

Article

Effect of Protrusions on the Falling Film Flow and Heat Transfer of Oily Wastewater Outside an Elliptical Tube

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Abstract

This study addresses the optimized design of falling-film heat exchanger tubes, aiming to enhance heat transfer efficiency and reduce thermal losses, thereby offering potential pathways for efficient green energy utilization. Ten tube models were established and analyzed using computational fluid dynamics (CFD) under constant heat flux conditions. The study investigated the effects of the position, number, and ellipticity (e) of external protrusions on the flow characteristics and heat transfer performance of oily wastewater. The simulation revealed that different protrusion configurations significantly influence hydrodynamic behavior and heat transfer mechanisms. It was found that introducing flow disturbances at an early developmental stage enhances the overall heat transfer performance of the external fluid. Specifically, for a tube with $e = 0.5$, the heat transfer coefficients (HTC) initially increases and then decreases with increasing Reynolds numbers (Re). This behavior is attributed to the reduction in flow stability caused by the protrusions at higher Re values, which promotes vortex shedding and leads to more complex flow patterns, thereby impairing heat transfer efficiency. Furthermore, as the number of protrusions increases, the overall HTC of the enhanced elliptical tube also follows a trend of an initial increase and then decrease. These results suggest the existence of an optimal protrusion density that enhances turbulence without incurring excessive resistance that would degrade thermal performance.

Keywords: falling film; protrusions; heat transfer; numerical simulation



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1. Introduction

Driven by the dual-carbon target, energy structure transformation is the key to sustainable economic and social development, and energy saving and efficiency enhancement has become a strategic focus. The falling film heat transfer technology, as a kind of efficient heat transfer method, is popular in the industrial field, and there are numerous uses for the horizontal tube falling film evaporator in the fields of seawater desalination, petrochemicals, chemical industry, refrigeration and so on [1].

Researchers have conducted numerous investigations into the heat and mass transport of fluids in smooth pipes with different parameters [2–6]. The heat transfer characteristics of different fluids flowing through heat transfer pipes have also attracted a lot of attention from researchers [7–9]. Among them, the liquid film spreading characteristics on the falling film tube are crucial for the liquid film heat transfer process [10–13]. Qi et al. [14] performed numerical calculations for modeling of circular as well as elliptical tubes, respectively. They found that the liquid film is more stable and has a thinner thickness outside of the elliptical tubes and is more stable as compared to the circular tubes. Zhao et al. [15] investigated the role of surface tension in the spreading process of liquid film and found that surface tension is critical to the flow process and that significant flow separation occurs in the absence of surface tension. Tahir et al. [16] demonstrated the effect of viscosity as well as surface tension on flow by investigating the changes in the liquid film in the presence of viscous action, surface tension, and their combined action. In papers by Fiorentino et al. [17–19], the local heat transfer coefficient (h_θ) is maximum in the impact zone in the upper part of the tube; then it decreases rapidly as the liquid film spreads above the tube. After that, in the developmental transition zone of the flow, h_θ exhibits a slow decreasing tendency with the gradual reduction in the temperature. Bajalan et al. [20] proposed that the variation in h_θ , with the parameter θ , is a result of the combined effect of both the film velocity, to which the velocity vector corresponds, and the film thickness. Specifically, on the one hand, h_θ increases with the film thickness, which explains why the heat transfer effect in the impact zone is most significant even when the film thickness is at its maximum value; on the other hand, h_θ also increases with the film velocity. In experimental research, Mu et al. [21] discovered that as spray density increases, the HTC first rises and then falls. Yang et al. [22] calculated dimensionless temperature profiles around the tube. They discovered that heat transmission at the microscopic level is influenced by both conduction and convection. Zhao et al. [23] found that there is a double paradoxical effect of increasing membrane flow rate the liquid flow rate is appropriately accelerated to facilitate heat transfer, but the thickening of the liquid film will hinder heat transfer.

The interaction of factors such as heat flux and tube diameter was fully considered in the design and construction of the evaporator of Edahiro et al. [24]. It was found that the heat flux as well as the tube diameter had a minimal effect on the total HTC. Hao et al. [25] constructed a three-dimensional pipeline with circular dimensions in order to investigate in depth the effects of different pipe sizing parameters and the oil content in oily wastewater on the fluid flow outside the pipe and quantitatively analyzed the correlation between different spray densities and pipeline sizes, etc., and the heat transfer outside the pipeline, thus revealing the profound influence of these parameters on the heat transfer characteristics through the exhaustive simulation calculation results. With the goal to examine the impact of core elements such as inlet height and oil content on the flow of oily wastewater outside a two-dimensional elliptical pipe, Zhang et al. [26] established a numerical model of a two-dimensional elliptical pipe, analyzed the flow and heat transfer behavior of oily wastewater in a complex pipeline environment by numerical computation, compared the simulation results with different combinations of parameters, and systematically revealed how these factors work together in the flow pattern, velocity distribution, temperature gradient, heat transfer efficiency, etc., of the oil-containing wastewater. The simulation results are compared and analyzed under different combinations of parameters, which systematically reveal how these factors synergistically affect the flow pattern, velocity distribution, temperature gradient and heat transfer efficiency of oily wastewater.

Cao et al. [27] designed an integrated structure, through the establishment of an experimental model, to study different parameter changes in the tube outside the spiral fins and smooth tube to compare the surface structure of the tube as well as the channel

baffle. The study found that the addition of joining in the flow field to join the structure of the baffle could effectively improve heat transfer. After establishing a heat exchanger tube bundle model with four different pipe shapes, He et al. [28] compared and analyzed the changes in the pipes' axial and circumferential heat transfer curves under various flow conditions. Ultimately, they discovered that elliptical tubes dominated the heat transfer effect under sheet flow conditions. Zhang et al. [29] examined the flow properties of elliptical tubes with various shapes under low Re flow circumstances by combining numerical calculations and tests and studied the flow characteristics and distribution characteristics of liquid film under the influence of wind speed. Nemati et al. [30] investigated the fluid flow as well as heat transfer in the tube with different parameters by modeling a circular tube with different fin shapes and using numerical simulation to derive the optimum fin parameters. Silk et al. [31] have experimentally investigated the better heat transfer performance of optimized geometries at low heat fluxes when three different fin shapes are added to a copper block at different spray angles. Ji et al. [9] developed an experimental model to compare the enhanced tube pool boiling, enhanced single-tube falling and smooth tube falling film evaporation process under different heat fluxes. Experimental data show that the enhanced tube exhibits superior heat transfer efficiency when the pool is at a boil. A rudimentary falling film plate numerical model was developed by Lu et al. [32] and they compared the heat transfer and dehumidification capacity of LiCl on smooth as well as rough plates by numerical simulation method; they found that the dehumidification enhancement with the rough plate could reach 23.8%. Åkesjö et al. [33] designed experiments to quantify the heat transfer of three types of tubes, smooth, corrugated and welded with ribs, in a single tube evaporator for different heat fluxes and found that Welded plates had higher heat transfer, followed by corrugated plates, and smooth plates had the smallest, and it was demonstrated through pilot-scale experiments that modified plates had better heat transfer performance. Lee [34] compared the heating rates of smooth tubes as well as porous coated tubes with copper powder sintered on the surface at low flow rates by designing experiments to compare the heating rates of smooth tubes and porous coated tubes with copper powder sintered on the surface through the measurements of heat transfer rates. Bock et al. [35] analyzed the effect of different materials and roughness on the buckling film heat transfer by polishing and then manually controlling the roughness of copper, soft steel and stainless steel tubes, experimentally testing the pressure and then calculating the HTC. The discovery was that the enhancement factor of falling film for all the tubes studied was positively correlated with the roughness. Cao et al. [36] developed a three-dimensional finned tube model and investigated the effect of liquid column spacing and other different physical properties on heat transfer. Numerical calculations led to the conclusion that fins away from the liquid column would have a higher HTC. In order to study the effect of porous coating on the heat transfer of the falling film tube, Zhao et al. [37] set up a three-dimensional numerical model, coupled with the Volume of Fluid (VOF) method, to analyze the heat transfer of the tube with different tube materials and different porosities. It was found that the smaller the porosity, the stronger the heat transfer and the better the heat transfer performance of copper and aluminum tubes. Zhao et al. [38] studied descending membranes with ribs of different shapes as well as slotted plates and found that increasing the original spacing attenuates the perturbation of the flow by the elements and that the perturbation of the flow is stronger for right-angled triangles.

Through the study of the above papers, we find that there are many studies on circular tubes and modified heat exchanger tubes or flat plates by researchers. However, there are few studies on elliptical tubes with raised structures. Particularly, the research is directed towards the shape as well as the location of the raised structure. Since elliptical

tubes perform better at heat transfer than circle tubes, research on liquid film flow and heat transfer properties outside of rough elliptical tubes is crucial for increasing energy efficiency and encouraging emission reduction and energy conservation.

2. Materials and Methods

The simulation was performed using Fluent 2022R1 software. This study employs a numerical model that idealizes complex oily wastewater as a homogeneous water-glycerol mixture, thereby focusing on fundamental hydro-thermal mechanisms; the results are thus directly applicable to oil-water systems dominated by physical properties (e.g., viscosity and surface tension) with negligible interfacial contamination. The numerical simulation in this research is based on the following assumptions:

(1) The fluid is incompressible, and the flow parameters are stable. (2) Laminar flow exists within the tube. (3) It is assumed that the oily wastewater is a mixture of water and glycerol, and there is no deposit formation on the pipe wall. (4) The rate of heat flux through the tube wall remains constant.

$$\frac{\partial(\rho C_P T)}{\partial t} + \text{div}(\rho C_P T \mathbf{V}) = \text{div}(k_{eff} \text{grad} T) + Q_T \quad (1)$$

$$\frac{\partial(\rho u)}{\partial t} + \text{div}(\rho u \mathbf{U}) = -\frac{\partial p}{\partial x} + \text{div}(\eta \text{grad} u) + S_u \quad (2)$$

$$\frac{\partial(\rho v)}{\partial t} + \text{div}(\rho v \mathbf{U}) = -\frac{\partial p}{\partial y} + \text{div}(\eta \text{grad} v) + S_v \quad (3)$$

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} = 0 \quad (4)$$

The fluids involved in the calculation process are gases and liquids, and the VOF model has the advantage of dynamically tracking the flow process. To precisely capture the liquid film characteristics and heat transfer, the VOF model is chosen for the simulation calculation.

$$\mu = \alpha_l \mu_l + \alpha_g \mu_g \quad (5)$$

$$\alpha_l + \alpha_g = 1 \quad (6)$$

$$\rho = \alpha_g \rho_g + (1 - \alpha_g) \rho_l \quad (7)$$

This study employs the RNG k- ϵ turbulence model.

$$\frac{\partial(\rho \kappa)}{\partial t} + \frac{\partial(\rho \kappa u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[\alpha \kappa \mu_{eff} \frac{\partial \kappa}{\partial x_j} \right] + G_k + G_b - \rho \epsilon - Y_M + S_\kappa \quad (8)$$

$$\frac{\partial(\rho \epsilon)}{\partial t} + \frac{\partial(\rho \epsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[\alpha_\epsilon \mu_{eff} \frac{\partial \epsilon}{\partial x_j} \right] + C_{1\epsilon}^* \frac{\epsilon}{\kappa} (G_k + C_{3\epsilon} G_b) - C_{2\epsilon} \rho \frac{\epsilon^2}{\kappa} - R_\epsilon + S_\epsilon \quad (9)$$

Surface tension significantly influences the capability of a liquid film to retain its original shape, so we choose the surface tension (CSF) model. Below are the equations.

$$\vec{F} = \delta \frac{\rho \kappa \nabla \alpha}{(\rho_l + \rho_g)/2} \quad (10)$$

Other parameters are calculated as follows.

When wall thermal resistance is neglected, the local heat transfer coefficient is defined by the local heat flux and the temperature difference between the tube wall and the liquid film.

$$h_{\theta} = \frac{q}{T_{w,\theta} - T_{\theta}} \quad (11)$$

$$h_{ave} = \frac{1}{\pi} \int_0^{\pi} h_{\theta} d\theta \quad (12)$$

$$Re = \frac{4\Gamma}{\mu} \quad (13)$$

For the spatial discretization, a first-order upwind scheme was applied to the turbulence and water vapor transport equations, while a second-order upwind scheme was used for the momentum and energy equations. Pressure was discretized using the PRESTO! scheme, and the coupling between pressure and velocity was handled by the PISO algorithm.

3. Case Description and Solution Method

3.1. Physical Model and Boundaries

This paper systematically investigates the mechanism by which protrusions influence the heat transfer performance of elliptical enhanced tubes. A comprehensive numerical analysis of key parameters, including the Re , number of protrusions, their distribution, and geometry, was conducted to reveal the dominant mechanisms governing the convective heat transfer enhancement. To accurately evaluate the independent effect of each parameter, factors such as fluid properties, inlet temperature, and wall thermal boundary conditions were maintained constant, thereby isolating the influence of the variables under consideration.

Figure 1 illustrates the geometric configuration of the physical models employed in this study, which comprise ten elliptical tubes with distinct enhancement elements. These elements, which are protrusions extending into the fluid domain, are designed to perturb the flow and enhance heat transfer performance. To balance computational efficiency with model representativeness, all geometries were simplified to two-dimensional axisymmetric models, an approach that retains the key flow and heat transfer characteristics. For instance, Figure 2 shows the computational domain for the Model 08 with five semicircular protrusions, including the fluid region, tube wall, and the corresponding boundary conditions.

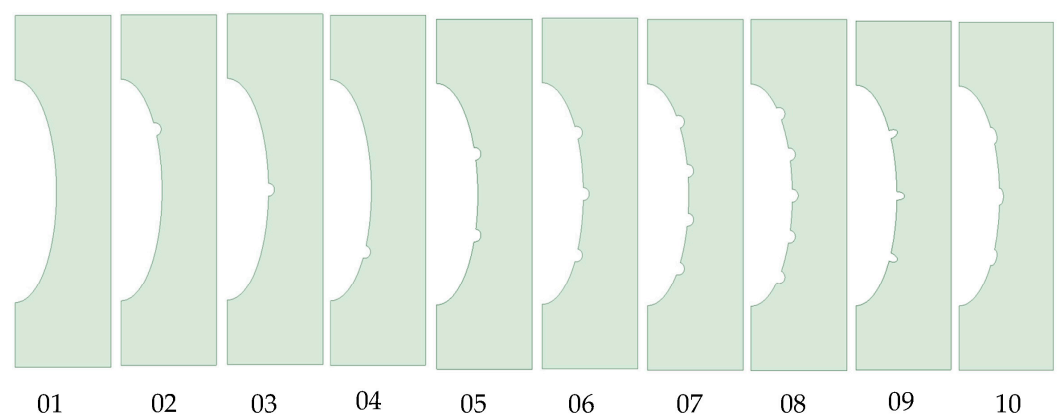


Figure 1. Model designations for enhanced tubes.

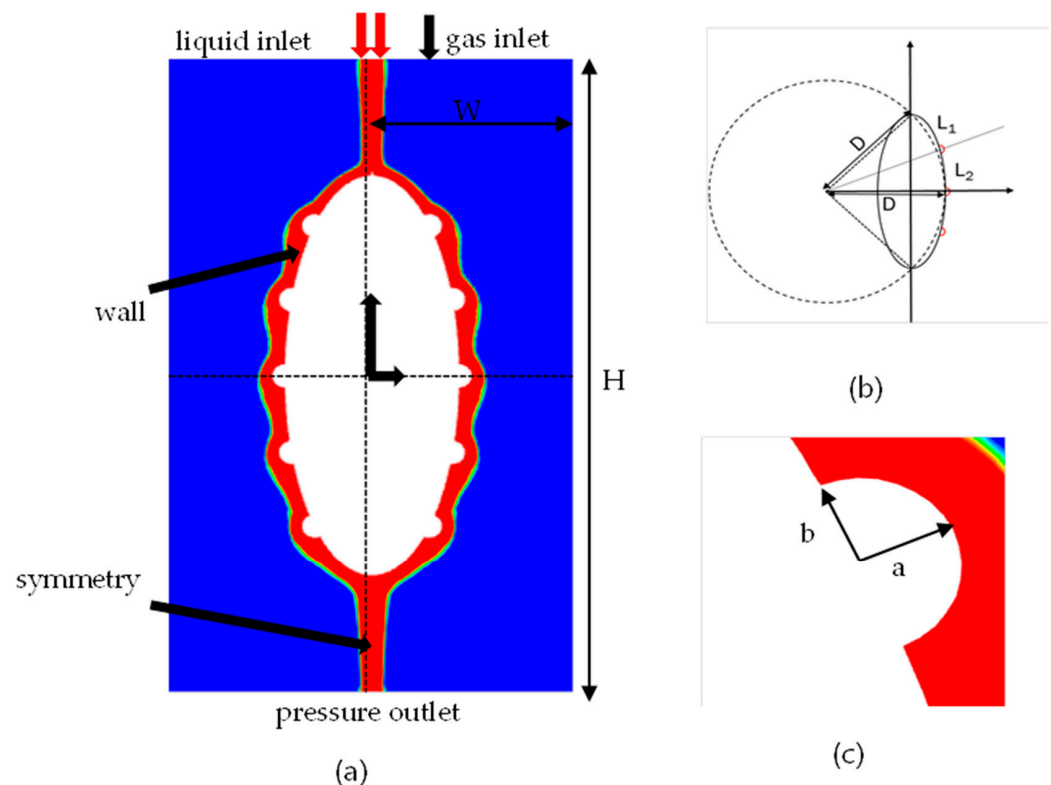


Figure 2. Boundary conditions and the physical model. (a) Tube computational domain; (b) Protrusion layout along elliptical tube; (c) Protrusion shape.

Figure 2a illustrates the geometric of the elliptical tube model and the computational domain. The tube has a major axis diameter of 34.5 mm and a minor axis diameter of 13.8 mm. The rectangular computational domain has a width (W) of 15 mm and a height (H) of 54.5 mm. Liquid is injected through the inlet that is 2 mm wide at the top, forming a falling film flow along the tube wall under gravity. To enhance computational efficiency and exploit geometric symmetry, only half of the channel is used as the computational domain, with a symmetry boundary condition applied on the symmetry plane. Oily wastewater enters from an inlet 10 mm above the tube, flows downward along the wall, and is discharged through the bottom outlet. As shown in Figure 2b, the protrusions are uniformly arranged on the tube with equal curvature. Figure 2c illustrates the cross-sectional geometry of a protrusion, defining the aspect ratio, $e = b/a$.

This study employs a uniform thickness condition at the film inlet. Although practical inlet conditions are often non-uniform, this simplification is standard in fundamental studies of falling film hydrodynamics [36,39]. This approach is adopted to isolate the effect of protrusion geometry on flow development and heat transfer, without the confounding influence of complex inlet distributions.

3.2. Grid System and Independence Validation

As illustrated in Figure 3, a structured grid was employed for the spatial discretization of the computational domain, with red borders and arrows highlighting the local grid. The mesh was refined near the tube wall to accurately capture the associated flow and heat transfer characteristics.

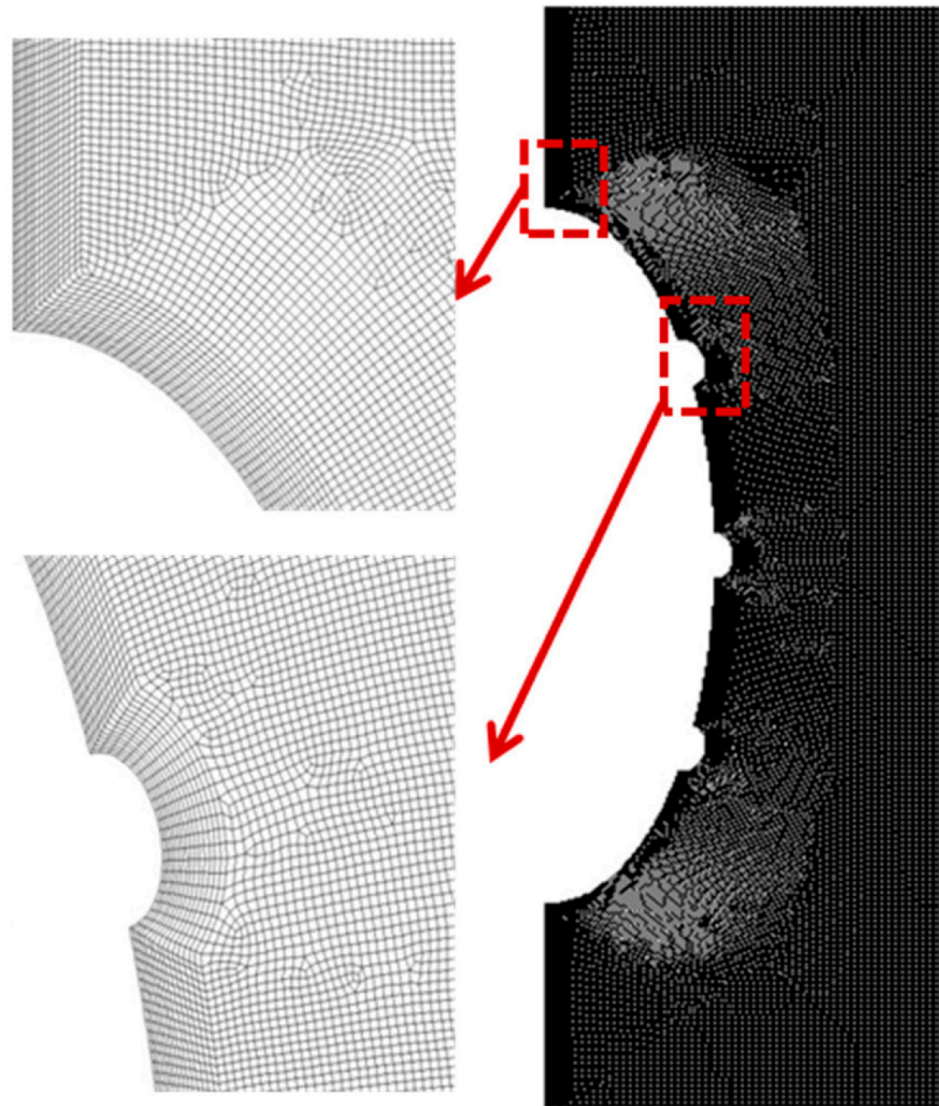


Figure 3. The mesh model.

3.3. Model Validation

Mesh independence was validated using model 01 as a representative case. As shown in Figure 4, the average HTC stabilized at a resolution of 16,935 elements. Further refinement to the finest mesh resulted in deviations of less than 5%, indicating that the 16,935 elements mesh satisfactorily meets the computational accuracy requirements. Therefore, considering a balance between computational efficiency and numerical accuracy, the mesh with 16,935 elements was selected for subsequent simulations of this tube. All other tube configurations employed a similar mesh resolution and identical control parameters to ensure result consistency and comparability.

A time-step independence study was performed to ensure the reliability of the numerical results, as summarized in Figure 5. Simulations were conducted with five different time steps. The results indicate that for a time step of $\Delta t = 5 \times 10^{-6}$ s, which corresponds to a maximum Courant number below 0.25, the variations in the local HTC become negligible (changes of less than 1%). Since a further reduction in the time step incurred significant computational overhead without materially affecting the solution, $\Delta t = 5 \times 10^{-6}$ s was adopted for all subsequent simulations, achieving a balance between computational efficiency and solution accuracy.

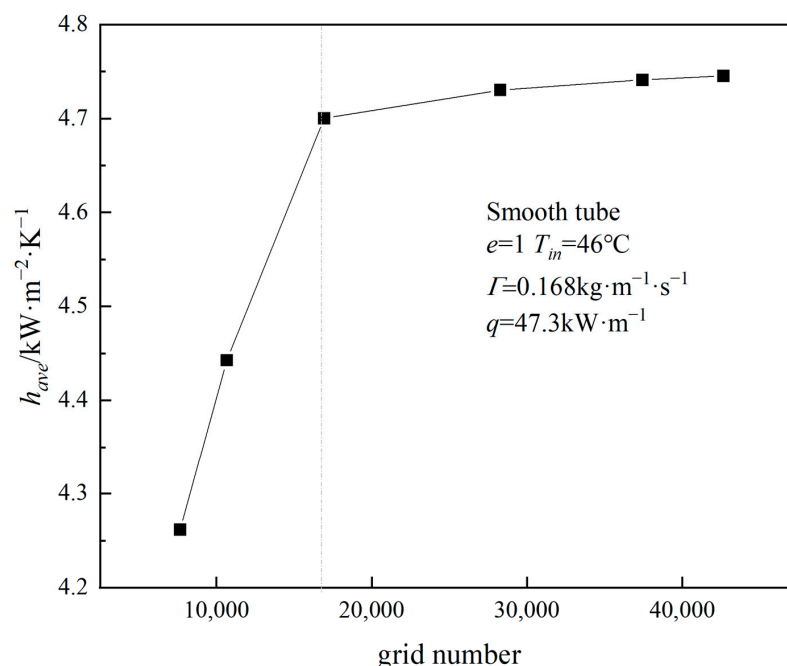


Figure 4. Grid independence verification.

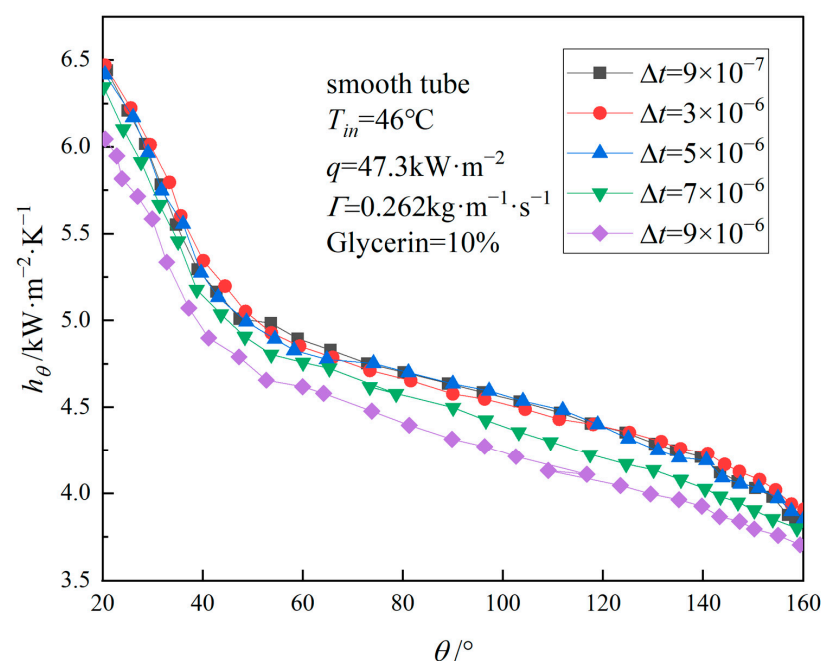


Figure 5. Time step independence verification.

The convergence of transient simulations is governed by residuals within each time step. As demonstrated by the convergence history shown in Figure 6, residuals for all equations exhibit decreasing trends, stabilizing below the preset convergence criteria. This robust convergence behavior confirms that each time step's solution is fully adequate.

This study developed a numerical model of a smooth elliptical tube and validated the computational methodology. For model validation, systematic comparisons were conducted against existing experimental data. Key parameters in the numerical model, including boundary conditions, material properties, and initial conditions, were aligned with the experimental conditions to ensure a fair comparison. As shown in Figure 7, the simulated liquid film thickness shows excellent agreement with the results published by Hou [40], and the HTC correlates closely with the data reported by Parken [41]. The model accurately

captures both the trends and the absolute values of these parameters. Quantitative analysis shows average deviations of 7.1% for the liquid film thickness and 6.5% for the HTC, both of which are within acceptable limits for engineering applications. The close agreement between the numerical and experimental results validates the present model and the selected parameter set. This provides a reliable foundation for subsequent studies, enabling the use of this model to analyze heat transfer and flow characteristics under various structural and operational conditions.

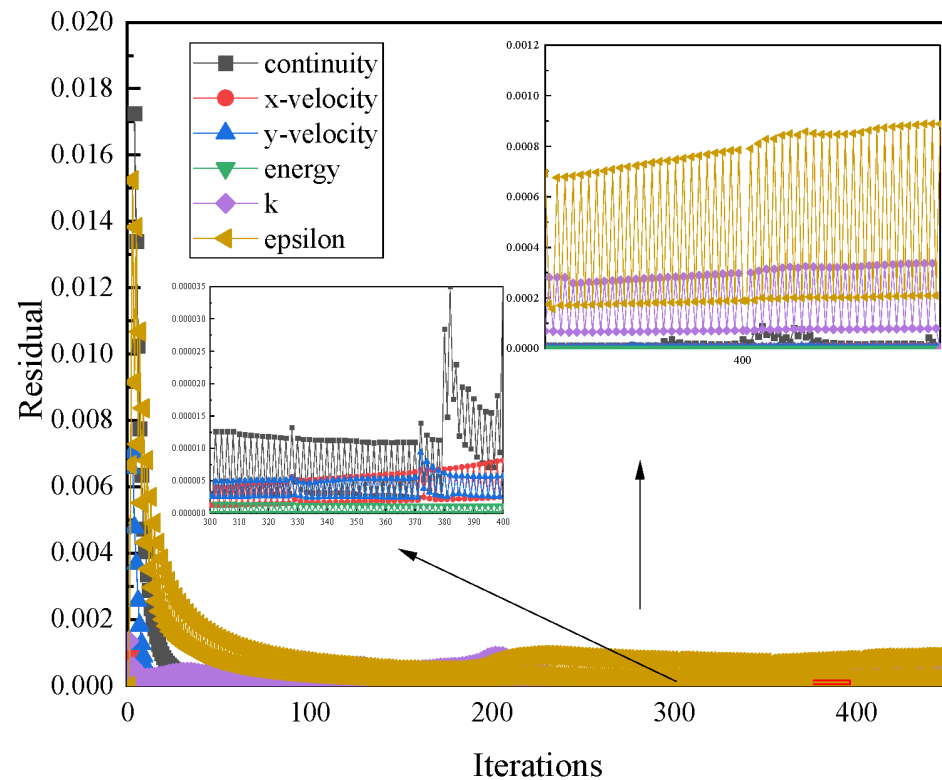
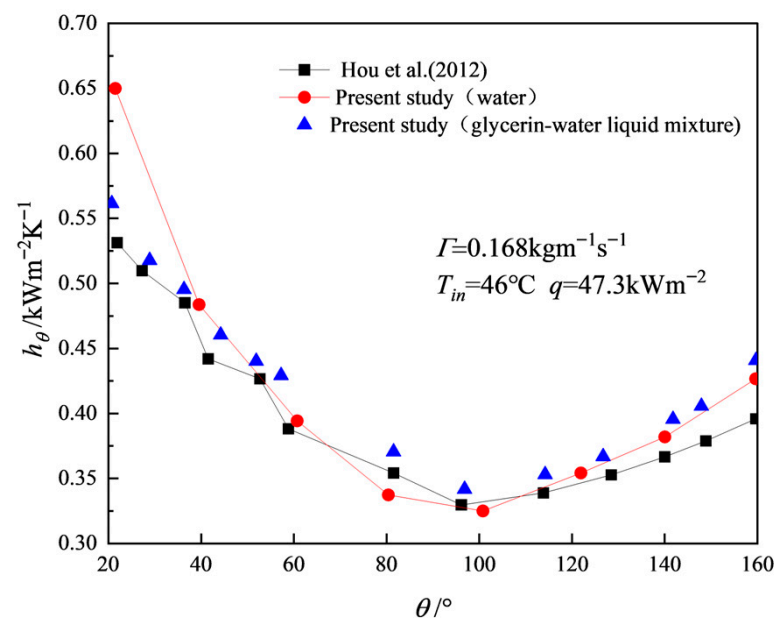


Figure 6. Convergence of the computational residuals.



(a)

Figure 7. Cont.

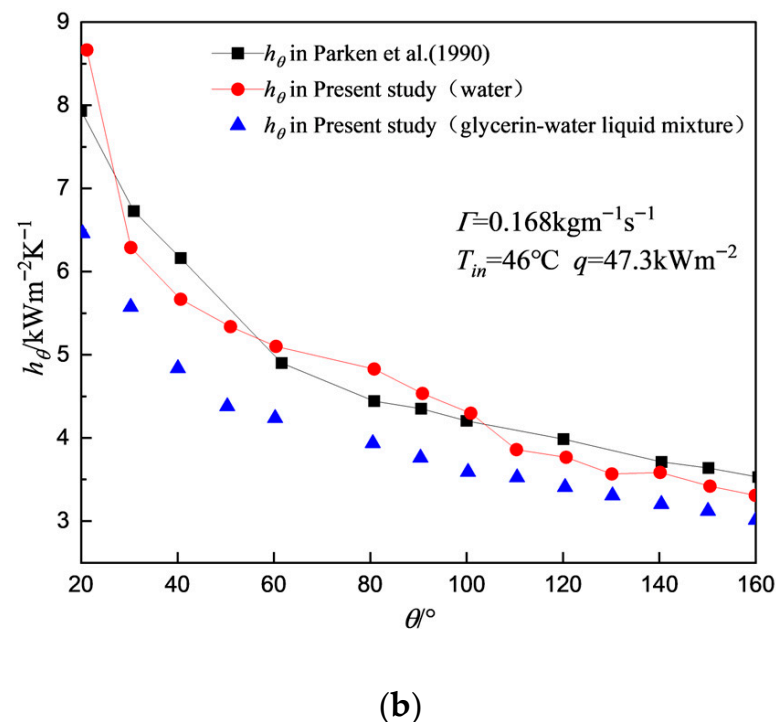


Figure 7. Validation of liquid film thickness and h_θ [40,41]. (a) Validation of h_θ ; (b) Validation of δ .

4. Results

4.1. Spatiotemporal Evolution of Falling Films over Enhanced Elliptical Tubes

The temporal evolution of the liquid film flow along the outer wall of the enhanced elliptical tube is illustrated in Figure 8. Figure 8a depicts the initial flow stage, where the oily wastewater descends along the tube wall under the combined action of gravity and wall adhesion. Subsequently, as shown in Figure 8b, the liquid stream impacts the wall surface due to inertial forces. After the impact, the liquid film continues to spread and flow downward, driven by gravity and adhesion. Upon encountering the first protrusion (Figure 8c), the flow undergoes a significant change. The protrusion, with its height perpendicular to the wall, acts as a physical barrier, obstructing the flow path and causing noticeable upstream film accumulation. During this process, part of the film's kinetic energy is converted into pressure energy, inducing flow separation. This enables the liquid film to reattach along the edge of the structure and continue flowing downstream. In Figure 8d, the liquid film becomes locally thicker on the protrusion surface. Owing to the Coanda effect, the film adheres to the protrusion's contour and follows its curvature. Simultaneously, the protrusion significantly enhances the waviness of the film, increasing flow instability. By the stage shown in Figure 8e, the leading edge of the liquid film experiences a sudden velocity decrease after passing over the protrusion. However, the trailing fluid continues to advance by inertia, leading to further accumulation upstream. Under the Coanda effect, the liquid film continues to move along the curved surface, exhibiting pronounced unstable oscillations near the protrusion. Despite the strong disturbance, no film rupture occurs due to the stabilizing effect of surface tension. Finally, as depicted in Figure 8f, the liquid film detaches from the separation zone downstream of the protrusion and continues its downward flow, completing a full transient flow cycle of the oily wastewater outside the enhanced elliptical tube.

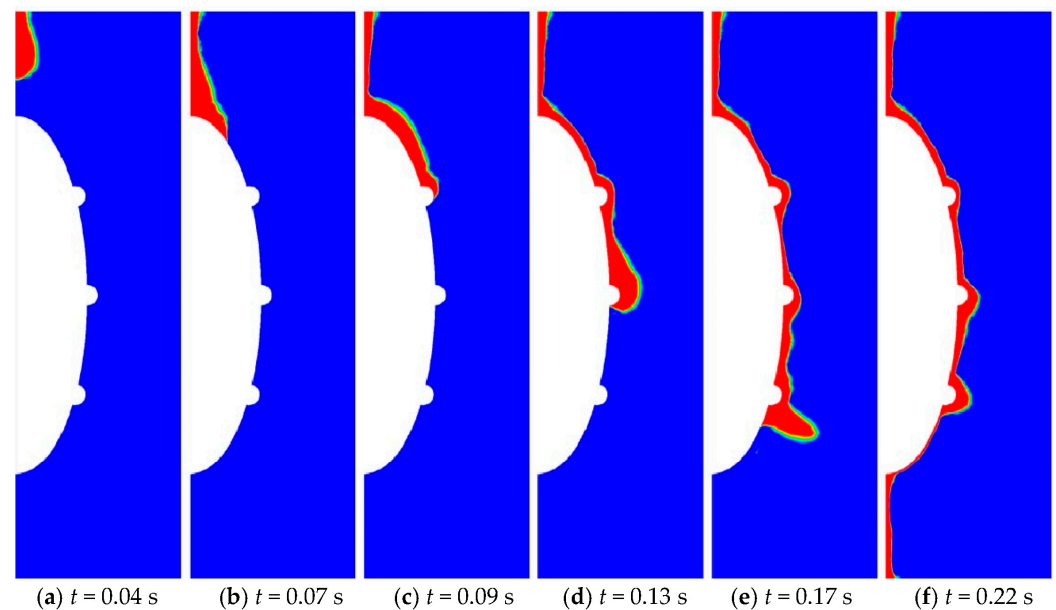


Figure 8. Transient liquid film evolution.

4.2. The Influence of Protrusion Position on Heat Transfer Performance

Figure 9 presents a comparison of the liquid film distributions over smooth and enhanced elliptical tubes with protruding structures at $Re = 1000$. As observed in Figure 9, the liquid film is thicker near the protrusions. The corresponding turbulent kinetic energy (TKE) contour plot (Figure 10) shows that regions of high TKE are predominantly located where the liquid film impinges on the tube crown and the protrusions. In smooth elliptical tubes, the crown and bottom exhibit regions of higher local TKE. In Tube 02, this protrusion is closest to the top of the tube. Its position disturbs the flow earlier and significantly increases the overall TKE level. In contrast, the protrusion on Tube 03 is located further down, and its resulting flow perturbations are mainly concentrated in the mid-to-lower sections. As shown in Figure 10, the region around the protrusion in Tube 02 exhibits significantly higher local TKE. The TKE at the corresponding location in Tube 03 is noticeably lower, whereas Tube 04 shows the lowest TKE levels near its protrusion. This correlation demonstrates that a more upstream protrusion position more effectively enhances liquid film disturbance, disrupts the hydrodynamic boundary layer, and intensifies heat transfer through the increase in turbulence intensity. In Tube 02, the protrusion is positioned closer to the top of the tube, where the incoming fluid has a thinner boundary layer, higher TKE, and lower flow stability. When this high-energy flow impinges on the protrusion, it induces more severe flow separation and shear layer instability, thereby significantly enhancing the production of local TKE. Compared to Tubes 03 and 04, Tube 02 exhibits a higher conversion efficiency of mean kinetic energy into TKE, with a corresponding reduction in energy loss through viscous dissipation.

The distribution of the local HTC, shown in Figure 11, corroborates the flow characteristics revealed by the TKE contours. A significant fluctuation is observed for Tube 02 near the circumferential angle $\theta = 30^\circ$. Tubes 03 and 04 exhibit pronounced fluctuations in the HTC near $\theta = 100^\circ$ and $\theta = 150^\circ$, respectively. This phenomenon is attributed to the disturbance generated by the protrusions on the enhanced elliptical tube wall: when fluid impacts these protuberances, the flow pattern is altered, turbulence intensity is augmented, and TKE rises, thereby enhancing heat transfer. In contrast, the flow structure near the smooth tube wall remains stable without significant flow separation or abrupt changes in TKE. Consequently, its HTC distribution remains relatively smooth and uniform.

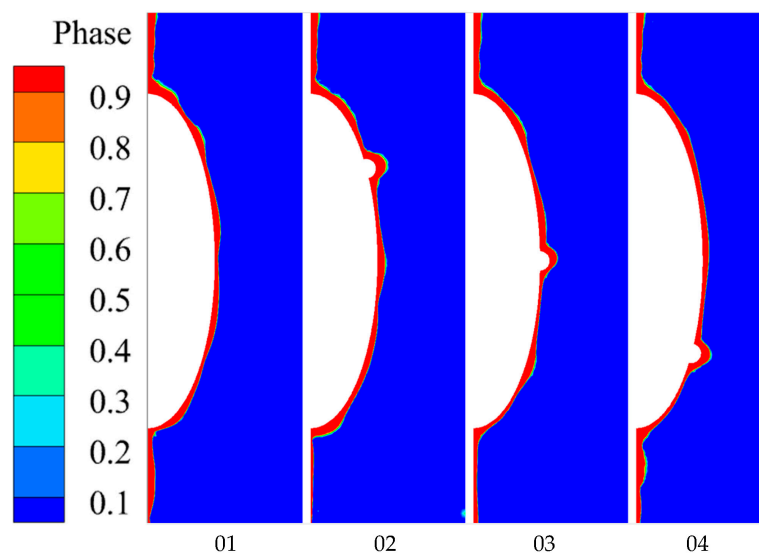


Figure 9. Effect of protrusion location on liquid film thickness distribution in enhanced tubes.

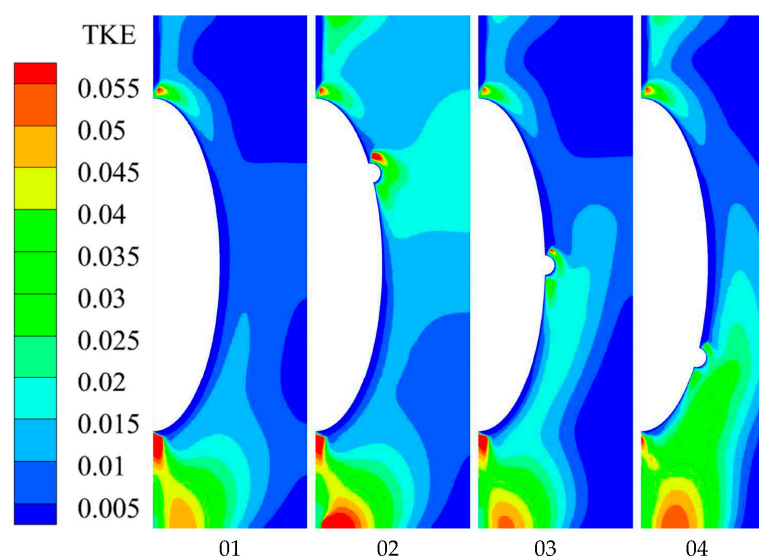


Figure 10. Effect of protrusion location on TKE distribution in enhanced tubes.

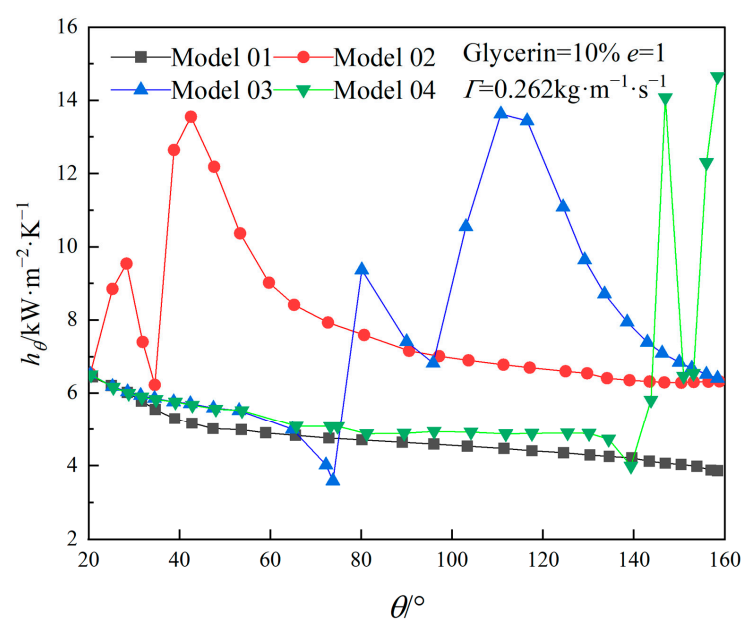


Figure 11. Distribution of Local HTC in enhanced tubes with different protrusion locations.

4.3. Influence of e and Re on Heat Transfer Performance

Figure 12 illustrates the variation in the average HTC for enhanced elliptical tubes of different e over a range of Re . Overall, the e of the protrusions has a diverse influence on thermal performance. For $e = 0.5$, the average HTC initially increases with the Re , and subsequently declines, indicating a clear non-monotonic trend. In contrast, tubes with external protrusions of $e = 1.0$ and $e = 1.5$ exhibited a monotonically increasing trend with the Re , with no observable inflection point.

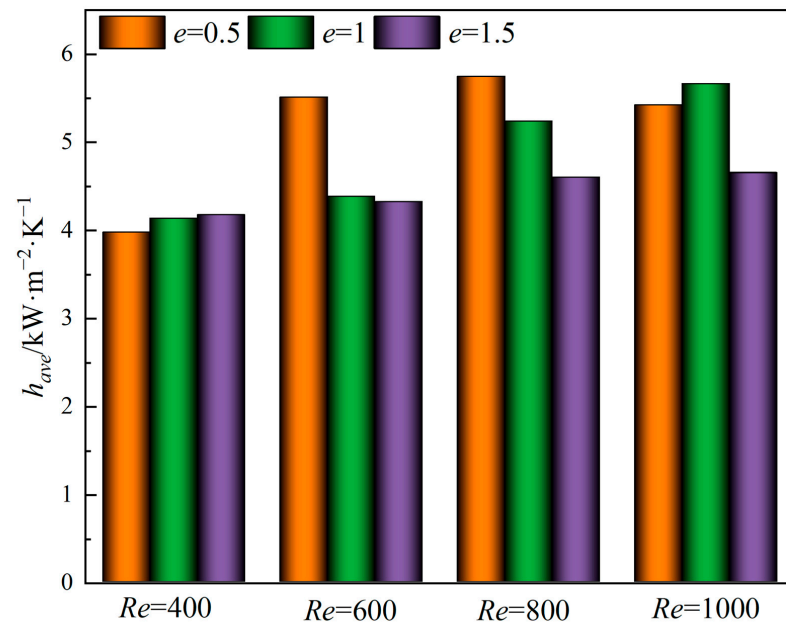


Figure 12. Influence of e and Re on the average HTC of enhanced elliptical tubes.

4.3.1. Impact of Re

Figure 12 illustrates the heat transfer performance of enhanced elliptical tubes with varying e . For $e = 0.5$, the HTC initially increases and then decreases with a rising Re . Due to this complex, non-monotonic response, the configuration with $e = 0.5$ was selected for an in-depth analysis of the influence of the Re on its flow characteristics to elucidate the underlying mechanisms. Figure 13 illustrates the liquid film distribution patterns for the $e = 0.5$ tube at various Re , clearly revealing variations in film thickness distribution and wetting behavior with Re .

As shown in Figure 13, the liquid film thickness distribution at $e = 0.5$ for different Re . The figure clearly shows that the protrusions, with a finite height, act as a significant physical barrier to the liquid film flow, causing pronounced upstream accumulation. At $Re = 400, 600$, and 800 , increasing fluid inertial forces progressively reduce the overall liquid accumulation. The film thickness upstream of the protrusions decreases markedly, while the thickness between protrusions increases, indicating that the enhanced fluid momentum is sufficient to overcome the flow resistance, leading to a more uniform distribution. At $Re = 400$, the flow exhibits a characteristic three-stage behavior near the protrusions: upstream accumulation, top spreading, and downstream reattachment with secondary accumulation due to flow separation. However, at $Re = 1000$, the film distribution shows greater complexity and spatial inhomogeneity. While the film spreads relatively uniformly over the first protrusion, significant accumulation occurs at the second and third protrusions. This phenomenon indicates that at higher Re , although global inertial forces dominate, the combined effects of multiple flow separations, vortex shedding, and free-surface instabilities lead the liquid film to exhibit significant transient behavior and spatial inhomogeneity as it propagates downstream.

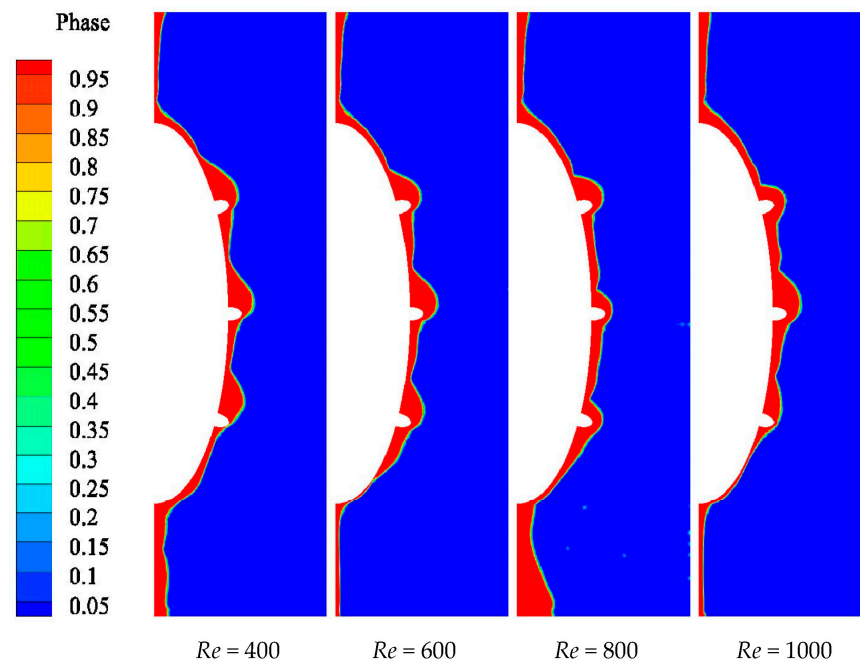


Figure 13. Liquid film distribution on enhanced elliptical tubes at different Re .

Figure 14 presents a velocity distribution that elucidates the mechanisms governing the liquid film distribution at different Re . A distinct low-velocity stagnation zone is observed upstream of the protrusion across all conditions, resulting from flow obstruction. As the Re increases from 400 to 800, the strengthened fluid inertial forces more effectively overcome the adverse pressure gradient. This results in a more vigorous impact on the protrusion, causing the stagnation zone to contract. Within this range, near-wall velocities increase markedly. The geometric constraint of the protrusion also induces a local acceleration zone upstream due to streamline compression. This acceleration enhances turbulent mixing, which effectively destabilizes the thermal boundary layer and thereby enhances heat transfer. However, when the Re increases further to 1000, the flow field undergoes a significant transformation. The heightened flow instability promotes energy dissipation in the core flow, which compromises the uniformity of the velocity distribution. The flow exhibits strong streamwise fluctuations, accelerating before decelerating substantially. This unstable behavior weakens the sustained disruption of the thermal boundary layer, leading to a net decline in overall heat transfer performance.

As shown in Figure 15, the distribution of the local HTC further corroborates the flow behavior inferred from the velocity field in Figure 12. The protrusions significantly disrupt the liquid film, causing pronounced fluctuations in the local velocity gradient that correlate well with the tube structure. At $Re = 400, 600$, and 800 , three regions of inferior heat transfer performance are observed near $\theta = 20^\circ, 70^\circ$, and 140° . These locations correspond to areas of thick liquid film accumulation, where the increased thermal resistance degrades heat transfer performance. Conversely, higher HTCs are observed near $\theta = 30^\circ, 90^\circ$, and 150° , which coincide with flow reattachment points, film splitting, or local acceleration zones. The enhanced flow disturbance in these regions promotes heat transfer effectively. When the Re increases to 1000, the local HTCs are significantly lower than those at $Re = 800$. This reduction is primarily attributed to excessive flow instability, where kinetic energy is dissipated turbulently rather than utilized for effective near-wall mixing. Concurrently, the markedly diminished fluctuation amplitude at positions such as $\theta = 90^\circ$ and 150° indicates a weakened disturbance effect from the protrusions and a potential shift in the flow separation point. The deterioration of the overall flow structure ultimately diminishes

heat transfer performance, demonstrating that beyond an optimal value, a higher Re can detrimentally affect heat transfer due to flow field degradation.

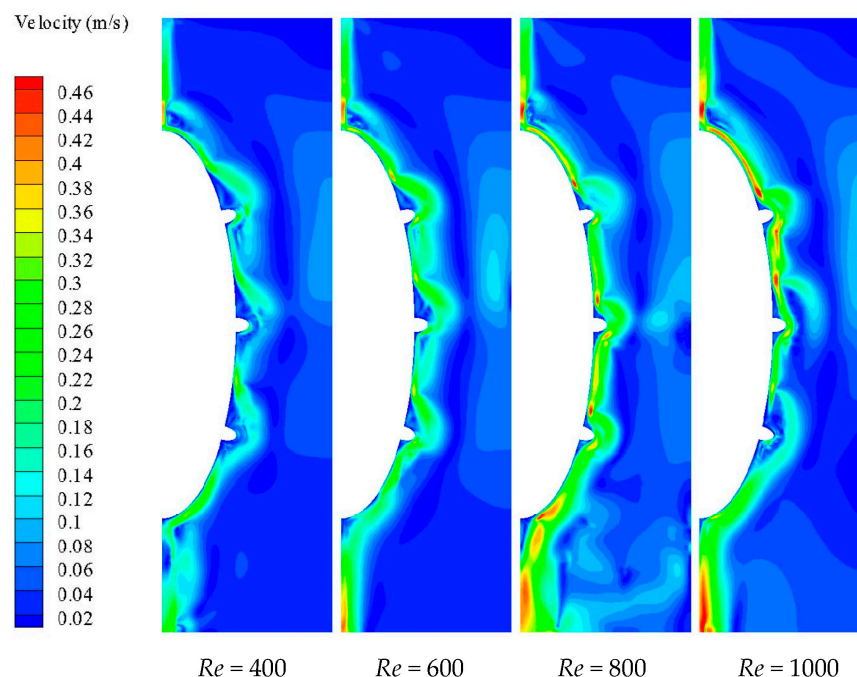


Figure 14. Velocity distribution on enhanced elliptical tubes at different Re .

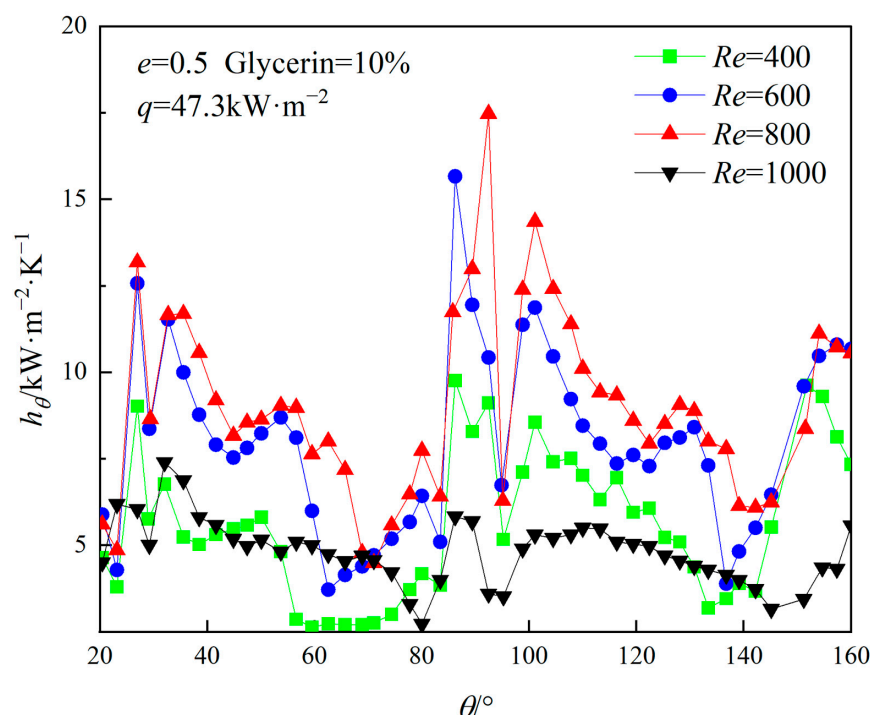


Figure 15. Influence of Re on the local HTC of enhanced elliptical tubes.

4.3.2. Impact of e

At $Re = 400$, different protrusion geometries on the elliptical tube's outer wall provide limited heat transfer enhancement, with no significant differences observed between the structures. This suggests that at low Re , flow remains viscosity-dominated, and the protrusions are unable to promote sufficient turbulent mixing, resulting in a limited contribution to overall heat transfer. However, as the Re increases to 600 and 800, increasing

fluid inertia amplifies the influence of e , making its effect on heat transfer performance pronounced. Within this range, models with three distinct protrusion configurations demonstrate clear and consistent differences in thermal performance. This highlights the critical role of geometry in regulating heat transfer mechanisms. Notably, variations in the local HTC further corroborate the substantial impact of protrusion geometry. To investigate the underlying mechanisms governing fluid flow and heat transfer, the flow field at $Re = 800$ was selected for detailed analysis. This condition captures the essential inertial-viscous interactions while avoiding the complexities of flow instability prevalent at higher Re .

Figure 16 illustrates the liquid film thickness distribution around the protrusions at a common flow time ($t = 0.91$ s) for $e = 0.5$, 1.0, and 1.5. At $e = 0.5$, significant liquid film accumulation occurs upstream of the protrusions, exhibiting a relatively thick film, while the liquid film between adjacent protrusions is significantly thinner, resulting in a non-uniform distribution over the entire tube surface. As e increases to 1.0, the accumulation diminishes both upstream and downstream, and the film thickness between protrusions increases, leading to a more uniform profile. At $e = 1.5$, liquid film accumulation is minimal, and the distribution is highly uniform around the protrusions, suggesting that this geometry minimizes flow disturbance and promotes greater stability. The corresponding velocity contours in Figure 17 further elucidate the underlying flow mechanisms. For $e = 0.5$, the protrusions induce intense near-wall velocity fluctuations and pronounced flow separation, resulting in a highly non-uniform velocity distribution. Large-scale flow vortices are present upstream of the protrusions. As the aspect ratio e increases, the vortex scale gradually decreases, flow disturbances weaken markedly, and the velocity distribution becomes more uniform. The flow structure is most stable at $e = 1.5$, exhibiting the lowest disturbance intensity. These findings demonstrate that, under the conditions of this study, a smaller e enhances liquid film disturbance, expands the turbulent region, and thus improves heat transfer performance. Conversely, a larger e facilitates a more uniform and stable liquid film and flow field but correspondingly reduces the disturbance intensity and the associated heat transfer enhancement.

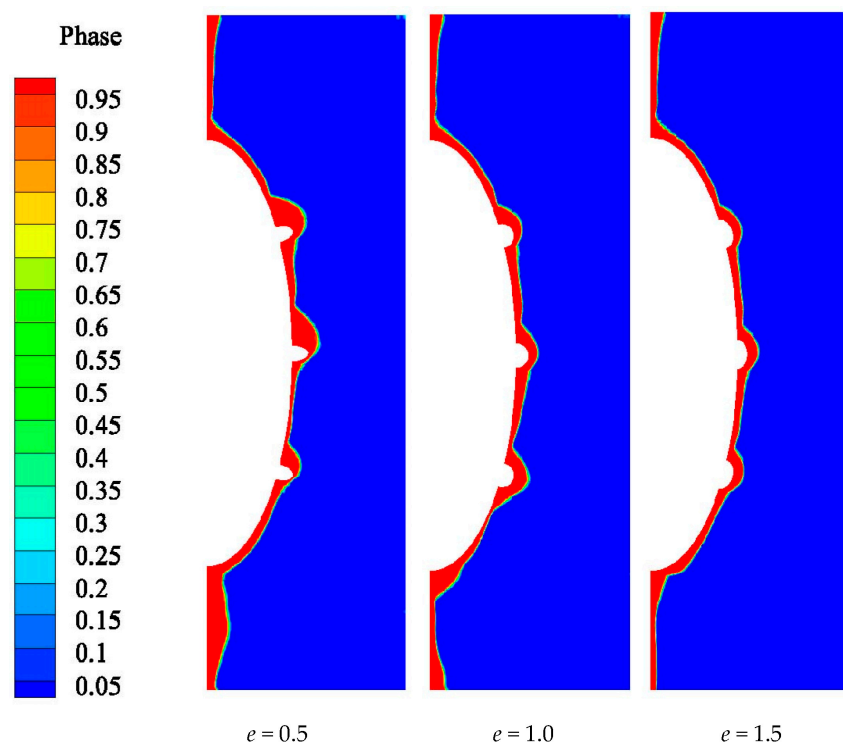


Figure 16. Effect of e on liquid film thickness distribution in enhanced tubes.

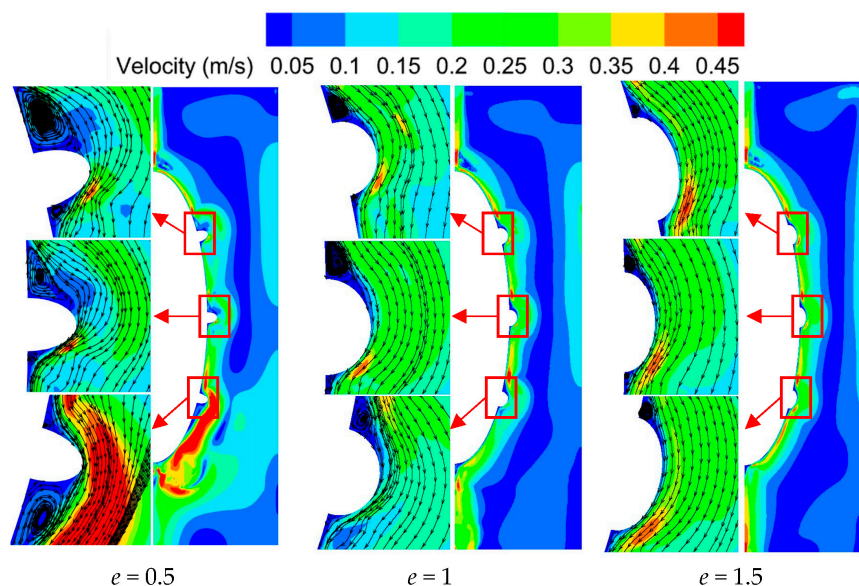


Figure 17. Fluid velocity vectors and vortex structures outside a tube with protrusions at different e .

Figure 18 illustrates the distribution of the local HTC under the corresponding operating conditions. It is evident that all three enhanced elliptical tubes with distinct protrusion geometries exhibit significant fluctuations in the local HTC near $\theta = 30^\circ$, 90° , and 150° . This phenomenon is closely correlated with the flow patterns depicted in the velocity contours of Figure 17, as the peak positions of these variations align precisely with the regions where the protrusions disturb the liquid film.

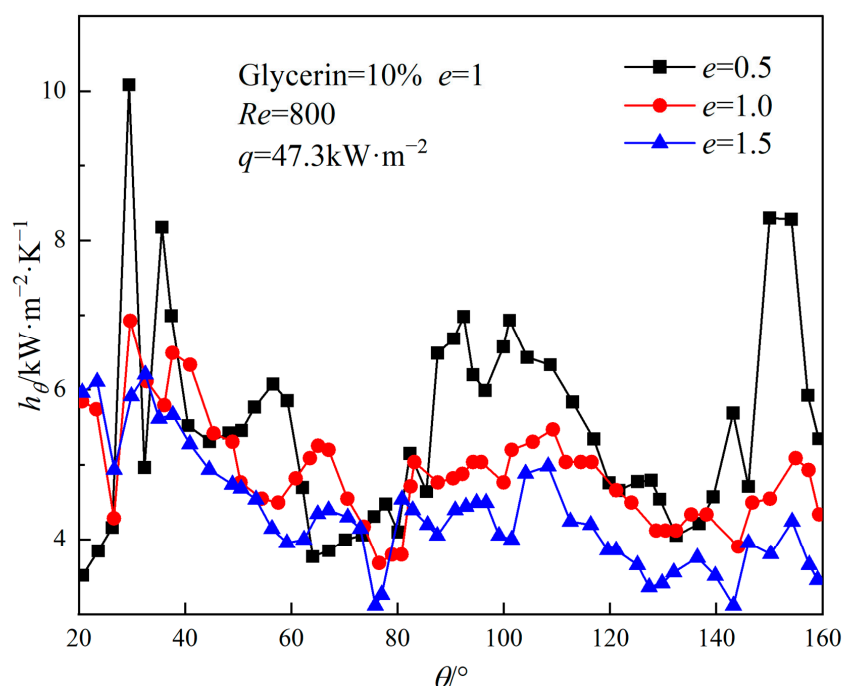


Figure 18. Distribution of Local HTC in enhanced tubes with different e .

Specifically, the protrusion structure with $e = 0.5$ induces the most intense liquid film disturbance due to its larger normal height, leading to more pronounced flow separation and vortex generation. Consequently, this case exhibits the largest amplitude of fluctuations in the local HTC in Figure 18. This demonstrates that the protrusion geometry enhances heat transfer by intensifying turbulent mixing and disrupting the thermal boundary layer, with the intensity of the disturbance directly influencing the magnitude of the HTC variations.

4.4. The Effect of the Number of Protruding Structures on Heat Transfer

Figure 19 compares the liquid film distribution on enhanced tubes with varying numbers of protrusions with that on a smooth elliptical tube, under $\Gamma = 0.262 \text{ kg} \cdot \text{m}^{-1} \cdot \text{s}^{-1}$. The results demonstrate that the number of protrusions significantly influences liquid film stability. Specifically, on Models 01, 04, and 05, the liquid film flow is more stable and exhibits a relatively uniform distribution, indicating good wall wetting. However, as the number of protrusions increases on Models 06, 07, and 08, liquid film instability intensifies markedly, observed as an uneven film distribution with pronounced local accumulation both upstream and downstream of the protrusions. This phenomenon is closely linked to the flow field's turbulent characteristics. As shown by the TKE distribution in Figure 20, an increasing number of protrusions significantly intensifies flow disturbances, elevating the overall TKE levels within the fluid. High TKE is predominantly concentrated around the protrusions and in the near-wall regions, reflecting more intense random fluctuations and energy exchange between fluid parcels. Although this enhanced turbulence can promote mixing, excessive disturbance degrades the liquid film's continuity and stability, leading to film rupture or local thickening, which ultimately diminishes heat transfer efficiency.

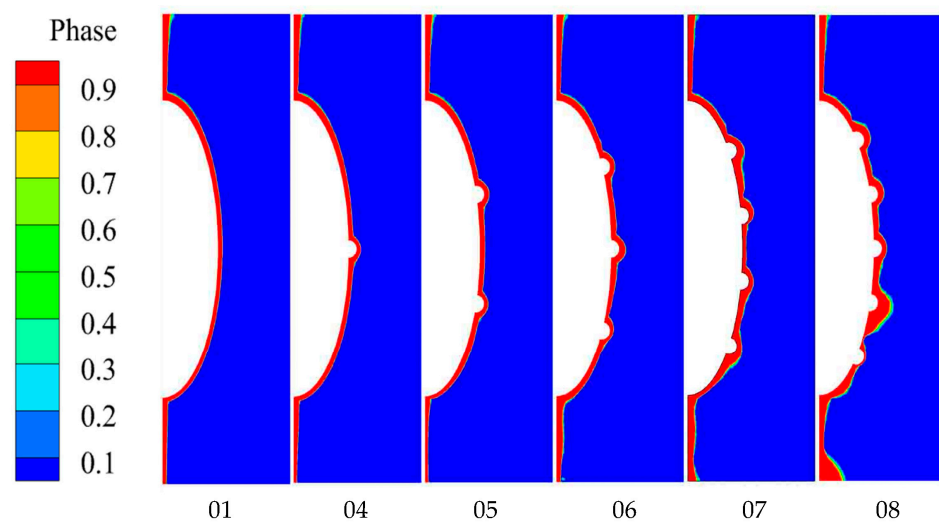


Figure 19. Liquid film thickness distribution on enhanced tubes with varying numbers of protrusions.

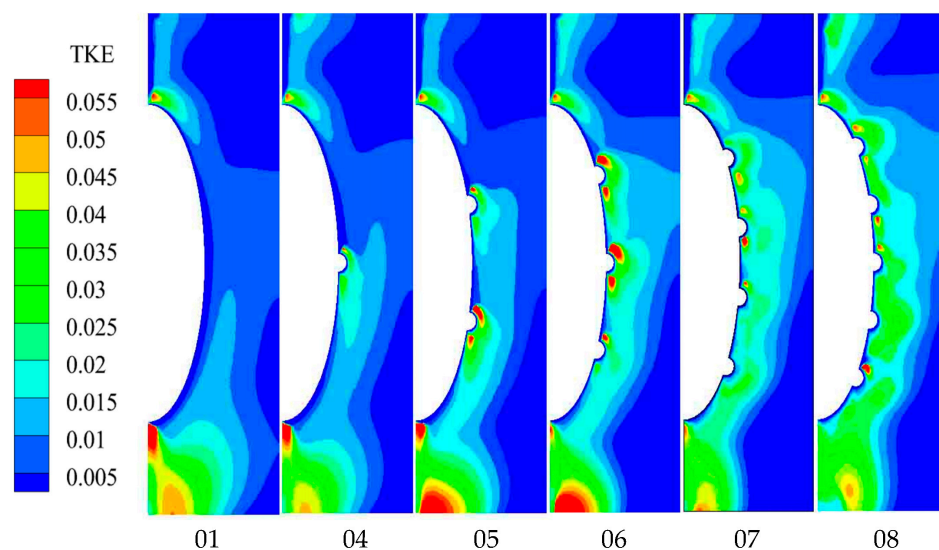


Figure 20. TKE distribution on enhanced tubes with varying numbers of protrusions.

This phenomenon is closely related to the turbulent characteristics of the flow field. As shown in the turbulent kinetic energy distribution in Figure 20, the fluid exhibits elevated turbulent kinetic energy at the inlet of the smooth elliptical tube due to impact effects and further increases at the outlet due to flow separation; along the remainder of the pipe wall, turbulent kinetic energy levels remain relatively low. For the enhanced elliptical tube featuring protrusions, fluid impact upon these structures increases upstream flow velocity, thereby elevating turbulent kinetic energy in this region. Downstream of the protrusions, flow separation induced by the reverse pressure gradient similarly causes TKE to rise. Notably, TKE distributions exhibit marked differences between configurations: in Model 04, high TKE values concentrate in the mid-to-lower regions near the protrusions, with TKE gradually decaying along the wall. Conversely, Model 07 displays localized high-TKE zones both upstream and downstream of the protrusions, exhibiting a more uniform overall distribution. This indicates heightened flow structure disturbance affecting the core flow field. Increasing the number of protrusions significantly intensifies flow disturbances, thereby enhancing overall turbulence intensity within the fluid. High TKE zones predominantly cluster around protrusions and near-wall regions, reflecting more intense random pulsations and energy exchange between fluid eddies in these areas. Whilst heightened turbulence aids mixing, excessive disturbance compromises liquid film continuity and stability, potentially triggering film rupture or localized thickening, thereby diminishing heat transfer efficiency.

Figure 21 presents the average HTC of the enhanced elliptical tubes, with results consistent with the TKE distributions shown in Figure 20. At $\Gamma = 0.262 \text{ kg} \cdot \text{m}^{-1} \cdot \text{s}^{-1}$, the average HTC of the enhanced tubes is at least 10% higher than that of the smooth elliptical tube. As the number of protrusions increases, the average HTC first increases and then decreases. Model 07 exhibits the optimal heat transfer performance.

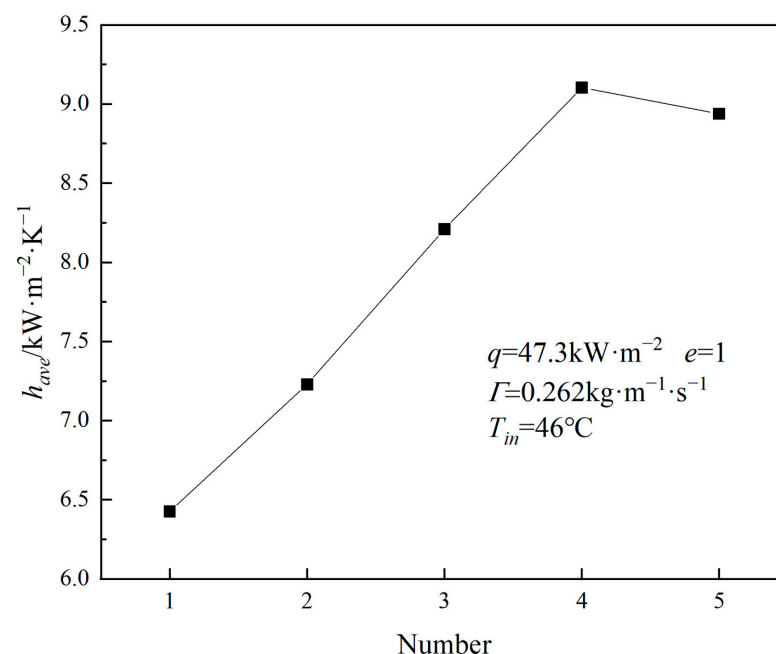


Figure 21. Effect on h_{ave} for different number of protrusions.

This non-monotonic trend can be explained by the flow interaction mechanisms: as the number of protrusions increases, the disturbance to the liquid film intensifies and the high-TKE region expands, promoting heat transfer. However, beyond a critical number of protrusions, excessive disturbance causes severe liquid film fluctuation and loss of continuity, leading to uneven film thickness and ultimately impairing heat transfer performance.

5. Conclusions

This study systematically investigates the influence of protrusions on the flow and heat transfer performance of an external liquid film by developing a two-dimensional enhanced elliptical tube model with protrusions. It examines the effects of three key parameters: the position, distribution, and geometry of the protrusions, on the flow structure and heat transfer characteristics. Through numerical simulations under multiple operational conditions and comparative analysis, the mechanisms by which protrusions disturb the liquid film flow and disrupt the thermal boundary layer are revealed. Based on the results, the following conclusions are drawn:

- (1) In the optimized design of the elliptical tube, the location of the protrusion outside the tube should be taken into account, so that when intervening in the liquid film flow outside the tube at an early stage, the disturbance effect of the protrusion on the liquid film flow can be used to accelerate turbulent mixing inside the liquid film and reduce the thermal resistance; this approach thereby more rapidly and effectively enhances the convective heat transfer intensity on the outer surface of the tube and improves the overall heat transfer performance at an early stage.
- (2) The heat transfer performance of enhanced elliptical tubes is strongly dependent on the number of protrusions on the tube exterior. As the number of protrusions increases, the heat transfer performance exhibits a non-monotonic trend, first increasing and then decreasing. For an ellipticity of $e = 1$, Model 07 exhibits the highest heat transfer performance.
- (3) Research on enhanced elliptical tubes revealed that different external enhancement elements exert divergent effects on heat transfer performance as the Re increases. At $e = 0.5$, the heat transfer coefficient decreases with rising Re . This behavior is primarily due to the more intense flow disturbances induced by the enhancement elements at higher Re , which result in substantially increased flow resistance and energy dissipation, ultimately diminishing the overall thermal efficiency.
- (4) The geometric parameters of protruding structures significantly influence fluid flow and heat transfer enhancement in elliptical tubes. When the perimeter of the protrusions is held constant, their effect on heat transfer is negligible at $Re = 400$. At $Re = 600$ and $Re = 800$, the heat transfer efficiency decreases with increasing ellipticity e . In contrast, under higher- Re conditions, the elliptical tube with $e = 1$ exhibits optimal heat transfer performance.

The findings from this two-dimensional model, which assumes an ideal liquid film distribution, represent the theoretical optimum for heat transfer performance. Any practical three-dimensional flow phenomena, most notably flow channeling, would lead to a deterioration in performance. Thus, the results herein provide a conservative reference for engineers, defining a performance ceiling that real-world systems are unlikely to exceed.

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Nomenclature

Greek characters

a	semi-major axis of an elliptical rough structure, mm	θ	Circumferential angle, °
b	semi-minor axis of an elliptical rough structure, mm	ρ_l	liquid phase density, $\text{kg}\cdot\text{m}^{-3}$
g	gravity acceleration, $\text{m}\cdot\text{s}^{-2}$	ρ_g	gas phase density, $\text{kg}\cdot\text{m}^{-3}$
h_θ	local heat transfer coefficient, $\text{kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$	μ_l	liquid phase dynamic viscosity, Pa·s
h_{ave}	average heat transfer coefficient, $\text{kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$	μ_g	gas phase dynamic viscosity, Pa·s
Re	Reynolds number	Γ	liquid film flow per unit length on one side of tube wall, $\text{kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$
q	heat flux, $\text{kW}\cdot\text{m}^{-2}$	δ	liquid film thickness, mm
T_{in}	liquid inlet temperature, K	σ	surface tension coefficient
H	liquid film distributor height, m	κ	free surface curvature, m^{-1}
P	pressure, Pa	β	The protrusions positions correspond to the angles,
t	times, s		
$\vec{F}$	surface tension, N		
e	Length–width ratio, $e = b/a$		
$T_{w,\theta}$	Temperature of the tube wall at the circumferential angle θ		
T_θ	Temperature of the falling film at the circumferential angle θ		

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