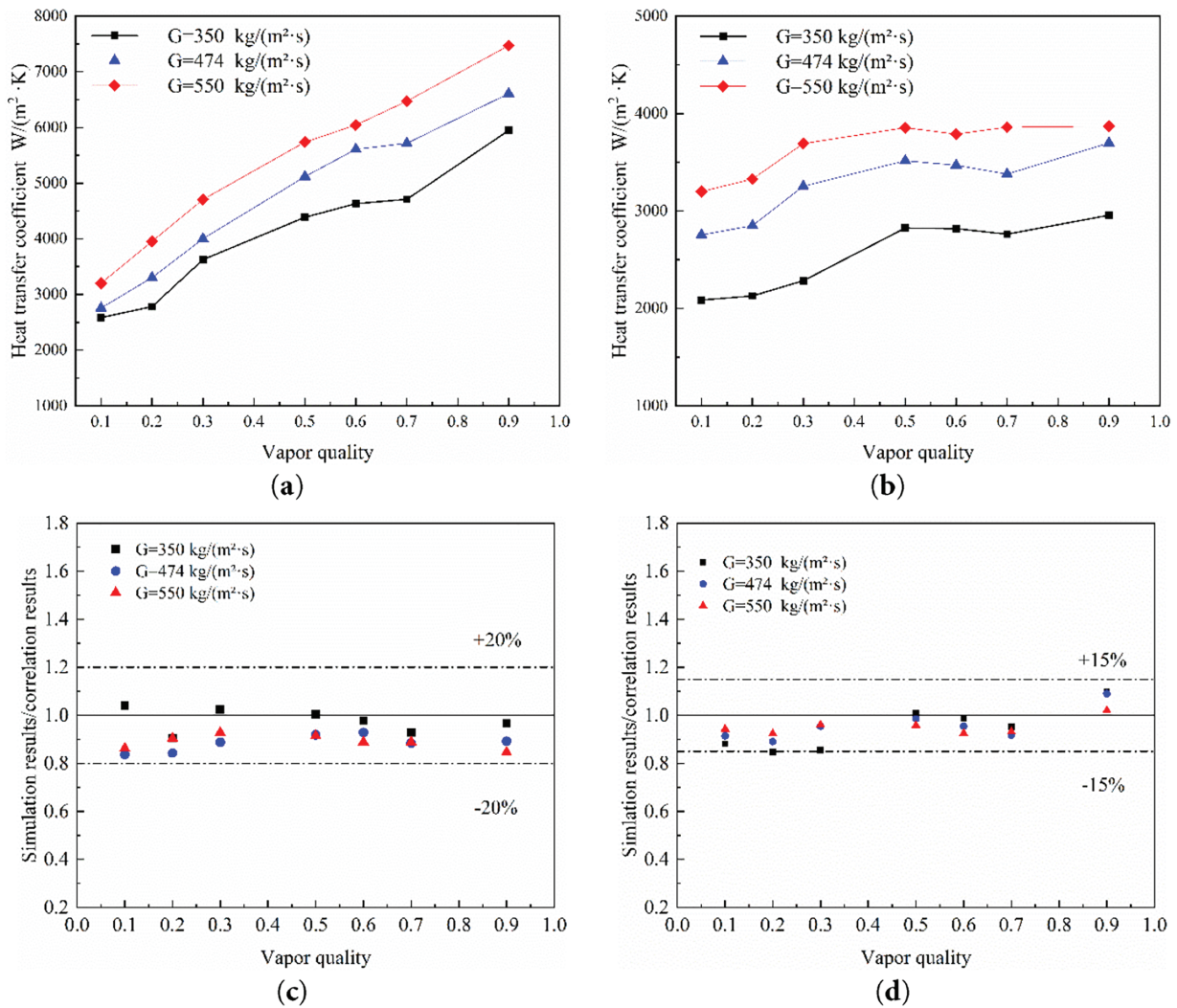


Figure 4: (a) Heat transfer coefficient without considering correction; (b) Heat transfer coefficient considering correction; (c) Comparison of heat transfer coefficient obtained from Boyko Correlation and numerical simulations; (d) Comparison of correction obtained from Boyko Correlation and numerical simulations.

Fig. 4c compares the simulation results obtained without accounting for the mixing effect with the predictions of the Boyko correlation [29], while Fig. 4d presents the corresponding comparison when the mixing effect is incorporated. Overall, the simulated heat transfer coefficients are slightly lower than the predicted values. Without considering the mixing effect, the deviation relative to the Boyko correlation [29] is within $\pm 20\%$, with smaller discrepancies observed at lower mass fluxes. After incorporating the mixing effect, the overall deviation decreases from $\pm 20\%$ to $\pm 15\%$, and improved agreement is achieved at higher mass fluxes and at vapor qualities close to 0.5. Accounting for the mixing-effect correction reduces the deviation because it introduces the additional heat/mass-transfer resistance that is inherently present in zeotropic multicomponent condensation, where volatile species accumulate near the liquid–vapor interface and form a diffusion boundary layer. Without this correction, the phase-change driving force is effectively overestimated, leading to a systematic overprediction of the heat transfer coefficient. Applying the correction factor lowers the effective heat transfer coefficient to reflect this degradation mechanism, thereby improving agreement and reducing the deviation to about 15%.

3.2 Simulated Frictional Pressure Drop under Stationary Conditions

The frictional pressure drop of hydrocarbon mixture refrigerants increases with both vapor quality and mass flow rate. As shown in Fig. 5a, at a fixed vapor quality, a higher mass flow rate yields a larger Reynolds number in the liquid film and consequently a higher frictional pressure drop. At a fixed mass flow rate, an increase in vapor quality reduces the mixture density, which enhances the shear exerted by the gas phase on the liquid film and leads to a further increase in pressure drop. According to existing research [35], the Fuchs correlation [36] provides a satisfactory prediction of the frictional pressure drop. The Fuchs correlation [36] is given by Eq. (12). Fig. 5b compares the simulation results with the values predicted by the Fuchs correlation and shows that most deviations are within $\pm 15\%$. The agreement is particularly good at a vapor quality close to 0.5, whereas larger deviations are observed at very low and very high vapor qualities.

$$\left(\frac{dp}{dl}\right)_{tp} = \left(\frac{dp}{dl}\right)_l + \varphi_{lv} \left[\left(\frac{dp}{dl}\right)_v - \left(\frac{dp}{dl}\right)_l \right]$$

$$\left(\frac{dp}{dl}\right)_v = \left[0.3164 \text{Re}_v^{-0.25} + 0.03 \left(\frac{d}{D}\right)^{0.5} \right] \frac{m^2}{2\rho_v d} \quad (12)$$

$$\varphi_{lv} = \varphi_{lv}(x, Fr_l, \rho_l \rho_v^{-1}) = \sum_{i=1}^n C_i (Fr_l, \rho_l \rho_v^{-1}) \cdot x^{i-1}$$

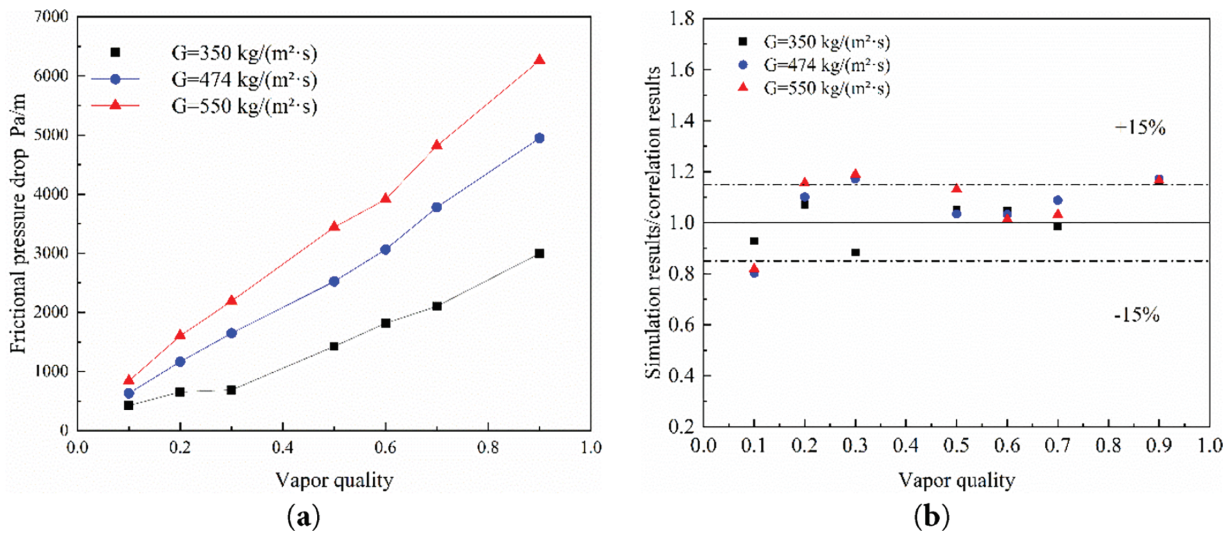


Figure 5: (a) Simulation results of frictional pressure drop; (b) Comparison of correction obtained from Fuchs Correlation and numerical simulations.

3.3 Simulation Results under Different Swaying Periods

Fig. 6 presents the velocity field distribution at the outlet of the measuring section for different rolling periods at the same swaying amplitude. The swaying amplitude is 3 m, and the swaying periods are 2 and 5 s. For a fixed amplitude, a shorter swaying period yields a higher characteristic velocity, and the overall flow field exhibits alternating acceleration and deceleration with a clear periodic character. The maximum velocity occurs near the full and half characteristic periods, for example at $t/T_c = 1$. At this instant, the velocity field for the case with a swaying period of 2 s is markedly smaller than that for the case with a swaying period of 5 s. At other times, such as $t/T_c = 1.25$, the difference between the two velocity field distributions becomes

less pronounced. The flow field distributions presented in the results section reflect the numerical solution prior to this empirical correction.

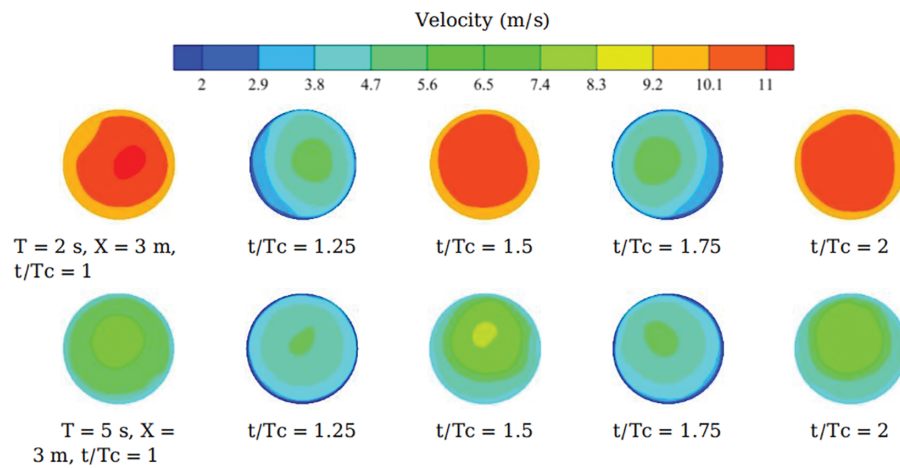


Figure 6: Velocity distribution under different swaying periods ($x = 0.5$, $G = 550 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $d = 10 \text{ mm}$).

Fig. 7 presents the cross sectional distribution of gas volume fraction at the outlet of the measuring section for two swaying periods at a constant amplitude $X = 3 \text{ m}$. The colour scale ranges from 0 (blue, liquid rich) to 1 (red, gas rich). For the shorter period $T = 2 \text{ s}$ (top row), the gas core in the central region (red and orange) occupies most of the cross section and is surrounded by a thin condensate film (blue and cyan). As the normalized time increases from $t/T_c = 1$ to 2, the sector with high void fraction clearly processes around the circumference, and the liquid film becomes strongly non uniform in the azimuthal direction. The interfacial ridge (green and yellow band) migrates rapidly, indicating a pronounced periodic deformation of the film. The snapshots exhibit a larger change in angular position between consecutive frames, which reflects stronger fluctuations at the shorter swaying period.

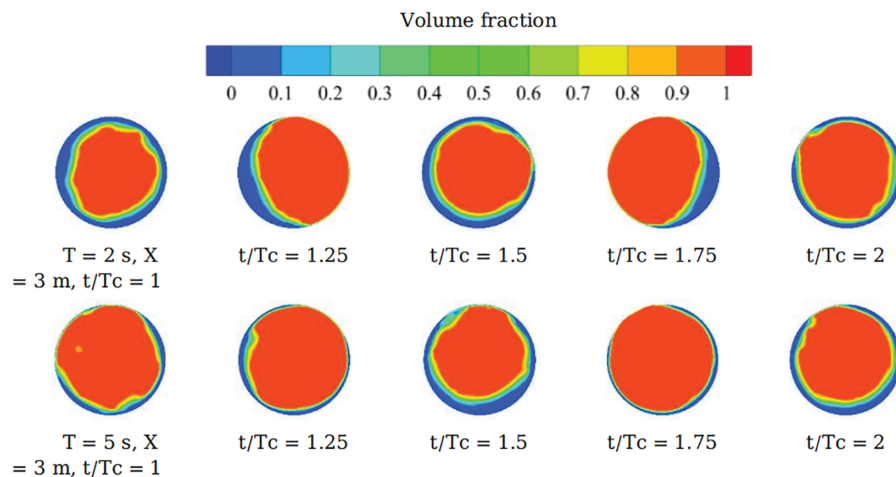


Figure 7: Volume fraction under different swaying periods ($x = 0.5$, $G = 550 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $d = 10 \text{ mm}$).

For the longer period ($T = 5$) s (bottom row), the interface evolves more slowly and smoothly. The condensate film remains relatively uniform around the circumference, and the high-void-fraction sector exhibits only a small angular shift between successive frames. A localized yellow region at ($t/T_c = 1$) suggests a transient disturbance or entrainment event, but it dissipates rapidly in the subsequent frames. At a fixed amplitude, the shorter period ($T = 2$) s induces stronger interfacial motion, a broader distribution of void fraction near the wall (i.e., a wider green-yellow band), and a more distinct cycle-to-cycle periodicity. By contrast, the longer period ($T = 5$) s provides more time for the film to relax, leading to a narrower interfacial band and reduced phase migration across the outlet plane. Taking the contour at ($t/T_c = 1.25$) as an example, the wall is fully wetted at this instant, and the liquid-film thickness on the right side is nearly zero. The imposed motion increases the film thickness on the left side, enlarging the overall liquid-film coverage area and thereby weakening heat transfer.

Fig. 8 compares the heat transfer coefficients under stationary and swaying conditions for different swaying periods. Relative to the stationary case, the swaying period exerts a clear influence on the heat transfer coefficient. Within the simulated range, certain operating conditions lead to an enhancement of heat transfer, whereas others result in a reduction, with the overall variation remaining within $\pm 25\%$. For a fixed swaying amplitude, a shorter roll period produces a more pronounced effect on the heat transfer coefficient.

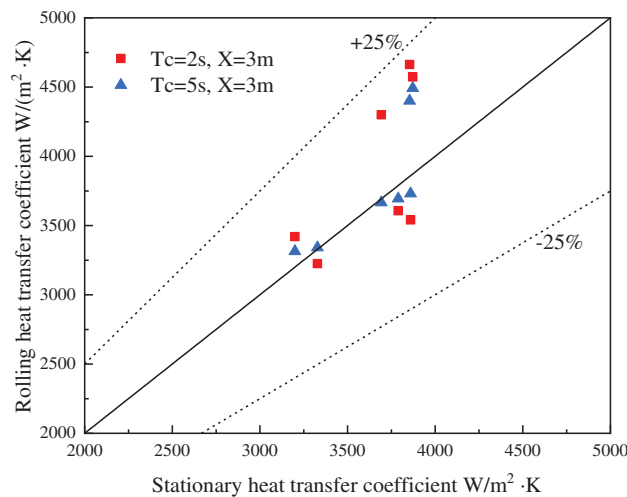


Figure 8: Heat transfer coefficient under different swaying conditions.

3.4 Simulation Results under Different Swaying Amplitudes

Fig. 9 shows the velocity field distribution at the outlet of the measuring section under different swaying amplitudes with the same swaying period. The swaying period is 2 s, and the swaying amplitudes are 2 and 3 m, respectively. When the period remains constant, a larger swaying amplitude results in a higher velocity. The overall flow velocity exhibits alternating increases and decreases, showing a distinct periodic variation. The maximum velocity occurs at the full and half periods, such as at $t/T_c = 1.5$, where the velocity field corresponding to a swaying amplitude of 3 m is significantly greater than that of 2 m. At other times, such as $t/T_c = 1.25$, the difference in velocity field distribution between the two cases becomes less pronounced.

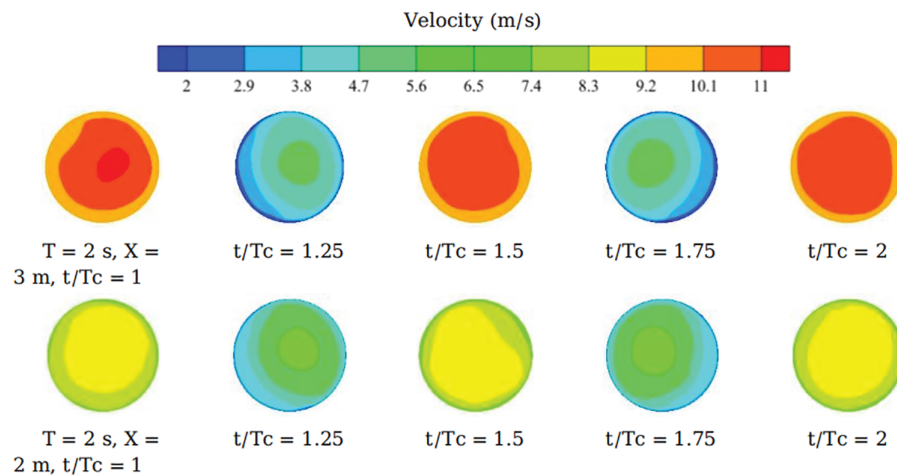


Figure 9: Velocity distribution under different swaying amplitude ($x = 0.5$, $G = 550 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $d = 10 \text{ mm}$).

Fig. 10 shows the distribution of gas volume fraction at the outlet of the measuring section for different swaying amplitudes at the same swaying period. When the period is held constant, a larger swaying amplitude leads to more intense fluctuations, and the overall gas volume fraction exhibits a clear periodic variation.

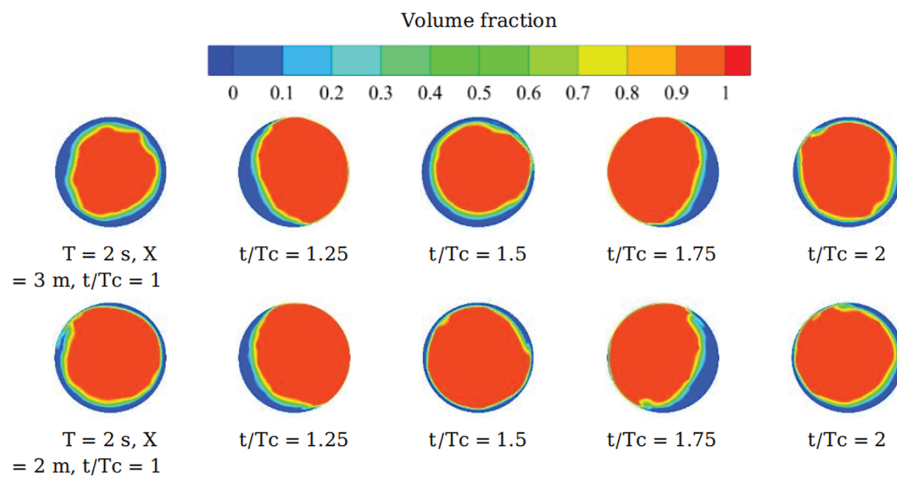


Figure 10: Volume fraction under different swaying amplitude ($x = 0.5$, $G = 550 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $d = 10 \text{ mm}$).

Fig. 11 compares the heat transfer coefficients under stationary and swaying conditions for different swaying amplitudes. Compared with the stationary case, roll motion has a noticeable effect on the heat transfer coefficient. Within the simulated conditions, some cases exhibit heat transfer enhancement, while others show a reduction, with the overall variation remaining within $\pm 25\%$. For a given swaying period, the influence becomes more pronounced as the swaying amplitude increases.

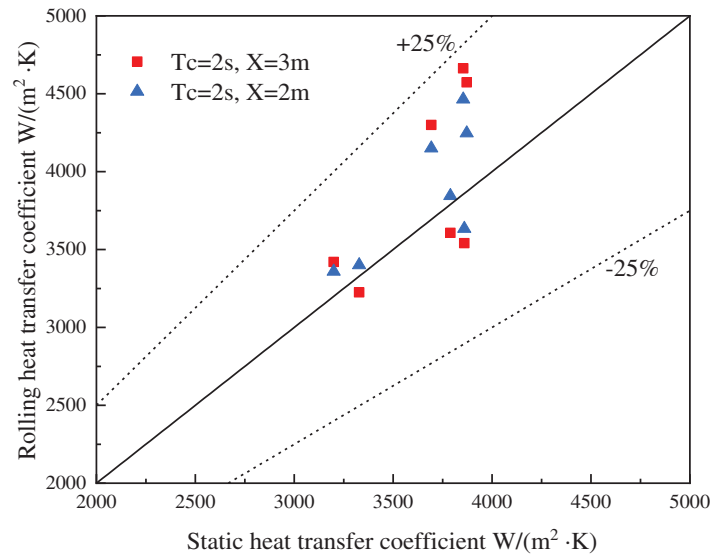


Figure 11: Heat transfer coefficient under different swaying amplitude.

4 Conclusions

This study employs numerical simulations to investigate the condensation of hydrocarbon mixtures in inclined tubes under both static and swaying conditions, yielding the following key findings:

1. The proposed numerical model is capable of simulating the condensation of mixed hydrocarbons in an inclined tube. When the mixing correction is included, the simulated heat transfer coefficients under stationary conditions show a relative deviation of less than 20% from the experimental data, with an average deviation of -0.76% . The simulated frictional pressure drop exhibits a relative deviation of less than 15%, with an average deviation of 4.5%. Deviations from the empirical correlation remain within 15%.
2. At a fixed vapor quality, increasing the mass flux enhances both the heat transfer coefficient and the frictional pressure drop. At a fixed mass flux, higher vapor quality results in increased heat transfer coefficients and greater frictional pressure drops.
3. Shorter swaying periods and larger amplitudes have a more pronounced impact on heat transfer, and the flow velocity exhibits a clear periodic variation. Depending on the motion phase, swaying may either enhance or suppress heat transfer. When swaying increases liquid-film waviness or thickens the condensate film, the resulting rise in thermal resistance reduces the heat transfer coefficient. Overall, the effect on the time-averaged heat transfer coefficient remains within 25%.

Outlook. Under swaying conditions, further research is required to elucidate the condensation behavior of mixed hydrocarbons in spiral (helical) coils and across different tube diameters. In particular, a quantitative assessment of how liquid-film dynamics contribute to heat transfer enhancement or deterioration remains to be developed.

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Nomenclature

A	The swaying amplitude, m
$F\sigma$	Surface tension, N
g	Gravity, m/s^2
h	The average enthalpy of the gas and liquid phases J/kg
h_{film}	Condensate film heat transfer coefficient, $\text{W}/(\text{m}^2 \cdot \text{K})$
P	Pressure, Pa
Pr	Prandtl number
Re	Reynolds number
r	The time relaxation factor, 1/s
S	Source item, $\text{kg}/\text{m}^3 \cdot \text{s}$
T	Temperature, K
T_c	Periodic, s
T_s	Saturation temperature, K
Z	Sensible heat ratio
α	Volume fraction
ρ	Density, kg/m^3
$\vec{u}$	The shared velocity of the two phases, m/s
μ	Dynamic viscosity of the mixture, $\text{pa} \cdot \text{s}$
λ_{eff}	The effective thermal conductivity between the gas and liquid phases $\text{W}/(\text{m} \cdot \text{K})$
θ	The angular displacement, $^\circ$
Φ	Heat and mass transfer resistance in vapor core for mixture $(\text{m}^2 \cdot \text{K})/\text{W}$
film	Condensate film
g	Gas
l	Liquid

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