Simulation and Experimental Study on Vacuum Negative Pressure Infusion of Working Fluid for Micro Heat Pipes

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ABSTRACT Optimum filling ratio as an important parameter has a great effect on the heat performance of heat pipes. Especially for micro heat pipes with only hundreds of microliters of internal volume, their optimum filling ratios are hard to be obtained without the filling accuracy of an order of microliter magnitude. This paper simulated the filling process of water as the working fluid, and the change regularities of the pressure, the velocity and the volume fraction of liquid water were investigated. The simulation results show that the flow pattern of plug flow is the key to leading to the infusion error. The liquid-vapor slugs caused the pulsation of pressure and velocity, which would make the flow flux uncontrollable accurately. The maximum relative error in simulated filling ratios reached -1.8%, of which -1.6% was due to the slug flows as against only 0.1% due to phase change. To eliminate the influence of slug flows formed under negative pressure, a novel charging method with a buffer pipeline was proposed. Then a test of five groups of different charging quantities (20 µL, 25 µL, 30 µL, 40 µL, and 50 µL) for two MHPs with about 100 µL internal volume was carried out, and the results suggest that the mean absolute error is below 1 µL.

INDEX TERMS Filling ratio, micro heat pipe, infusion method, heat performance.

NOMENCLATURE

\[ W \quad \text{Total work done by external forces (J).} \]
\[ P_i \quad i = 1,2; \text{pressure of working fluid at position} \]
\[ S \quad \text{i (Pa).} \]
\[ E_k \quad \text{Inner bore area of silicone tube.} \]
\[ E_p \quad \text{Kinetic energy (J).} \]
\[ m \quad \text{Potential energy (J).} \]
\[ v_i \quad \text{Mass (Kg).} \]
\[ \Delta t \quad \text{Velocity of working fluid at position i (m/s).} \]
\[ h \quad \text{Time interval (s).} \]
\[ \rho \quad \text{Height (m).} \]
\[ g \quad \text{Density of working fluid (Kg/m}^3). \]
\[ t \quad \text{Acceleration of gravity (m/s}^2). \]
\[ K \quad \text{Time (s).} \]
\[ T \quad \text{Thermal conductivity (W/(m-K)).} \]

\[ Q \quad \text{Heat flux (W).} \]
\[ L \quad \text{Length (m).} \]

Subscript

\[ \text{i} \quad \text{Position in the direction of silicone tube length.} \]
\[ \text{eff} \quad \text{Effective.} \]

I. INTRODUCTION

The trend toward higher circuit integration and higher power consumption results in an increase in the heat flux, which may cause the operating temperatures of electronic components to exceed the desired temperature level [1]. According to the statistics, assuming that the temperature was beyond the normal operating temperature by 10°C, the performance would decrease by a half [2]. As a consequence, the cooling devices should be smaller in size and effective in heat transport. In recent years, passive thermal solutions without
input drive power have become more popular, including heat pipes, vapor chambers, and so on. The common advantages of these components are utilizing the latent heat of phase change to transfer heat effectively, and the working fluid inside is periodic flowing relying on the wick structure. Some investigators have focused on applications of heat pipes in high-power LEDs [3], heat exchangers [4], [5], heat recovery systems [6], cell cryopreservation, and so forth. For example, due to the low thermal resistance and high capillary force of flat plate heat pipe (FPHP), the highest temperature of FPHP integrated with a 120W LED module could be decreased by 28%, as compared to a commercial substrate [3]. Especially, micro heat pipe (MHP) which differs from a conventional heat pipe has been attracting more and more attention, because it does not contain a wick instead of the sharp edges of micro-grooves or is in form of a single closed serpentine tube to circulate the working fluid [7], [8]. A novel skew-grooved structure was proposed for MHPs to improve the capillary force of the grooved wick and then promote the return of the condensate to the evaporator [9]. Radial grooves, star grooves, and rhombus grooves MHPs were designed and fabricated to allow liquid and vapor flow separation to reduce viscous shear force [6], [10], [11]. These three MHPs were all composed of three layers, and their heat transfer performance was improved by 27% more than that of the plain wafer. Compared with conventional heat pipes, wickless MHPs possess much more different cross-section geometries and smaller overall sizes, which is in favor of integration with the cooling objects.

The main factors that affect the heat transfer performance of MHPs, as well as conventional heat pipes, include groove/wick structures, the material of working fluid, operating degrees of inclination, and especially filling ratios. For example, the combined effect of the inclination angle and the filling ratio at different heat input levels were investigated, and the results show that the vertical operation was much more efficient than the horizontal operation and that 0.5 is the optimal filling ratio in comparison to the filling ratios of 0.7 and 0.9 [12]. For a closed-loop pulsating heat pipe, the lowest temperature was obtained at a nearly 70% filling ratio [13]. Hussein et al. [14] investigated three different distilled water filling ratios of 10%, 20% and 35% over wickless heat pipes with different cross-sectioned geometries, and the experimental results indicate that the optimum water filling ratio of the elliptical cross section wickless heat pipe is about 10%, while the circular cross section one is very close to 20%. Shi et al. [15] did experimental research on the heat transfer coefficient of radial heat pipes, and obtained the relationship between heat pipe equivalent heat resistance and working temperature, heat flux and filling ratio. Jobb et al. [16] investigated the quantity of working fluid in the gravitational heat pipe, and the individual results show that the most appropriate level of working media is between 20%-25%. Pouryoussefi et al. [17] simulated the chaotic flow in a 2D closed-loop pulsating heat pipe and proved all of the Lyapunov exponents were positive for the temperature interval between 120°C and 180 °C at the filling ratio of 75%, which was a signature of chaos in these operating conditions. Tharayil et al. [18] investigated the heat transfer performance of a novel miniature loop heat pipe(mLHP) using distilled water as the working fluid for the filling ratios of 20%, 30% and 50% in a heat load range of 20–380W. The experimental study shows that the filling ratio has significant impact on the heat transfer performance of mLHP and that the filling ratio of 30% is identified as the optimum filling ratio. Aly et al. [19] concluded that the evaporation and condensation heat transfer coefficients increase as filling ratio increases when the filling ratios are 20, 40, 60 and 80%, respectively, for a helically-micro-grooved heat pipe. Shi et al. [20] found that an oscillating heat pipe with filling ratio of 50% started up earlier than that with 70% when heating input was 159.4W, while ethanol or acetone is used as the working fluid. Yue et al. [21] established a CFD model of the evaporator of a microchannel separate heat pipe to study its heat transfer characteristics and flow mechanisms under different filling ratios and found that the optimal refrigerant filling ratio was from 68% to 100%. The experiment results of Kusuma’s study [22] show that the filling ratio of 60% resulted faster overshoot phenomenon inside the vertical straight wickless-heat pipe with a lower temperature source. Markal and Aksoy [23] investigated the effect of six different filling ratios (10%, 25%, 40%, 55%, 70% and 85%) on the thermal performance of a closed loop pulsating heat pipe, and the optimum thermal performance is obtained for the filling ratio of 40% in existing conditions. Wu et al. [24] found that the CO2 filled heat pipe at the filling ratio of 40% has the optimal heat transfer behavior with the smallest thermal resistance of 0.123K/W. Recently, Abdulshaheed et al. [25] investigated four different filling ratios of DI water 3%, 5%, 10%, and 15% to determine the optimum configuration, and it was found that the optimum filling ratio is 5%, with the lowest thermal resistance of 0.019 K/W. Beiginaloo et al. [26] compared numerical and experimental results, and the averaged optimum values for the mentioned parameters (aspect ratio, heat load, and filling ratio) were 0.8325, 246 W, and 85%, respectively. Flexible branch heat pipes with the 15% filling ratio of working fluid have the smallest thermal resistance while the pipes with the 45% filling ratio have the best anti-gravity performance [27]. Besides, Guo et al. [28] focused mainly on the effects of heat input, filling ratio, inclination angle, tube diameter and coolant temperature on the thermal performance of a wraparound heat pipe charged with R134a, and the study results showed that thermal resistance decreases with the increase of heat input when the filling ratio is larger than 40%, that an optimal filling ratio for the heat pipe with the best performance ranges 50% to 60%, and that the best performance of heat pipe is observed for a 22° inclination angle, an outer diameter of 16 mm, and a filling ratio of 50%.

From the earlier studies, it can be concluded that filling ratios have a great effect on the thermal performance of all heat pipes and that the optimal filling ratios for different heat pipes are not all the same, so it is significant to study and
explore reasons for not having a uniform optimum filling ratio for each heat pipe can be attributed to the diversity of geometry parameters, working fluid materials, inclination angles, heat loads, and so on. Towards traditional heat pipes, their feature dimensions are big enough as well as they are made of metal materials such as copper or aluminum [2], [29], so it is easy for them to be pumped to a high vacuum degree, to control the accurate filling ratio, and then to be sealed. For example, the cold welding sealing method is used to copper-water heat pipe ends. Nevertheless, due to the heat pipe dimensions and manufacturing materials, the charging methods for micro-miniature heat pipes differ from those of conventional ones, in the other words, largescale methods have not been suitable for MHPs anymore. Therefore, it is necessary to get a deeper insight into the flow patterns and flow characteristics’ effects on filling ratios for microscale heat pipes.

Considering the above, little information about how working fluid filled has been introduced in Known literature, so the primary objective of the present work is to propose a novel high-precision method for microlitre magnitude charging. Actually, phase changes of working fluid in the early stages of the charging process are inevitable, which has a significant impact on the amount control of working fluid infusion. Based on this consideration, a 2D CFD infusion simulation with a vacuum negative pressure is employed to study the variation regularity of the working fluid flow rates, flow patterns, and pressure over the charging process. And then a stable infusion apparatus was designed for accurate charging. Finally, the heat transfer performance and optimum filling ratio of two types of MHPs were evaluated based on this present method.

II. PHYSICAL MODEL
Because of a fixed and constant amount of liquid pumped by each revolution and good sealability, the peristaltic pump is a good alternative to driving working fluid. The preliminary scheme is conducted as shown in Fig.1.

A. THEORETICAL ANALYSIS
Fig.2 shows the working principle of working fluid transferring in the silicone tube of the peristaltic pump. According to the work-energy principle, the total work done by external forces equals the increment of mechanical energy. Moreover, total work done by external forces, kinetic energy increment, and potential energy increment can be obtained by Eq. (1), Eq. (2), and Eq. (3), respectively.

\[ W = P_1 S_1 v_1 \Delta t - P_2 S_2 v_2 \Delta t \]  
\[ \Delta E_K = \frac{1}{2} m v_1^2 - \frac{1}{2} m v_2^2 \]  
\[ \Delta E_P = m g h_2 - m g h_1 \]

Therefore, Eq. (4) can be deduced.

\[ P_1 + \frac{1}{2} \rho v_1^2 + \rho g h_1 = P_2 + \frac{1}{2} \rho v_2^2 + \rho g h_2 \]  

For example, there is nearly no change for flow rate \( v \) in a silicone tube under normal pressure, in other words, \( v_1 \) is equal to \( v_2 \), so the kinetic energy increment is zero; also under normal pressure, assuming that the value of \( h_2-h_1 \) is about 10 cm and that \( \rho_1 \) equals \( \rho_2 \) because of the incompressibility, the pressure \( P \) would reduce only by 1kPa using Eq. (4). The pressure difference can be ignored relative to the standard atmospheric pressure of \( 1.01 \times 10^5 \)kPa. However, when the right side of the silicone tube is directly connected to the MHP, which has been pumped to a very low pressure by the vacuum pump, the differential pressure between \( P_2 \) and \( P_3 \) is very large, which means the velocity \( v_3 \) is much greater than the \( v_2 \), according to Eq. (4). At this point, it will be difficult to control the actual filling amount by the peristaltic pump. There are mainly three reasons for this problem. Firstly, the actual flow flux is linearly related to differential pressure in accordance with Poiseuille’s formula as the kinematic coefficient of viscosity and radius of the tube have been determined. The second one is that differential pressure has a great effect on the flow velocity of working fluid according to Eq. (4). It will result in the flow velocity arising dramatically as soon as the working fluid gets into the negative pressure tube. Especially, negative pressure could bring down the saturated vapor pressure and then make the working fluid phase change easily, which would make the flow more complicated. For these reasons, it is obvious that the drastic changes in pressure and flow velocity in the tube become significant factors affecting accurate infusion control.
B. CFD SIMULATION

To get the change regularities of flow velocity and pressure variation of working fluid during infusion, 2D simulation analyses were performed through ANSYS Fluent. The physical model geometry and dimension are shown in Fig.3. The computational domain of this model was based on the interior of the silicone tube with an inner diameter of 0.8mm and a length of 20mm and the interior space of an equivalent MHP with a length of 20mm and a width of 5mm. In this model, there is only an entrance for the working fluid on the left side of the silicone tube, while the others are walls. And the entrance can be seen as the position of working fluid flowing out from the peristaltic pump with a flow speed of 4mm/s.

Before the simulation, ICEM CFD was used to mesh this model, and mesh refinement proceeded near the boundary, as shown in Fig.4. The total number of meshes reached 28000.

Water was employed as the working fluid in the following simulation, and both liquid water at 20° with a constant density of 998.2kg/m³ and vapor as the ideal gas would be used as the fluid materials. In Fluent general setup, the solver type of Pressure-Based and the 2D Planar space were chosen. Then, the VOF model was selected with two phases for the primary phase as liquid water and the secondary phase as vapor. Besides, surface tension force modeling with a coefficient of 0.072N/m and evaporation-condensation of Lee model were considered. In saturation properties, the saturation temperature was set to be pressure-dependent, while the relationship between temperature and pressure was provided by a tabular file. In order to improve the computational stability, the absolute pressure of the fluid domain full of vapor was initialized to 300Pa.

Moreover, the velocity inlet with a duration of 13s is controlled by a function expression about time which could be described as shown in Table 1. The purpose of stage 1 was to fill the silicone tube domain with working fluid. And then standby for 1s, it is to obtain the variations of fluid flow in the tube. Immediately following quantitative filling to the MHP domain for 5s, it is to investigate the stabilization of infusion amount in the following 2s.

![FIGURE 3. Model geometry and dimension.](image)

![FIGURE 4. Meshing details of the model.](image)

| STAGE NO. | TIME | VELOCITY INPUT | FUNCTION |
|-----------|------|----------------|----------|
| 1         | 0≤t≤5s | 4mm/s          | Pre-filling |
| 2         | 5s≤t≤6s | 0mm/s          | Standby   |
| 3         | 6s≤t≤11s | 4mm/s         | Quantitative filling |
| 4         | 11s≤t≤13s | 0mm/s          | Standby   |

In order to improve the computational stability, the absolute pressure of the fluid domain full of vapor was initialized to 300Pa.

TABLE 1. Boundary condition of velocity input.

The filling ratio is obtained according to Eq. (5).

$$ FR = \frac{A_{L-MHP}}{A_{MHP}} \times 100\% $$

where $FR$ is the filling ratio; $A_{L-MHP}$ means the area in which the volume fraction of working fluid inside the MHP is greater than 0.95; $A_{MHP}$ means the area of the MHP.

III. SIMULATION RESULTS

A. TRANSIENT SIMULATION RESULTS OF SILICONE TUBE

In this section, both the transient variations of velocity and pressure and the distributions of liquid volume fraction with the time changing are investigated.

Fig.5 shows that the flow velocity increased dramatically as soon as the water got into the negative pressure tube. And the velocity mainly varied from 100 mm/s to 600 mm/s, which means the average velocity increased nearly by sixty times compared to the inlet initial velocity. However, the average velocity immediately reduced from about 240mm/s at $t = 0.1s$ to 90mm/s at $t = 1.0s$. Apparently, rapid changes in velocity are inseparable from the influence of differential pressure. It can be observed from Fig.6 that the pressures were rising rapidly at the time corresponding to the velocity changes and that the pressure uniformities in the first three moments were very good except for the fourth with local pressure increased. More importantly, the pressures were always under the saturated pressure corresponding to 20° in this process, so there must be phase change situations of liquid water to vapor, and that is why water inflow is barely visible.

As the pressure continued to increase, under the action of surface tension, small water droplets appeared little by little as shown in Fig.7(a). The locations of the water droplets were exactly where the local pressures changed in Fig.6(d). After that, the droplets were constantly gathering and forming several liquid plugs which were separated by vapor plugs as shown in Fig.7(b). Compared to Fig.7(b), it can be found
in Fig.7(c) that there still existed evaporation to form vapor plugs and that the sizes of all liquid plugs were slightly reduced. Besides, Fig.7(c) illustrates that the fluid entering later became continuous following after a period of slug flow.

To further reveal the effects of pressure on flow characteristics, the horizontal centerline of the silicone tube domain was selected, and both the average pressure and the average velocity on this line with time changed were plotted together as shown in Fig.8. It can be seen that the pressures in vapor plugs were higher than that in liquid plugs. Since the vapor and liquid plugs were alternated along the flow direction, the pressure exhibits a similar oscillation. Moreover, this oscillation has an impact on the flow velocity as well, in other words, pulsation is the main form of flow in this stage. Focusing on vapor-liquid interfaces, the pressure of the vapor side was always higher than that of the liquid side so differential pressure was formed, and this differential pressure was more similar to a drag force. When the drag forces on both sides of a vapor or liquid plug were unbalanced, this plug would move at a high speed.

Throughout the entire simulation results of the silicone tube, the flow mainly underwent three stages of flow pattern changes: vapor flow, slug flow, and continuous flow. Nevertheless, pressure is the most fundamental influencing factor. From Fig.9, the changes of average pressure in the silicone tube domain over time can be intuitively observed. Within the first 2.5s, the difference between the pressure in the silicone tube and saturated pressure was so high that the vapor was generated continuously during flow and different flow patterns appeared. Subsequently, the flow becomes continuous as the average pressure gradually approached saturated pressure. In the first standby stage, there were almost no new bubbles produced because the average pressure was above the saturated pressure. During quantitative filling, the flow with a small bubble became nearly continuous in the case.
of tiny average pressure fluctuations. In the second standby stage, there were no noticeable changes happened except for average pressure gradually approaching saturated pressure.

B. TRANSIENT SIMULATION RESULTS OF MHP

Fig. 10(a) shows that there appeared no noticeable vapor in the MHP domain until at \( t = 0.5 \text{s} \). Similar to Fig. 7(a), when liquid droplets appeared in the silicon tube, some small droplets also appeared in the MHP as shown in Fig. 10(b). According to Fig. 10(c), the first liquid plug within the silicone tube was going to enter the MHP at the time of \( t = 2.5 \text{s} \). However, there were only a few liquid plugs that flowed into MHP by the end of pre-filling at \( t = 5 \text{s} \) as shown in Fig. 10(d). From Fig. 10(e) and (f), the last vapor plug which was followed by continuous liquid water started to enter the MHP at the moment of \( t = 7.7 \text{s} \), and the distribution of liquid volume fraction in the MHP domain was given at \( t = 11 \text{s} \) when the entire infusion process was completed.

Through the above analysis, although continuous flow was eventually achieved, the actual amount of infusion was yet unclear. Eq. (5) was used to calculate the filling ratios corresponding to the different simulation times. Whereas, the theoretical filling ratio with time changing can be described as:

\[
FR = \begin{cases} 
0\% & 0 \leq t < 6 \text{s} \\
\left( \frac{16}{3} t - \frac{96}{5} \right) \times 100\% / A_{MHP} & 6 \text{s} \leq t < 11 \text{s} \\
16 \times 100\% / A_{MHP} & 11 \text{s} \leq t \leq 13 \text{s} 
\end{cases}
\]

(6)

Fig. 11 illustrates that the value of the simulation filling ratio was still mostly lower than the theoretical filling ratio, during quantitative filling of 6s to 11s, although the preinfusion had been conducted before this. Moreover, the simulation filling ratio reached the average of 14.2\%, as against 16\% of the theoretical filling ratio and the relative error was near -1.8\%. If the simulation scheme considering the connections of valves and sealing parts were applied to practice, the error would be further greater. Back to Fig. 10, 2.5s and 7.7s were the time of the first and the last liquid plugs flowed into the MHP, respectively, and the corresponding relative errors were 0.1\% and -1.6\%. Obviously, the slug flow was the main factor affecting the filling ratio, whereas the effect of phase change on MHP was negligible. The error oscillation after 7.7s was mainly caused by the shape change at the joint of the silicon tube and the MHP and the absolute value of this relative error was around 0.5\%.

IV. EXPERIMENTAL SECTIONS

According to the simulation results above, the key to precise infusion for MHP is to get a continuous constant velocity flow. Then, there are still two issues that need to be solved:
one is large differential pressures causing an increase in flow rate, which may make the flow flux uncontrollable, and the other is to eliminate slug flow to achieve quantitative infusion.

![Figure 12](image_url)

**FIGURE 12. Solution to the large differential pressure and slug flow.**

As can be seen in Fig. 12, there are two pipes connected between the MHP and the working fluid supply source, but an obstacle is used to switch the conduction of these two pipes. The flow resistance of slug flow is so high that the liquid does not enter the MHP for a short period of time, meanwhile, the inner pressure of the MHP is increasing under the influence of evaporation. When inner pressure is close to saturated pressure, the incoming flow becomes continuous. Then, the obstacle is switched to the other pipe in order to prevent the slug flow from flowing into the MHP. Since the liquid is incompressible and the vapor can be compressed, the liquid working fluid can be steadily charged into the MHP.

**A. INFUSION APPARATUS AND MANIPULATION**

Based on the above design, a modified MHP charging method using the peristaltic pump is presented based on our previous work [30], as shown in Fig. 13. The detailed operation sequences can be described as: 1) connect all parts as shown in Fig. 13; 2) turn on the peristaltic pump and control working fluid flowing to the seal section as shown in Fig. 2; 3) turn on the vacuum pump and pump the whole pipeline to predetermined vacuum degree, and then close two-way valves of #1, #3 and #4; 4) set rotation speed of the peristaltic pump and start the peristaltic pump to transfer working fluid through the route of tee joint #1, tee joint #2, two-way valve #2, tee joint #3 and MHP, then immediately stop the peristaltic pump until the working fluid flows steadily in the silicone tube (working fluid supply line); 5) close two-way valve #2 and open two-way valve #3, then start the peristaltic pump again to allow working fluid flowing to the cut position and stop the peristaltic pump; 6) set charging time for the peristaltic pump and start the pump for quantitative charging of working fluid until the setting time is up; 7) close two-way valve #3 and cut the left and the right PP pipes using electric heating scissor, respectively.

**B. CALIBRATION FOR INFUSION QUANTITY**

To verify the feasibility and to measure the charging precision, the calibration for different infusion quantities was carried out. In this study, the flow rate of the selected peristaltic pump (BT100-1F/YZ15, Konap, China) ranges from 0.006 mL/min to 350 mL/min with an accuracy of 0.5%. The external diameter and the inner diameter of the silicone tube are 4 mm and 0.8 mm, respectively. The pumping speed and ultimate pressure of the vacuum pump (RVD-2, KYKY, China) are 2.2L/s and 0.5Pa, respectively. The working fluid is DI water after degassing. The external diameter of PP (Propene Polymer) pipe connected to MHP is 0.9mm while the inner diameter is about 0.6mm. The softening temperature of the PP pipe is around 135°C, so the temperature of the electric heating scissor is controlled between 133°C and 137°C to cut the PP pipe off. Considering that the inner volume of MHPs is usually no more than 100µL and that the filling ratio of most MHPs ranges from 20% to 60%, the infusion quantities for the test are from 10µL to 70µL with an increment of 10µL. Eventually, the actual quantity of infusion is calculated by using an analytical balance (ME204, Mettler-Toledo, China) with the accuracy of 0.1mg to weigh the MHP before and after charging.

Every perfusion quantity was repeatedly charged three times, and the measured and the fitting results are shown in Fig. 14 when the rotation speed of the peristaltic pump is 2 r/min. The results indicate that the actual quantity increases with the rise of setting quantity and that the concentricity of different infusion quantities shows well. However, due to the non-uniformity between rotation speed and infusion quantity, the actual quantity should be calibrated according to measured data. The linear fitting method was used and the standard deviation of fitting reached 0.9996, so the charging system has great infusion linearity.

**C. EXPERIMENTAL TEST FOR HEAT PERFORMANCE**

Based on the high charging infusion, the effects of different filling ratios as well as the groove structure on the heat performance of MHPs were investigated. In this paper, two types of MHPs with different groove structures were chosen and the detailed dimensions were shown in Table 2. Each MHP consists of a Si substrate and a glass cover, and both thicknesses were 1mm and they were attached together using electrostatic bonding.
Then the MHPs were charged with degassed DI water using the charging method above, five couples of different charging quantities (20µL, 25µL, 30µL, 40µL, 50µL) for each MHP were carried out, and the results were shown in Fig. 15. The mean absolute error is below 1µL, so the charging method is suitable for high-precision infusion.

After charging the working fluid, the heat performance of both MHPs was tested through the testing system as shown in Fig. 16. The experiments were conducted in a vacuum chamber with a vacuum degree of less than 10Pa, which was to avoid or diminish heat convection. The temperatures of three sections, including the evaporation section, adiabatic section, and condensation section, were measured and recorded in the whole process of heat transfer. K-type thermocouples (TT-K-36-SLE, OMEGA, USA) were used and calibrated by the thermodetector (735-2, TESTO, Germany) with a precision of ±0.3°C. Each MHP was heated by a ceramic heater with the dimension of 7.0mm(Length) × 5.0mm(Width) × 1.3mm(Height), and the input power, ranging from 1W to 8W with an interval of 1W, was controlled by PID algorithm. A thermostatic bath (SLDC-0510, SHUNLIU, China) was used to control the condensation temperature through a cooling block with an area of 3mm × 6mm. In order to enhance the heat transfer, thermal silicone grease with a heat conductivity coefficient of higher than 2W/(m·K) was filled between the heater and the evaporation section and between the cooling block and the condensation section. For every single measurement, the stop condition was that the temperature of the evaporation section was higher than 90°C or the maximum power beyond 8W.

### V. RESULTS AND DISCUSSION

Fig. 17 shows the temperature gradient variations of evaporation sections of both MHP #1 and MHP #2. It was clearly observed that there was not much difference in temperature distribution between MHP #1 and MHP #2 when the power input was less than 5W. However, when the power input is more than 5W, the temperature gradient of MHP #2 was more sensitive to the filling ratio than MHP #1. Considering the temperature below 80°C, the optimum filling ratio for MHP #2 should be between 25% and 33%, while the optimum filling ratio for MHP #1 should only be near 30%.

Also, the changes in the effective thermal conductivity ($K_{eff}$) for two MHPs versus power input were investigated.
$K_{\text{eff}}$ can be obtained by

$$K_{\text{eff}} = \frac{Q L_{\text{eff}}}{(T_e - T_c) A}$$  \hspace{1cm} (7)

where $Q$ is the heat flux; $L_{\text{eff}}$ is the effective length between evaporation and condensation; $T_e$ and $T_c$ are the temperatures of evaporation and condensation, respectively; $A$ means the cross-section area of the Si substrate.

In Fig. 18, it can be seen that $K_{\text{eff}}$ of MHP #1 with filling ratios of 20% and 25% had a slight downward trend with power input increasing. And yet, the other three filling ratios of 30%, 40% and 50% raised abruptly at the power input of 5W, 6W and 5W, respectively, in other words, their enhanced heat transfer started at that time. For MHP #2, the filling ratio of 50% showed the best heat performance, starting up at the power input of 2W, but began to degrade subsequently. Then, the filling ratios of 30%, 25% and 40% worked obviously at the power of 4W, 5W and 6W, respectively. Nevertheless, the filling ratio of 20% performed the worst a little like MHP #1.

It can be observed that these two MHPs with an internal volume of about 100 $\mu$L performed completely different heat performances, notwithstanding that the filling amounts were all no more than 50 $\mu$L. Therefore, the precise infusion method proposed in this paper can make a significant contribution to analyzing the effects of micro-filling amounts or ratios on heat performance.

**VI. CONCLUSION**

In order to solve the problem that the infusion accuracy under vacuum negative pressure is difficult to control, a high-precision charging method using a peristaltic pump is proposed to implement a continuous inflow of working fluid. Through a 2D CFD simulation, the variation regularities of velocity, pressure and volume fraction of liquid and the influencing factors of infusion error are investigated. A set of comparative experiments of thermal performance are conducted to verify the practicability of this charging method. The conclusions are as follows:

1. According to the simulation results, the flow velocity increases dramatically as soon as the working fluid of liquid water gets into the negative pressure tube and subsequently reduces to a stable value with the increase of pressure.
Before the pressure reaches the saturation pressure of the working fluid, several liquid plugs separated by vapor plugs are formed in the infusion tube. And these plugs have great effects on the velocity and the pressure. When the pressure is close to the saturation pressure, there is no new vapor or liquid plug produced and the flow becomes continuous.

(3) By comparison of theoretical and simulated filling ratios, it is found that the slug flow is the main factor affecting the filling ratio, whereas the effect of phase change on MHP filling ratios was negligible. And the oscillation of the filling ratio relative error caused by the shape change at the joint of the silicon tube and the MHP is around 0.5% and is negligible.

(4) An experimental infusion apparatus with a buffer pipeline is set up to carry out the working fluid filling in microliter magnitude. And it is proved that the buffer pipeline is able to effectively eliminate the influence of slug flow on the infusion amount by a set of comparative experiments on heat performance.

REFERENCES

[1] Y. J. Youn and S. J. Kim, “Fabrication and evaluation of a silicon-based micro pulsating heat spreader,” Sens. Actuators A, Phys., vol. 174, pp. 189–197, Feb. 2012.

[2] Y. Deng, Z. Quan, Y. Zhao, and L. Wang, “Experimental investigations on the heat transfer characteristics of micro heat pipe array applied to flat plate solar collector,” Sci. China Technol. Sci., vol. 56, no. 5, pp. 1177–1185, May 2013.

[3] J. C. Hsieh, H. J. Huang, and S. C. Shen, “Experimental study of microrectangular grooves structure covered with mesh layers on performance of flat plate heat pipe for LED lighting module,” Microelectron. Rel., vol. 52, no. 6, pp. 1071–1079, Jun. 2012.

[4] Z. Liu, Z. Wang, and C. Ma, “An experimental study on heat transfer characteristics of high temperature heat exchanger with latent heat storage—Part I: Charging only and discharging only modes,” Energy Convers. Manage., vol. 47, nos. 7–8, pp. 944–966, May 2006.

[5] X. Liu, G. Fang, and Z. Chen, “Dynamic charging characteristics modeling of heat storage device with heat pipe,” Appl. Thermal Eng., vol. 31, nos. 14–15, pp. 2902–2908, Oct. 2011.

[6] J.-S. Chen and J.-H. Chou, “Cooling performance of flat plate heat pipes with different liquid filling ratios,” Int. J. Heat Mass Transf., vol. 77, pp. 874–882, Oct. 2014.

[7] H. Xu, P. Zhang, L. Yan, D. Xu, W. Ma, and L. Wang, “Thermal characteristics and analysis of microchannel structure flat plate pulsating heat pipe with silver nanofluid,” IEEE Access, vol. 7, pp. 51724–51734, 2019.

[8] P. Chen and Z. Pan, “Heat transfer analysis of flat heat pipe with enhanced microchannel shape,” IEEE Access, vol. 9, pp. 120833–120843, 2021.

[9] J.-H. Wu, Y. Tang, and L.-S. Lu, “Capillary force of a novel skew-grooved wick structure for micro heat pipes,” J. Central South Univ. Technol., vol. 18, no. 6, pp. 2170–2175, Dec. 2011.

[10] S.-W. Kang and D. Huang, “Fabrication of star grooves and thombus grooves micro heat pipe,” J. Micromech. Microeng., vol. 12, no. 5, pp. 525–531, Sep. 2002.

[11] S.-W. Kang, S.-H. Tsai, and H.-C. Chen, “Fabrication and test of radial grooved micro heat pipes,” Appl. Thermal Eng., vol. 22, no. 14, pp. 1559–1568, Oct. 2002.

[12] M. Mameli, V. Mannino, S. Filipescu, and M. Marrone, “Thermal instability of a closed loop pulsating heat pipe: Combined effect of orientation and filling ratio,” Expierim. Thermal Fluid Sci., vol. 59, pp. 222–229, Nov. 2014.

[13] M. Aboutaleb, A. M. N. Moghaddam, N. Noorbakhsh, and M. B. Shafii, “Experimental investigation on performance of a rotating closed loop pulsating heat pipe,” Int. Commun. Heat Mass Transf., vol. 45, pp. 137–145, Jul. 2013.

[14] H. M. S. Hussein, H. H. El-Ghetany, and S. A. Nada, “Performance of wickless heat pipe flat plate solar collectors having different pipes cross sections geometries and filling ratios,” Energy Convers. Manage., vol. 47, nos. 11–12, pp. 1539–1549, Jul. 2006.

[15] C. Shi, Y. Wang, and C. Xu, “Experimental study and analysis on heat transfer coefficient of radial heat pipe,” J. Thermal Sci., vol. 19, no. 5, pp. 425–429, Oct. 2010.

[16] M. Jobb, P. Nemec, L. Kosa, and M. Malcho, “Influence of working fluid amount and working position gravitational heat pipe on thermal performance,” presented at the 19th Appl. Experiment, Numer. Methods Fluid Mech. Energetics, 2014.

[17] S. M. Pourouyssefi and Y. Zhang, “Numerical investigation of chaotic flow in a 2D closed-loop pulsating heat pipe,” Appl. Therm. Eng., vol. 98, pp. 617–627, Apr. 2016.

[18] T. Tharayil, L. G. Asivatham, V. Ravindran, and S. Wongwises, “Effect of filling ratio on the performance of a novel miniature loop heat pipe having different diameter transport lines,” Appl. Thermal Eng., vol. 106, pp. 588–600, Aug. 2016.

[19] W. I. A. Aly, M. A. Elbalshouny, H. M. A. El-Hameed, and M. Fatouh, “Thermal performance evaluation of a helically-micro-grooved heat pipe working with water and aqueous Al2O3 nanofluid at different inclination angle and filling ratio,” Appl. Thermal Eng., vol. 110, pp. 1294–1304, Jan. 2017.

[20] W. Shi and L. Pan, “Influence of filling ratio and working fluid thermal properties on starting up and heat transferring performance of closed loop plate oscillating heat pipe with parallel channels,” J. Thermal Sci., vol. 26, no. 1, pp. 73–81, Feb. 2017.

[21] C. Yue, Q. Zhang, Z. Zhai, and L. Ling, “CFD simulation on the heat transfer and flow characteristics of a microchannel separate heat pipe under different filling ratios,” Appl. Thermal Eng., vol. 139, pp. 25–34, Jul. 2018.

[22] M. H. Kusuma, M. Juarsa, N. Putra, A. R. Antarkisawan, M. Subekti, and S. Widodo, “The filling ratio effect on the overshoot phenomenon of vertical straight wickless-heat pipe with low temperature source,” presented at the 3rd Int. Conf. Nucl. Energy Technol. Sci. (ICONETS), 2019.

[23] B. Markal and K. Aksoy, “The combined effects of filling ratio and inclination angle on thermal performance of a closed loop pulsating heat pipe,” Heat Mass Transf., vol. 57, no. 5, pp. 751–763, May 2021.

[24] Y. Wu, J. Jia, D. Tian, and Y. K. Chua, “Heat transfer performance of microgroove back plate heat pipes with working fluid and heating power,” J. Thermal Sci., vol. 29, no. 4, pp. 982–991, Aug. 2020.

[25] A. A. Abdulshaheed, P. Wang, G. Huang, Y. Zhao, and C. Li, “Filling ratio optimization for high-performance nanoengineered copper-water heat pipes,” J. Thermal Sci. Eng. Appl., vol. 13, no. 5, Oct. 2021, Art. no. 051025.

[26] G. Beigianaloo, A. Mohabbi, and M. M. Afsahi, “Combination of CFD and DOE for optimization of thermosyphon heat pipe,” Heat Mass Transf., vol. 58, no. 4, pp. 561–574, Apr. 2022.

[27] J. Huang, J. Xiang, X. Chu, W. Sun, R. Liu, W. Ling, W. Zhou, and S. Tao, “Thermal performance of flexible branch heat pipe,” Appl. Thermal Eng., vol. 186, Mar. 2021, Art. no. 116532.

[28] C. Guo, T. Wang, C. Guo, Y. Jiang, S. Tan, and Z. Li, “Effects of filling ratio, geometry parameters and coolant temperature on the heat transfer performance of a wraparound heat pipe,” Appl. Thermal Eng., vol. 200, Jan. 2022. Art. no. 117724.

[29] L.-L. Jiang, Y. Tang, H.-Y. Wu, W. Zhou, and L.-Z. Jiang, “Fabrication and thermal performance of grooved-sintered wick heat pipes,” J. Central South Univ., vol. 21, no. 2, pp. 668–676, Feb. 2014.

[30] C. Li, Y. Luo, C. Zhou, Q. Shan, and X. Wang, “Charging method of micro heat pipe for high-power light-emitting diode,” Micro Nano Lett., vol. 10, no. 10, pp. 518–522, Oct. 2015.
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