

Article

Visualization Investigation of Heat Transfer Behavior in a Flat-Tube Shaped Heat Pipe

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Abstract: Unveiling the heat transfer behavior of solar collectors in concentrating solar thermochemical energy storage is crucial for harnessing full-spectrum solar light. In this study, a glass Flat Tube-Shaped Heat Pipe (FT-SHP) was developed, and a visualization experimental platform was established to investigate its internal operation mechanisms and heat transfer characteristics. The results revealed that the liquid filling ratio (FR) significantly affects the heat transfer performance, with an optimal value identified as 25%. As the heat flow temperature in the evaporation section increased, both the Bubble Growing Frequency (BGF) and Droplet Condensation Reflux Period (DCRP) decreased, leading to a reduction in thermal resistance. Conversely, an increase in the cooling flow rate resulted in opposite trends in BGF and DCRP within the tube, while both the Reynolds (Re) number and thermal resistance decreased. As such, an empirical correlation between thermal resistance and Re number was derived, demonstrating a nonlinear relationship between thermal resistance, BGF, and DCRP. These findings provide important insights for the design of heat pipes, with the potential to enhance the efficiency and reliability of solar collectors.

Keywords: solar thermochemical energy storage; solar collector; heat pipe; visualization



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1. Introduction

Concentrated solar power (CSP) technology has developed over the past 40 years and is considered a promising alternative to traditional coal-fired power plants, playing a crucial role in reducing dependence on fossil fuels and mitigating greenhouse gas emissions. Recently, CSP technology has seen significant advancements, with the ability to reach temperatures exceeding 500 °C [1,2], making it compatible with existing thermal chemical processes such as methane dry reforming and high-temperature hydrogen production [3]. Solar high-temperature thermochemical energy storage, also known as a ‘chemical heat pipe’, refers to the use of chemical reactions to convert concentrated solar energy into chemical energy, thereby achieving energy conversion, storage, and transportation. This method is expected to become an important candidate for large-scale solar energy utilization.

A critical component in this technology is the solar high-temperature collector, or heat absorber, which converts solar radiation into thermal energy. However, these collectors are subject to high temperatures and extremely uneven radiation, leading to local hotspots and potential equipment damage, necessitating robust design solutions [4,5].

Heat pipes, known for their efficient heat transfer based on phase change, are widely used in industries such as chemical engineering, metallurgy, aerospace, and solar concentration [6–8]. Flat heat pipes, in particular, have a unique structure that results in very

low-temperature gradients and can withstand higher heat flux densities [9]. In a previous study, we developed a novel solar collector by coupling the heat pipe with the flat heat pipe, named the Flat-Tube Shaped Heat Pipe (FT-SHP) [10]. This innovative design leverages the efficient heat transfer and flattening capabilities of flat heat pipes to reduce local hotspots exposed to high solar concentration, thereby enhancing heat absorption and transfer efficiency.

The operation of heat pipes involves complex multiphase flow and phase change heat transfer processes. Currently, research on heat pipes primarily relies on experimentation supplemented by theoretical analysis. With the expansion of the application scope of heat pipes, various types of heat pipes with different physical parameters and operating conditions have emerged, complicating the study of internal flow and heat transfer mechanisms. Existing research often lacks precise mathematical descriptions and predictions of heat pipe behaviors. With advancements in computer performance and numerical computing technology, numerical simulation has become a vital tool for studying heat pipes. However, these simulations often rely on assumptions that can reduce their reliability. Additionally, most experiments cannot directly capture flow and heat transfer parameters inside heat pipes, leaving many phenomena unexplained. Therefore, visually observing the phase change heat transfer behaviors inside heat pipes is crucial for understanding heat and mass transfer processes.

Common visualization methods mainly consist of two approaches: simulation through computational fluid dynamics (CFD) techniques and experimental visualization using transparent materials or perspective imaging technology. CFD technology plays a crucial role in addressing complex flow problems in many engineering applications due to its universality, flexibility, accuracy, and efficiency [11,12]. For CFD simulation of the phase change heat transfer process inside heat pipes, the model used to determine the heat and mass transfer rate at the gas–liquid interface is crucial. The CFD simulation research of medium and low-temperature heat pipes generally adopts the Lee phase transition model, which is derived from the kinetic theory of gases under constant pressure conditions [13]. It does not consider the influence of initial boiling superheat. To rectify this difficulty, the author's team proposed the Pressure-based Phase Change Model (PPC) for the phase change heat transfer characteristics of coupled vaporization-condensation in heat pipes [14,15]. The PPC model relates the phase change temperature and the heat and mass transfer rate equation at the phase change interface to the internal pressure of the heat pipe. For the nucleation process of the working fluid at the heating surface, the consideration of initial boiling superheat is also introduced. Based on simulation research and experimental verification of the heat transfer process of ammonia heat pipes, it has proven to improve rationality and accuracy using the PPC model [15]. Additionally, CFD methods have also been applied to other types of heat pipes, such as two-phase loop thermosyphons, oscillating heat pipes, and rotating heat pipes.

The author's team is attempting to extend the application of the PPC model to simulate sodium heat pipes [15]. High-temperature heat pipes utilize alkali metals such as sodium and potassium as working fluids. These metals exhibit high thermal conductivity and an extremely low saturation vapor pressure, necessitating a high degree of superheat for boiling. Consequently, their phase change heat transfer characteristics differ significantly from those of medium and low-temperature working fluids. Current numerical simulation studies on high-temperature heat pipes have significantly simplified the internal heat and mass transfer processes [16–19], thereby failing to represent their operational mechanisms fully.

Alternatively, visualization experimental research uses perspective imaging technologies, including particle image velocimetry (PIV) [20,21], electrical capacitance tomography (ECT) [22], and neutron radiography (NR) [23,24], as well as transparent materials [25]. Among them, visualization using transparent materials is commonly used to investigate

two-phase flow and heat transfer processes in heat pipes. Smith et al. [26,27] investigated the effect of confinement on the boiling regimes of a small-diameter thermosyphon with different working fluids (water, ethanol, and HFE-7000). A flow regime map was developed based on the observed flow visualization. Through visualization, He et al. [28] investigated the two-phase flow mechanism of mini-diameter loop thermosyphon under different heat loads and filling ratios. A nylon pipe was installed above the evaporator to observe the flow pattern of the working fluid. The results showed that slug flow, churn flow, and annular flow were observed at lower filling ratios and only single-phase flow when the filling ratio reached 91%.

The visualization method provides an intuitive and feasible way to study the two-phase flow of working fluid in heat pipes, providing significant insights into the heat and mass transfer mechanisms. While current visualization experiments can qualitatively explain some heat transfer phenomena, they struggle to quantitatively reveal the heat transfer laws specific to heat pipes [29,30]. The development of dynamic observation of bubble and droplet behavior at the microsecond scale can provide more direct validation for simulation work. Combined with machine learning, it is necessary to propose additional models and revise phase transition models to improve simulation accuracy. However, there is a big lack of work in this area.

Herein, a glass-made FT-SHP was designed and fabricated to explore its internal heat transfer mechanism visually. By adopting the bubble growing frequency (BGF) and the droplet condensation reflux period (DCRP) as indicators, the experimental performance of the FT-SHP was evaluated. The evaporation and condensation phenomenon of the working fluid in this heat pipe was investigated, and its temperature distribution and thermal resistance were compared. The correlation between the start-up characteristics, temperature uniformity, and heat transfer performance of the FT-SHP was obtained. The direct visualization exploration of microsecond scale gas–liquid transfer behavior will promote the solution of a series of challenges in the CFD simulation of complex high-temperature heat pipes.

2. Experimental

2.1. Construction of FT-SHP

The FT-SHP was fabricated using high borosilicate glass. A heating water tank was coupled with the bottom plate of FT-SHP, with the structural diagram shown in Figure 1a,b.

The FT-SHP was mainly composed of a heat pipe and a vacuum valve. The heat pipe includes four vertical pipe fittings of $\Phi 18 \times 2$ mm and a large diameter plate pipe of $\Phi 137 \times 5$ mm. For the hot plate part, the large-diameter pipe fitting heats the lower bottom plate (hereinafter referred to as the evaporation section plate) through the heating water tank to realize the evaporation section function of the heat pipe. The upper surface of the vertical small pipe fittings and the large-diameter pipe fitting serve as the condensation section of the heat pipe through cooling, referred to as condensation pipe and condensation plate, respectively.

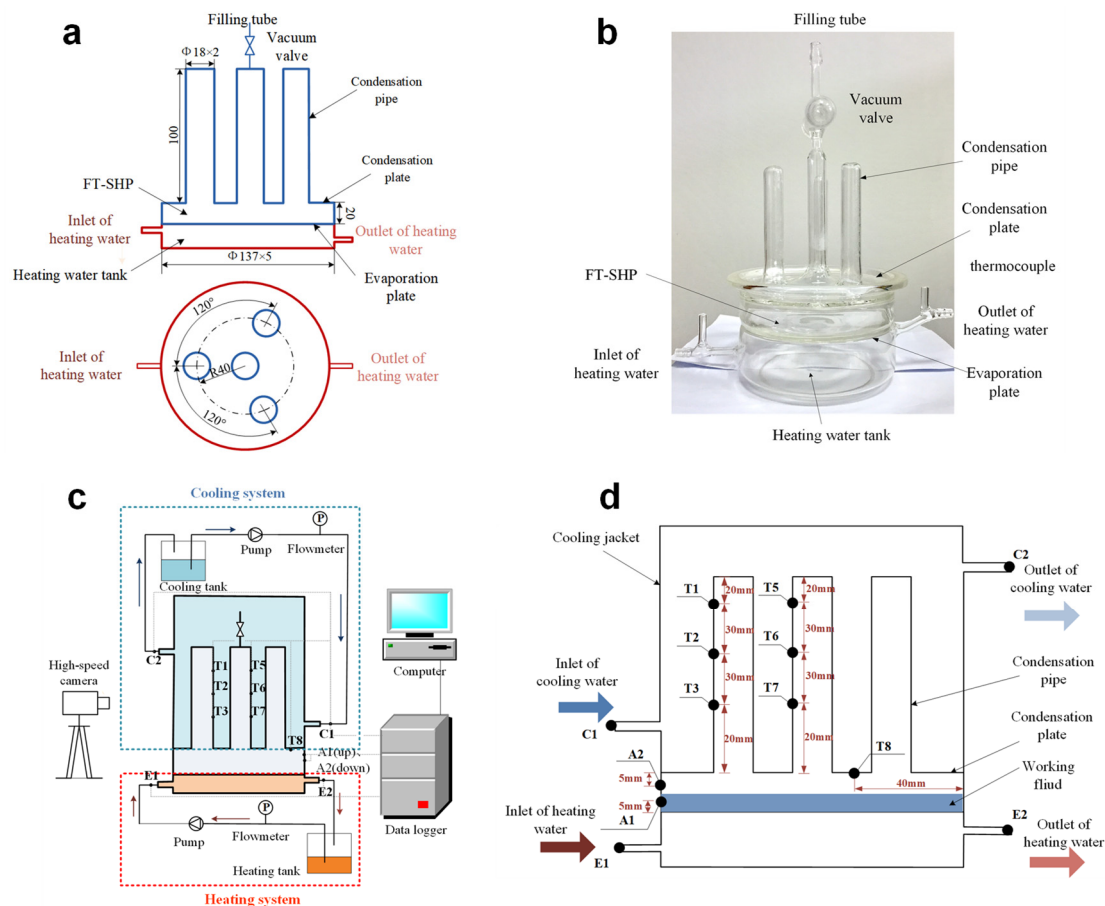


Figure 1. Structure diagram and photo of the FT-SHP made of glass. (a) Diagram and (b) Photo. (c) Schematic view of the experimental set-up for the FT-SHP. (d) Arrangement of temperature monitoring points.

2.2. Experimental System and Test Profiles

A schematic view of the visualization experimental system for the FT-SHP is shown in Figure 1c. The system is divided into the FT-SHP, heating system, cooling system, and data–image acquisition instrument. Hot water was used to heat the evaporation plate. A large diameter pipe fitting, with an outer diameter of 137 mm and an inner diameter of 127 mm, served as the cooling jacket. The heating water tank and cooling water jacket were connected to thermostatic baths, which were used to control the inlet temperature of the cooling and heating water. Two flowmeters were used to measure the flow rates of heating and cooling water, respectively. Fourteen T-type thermocouples were used to monitor the outer wall and water temperatures.

The distribution and location of temperature monitoring points are shown in Figure 1d. Nine thermocouples were used to monitor the temperature of the condensation section: T_1 – T_3 at the condensation edge pipe, T_5 – T_7 at the condensation center pipe, T_8 at the condensation plate, and A_1 and A_2 at the adiabatic section. C_1 and C_2 were used to measure the inlet and outlet temperature of cooling water. E_1 and E_2 were used to measure the inlet and outlet temperature of heating water. Data were recorded by a data logger and then transferred to a computer. A vacuum pump was used to evacuate the non-condensable gas inside the heat pipe before the experiment. A high-speed camera was used to capture the two-phase flow phenomena of the working fluid in the FT-SHP at a frequency of 1500 fps.

To analyze the steady-state operating characteristics of the heat pipe, the bubble growing frequency (BGF) of the condensate vapor inside the pipe and the droplet condensation reflux period (DCRP) in the evaporation section liquid pool were proposed as evaluation

indicators. Given that the period of bubble generation, growth, and rupture during the steady-state stage may be very short, the high-speed camera's shooting speed was adjusted to 1500 fps.

2.3. Performance Evaluation

Total Thermal Resistance R Was Used to Evaluate the Thermal Performance of FT-SHP, and It Was Calculated Using the Following Equation:

$$R = (T_e - T_c) / Q_e \quad (1)$$

where T_e and T_c are the time-mean values of the wall temperature at the evaporator (E_1 and E_2) and condenser (T_1 – T_8), respectively. Q_e is the practical heat transfer power of the evaporator, and it can be calculated by

$$Q_e = C_p \rho V_e (E_1 - E_2) \quad (2)$$

where c_p denotes the specific heat of heating water, ρ is the density of water, and V_e represents the flow rate of heating water. E_1 and E_2 are the water temperature at the inlet and outlet of the heating water tank, respectively. To ensure the visualization effect of the heat pipe, the circumferential side of the evaporation section was not insulated. The circumferential side of the heating jacket undergoes convective heat transfer with the air. However, considering the ignorable small area of the circumferential side and the relatively small temperature difference with the environment, the estimated heat loss is negligible. Therefore, the practical heat transfer power in the evaporation section can be calculated based on the inlet and outlet temperatures of water.

Reynolds number R_e could indicate the flow regimes of the inner vapor of working fluid, and R_e was obtained using the following equation:

$$R_e = d u \rho / \mu = 4W / \pi d \mu \quad (3)$$

$$W = Q_e / \gamma \quad (4)$$

where d is the pipe diameter, u is the velocity, μ is the viscosity of working fluid (water), W is the mass flow of working fluid, and γ is the latent heat of vaporization of water vapor at working temperature.

3. Results and Discussion

3.1. Visualization Characteristics in Stable Operation

When the heat pipe operated stably, the working fluid in the evaporator region absorbed energy and evaporated, leading to the generation, growth, and rupture of bubbles. Simultaneously, droplets formed in the condenser region. The droplet condensation reflux period (DCRP) involves four stages: generating, gathering, growing, and dripping. The single bubble growth and condensation phenomenon in the FT-SHP with a filling ratio (FR) of 20% are shown in Figure 2a. The inlet temperature and flow rate of the heating water were controlled at 90 °C and 1000 mL/min, respectively.

Additionally, the process of steam condensation includes four stages: droplet formation, merging, growth, and detachment. Understanding these stages and their interactions is fundamental for revealing the droplet condensation heat transfer mechanism. Thus, recording the condensation reflux cycle of heat pipes is crucial for analyzing the evaporation and condensation rates during stable operation. Figure 2b illustrates the condensation process of a single droplet. Initially, a 3 mm diameter droplet in the middle pipe begins to descend, absorbing smaller droplets along its path. After 0.5 s, the droplet grows to 4 mm

in diameter. At 1.6 s, the droplet continues to increase in size while remaining in the pipe. Finally, at 4.1 s, the droplet falls to the evaporator. The entire period from droplet generation to condensation reflux at a specific position is approximately 144 s, significantly longer than the bubble growth period. The stable generation and disappearance of individual droplets make the DCRP in the condenser a reliable metric for quantitatively assessing visualization phenomena.

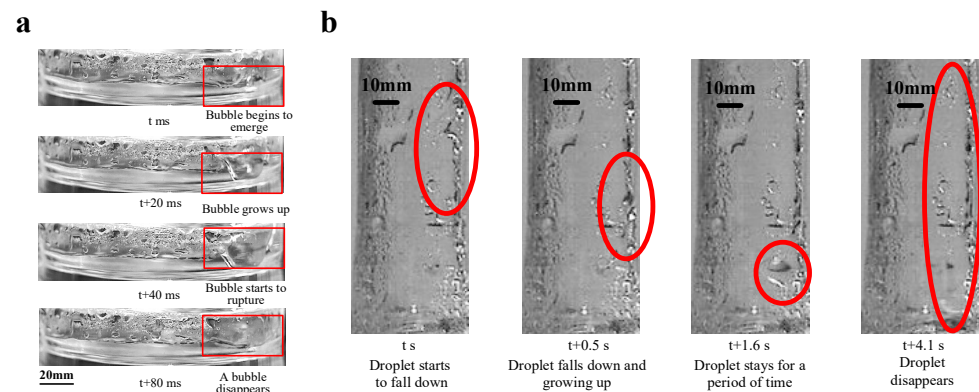


Figure 2. (a) The generation and rupture of bubbles. (b) Condensation process of a single droplet in the condenser pipe.

3.2. Heat Transfer Performance of FT-SHP

The performance of FT-SHP under varying factors, including filling ratio (FR), cooling flow rate, and heating temperature during steady-state operation, was investigated. The filling ratio, in particular, is a pivotal determinant of heat transfer efficacy within a heat pipe. Given the confined volume within the heat pipe, even minor alterations in the liquid filling ratio can cause substantial variations in the internal distribution of the working fluid. At extremely low filling ratios, there is a risk of vapor dry-out within the evaporator segment, impeding efficient heat transfer. Conversely, excessively high filling ratios pose challenges in bubble generation, consequently hampering heat transfer efficiency. Therefore, meticulous adjustment of the filling ratio is imperative to balance these factors and ensure optimal heat transfer performance.

As illustrated in Figure 3a, the DCRP exhibited no significant increase until the FR reached 25%. Beyond this threshold, the DCRP experienced a rapid surge. Below the 25% FR, the accelerated evaporation rate of the working fluid led to an accumulation of excessive steam in the condenser, resulting in a decline in DCRP as the FR increased. In contrast, a FR higher than 25% saw a reduction in steam within the condenser due to the presence of excess liquid in the evaporator, thereby prolonging the DCRP. Additionally, the fluctuation trend of the DCRP at 25% FR was more gradual compared to that at 45% FR. Furthermore, an increase in heating temperature correlated with a decrease in DCRP, indicating enhanced heat transfer performance. This analysis underscores the intricate interplay between the filling ratio, heating temperature, and the DCRP, providing valuable insights into optimizing heat transfer efficiency.

It can also be seen from Figure 3a that the BGF increased with an increase in FR at the same heating temperature. The increasing rate of BGF was the highest when the FR increased from 20% to 25%, with a sharp rise from 52 to 75.3 ms at a heating temperature of 90 °C. This can be attributed to the decreased evaporation rate and the extended generation of the bubble as the filling ratio increased.

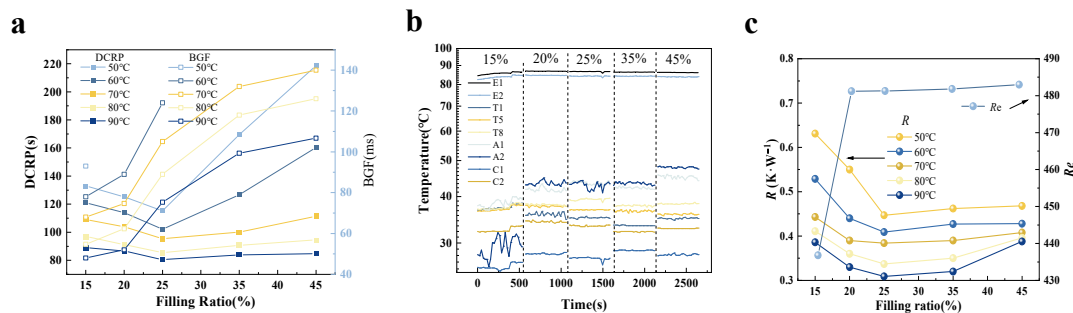


Figure 3. (a) Variations of DCRP and BGF under different filling ratios. (b) Temperature variations at steady state under different filling ratios. (c) The effect of filling ratio on the thermal resistance and Re number at steady state.

The heat transfer performance of the FT-SHP was analyzed by comparing the temperature variations at various monitoring points under different filling ratios during steady-state operation (Figure 3b). Specifically, points T_1 and T_5 were selected to compare temperatures at the condensation center pipe and the condensation edge pipe, respectively. The temperature difference between the inlet and outlet of the heating water remained largely consistent across different filling ratios, indicating that the actual heat transfer power was similar. Compared with other points, the temperatures at A_1 and A_2 in the adiabatic section exhibited larger fluctuations, attributed to the continuous formation of boiling bubbles in the evaporation section during steady-state operation. The temperature fluctuation at A_1 and A_2 hit the biggest when the FR was 15%. When the FR exceeded 25%, a smaller high-frequency fluctuation at A_1 and A_2 was obtained, indicating that a stable boil was established. Additionally, the smallest temperature difference at A_1 and A_2 was observed at a FR of 25%.

To further analyze the heat transfer performance of FT-SHP and explore the characteristics of dimensionless standard numbers, the Reynolds number (Re) of steam flow inside FT-SHP was calculated (Figure 3c). The results indicated that Re increased when the FR was below 25%. However, Re remained almost unchanged when the FR was above 25%. Furthermore, when the heating temperature was fixed, the lowest thermal resistance (R) was observed at a FR of 25%. When the heating temperature was set to 90 °C, the lowest thermal resistance of 0.31 K/W was obtained. Therefore, this analysis reveals that the filling ratio significantly influences the thermal performance during steady-state operation, with an optimal filling ratio of 25%.

Different levels of cooling flow rates can have various effects on heat transfer performance. Increasing the cooling flow rate generally enhances heat transfer efficiency, as more cooling medium flows over the surface of the heat pipe, removing more heat. This increase typically results in a decrease in the surface temperature of the heat pipe and the temperature of the heat transfer medium. However, excessively high cooling flow rates may lead to negative effects, such as increased system energy consumption, higher pump power requirements, and elevated system costs. Additionally, excessively high flow rates may induce unstable flow within the heat pipe, thereby decreasing heat transfer efficiency. Therefore, exploring the impact of different cooling flow rate levels on heat transfer performance is crucial. As such, the thermal performance of the FT-SHP under different cooling water flow conditions was investigated. The cooling flow rate was increased from 300 to 600 mL/min with an increment of 100 mL/min (Figure 4a). As the cooling flow rate increased, the DCRP declined, primarily because of the heightened heat transfer efficiency during condensation. Conversely, the BGF showed a rise with the escalating cooling flow rate, reaching 120 ms at a rate of 600 mL/min. It is important to note that a

longer BGF implies a diminished impact of bubbles on disrupting the boiling process of the working fluid, thereby promoting greater stability in the evaporation process. This trend underscores the intricate interplay between cooling flow rates and the dynamic behaviors of heat transfer mechanisms within the FT-SHP. Exploring these relationships further could offer insights into optimizing cooling strategies for enhanced heat transfer performance.

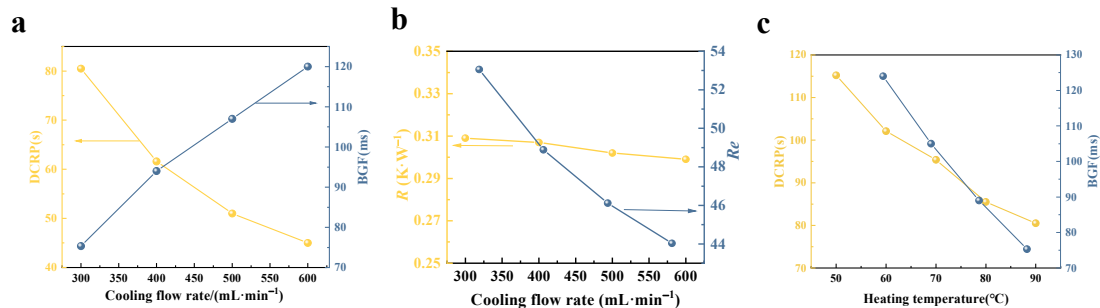


Figure 4. (a) Variations of DCRP and BGF at steady state under different cooling flow rates. (b) The effect of cooling flow rate on the R and Re number at steady state. (c) Variations of DCRP and BGF at steady state under different heating temperatures.

Additionally, the Re number exhibited a downward trend with the augmentation of the cooling flow rate (Figure 4b), indicating a reduction in the disturbance of the working fluid within the liquid pool. Moreover, the thermal resistance (R) demonstrated a gradual decline as the cooling flow rate increased, albeit at a slow pace. Notably, the maximum deviation in thermal resistance was minimal, approximately 0.009 K/W, underscoring the marginal impact of variations in cooling flow rates on thermal resistance. This nuanced relationship highlights the intricate balance between cooling flow dynamics and thermal performance optimization. Further analysis of these trends could offer valuable insights into fine-tuning cooling strategies to achieve optimal heat transfer efficiency.

As the heating temperature increases, the evaporation rate of the working fluid inside the heat pipe also increases, thereby enhancing the heat transfer performance. This occurs because higher temperatures accelerate the phase change process of the working fluid, causing more liquid to evaporate into vapor and thereby increasing the heat transfer within the heat pipe. Additionally, higher heating temperatures can also increase the temperature gradient inside the heat pipe, thereby promoting more efficient heat transfer. However, excessively high heating temperatures may also lead to some issues, such as increasing pressure on the heat pipe, and may even cause damage to the heat pipe material or uneven thermal expansion. Therefore, when designing and operating heat pipes, it is necessary to consider comprehensively the impact of heating temperature on heat transfer performance to achieve optimal heat transfer efficiency.

The heat transfer performance of the FT-SHP at steady state under different heating temperatures (50–90 °C) was measured. As illustrated in Figure 4c, both DCRP and BGF exhibited a decreasing trend with rising heating temperatures. This phenomenon can be attributed to the elevated heating water temperature, which leads to heightened heat transmission, consequently increasing the heat flux density within the working fluid of the heat pipe. Consequently, this rise triggers an augmentation in the vaporization core on the wall and accelerates the frequency of bubble generation, intensifying vaporization within the evaporation section and amplifying bubble disturbance. This intricate interplay between heating temperature and vaporization dynamics underscores the importance of understanding their impact on the overall heat transfer performance of the heat pipe.

With the continuous increase in the heating water temperature, all temperature monitoring points showed a significant upward trend (Figure 5a). It is obvious that the tem-

perature difference between the inlet and outlet of the evaporation and cooling sections increased, indicating that the practical heat transfer power of the heat pipe increased with the increase in heating temperature. When the heating temperature was below 70 °C, a drastic temperature fluctuation at adiabatic section A₂ was observed, possibly due to geyser boiling occurring in the evaporator. When the heating temperature increased to 90 °C, the temperature fluctuations of A₂ were more frequent with a small amplitude, indicating stable boiling was established.

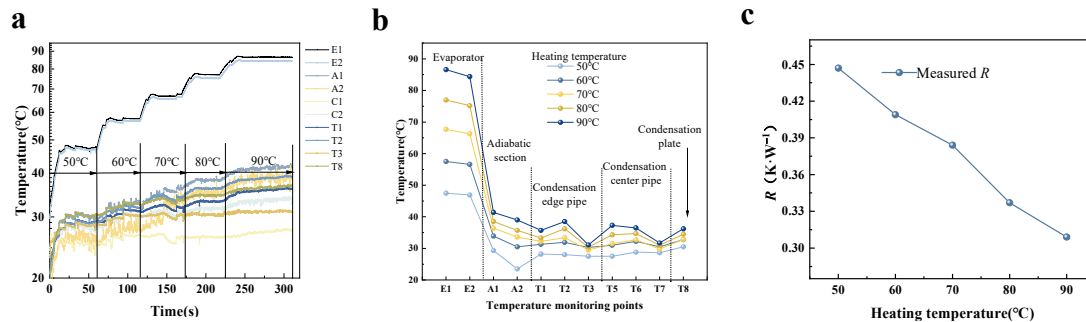


Figure 5. (a) Temperature variations of monitoring points at steady state under different heating temperatures. (b) Temperature distribution of the FT-SHP under different heating temperatures. (c) The effect of heating temperature on R at steady state.

Furthermore, Figure 5b depicts the temperature distribution within the heat pipe, revealing a consistent upward trend in temperature across all points with increasing heating temperature. Moreover, the temperature differential between A₁ and A₂ gradually diminished with rising heating temperatures, underscoring the dynamic nature of temperature variation within the heat pipe with changing heating conditions.

Conversely, R exhibited an almost linearly decreasing trend with the rising heating water temperature. Notably, as the heating water temperature reached 90 °C, the thermal resistance dropped to its lowest point, approximately 0.309 K/W (Figure 5c). This observation underscores the critical role of heating temperature in optimizing thermal performance, suggesting potential avenues for enhancing heat transfer efficiency in the FT-SHP.

3.3. Relationships Between Heat Transfer Performance and Visualization Phenomena

In order to investigate the effects of flow and visualization parameters such as Re number, BGF and DCRP on heat transfer performance of R , an analysis and prediction method was used. A back-propagation (BP) neural network model was established to predict the visualization thermal performance of the FT-SHP. The model simulated thermal resistance, visualization performance parameters, and dimensionless flow numbers and predicted their relevance. The results indicate that a 4-12-4 BP neural network model structure is optimal, with prediction errors below 5% after a training time of 2000 iterations. The results are shown in Figure 6.

As shown in Figure 6a, R decreased significantly with increasing Re . This can be explained by the fact that higher Re values result in more vigorous fluid flow within the FT-SHP, which enhances heat transfer and thus reduces thermal resistance. In addition, a correlation between R and Re was fitted, as depicted by the red solid line in Figure 6b. The fitting correlation can be expressed as follows:

$$R = 3.45Re^{-0.37} \quad (5)$$

It can also be seen from Figure 6c that the relative error of measured R and predicted R was below 20% in most cases, indicating a benign reliability.

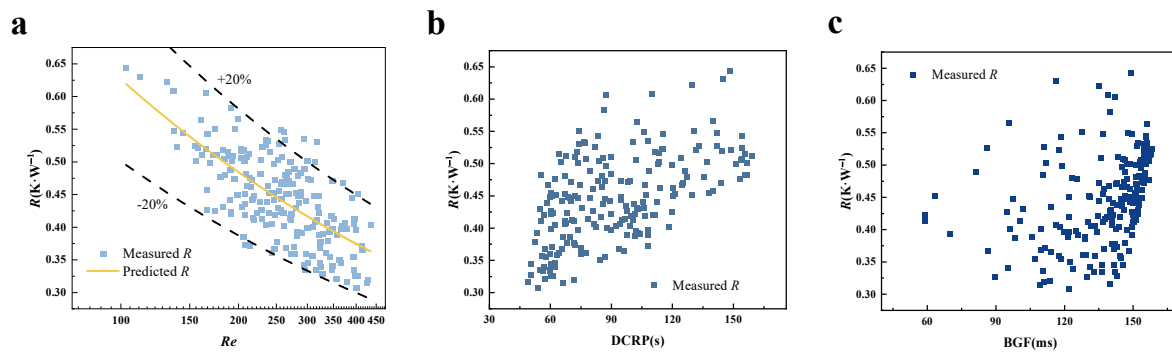


Figure 6. (a) R varies with Re . (b) R varies with the DCRP. (c) R varies with the BGF.

4. Conclusions

In this study, we designed a glass FT-SHP and established a visualization experimental platform to investigate its operational mechanisms and heat transfer characteristics. The BGF and DCRP were used as key parameters to evaluate the performance of the heat pipe. Our findings revealed that the BGF ranged from 90 to 150 ms, and the DCRP ranged from 50 to 150 s, while the overall thermal resistance was less than 0.6 K/W . As the liquid filling ratio increased, the BGF exhibited a lengthening trend, while the DCRP and thermal resistance initially decreased and then increased, identifying an optimal liquid filling ratio of 25%. Increasing the cooling flow rate resulted in opposite trends for BGF and DCRP, while both the Re number and thermal resistance decreased. Furthermore, as the heat flow temperature in the evaporation section increased, the BGF and DCRP shortened, leading to reduced thermal resistance. Notably, during heating temperatures above 70°C , the heat pipe entered an efficient operational stage. We calculated the Re numbers for the heat pipes and found them to be lower than 500, indicating a laminar flow. With increasing Re numbers, thermal resistance decreased, thus improving heat transfer performance. An empirical correlation between thermal resistance and Re number was derived, showing a deviation within 20%. The thermal resistance displayed a nonlinear relationship with BGF and DCRP, increasing with DCRP, though the relationship with BGF remained unclear. These conclusions establish a semi-quantitative correlation between micro gas–liquid dynamics phenomena and macroscopic performance in heat pipes, providing crucial guidance for optimizing the design of heat pipes, with the potential to significantly enhance the efficiency and reliability of solar collectors.

Author Contributions: J.L.: Investigation, Methodology, Resources, Writing—original draft. R.W.: Investigation, Methodology, Visualization. T.X.: Investigation, Methodology, Writing—original draft. H.C.: Formal analysis, Project administration, Supervision, Writing—review & editing. All authors have read and agreed to the published version of the manuscript.

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Data Availability Statement: Please refer to the attached content of Visualization Investigation of Heat Transfer Behavior in a Flat-Tube Shaped Heat Pipe-SI for relevant data.

Conflicts of Interest: Author Jue Li was employed by the company CIMC Enric Engineering Technology Co., Ltd. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

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