

Design and Heat Transfer Analysis of a Novel Micro Heat Pipe Based on Inner Wall Rib Structure

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Abstract: With the increasing power density of electronic and industrial devices, efficient heat dissipation technology has become especially important. This is particularly true with the growing use of high-power LED lights in new energy vehicles, where compact and efficient thermal management solutions are crucial for the performance and longevity of these devices. In this study, a micro heat pipe structure with efficient heat transfer, simple structure, light weight, and small size is designed, making it particularly suitable for applications in confined spaces. By adding annular micro-ribs to the internal side walls of the evaporator and condenser sections of the phase-change micro heat pipe, it was found that, compared to the ribless design, the ribs increased fluid disturbances, promoted a more effective phase-change heat transfer process, and significantly improved the overall heat dissipation efficiency. Through fluid simulation testing of the ribs with different heights, widths, and spacings, it was found that smaller rib spacing significantly enhanced the heat transfer efficiency in the condenser section, while the height and width of the ribs, although complex in their effect on the evaporator section, were also key factors in improving heat transfer performance. Experimental results also showed that under high power conditions, the temperature in the condenser section of the ribbed heat pipe was significantly lower than that of the ribless heat pipe, with a temperature difference exceeding 20°C, confirming the significant enhancement of heat transfer performance due to the ribs. This study provides new insights and a technical foundation for optimizing heat pipe heat dissipation performance, especially in thermal applications for high-power devices.

Keywords: Micro heat pipe, Ribs, Heat transfer enhancement, Computational fluid dynamics

1. Introduction

1.1. Research Background

With the rapid development of modern technology, the application of high-power electronics and industrial equipment has become increasingly widespread. The rising power density has made thermal management a critical challenge, directly influencing system stability, performance, and lifespan. Without effective heat dissipation, equipment may suffer from overheating, performance degradation, or even irreversible damage. Therefore, optimizing thermal management for high-power systems is crucial in their design and manufacturing processes.

In space-constrained applications, traditional cooling methods, such as fans, heat sinks, or liquid cooling systems, often face limitations in size and efficiency. Micro heat pipes have emerged as a compact and efficient thermal component for addressing these challenges. By leveraging the principle of phase-change heat transfer, micro heat pipes can efficiently transport heat within confined spaces, helping maintain operational temperatures and extending equipment lifespan.

Thanks to their lightweight, compact size, and noiseless operation, micro heat pipes are highly effective for cooling high-power, small-scale devices. Typical applications include high-density servers, power amplifiers, and power electronics in electric vehicles. These scenarios demand high thermal performance in limited spaces, making micro heat pipes an ideal choice for modern thermal management needs.

1.2. Current Research Status

To address various thermal challenges, researchers have extensively studied the applications of heat pipe technology. A typical heat pipe consists of three sections: an evaporator, an adiabatic section, and a condenser. Heat is absorbed by the working fluid in the evaporator, where it vaporizes. The vapor travels to the condenser, releases heat, and condenses back to liquid, which then returns to the evaporator via capillary or gravitational forces. This cycle efficiently transfers heat from the source to the sink, ensuring stable operation. Compared to traditional cooling methods, heat pipes offer simplicity, compactness, and lightweight construction, making them suitable for complex and space-constrained environments.

Although heat pipes demonstrate excellent thermal performance, increasing demands for higher efficiency and stricter working conditions necessitate further optimization. Researchers have explored various approaches to improve heat pipe performance, such as geometric modifications to design annular or flattened heat pipes that extend heat transfer paths and adapt to diverse applications. Additionally, advanced working fluids like nanofluids or novel refrigerants have been introduced to enhance phase-change heat transfer. Furthermore, internal microstructures or superhydrophobic coatings have been added to improve liquid return and evaporation processes. These strategies have been evaluated using computational fluid dynamics (CFD) models to simulate two-phase flow and heat transfer under varying conditions, enabling optimized designs validated by experiments. The combined efforts of numerical simulations and experimental validation have significantly enhanced heat pipe performance [1-3].

Recent research has focused on applying optimized heat pipe designs to high-power devices. For example, researchers have developed specialized heat pipe structures tailored for high-power systems and verified their performance under diverse conditions. Compact heat pipe clusters have also been designed to reduce junction temperatures in confined spaces. Advanced packaging and interconnection strategies have further improved cooling performance and space utilization, significantly outperforming traditional cooling modules [4-7].

Ribs, as common heat exchange components, can effectively enhance fluid disturbance and improve heat transfer. Extensive studies have investigated the effects of rib shapes, arrangements, and geometric parameters on thermal performance. Research ranges from conventional circular and square ribs to optimized streamlined and trapezoidal designs, as well as staggered and aligned arrangements. While rib-based optimization is well-established, their application to phase-change heat pipes remains underexplored. Given the demand for compact designs in practical scenarios, the potential of ribs in enhancing phase-change heat pipe performance warrants further investigation [8-12].

1.3. Objective of This Study

In conclusion, while rib optimization has been widely studied, its application to phase-change heat pipes, aiming to enhance thermal performance without increasing space requirements, remains underexplored. This study investigates the feasibility of incorporating annular ribs on the internal walls of the evaporator and condenser sections to improve the thermal performance of heat pipes. The proposed design achieves significant heat transfer enhancements without altering the overall structure or increasing space consumption, meeting the demands for compact and efficient cooling solutions in high-power devices.

2. Numerical Simulation Methodology

2.1. Fluid Dynamics Governing Equations

2.1.1. Conservation Equations

To simulate the fluid flow inside the heat pipe, the changes in mass, stress, and energy of each fluid element need to be considered. Since this study focuses on phase-change heat pipes, the interphase interactions are represented using a volume fraction a_k , which ranges from 0 to 1 and indicates the fraction of volume occupied by each phase. To track the position and shape of the fluid interface, this study employs the standard Volume of Fluid (VOF) model, which captures changes in the volume fraction over time and with the fluid velocity field u . The transport equation for the volume fraction is expressed as:

$$\frac{\partial a_k}{\partial t} + u \cdot \nabla a_k = 0$$

In addition, this study incorporates mass and energy source terms from the Lee model to account for phase change processes. The mass source term S_M represents the rate of mass change per unit volume per unit time during the phase change. It is calculated as: $S_M = k(T_s - T_e)^n * \rho$, where T_s is the saturation temperature, T_e is the ambient temperature, and k and n are parameters related to the material properties and heat pipe design. Specifically, k is the heat transfer coefficient, and n reflects the nonlinear influence of temperature differences. Together, these parameters represent the phase-change rate, and multiplying them by the density ρ of the phase-change material yields the source term. With the inclusion of the mass source term, the continuity equation representing mass conservation becomes:

$$\frac{\partial(a_k \rho_k)}{\partial t} + \nabla \cdot (a_k \rho_k u) = S_M$$

For momentum conservation within a unit volume, the Navier-Stokes equation is employed to describe the velocity and pressure fields for each phase. When considering interphase interactions, the momentum equation can be expressed as:

$$\frac{\partial(a_k \rho_k u_k)}{\partial t} + \nabla \cdot (a_k \rho_k u_k u_k) = -a_k \nabla p + a_k \rho_k g + \nabla \cdot (a_k \tau_k) + M_k^{kl}$$

Here, u_k represents the velocity vector of phase k , τ_k denotes the stress tensor within phase k , and M_k^{kl} describes the interaction forces between phase k and other phases l . These terms collectively govern the momentum changes in the two-phase flow.

For energy conservation, the energy changes due to phase transitions are similarly represented by the energy source term S_E introduced in the Lee model: $S_E = k(T_s - T_e)^n \left(h + \frac{1}{2} c_p (T_s - T_e) \right)$. In this equation, h denotes the latent heat of phase change, and c_p is the specific heat capacity. Incorporating the energy source term, the energy conservation equation becomes:

$$\frac{\partial}{\partial t}(a_k \rho_k e_k) + \nabla \cdot (a_k \rho_k e_k u) = -\nabla \cdot (a_k q_k) + a_k \Phi_k + a_k \rho_k \dot{q}_k + S_E$$

On the left-hand side, the equation represents the rate of energy change per unit volume over time and energy transport due to convection. The right-hand side includes heat conduction (via the heat flux vector q , viscous dissipation Φ , and external heat sources $\rho \dot{q}$.

2.1.2. Turbulence Model

Given the focus on heat pipe cooling under high power input, where high Reynolds numbers often occur, the flow within the heat pipe is assumed to be turbulent. In turbulent flow, fluid velocity and other physical quantities (such as pressure, density) fluctuate dramatically over time and space. To simplify this complex flow behavior, the Reynolds Averaged Navier-Stokes (RANS) method is typically used to handle turbulence. This method decomposes instantaneous flow variables into time-averaged and fluctuating components, namely: $u_i = \bar{u}_i + u_i'$

Using this averaging approach, the conservation equations for mass and momentum can be described by RANS decomposition:

$$\begin{aligned} \frac{\partial \bar{\rho}}{\partial t} + \frac{\partial(\bar{\rho} \bar{u}_i)}{\partial x_i} &= 0 \\ \frac{\partial(\bar{\rho} \bar{u}_i)}{\partial t} + \frac{\partial(\bar{\rho} \bar{u}_i \bar{u}_j)}{\partial x_j} &= -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\bar{\tau}_{ij} - \bar{\rho} u_i' u_j' \right] + \bar{\rho} g_i \end{aligned}$$

Here, $-\bar{\rho} u_i' u_j'$ is the Reynolds stress term calculated by the turbulence model. To detailedly describe turbulence characteristics, many classical turbulence models are widely applied. The simulation in this study uses the SST $k - \omega$ model, which combines the $k - \omega$ and $k - \epsilon$ models, excelling in handling near-wall flow and providing good predictions in the far-field region. The transport equations for turbulent kinetic energy k and turbulence dissipation rates ϵ and ω used in the model are as follows:

$$\begin{aligned} \frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_i)}{\partial x_i} &= P_k - \beta^* \rho k \omega + \frac{\partial}{\partial x_j} \left[(\mu + \sigma_k \mu_t) \frac{\partial k}{\partial x_j} \right] \\ \frac{\partial(\rho \epsilon)}{\partial t} + \frac{\partial(\rho \epsilon u_i)}{\partial x_i} &= C_1 \frac{\epsilon}{k} P_k - C_2 \rho \frac{\epsilon^2}{k} + \frac{\partial}{\partial x_j} \left[(\mu + \sigma_\epsilon \mu_t) \frac{\partial \epsilon}{\partial x_j} \right] \\ \frac{\partial(\rho \omega)}{\partial t} + \frac{\partial(\rho \omega u_i)}{\partial x_i} &= \alpha \frac{k}{\omega} P_k - \beta \rho \omega^2 + \frac{\partial}{\partial x_j} \left[(\mu + \sigma_\omega \mu_t) \frac{\partial \omega}{\partial x_j} \right] + 2(I - F_1) \rho \sigma_\omega^2 \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \end{aligned}$$

The meanings of the parameters in these transport equations are as follows: P_k represents the turbulence production term, indicating turbulent kinetic energy generation due to mean velocity gradients; μ_t is the turbulent viscosity coefficient, describing the contribution of turbulence to momentum transfer; α controls the relationship between turbulent energy and specific dissipation rate ω in the turbulence dissipation rate equation; β and β^* regulate the dissipation rates of turbulent kinetic energy and turbulent energy, respectively; σ_k and σ_ω are the inverse Prandtl numbers, affecting the diffusion of turbulent kinetic energy k and specific dissipation rate ω ; σ_ϵ is the inverse Prandtl number affecting the diffusion of turbulence dissipation rate ϵ ; F_1 is a blending function, used to switch model characteristics between near-wall and far-field regions, ensuring the use of $k - \omega$ model characteristics near the wall and $k - \epsilon$ model characteristics in the far-field.

2.2. Simulation Setup and Case Description

2.2.1. Computational Geometry

To reduce computational cost while accurately reproducing the two-phase flow and heat transfer phenomena in a 3D copper heat pipe, an axisymmetric 2D model was employed. The heat pipe consists of three sections: an evaporator (100 mm), an adiabatic section (150 mm), and a condenser (100 mm), with a radius of 5 mm. Gravity is set toward the evaporator to utilize gravity for liquid return.

To systematically evaluate the effects of rib geometrical parameters on heat pipe performance, three parameters were selected: rib spacing (S), rib height (h), and rib width (w). three parameters assigned with three different values, resulting in a total of 27 simulation cases via full factorial design. Additionally, a ribless heat pipe was simulated under identical conditions as a baseline for performance comparison.

Table 1: Geometrical Parameters.

Parameter	Values (mm)
Rib spacing(S)	2/5/10
Rib height(h)	0.2/0.5/1
Rib width(w)	0.5/1/2

2.2.2. Boundary Conditions

To simulate the heat transfer under high-power conditions, a constant heat flux of $100,000 \text{ W/m}^2$ was applied in the baseline case (ribless heat pipe), representing typical operational conditions for high-power systems. For ribbed designs, the heat flux was adjusted proportionally to the ratio of the internal surface area of each case to the ribless design, ensuring a consistent total heat input across cases.

The condenser sidewall was set at a constant temperature of 298.15 K (25°C) to represent ambient conditions. The adiabatic section and the heat pipe bottom were insulated with no heat flux. The heat pipe was filled with water and vapor, with the evaporator initially filled entirely with liquid water and the remaining sections containing vapor at a pressure of 1000 Pa, approximating a vacuum state. The working fluid was treated as incompressible, and no-slip boundary conditions were applied to all walls.

To ensure accuracy, mesh independence tests were conducted and the grid size of 0.0001 mm was selected since this size provided results comparable to finer grids while maintaining computational efficiency.

2.2.3. Numerical Methods

The simulations employed a transient, double-precision solver to ensure high accuracy. A time step of $5\text{e-}04$ was chosen to capture transient flow behaviors effectively. For precise interface tracking, the Volume of Fluid (VOF) model was used with a cutoff value of $1\text{e-}08$, and the sharp/dispersed reconstruction method was applied, where "sharp" was used for distinct interfaces and "dispersed" for diffuse ones.

Low Reynolds number corrections and production limiters were activated in the turbulence model. These settings enhanced accuracy in near-wall regions and prevented excessive turbulence energy generation.

The pressure-velocity coupling was handled using the Pressure-Implicit with Splitting of Operators (PISO) scheme, which improves convergence speed through multiple corrections per time step. Discretization was performed using a bounded second-order implicit method to balance accuracy and stability. Each time step was iterated up to 20 times, with residuals converging below $1e-04$ for flow variables and $1e-06$ for energy.

3. Numerical Simulation Results and Analysis

3.1. Unsteady Heat Pipe Flow and Heat Transfer Process

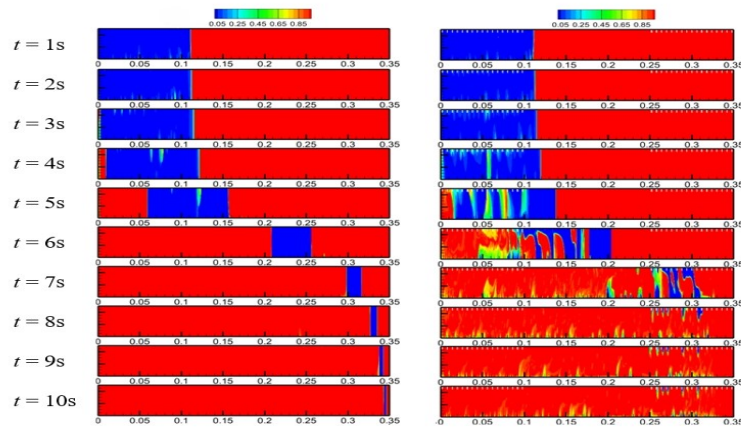


Figure 1: Vapor volume fraction distribution with 1-second intervals. The left side is normal heat pipe and the right side is heat pipe with intermediate rib parameters ($W = 1\text{mm}$, $S = 5\text{mm}$, $h = 0.5\text{mm}$)

As shown in Figure 1, there are significant differences in the distribution of the gas phase volume fraction between the ribbed and ribless heat pipes. In the ribless heat pipe (left side), the gas phase extends evenly from the bottom to the top, with a smooth gas-liquid interface and stable flow and expansion. Due to the continuous heat input, this stable heat transfer process makes it difficult for the vapor to effectively condense, leading to a significant reduction in liquid water return and hindering the circulation process.

In contrast, in the ribbed heat pipe (right side), the gas phase distribution is more irregular. The ribs introduce significant disturbances in the expansion process, causing large fluctuations and instability in the gas-liquid interface. After 6 seconds, it becomes evident that this complex flow structure leads to significant variations in the gas phase volume fraction across different regions, with alternating areas of concentrated and sparse gas phase. This complex flow structure allows the condenser to begin transferring heat earlier, improving the heat transfer efficiency and allowing for proper liquid return.

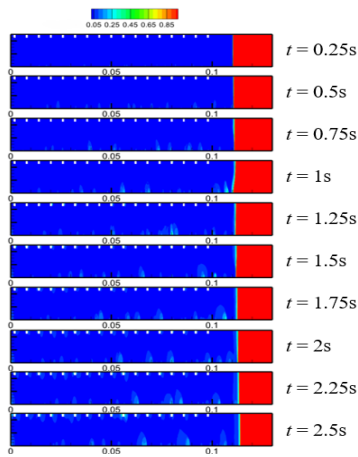


Figure 2: Vapor volume fraction in the evaporator of the ribbed pipe

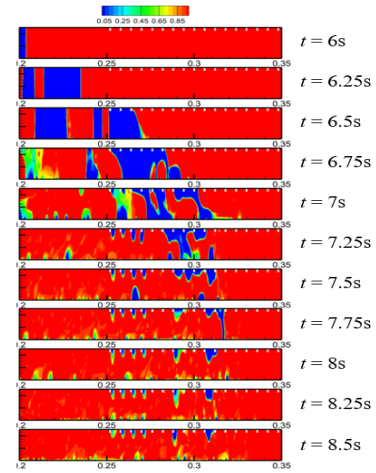


Figure 3: Vapor volume fraction in the condenser of the ribbed pipe

Since the operating processes of different cases are similar, the analysis in this study focuses on the ribbed heat pipe with intermediate parameter values, as shown in Figures 2 and 3.

Figure 2 shows the behavior in the evaporator and part of the adiabatic section during the first 2.5 seconds of operation. The bubble formation rate is slow in the early stages, and bubbles are only observable when they move to the center and merge into larger bubbles. After about one second, as the temperature rises, small bubbles start forming at the interface between the ribs and the wall, with some detaching. These bubbles then converge in the center of the heat pipe and gradually move toward the condenser, eventually detaching from the liquid water surface.

Figure 3 shows the behavior in the condenser section between 6s and 8.5s. In the first half-second, a liquid column is seen rising from the hot end to the condenser. Subsequently, the vapor in the liquid column starts to condense, and the vapor that penetrated the column and reached the condenser also begins to liquefy and return, eventually breaking the liquid column. After that, some of the condensed liquid adheres to the wall and forms a liquid film, while the remaining portion returns in the form of droplets from the center region.

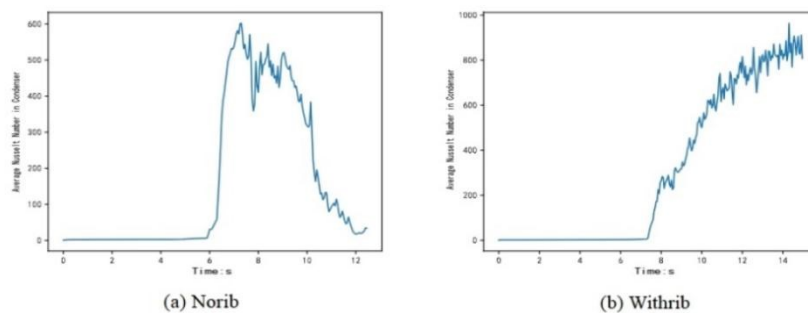


Figure 4: Nusselt number in the condenser section

Figure 4 illustrates the variation of the Nusselt number in the condenser section over time for both cases. The Nusselt number is a dimensionless parameter used to describe the enhancement of convective heat transfer relative to conductive heat transfer. It is expressed as $Nu = \frac{hL}{k}$, where h represents the convective heat transfer coefficient in the condenser section, L is the characteristic

length (typically the length of the condenser section or the thickness of the liquid film), and k is the thermal conductivity of water. A higher Nusselt number indicates greater heat transfer efficiency in the condenser section. During the early stages of operation, when the vapor has not yet reached the condenser, the Nusselt number is close to zero. Afterward, both data sets show an increasing trend. In the ribless case, the steam is blocked by the liquid column, causing more steam to accumulate and reach the condenser quickly, resulting in a significant increase in the Nusselt number starting at 6s. The ribbed case shows a noticeable increase in condensation heat transfer at around 7.5s. However, the ribless case experiences a dry-out phenomenon after 10s, with the Nusselt number dropping from a peak of over 500 to less than 100. In contrast, the ribbed case exhibits a more stable increase with minor fluctuations, reaching around 900 at 14s.

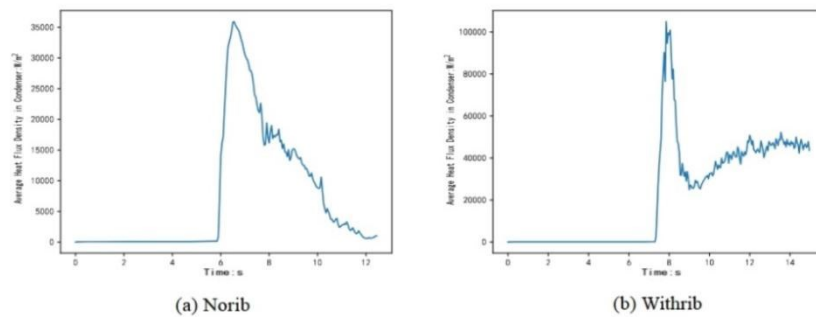


Figure 5: Heat flux density in the condenser section

Figure 5 further reveals the time points at which the vapor pushes the liquid column to the condenser. At this point, a large amount of vapor contacts the condenser wall and exchanges heat with the liquid column and the liquid film left by the vapor that passed through it, leading to a significant instantaneous heat transfer. However, in this phase, the ribless case experiences local overheating, with a heat flux density of only 35,000 W/m^2 , while the ribbed case reaches 100,000 W/m^2 . Subsequently, the heat flux density decreases significantly. In the ribbed case, the heat flux density stabilizes at around 45,000 W/m^2 , while in the ribless case, the heat flux density drops drastically to less than 5,000 W/m^2 due to overheating.

The thermal resistance of the heat pipe is calculated by the temperature difference between the evaporator and condenser sections divided by the input power at the evaporator, and it serves as a key parameter for evaluating heat transfer efficiency. A higher thermal resistance indicates lower heat transfer efficiency. By calculating the thermal resistance of both heat pipe structures under steady-state conditions, it was found that the thermal resistance of the ribless structure is 15.9, significantly higher than the ribbed structure's 9.2, indicating that the ribbed design performs better in terms of heat transfer efficiency.

3.2. Influence of Rib Geometrical Parameters on Heat Pipe Heat Transfer Performance

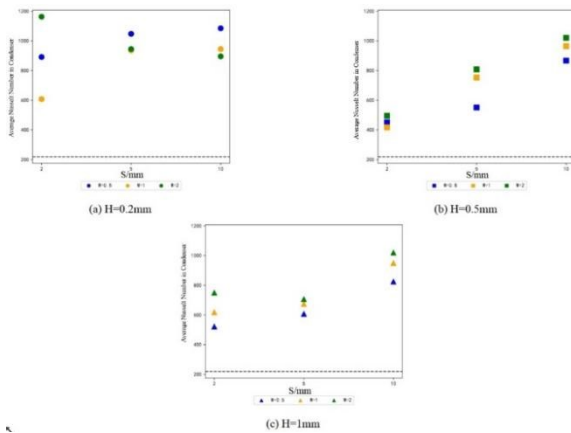


Figure 6: Nusselt number in the condenser section for different cases at steady state

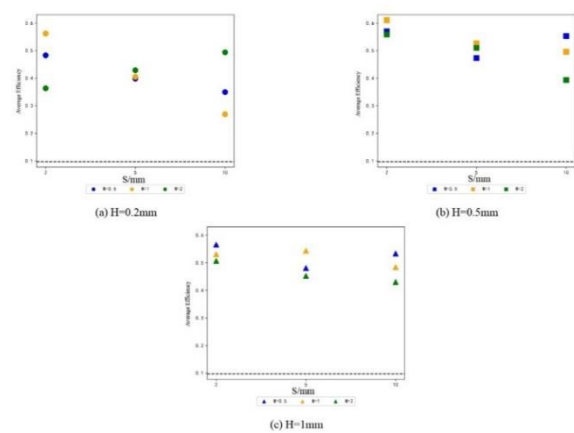


Figure 7: Overall heat transfer efficiency for different cases at steady state

As shown in Figure 6a, when the rib height h is small (0.2 mm), for the cases with $w=0.5$ and $W=1$, the Nusselt number increases with the spacing S , but the growth rate slows down after $S>5$ mm. In contrast, the case with $W=2$ exhibits the opposite trend, with the Nusselt number first decreasing rapidly and then slowing down. However, the Nusselt number in this case is generally higher, with values around 1000, except for the case where $S=2$ and $w=1$, which just exceeds 600.

When the rib height is $h=0.5$ mm (as shown in Figure 6b), for all different values of w , the Nusselt number increases with the spacing S : for $S=2$, the Nusselt numbers of the three cases are between 400 and 600, and for $S=10$, they range from 800 to 1000. However, the case with $w=0.5$ shows a slow start and faster growth afterward, while the other two cases increase almost at a uniform rate.

In Figure 6c, where the rib height is $h=1$ mm, the Nusselt numbers for all cases show a slow-to-fast increase with the spacing, from 500-700 at $S=2$ to 800-1000 at $S=10$. However, for the case with $w=2$, there is a slight decrease in values between $S=2$ and $S=5$, but it increases significantly when $S=10$.

The case with $w=2$ consistently shows the highest Nusselt number, followed by the case with $w=1$. The case with $w=0.5$, although having lower overall values, shows a Nusselt number that is 50-100 higher than the other two values when the rib height is $h=0.2$ mm.

The Nusselt numbers for all 27 cases in general show distinct performance (with a minimum just above 400 and a maximum near 1200), indicating that rib geometrical parameters play a critical role in introducing disturbance, which affects the heat transfer performance. The ribless case (shown by the dashed line) has a significantly lower value due to the dry-out condition under high power, with the Nusselt number just slightly above 200, which shows the great effect that ribs create.

The heat pipe's heat efficiency η measures the performance in transferring heat. The specific value is the ratio of the heat transfer in the condenser section to the heat transfer in the evaporator section. The heat transfer is calculated by multiplying the convective heat transfer coefficient, effective heat transfer area, and the temperature difference between the surface and the medium.

As shown in Figure 7a, when the rib height is small, the heat efficiency shows varying trends with increasing spacing S . For the cases with $w=0.5$ and $w=1$, the heat transfer decreases, while for the case with $w=2$, the trend is reversed, and the heat transfer increases.

In Figure 7b, when the rib height is moderate ($h = 0.5$ mm), for all values of w , the heat transfer at $S=2$ is relatively high, around 0.6, and then slightly decreases as the spacing increases. Notably, for

$w=0.5$, the efficiency increases again at $S=10$ to around 0.55, while for the other two cases, it decreases to approximately 0.4–0.5.

In Figure 7c, when the rib height is high ($h = 1 \text{ mm}$), the heat transfer for all three w cases is still above 0.5 at $S=2$, and then it only experiences minor fluctuations as the spacing increases, with values dropping by only 0.05 or less.

Overall, the variation in w does not lead to significant numerical differences. When $h=0.2\text{mm}$, the trends are more erratic, and the numerical differences are larger, resulting in a minimum efficiency as low as 0.3. However, for $h=0.5\text{mm}$ and $h=1\text{mm}$, the trends are more consistent, and the heat efficiency generally remains around 0.5. Interestingly, at these rib heights, the Nusselt number and heat efficiency exhibit slightly opposite trends: the Nusselt number increases with S , but the heat efficiency decreases. This suggests that while the convective heat transfer capability in the condenser improves with increasing spacing, the actual heat transfer amount becomes lower. The ribless case shows a significantly lower heat efficiency (only 0.1), which further emphasizes the importance of ribs in enhancing the heat transfer performance of the heat pipe.

3.3. Flow Mechanisms of Rib-Enhanced Heat Transfer

To better understand the factors influencing the heat transfer performance of ribs, the following content proposes hypotheses based on the quantitative analysis from the previous section, attempting to explain the underlying mechanisms.

First, the effect of rib width on heat transfer is minimal, and the reason is clear: fluid disturbances mainly occur when the fluid strikes or passes over the ribs, and rib width does not significantly affect the disturbance pattern. In contrast, the width of the ribs is more related to the total amount of obstacles encountered by the fluid and the flow passage area occupied by the ribs, but it does not have a substantial impact on heat exchange performance. Rib width changes are more likely to affect the space occupation ratio of the ribs and the mechanical strength of the heat pipe.

When the rib height is small, there is no obvious trend in the enhanced heat transfer effect, and the reason for the poor performance is likely due to the limited disturbance effect. Although the ribs can induce turbulence, the effect merely prevents the dry-out phenomenon that occurs without ribs, but it does not significantly enhance heat transfer performance.

The contradictory trends in Nusselt number and heat efficiency indicate that, despite better overall heat transfer with the surface, efficient heat exchange with the external medium is not achieved. This phenomenon could be explained by local overheating: when the rib spacing is large, the fluid flows smoothly, maintaining uniform convective heat transfer, and the larger flat surface at the bottom causes phase separation and heat accumulation. These effects result in a higher Nusselt number, but they limit the effective use of latent heat during phase change, causing more heat to accumulate or directly conduct to the prominent part of the ribs rather than dissipating efficiently through the rib sides or the ribless area.

To validate this hypothesis, the study performed velocity streamline analysis on cases with rib spacings of 2mm and 10mm under steady-state conditions, with rib height and width set to their middle values.

From Figures 8 and 9, it can be seen that a larger rib spacing indeed reduces flow resistance by providing appropriate disturbance, and the flow is not that chaotic as that caused by frequent disturbances at smaller rib spacing, providing stabler overall convective heat transfer performance. However, the small eddies generated at smaller spacings lead to stronger fluid mixing since the high power input make fluid pushes other more frequently, which improves overall heat transfer efficiency.



Figure 8: Velocity image for $S=2\text{mm}$

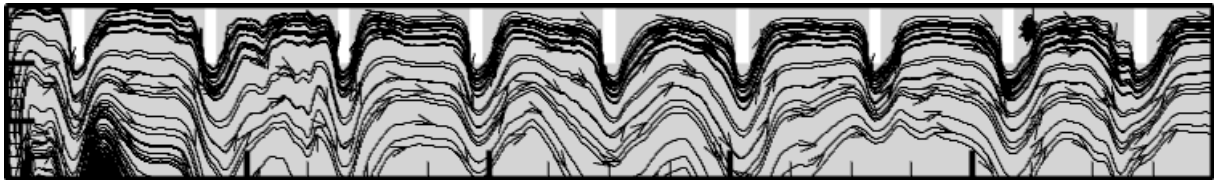


Figure 9: Velocity image for $S=10\text{mm}$

4. Experiment evaluation



Figure 10: Actual Pipe

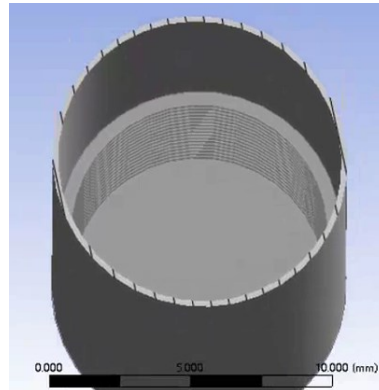


Figure 11: Cross-section of the ribbed heat pipe

To verify the impact of ribs on the heat dissipation performance under actual high-power conditions, this study conducted experiments comparing the temperatures of ribbed and ribless heat pipes. Both heat pipes had the same specifications: an inner diameter of 10 mm, a length of 350 mm, and a wall thickness of 2 mm. The ribbed heat pipe was created by adding annular ribs to the ribless heat pipe, with rib spacing of 5 mm, a height of 0.5 mm, and a width of 1 mm. The heat pipes were filled with water to match the volume fraction used in the simulations, with the water level around 10 mm, and were deployed after vacuuming.

The heat pipe was heated using a heating tube with a heat flux input capacity of $30,000 \text{ W/m}^2$, with the highest temperature during the experiment controlled at 250°C . Under an ambient temperature of 26.8°C , the experiment recorded temperatures at multiple points in the condenser section every 10 seconds. Three repeated experiments were conducted for both the ribless and ribbed heat pipes, with each experiment lasting for 12 minutes. Finally, the data was averaged and analyzed, resulting in the following temperature variation trends:

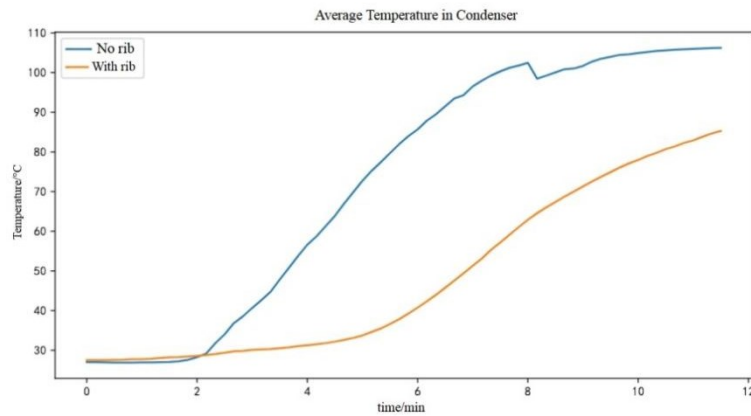


Figure 12: Variation of average condenser section temperature over time

As shown in Figure 12, starting from the same initial temperature, the ribbed heat pipe began transferring heat to the condenser section earlier due to disturbances caused by the ribs, and it maintained a more stable temperature rise by continuously removing heat at a higher efficiency. In the end, both heat pipes reached a stable temperature, with the average condenser section temperature of the ribbed heat pipe being over 20°C lower than that of the ribless heat pipe. This demonstrates that ribs significantly enhance the heat dissipation performance of the heat pipe.

5. Conclusion

This study has developed an innovative micro heat pipe with built-in micro-ribs to significantly enhance its heat transfer performance. By constructing a detailed numerical simulation model of the phase-change micro heat pipe, this research systematically analyzes the effects of the geometrical parameters of the micro-ribs, including height, width, and spacing, on the flow characteristics and thermal efficiency during the phase-change process. The study reveals that the micro-rib design effectively increases fluid disturbances in both the evaporator and condenser sections, thereby promoting a more efficient phase-change heat transfer process and significantly improving overall heat dissipation efficiency.

Specifically, it was found that smaller rib spacing could significantly improve the heat transfer efficiency in the condenser section. By optimizing the layout of the micro-ribs, the heat exchange in the condenser process can be effectively enhanced. In the evaporator section, the influence of the rib height and width on heat transfer performance is more complex and requires a comprehensive consideration to achieve the best effect. Notably, in the evaporator section, the micro-ribs enhance fluid disturbance and mixing, improving the distribution and transfer of heat.

To verify the accuracy and practical effectiveness of the numerical simulation results, physical micro heat pipes were fabricated and a series of experiments were conducted. Experimental data show that the micro heat pipe with built-in micro-ribs demonstrated significant heat transfer enhancement in practical applications. Compared to the traditional ribless design, its heat dissipation efficiency was significantly improved. This result not only validates the theoretical advantages of the micro-rib design but also demonstrates its immense potential in practical applications, providing a new solution for the development of efficient heat pipe technologies.

Overall, through systematic numerical simulations and experimental validation, this study demonstrates the innovative application of micro-ribs in micro heat pipe design and highlights their importance in enhancing heat transfer performance. These research findings provide valuable references for the design and optimization of future efficient thermal management systems and have broad application prospects.

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