Optimizing Heat Pipes With Partially-Hybrid Mesh-Groove Wicking Structures and Its Capillary-Flowing Analysis by Simulation

Guanghan Huang

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OPTIMIZING HEAT PIPES WITH PARTIALLY-HYBRID MESH-GROOVE WICKING STRUCTURES AND ITS CAPILLARY-FLOWING ANALYSIS BY SIMULATION

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DEDICATION

To my parents, Youxiang Huang and Shuimei Zeng, for their endless love, encouragements and support. To my wife and son, Xiaoyi Zhou and Jingxian Huang, for their accompany and tolerance. My family’s support makes me who I am, where I am.
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ABSTRACT

Heat pipes are known as efficient two-phase heat transfer devices and widely utilized in thermal management of power plants and electronics. The hybrid mesh-groove wick promises to attain a higher thermal performance of the heat pipe by balancing the permeability and capillarity. However, traditional fully hybrid mesh-groove wick presents considerable condensation thermal resistance due to the condensed quiescent working fluid and thick, saturated wick.

In this study, a novel partially hybrid mesh-groove wick has been proposed to enhance the evaporation of L-shaped copper-ethanol heat pipes. L-shaped heat pipe promotes high-efficient draining of condensed liquid by gravity, while traditional straight-shaped heat pipe encounters unwanted flooding in the horizontal condensation section. The experiment has been conducted to optimize the coverage length of the hybrid mesh-groove wick, including grooved wick, partially hybrid mesh-groove wick, and fully hybrid mesh-groove wick. The results demonstrate that the partially hybrid wick substantially outperforms the grooved wick and fully hybrid wick. The highly efficient capillary evaporation enabled by the mesh-groove hybrid wick is the main factor. However, such hybrid wick cannot be applied in the condensation section owing to thick liquid film confined by the hybrid wick.

Besides, this study has also investigated the effect of mesh-layers number and charging ratio on the thermal performance of the L-shaped partially hybrid mesh-groove wicked heat pipe (LPHHP). Furthermore, a regression analysis method in machine
learning has been explored to optimize the charging ratio of the LPHHP. The optimal charging ratio is in a region influenced by the heat load rather than a constant value as a traditional view.

Also, the 3D capillary flowing process of both groove and hybrid wick has been simulated using the CFD method. Simulation results present that the capillary filling process can be significantly enhanced by the mesh-groove hybrid wick in comparison with the groove wick. The enhanced mechanism of the hybrid wick is an extra meniscus on the upper wire surface, resulting in higher capillary pressure. Lastly, a bioinspired contracting mesh-groove hybrid wick has been proposed for the fastest capillary filling. The channel width ratio has been optimized using the CFD method. The optimal width ratio is demonstrated to be 0.5, with a normalized time of 78.2% compared to the straight-channel hybrid wick.
TABLE OF CONTENTS

DEDICATION .................................................................................................................. iii
ACKNOWLEDGEMENTS ............................................................................................... iv
ABSTRACT ...................................................................................................................... vi
LIST OF TABLES ............................................................................................................. ix
LIST OF FIGURES .......................................................................................................... x
LIST OF SYMBOLS ........................................................................................................... xiv
LIST OF ABBREVIATIONS .............................................................................................. xvii

CHAPTER 1 INTRODUCTION ......................................................................................... 1
  1.1 Fundamentals of a heat pipe ..................................................................................... 2
  1.2 The wicking structure of heat pipes .......................................................................... 5
  1.3 The literature review of hybrid wick of heat pipes ....................................................... 7
  1.4 The hybrid mesh-groove wicks of heat pipes ............................................................... 13
  1.5 The optimizing study of heat pipe parameters ........................................................... 16

CHAPTER 2 EFFECT OF HYBRID WICK COVERING LENGTH ON THE THERMAL PERFORMANCE OF THE L-SHAPED HEAT PIPES ...................................................... 19
  2.1 Development of L-shaped heat pipes with hybrid wicks .............................................. 19
  2.2 Experimental facility and methodology ...................................................................... 24
  2.3 Data reduction and uncertainty analysis ....................................................................... 30
  2.4 Result and discussion .................................................................................................. 33
  2.5 Conclusion .................................................................................................................. 37
CHAPTER 3 THE OPTIMAL MESH-LAYERS NUMBER AND CHARGING RATIO OF THE L-SHAPED HEAT PIPES WITH PARTIALLY-HYBRID MESH-GROOVE WICKING STRUCTURE ............... 38

3.1 Experimental methodology and data analysis ................................................................. 38

3.2 Results and discussion .................................................................................................. 41

3.3 Conclusion .................................................................................................................... 49

CHAPTER 4 THE OPTIMAL DESIGN OF GROOVE FOR CAPILLARY FLOWING AND ITS MECHAMISM ANALYSIS BY SIMULATION METHOD ....................... 51

4.1 The introduction of theory for capillary-flowing simulation in literature ........ 52

4.2 The selected models for capillary filling in this study ................................................. 56

4.3 Validation of simulation by analytical modeling ......................................................... 60

4.4 Model setup for capillary filling of grooves in this study .......................................... 63

4.5 The effect of contact angle on the capillary filling of groove ................................. 65

4.6 The gravitational effect on the capillary filling of groove ........................................ 67

4.7 The optimal groove shape for the fastest capillary filling .................................. 71

4.8 Conclusion .................................................................................................................. 76

CHAPTER 5 THE ENHANCED CAPILLARY PERFORMANCE OF MESH-GROOVE HYBRID WICK AND THE OPTIMIZATION OF CONTRACTING HYBRID WICK ...... 77

5.1 The enhanced capillary filling of mesh-groove hybrid wick compared to groove ................................................................................................................. 77

5.2 Optimizing the hybrid wick with a bioinspired contracting channel for the fastest capillary filling ...................................................................................... 81

5.3 Conclusion .................................................................................................................. 90

CHAPTER 6 CONCLUSIONS AND FUTURE WORKS ............................................ 92

REFERENCES ................................................................................................................. 97
LIST OF TABLES

Table 2.1 Main parameters of the L-shaped heat pipe ........................................... 21

Table 2.2 Specific parameters of groove and mesh of hybrid wick ............................. 23

Table 3.1 The charging amounts of the L-shaped partially hybrid wicked heat pipes ................................................................. 40

Table 4.1 The Bond numbers with various liquid heights and gravities ......................... 69
LIST OF FIGURES

Figure 1.1 Schematic of the working mechanisms of a heat pipe [7]................................. 3

Figure 1.2 Wicking structures of heat pipes: (a) micro grooves [23], (b) sintered powders [22], (c) woven screen mesh [24], and (d) sintered fibers [25] ....................... 6

Figure 1.3 The hybrid wicking structures of heat pipes. (a) The grooved porous structure, (b) the combined grooves with sintered mesh, (c) the combined grooves with a sintered powder, and (d) the bi-porous wicking structure.................. 8

Figure 1.4 Strip-shaped porous wick for flat plate heat pipe........................................... 10

Figure 2.1 Schematic of three configurations of wicks in L-shaped heat pipes: (a) grooved wick, (b) fully hybrid wick in both the evaporation and the condensation region, and (c) partially hybrid wick only in the evaporation region. ................................................................. 20

Figure 2.2 Schematic of liquid film distributions on the grooved wick and hybrid mesh-groove wick as evaporator and condenser, respectively. (a) The grooved wick with partial dry-out and the hybrid wick with fully saturated liquid in the horizontal evaporation region. (b) The highly efficient liquid draining of grooved wick and confined liquid film of hybrid wick in the vertical condensation region.......... 22

Figure 2.3 The top-view SEM image of the hybrid wicks developed in this study. (a) SEM image of microchannels. (b) Hybrid wick with one-layer mesh. (c) Hybrid wick with double-layer mesh. (d) Intermeshing between individual wires. (e) The formed sintered neck is connecting wire and fin tip. (f) Zooming in of double-layers mesh ............... 24

Figure 2.4 The experimental and theoretical liquid volumes of the saturated hybrid wick as a function of mesh-layers number................................. 26

Figure 2.5 Schematic of the L-shaped heat pipe testing setup in this study. ................. 28

Figure 2.6 Thermocouple arrangement on the L-shaped heat pipe surface for characterization. ........................................................................................................... 30

Figure 2.7 The heat transfer coefficient ($h_{in}$) of heat loss as a function of the temperature difference between the thermal insulation surface temperature and ambient air temperature ........................................ 32
Figure 2.8 The steady-state wall temperature distributions of three heat pipes with various coverage lengths of hybrid wick at the heat loads of (a) 60 W, (b) 80 W, and (c) 100 W. (d) Comparison of thermal resistances between the hybrid wick heat pipes and grooved heat pipe.

Figure 3.1 The SEM images of the hybrid mesh-groove wicks with (a) one, (b) two, (c) three, and (d) six layers of mesh in cross-sectional views.

Figure 3.2 The temperature distributions of the LPHHPs with various mesh-layers numbers at the heat loads of (a) 40 W and (b) 110 W. The effective thermal conductivity of the LPHHPs at the heat load ranges of (c) 10-70 W and (d) 80-140 W.

Figure 3.3 Effect of charging ratio on the thermal resistance of the LPHHPs with (a) one, (b) two, and (c) three layers of mesh. The charging ratio ranged from 50% to 300% of the void volume of the wick.

Figure 3.4 Coupling effect of charging ratio and heat load on the effective thermal conductivity of the LPHHPs with (a) one, (b) two, and (c) three layers of mesh. (d) The maximum keff achieved by the LPHHPs as a function of charging ratio and heat load.

Figure 3.5 (a) The non-linear regression analysis of the keff using the Random Forest Algorithm of machine learning. (b) A map of the optimal charging ratio region versus heat load.

Figure 4.1 Configuration of a microchannel with a liquid reservoir on the left.

Figure 4.2 The physical model and boundary setting for validation of 2D CFD simulation with an analytical model. The applied CFD software is Comsol 5.5.

Figure 4.3 The validation of the CFD result using the Washburn model.

Figure 4.4 The physical model setting of capillary filling in (a) Groove wick and (b) mesh-groove hybrid wick.

Figure 4.5 The mesh of (a) groove wick and (b) mesh-groove hybrid wick.

Figure 4.6 The filling length and liquid front shape comparison of grooves with various contact angles (0°, 30°, 60°, and 90°) at the time of (a) 0.5 ms, (b) 1 ms, and (c) 1.5 ms.

Figure 4.7 The (a) pressure and (b) velocity distributions of capillary filling on grooves with various contact angles of 0°, 30°, and 60°. The cut plane is at the bottom and the results are at the moment when the liquid front reaches 0.5 mm.
Figure 4.8 The filling length and liquid front shape of grooves under various gravities of -10g, 0g, 5g, and 10g. The selected times are (a) 1 ms, (b) 3 ms, and (c) 5 ms.............................................................. 70

Figure 4.9 The (a) pressure and (b) velocity distributions of capillary filling on grooves with various gravities of -10g, 0g, 5g, and 10g. The cut plane is at the bottom and the results are at the moment when the liquid front reaches 0.3 mm. .............................................................. 70

Figure 4.10 The dimensions and the hydraulic diameter of the grooves with various shapes .................................................................................................................. 72

Figure 4.11 The filling length and liquid front shape of grooves with various shapes of triangle, trapezoid, rectangle, and dovetail slot. The selected times are (a) 0.3 ms, (b) 0.6 ms, and (c) 0.9 ms.......................................................... 74

Figure 4.12 The (a) pressure distribution and (b) velocity distribution of flowing liquid in grooves with various shapes of triangle, trapezoid, rectangle, and dovetail slot. ........................................................................ 75

Figure 4.13 The pressure and velocity distribution of grooves with various shapes at the filling time of 0.6 ms. .................................................................................. 75

Figure 5.1 The (a) three-dimensional view, boundary conditions setting, and (b) generated mesh of the mesh-groove hybrid wick. The structure and significant parameters are presented. The fluid flows from left to right along the Z-axis. ........................................................................... 78

Figure 5.2 The enhanced capillary flow of hybrid wicks (with contact angles of 0°, 30°, and 60°) compared to the grooved wick (with contact angles of 60°). The selected times are (a) 0.5 ms, (b) 1 ms, and (c) 1.5 ms. ............. 79

Figure 5.3 The liquid front shape of (a) hybrid wick and (b) groove at the contact angle of 60°. Extra menisciuses are induced by the wires, resulting in improved driven pressure.................................................................. 80

Figure 5.4 Pressure distribution of (a) hybrid wick and (b) groove wick. The cross-section is parallel to the Z-axis and perpendicular to the bottom wall..... 81

Figure 5.5 The contracting capillary-radius structure inspired by nature. (a) The contracting branches structure for capillary-evaporation of leaves in nature. (b) The fastest capillary transportation is achieved by reducing pore radius [102]. ....................................................................................... 82

Figure 5.6 The 3D structures of contracting hybrid wicks with various width ratios ranging from 0.1 to 2. ....................................................................................... 83
Figure 5.7 The filling length and liquid front comparison of hybrid wicks at various width ratios (m) ranging from 0.1 to 2. The selected times are (a) 0.6 ms, (b) 1 ms, (c) 1.4 ms, (d) 1.8 ms, (e) 2.2 ms, and (f) 2.6 ms. ................. 85

Figure 5.8 The filling length-time curves of hybrid wicks with various m value. ........ 86

Figure 5.9 The moving-stop of liquid at capillary filling of hybrid wick due to absorption of liquid into micro-hole ................................................................. 86

Figure 5.10 The normalized filling times of hybrid wick versus the width ratio (m). .......................................................... 87

Figure 5.11 Comparison of pressure field of hybrid wicks with various width ratios at the filling length of 0.5 mm. .................................................. 88

Figure 5.12 Comparison of pressure field of hybrid wicks with various width ratios at the filling length of 0.7 mm. .................................................. 89

Figure 5.13 Comparison of the velocity field of hybrid wicks with various width ratios at (a) uniform scale and (b) non-uniform scale.............................. 89

Figure 6.1 The hybrid inter-connected groove/mesh wick for capillary evaporation under point heat sources......................................................... 96
LIST OF SYMBOLS

$A$  Area, $m^2$

$A_p$  The cross-sectional area of the pipe, $m^2$

$Bo$  Bond number, dimensionless

$Ca$  Capillary number, dimensionless

$d$  Wire diameter, m

$D_h$  Hydraulic diameter, m

$F_s$  Volumetric force, $N/m^3$

$g$  Gravity, $m/s^2$

$h$  Heat transfer coefficient, $W/(m^2\cdot K)$

$H$  Height, m

$k$  Effective thermal conductivity, $W/(m\cdot K)$

$l$  Length of liquid, m

$L$  Length, m

$m$  Channel width ratio, dimensionless

$M$  Mesh number, dimensionless

$n$  Interface normal, dimensionless

$p$  Pressure, Pa
$P$  Perimeter, m

$Q$  Heat load, W

$r$  Capillary radius, m

$R$  Thermal resistance, °C/W

$t$  Time, s

$T$  Temperature, °C

$V$  The volume of liquid, m$^3$

$V$  Velocity of the mixture, m/s

$u$  Flow velocity, m/s

$V_{ws}$  The volume of working fluid required to saturate the wick, m$^3$

$w$  Width, m

$x$  Wetting length, m

Greek symbols

$\alpha$  Volume fraction, dimensionless

$\varepsilon$  Porosity, dimensionless

$\delta$  Mesh thickness, m

$\phi$  Charging ratio, dimensionless

$\rho$  Density, kg/m$^3$

$\mu$  Viscosity, N.s/m$^2$

$\sigma$  Surface tension, N/m
θ Contact angle, °
**LIST OF ABBREVIATIONS**

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<th>Abbreviation</th>
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<tr>
<td>A</td>
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<td>a</td>
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<td>Condense</td>
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<td>The lower point of evaporation</td>
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<td>Effective</td>
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<td>G</td>
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<td>Real power</td>
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<td>s</td>
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CHAPTER 1 INTRODUCTION

Heat pipes are widely utilized in the application of thermal conversion and management, including aerospace engineering, thermal power plants, electronic device cooling, and solar collectors, etc. For example, more than one hundred heat pipes are required for a telecommunication satellite [1]. High heat transfer rates are achieved by heat pipes over a long distance, with exceptional flexibility, easy control, simple fabrication, and minimal temperature difference, not to mention, all of this without any external driving power applied.

There are various designs of heat pipe, including rotating, pulsating, sorption, loop, thermal diodes, non-circular shaped, and variable conductance heat pipes [2]. By dimension, heat pipes can be classified to be a miniature heat pipe or micro heat pipe according to the dimensional magnitude of hydraulic diameter. Firstly, the micro heat pipe concept was firstly proposed by Cotter [3] for the application of electronic devices cooling. The micro heat pipe is generally characterized as a heat pipe unit in which the average curvature of the liquid-vapor interface is in the same magnitude to the reciprocal of the wicking-channel hydraulic radius[4]. Typically, a micro heat pipe has convex and cusped cross-sections (for example, a polygon shape) with a hydraulic diameter in a range of 10–500 μm [5]. Secondly,
a miniature heat pipe is classified as a heat pipe with a flowing-channel hydraulic diameter in a range of 0.5 to 5 mm [6]. Miniature heat pipes can be designed and fabricated based on 1D capillary structure (axially grooved wick), 2D capillary structure (interconnected cross grooves), or wire meshes configuration. However, the concepts of miniature and micro heat pipes are not always correctly indicated in the current open literature. For instance, miniature heat pipes with micro-grooves structure are sometimes referred to micro heat pipes in an improper way [7].

Many different types of novel heat pipes are developed rapidly in recent years to address thermal management issues of electronics [8-10], solar energy [11-13], and lots of other applications [10, 14, 15]. Astrogated heat pipe development is motivated to meet the needs of microelectronic components in space, such as compression in a limited space, high density of power, and lightweight. According to the NASA annual report [16], reducing one pound of a spacecraft weight is able to help save the launch costs up to $10,000 US dollars.

1.1 Fundamentals of a heat pipe

A heat pipe is known as an efficient two-phase heat transfer device with excellent temperature uniformity and high reliability [17, 18]. The operation mechanism of a heat pipe [7, 19] can simply be explained based on a wicked cylindrical geometry with working fluid inside, as shown in Figure 1.1. However, the shape, material, and dimension of the heat pipes can be various in a wide range. Heat pipes consist of a closed chamber (shell and end
caps in two sides), a wicking structure, and charged working liquid circulating in an equilibrium state with evaporated vapor. The overall heat pipe can be divided into three sections based on heat flow direction: the evaporator section, the adiabatic section and the condenser section. Noted that some heat pipes have no adiabatic section and could have multiple evaporation regions or condensation regions depending on specific designs and applications.

![Diagram of heat pipe](image)

Figure 1.1 Schematic of the working mechanisms of a heat pipe [7].

The heat transfer path and circulation of a heat pipe can be described as follows. The heat that is implemented to the outside shell of the evaporation section is transferred through the wall of heat pipe by conduction first and then the wicking structure. At the interface of the wicking structure and vapor region, working fluid evaporate to vapor and diffuses to the vapor core, at which the pressure will be increased. The arisen vapor pressure on the
evaporation side is the only force to drive the vapor from the evaporation section to the condensation region. Then, the vapor condenses to liquid phase and flow back to the evaporation region, releasing large amount of latent heat. On the other hand, the condensed liquid in condensation region will flow back to the evaporation region by the driving capillary pressure induced by various menisci along the wicking structure. With this circulating loop, the heat pipe is able to continuously transfer the evaporation/condensation latent heat back and forth between the evaporation region and the condensation region.

The capillary pressure is the major driving force to circulate the liquid from the condensation section to the evaporation section. The menisci at the liquid-vapor interface are significantly curved in the evaporator region because the liquid is receding into the wicking structures during evaporation. On the other hand, the menisci are almost flat in the condensation region, which indicates high hydraulic pressure. The capillary pressure along the length of heat pipes is induced by the changing curvature of menisci in the wick along the heat pipe. This capillary pressure gradient drives the working fluid against the pressure losses of liquid or vapor, and the volumetric forces such as acceleration or gravity.

The flowing friction is the main reason causing the pressure drop along the wicking structure [7]. However, in more complicated wicking structures, i.e., sintered mesh and fibers, the location of both the vapor and liquid is difficult to be tested experimentally or
predicted theoretically (using Darcy’s law). Transparent heat pipes are proposed and designed to visualize the flowing motion and phenomenon [20].

1.2 The wicking structure of heat pipes

The heat pipe thermal performance is primarily affected by wicking structures, which determine the working fluid renewal efficiency. The wicking structures are the core components of a heat pipe with two-phase flow, which provide the capillary pressure to drive the circulation loop of the working fluid and the liquid-vapor interface for phase changes. The thermal performance of the heat pipes, including the start-up response, isothermal performance, and the heat transfer limit, mainly rely on the wicking structures. Various types of wicking structures have been proposed to enhance the heat pipes thermal performance. The most typical wick types are parallel grooves, sintered meshes, metal powders, and fibers [19], as shown in Figure 1.2. A powder wick, mesh wick, and fiber wick are classified as sintered wicks, which have excellent anti-gravity capability, large capillary forces, and low cost. Sintered-powders wicking structures are applied to approximately 80% of conventional heat pipes according to statistics [21]. The permeability and capillary pressure of a wick are two key properties for heat pipes, which determine the capillary limit and thermal resistance of heat pipes [22]. The capillary pressure and permeability of various wicking structures are substantially different.
Microgroove wicking structures are effective for heat pipe because of the lighter weight, lower thermal resistance and high permeability. The types of grooves reported in open literature vary from rectangular [28], trapezoidal [29], “Ω” shaped grooves [30], triangular [26] and V-shaped [27]. At present, several techniques have been utilized to fabricate micro-grooves, including the ploughing–extrusion process [32-34], laser micromachining [38], electro-discharge machining [35], high-speed spin-forming process [31], and the plasma etching [36, 37]. Several studies [39-41] have reported that groove alignment is the critical factor to achieve higher heat flux before dry-out.

The sintered powder wicks are normally mono-porous wick, which will produce high hydraulic resistance at high heat flux, causing dry-out in the evaporate section. The best mono-porous powder wicks have a critical heat flux at 300 W/cm² [42]. Besides, the sintering quality of powder wick is affected by two key parameters, i.e., sintered temperature and atmosphere. During sintering, the temperature should be set between the fully melting point and the value of half melting point. Better mechanical strength of sintered wick could be achieved by higher sintering temperature. The sintering atmospheres of various materials
make a big difference. For example, a reducing atmosphere is needed for sintering nickel powders to restore the particles surface [43].

The flexible sintered fibers and meshes have been widely used in thin flat plate heat pipes [44] in actual production. Firstly, the performance of porous mesh can be enhanced by surface treatment. For example, the capillary and thermal performance of a copper mesh wick was improved by the process of chemical deposition and sintering [45]. The capillary force of mesh was also significantly enhanced by oxidization, as reported by Yang et al. [46]. A composite braided wire wicking structure, which was composed of two wires with different diameters, was oxidized to promote the capillary performance. Secondly, sintered fibers are also typically used in a flat plate heat pipe, as firstly proposed by Ahamed et al. [47] in 2011. Copper fibers can be developed to provide energy-efficient and space-limited thermal management solutions for electronic devices with high power density [48].

1.3 The literature review of hybrid wick of heat pipes

The wicking structures exhibiting both high liquid permeability and larger capillary force are required for heat pipe with good thermal performance. The grooved wicks have higher permeability but lower capillarity, while the sintered powders or meshes can generate higher capillary pressure but have lower permeability [49]. Thus, a hybrid wicking structure, which consists of two or more types of single wicking structure, was proposed to well balance the high capillary pressure and high permeability. At
present, there are four typical types of composite wicking structures utilized in heat pipes, i.e., the grooved-shape porous structure (Figure 1.3 (a)), the combination of grooves and sintered powder/mesh/fiber (Figure 1.3 (b) and (c)), the combination of two porous structures, and the bi-porous wicking structure (Figure 1.3 (d)).

![Figure 1.3](image)

Figure 1.3 The hybrid wicking structures of heat pipes. (a) The grooved porous structure, (b) the combined grooves with sintered mesh, (c) the combined grooves with a sintered powder, and (d) the bi-porous wicking structure.

**1.3.1 The grooved shaped porous wicks**

The groove-shaped porous wicking structures have been investigated in recent years. Popova et al. [50, 51] developed an innovative wicking structure, in which multiple parallel rectangular micro-channels were fabricated by machining on the surface
of the porous sintered copper powders. The thermal performance of the flat-plate heat pipe with this wick was tested experimentally under various working conditions, and the results presented that the flat heat pipe achieved a higher thermal performance than that of a flat copper substrate. Kamenova et al. [52] designed and fabricated a wicking structure by sintering the commercial copper powder on one side of the heat-pipe shells, and grooves with width of 0.5 mm were machined on the surface of the porous wick to provide vapor passage. Results showed that the maximum heat transfer rate of heat pipe with this wick could be up to approximately 52 W under the optimal water charging amount of ~340 µl, and the gravity does not make a big difference on its functioning. Ponnappan et al. [53] developed and tested a groove-shaped screen wick in a miniature heat pipe with enhanced thermal performance and ease of fabrication.

Li and Lv [54, 55] fabricated and investigated a strip-shaped porous wicking structure, which was fabricated by sintered copper powder with diameters between 50 and 100 µm, as presented in Figure 1.4. Results showed that a heat load of 120 W can be dissipated by the heat pipe using the wick in a horizontal level with a total thermal resistance of 0.196 °C/W. In addition, the effective thermal conductivity of the proposed flat-plate heat pipe was higher than 4 times that of a copper plate under natural convection conditions. As reported by the same research group in a subsequent study [56], a wicking structure was formed by sintering five staggered super-hydrophilic copper
meshes. Rectangular-shaped channels were machined on the surface of the entire mesh screen using electrode cutting and function as the vapor passages. It was demonstrated that a high heat flux of 490 W/cm² can be managed by this heat pipe with the proposed wick.

![Image of strip-shaped porous wick for flat plate heat pipe.](image)

**Figure 1.4** Strip-shaped porous wick for flat plate heat pipe.

### 1.3.2 The grooved/sintered porous wicks

The combined grooves with sintered porous wick have been applied to enhance heat pipe performance. The fiber-groove hybrid wick and powder-groove hybrid wick have shown significant enhancement on both capillary performance and heat transfer rate. Firstly, for the fiber-groove hybrid wick, Singh et al. [57] developed a hybrid fiber wicking structure on a flat miniature heat pipe applied in thermal management of handheld mobile device. The hybrid fiber structure was fabricated by sintering copper fibers on the axial grooves along the inner shell of the heat pipe. In comparison with the conventional wicking structures, the proposed wick can provide an optimal combination
of capillary force and permeability, achieving larger heat transfer capability and minimal axial temperature gradient. At an heat load of 5 W, a total thermal resistance of 4.5 °C/W was achieved by the proposed heat pipe.

Besides, R. Singh et al. developed a hybrid fiber-groove wicking structure by sintering copper fibers onto the groove surface along the axial direction of the heat pipe [57]. The thermal performance of heat pipes with grooves wick and meshes wick was also investigated for comparison. Results showed that the thermal resistance of the hybrid fiber wick (0.44 K/W) is much lower than those of the groove wick (4.5 K/W) and the screen mesh (1.09 K/W).

Secondly, for the powder-groove hybrid wick, Li et al. [58] developed a novel bilateral sintered arch-shaped grooved-powder wicking structure (BSGW) and a single sintered arch-shaped grooved-powder wick (SSGW). The results indicated that the maximum heat transport capacity of the thin flat plate heat pipe with SSGW and BSGW are 12 W and 13 W, respectively. Yong Li et al. [23] proposed a mathematical model of heat pipe with a sintered powder-grooved hybrid wick. The theoretical results have been compared with experiments. The results show that either boiling heat transfer or conduction are able to occur in the evaporator, while only conduction occurs in the condenser. The total thermal resistance of the hybrid-wick heat pipe varies from 0.02 to 0.56 K/W. Daxiang Deng et al. [49] developed a hybrid wick by combining micro
V-shaped grooves and copper powder by means of sintering. The hybrid wick was experimentally examined and compared with normal grooved and sintered wicks. Experimental results presented that the composite wicks in this study enhanced both the capillary performance and the permeability compared to the mono-sintered wicks. Moreover, the composite wick presented much larger capillary force than that of the grooved wicks. In addition, Wang and Catton [59] developed a hybrid wicking structure by covering a thin porous material on the surface of triangular groove. Theoretical analysis results presented that it not only increased the evaporating surface area but also promoted the capillary pressure. Thus, the hybrid wick significantly enhanced the evaporative heat transfer compared to the groove wick without porous layers.

1.3.3 The combination of two types of porous wicks

The hybrid wick can also be formed by combining two types of porous structures with various pore sizes. Zhou et al.[60] fabricated a novel composite wicking structure, i.e., sintered mesh-copper foam wick, which was fabricated by sintering copper mesh onto the copper foam. A maximum heat transfer limit of 5 W was demonstrated by the experimental result. In addition, Franchi and Huang [61] developed and investigated composite wicks fabricated by sintering fine metal powders onto coarse-pore copper mesh. The homogeneous wicking structures significantly enhanced the thermal performance of the heat pipes.
1.3.4 The bi-porous wicks

The bi-porous structure has two distinct pore sizes and they are separated from each other. The bi-porous wick includes two types: the first one is called bi-disperse structure which consists of small-particles clusters, and the second one is made of large particles coated with small holes or pores on the surface. A bi-porous wick is able to overcome issues caused by high heat flux challenges [62]. This wick can achieve better capillary limit or heat transfer capacity whereby the finer pores enhanced capillary pressure, while the larger pores effectively reduce the liquid flowing resistance on transportation. Semenic and Catton [42] demonstrated that the biporous wicking structure achieve a higher heat transfer limit or critical heat flux than the mono-porous wicking structures. This is because they can form evaporative menisci both on top of the wick surface and inside the wicking structure. Specifically, the heat transfer coefficient of evaporation of the bi-porous wick achieve a maximum value of 64,000 W/m²K, which is approximately six times that of the mono-porous wick [63].

1.4 The hybrid mesh-groove wicks of heat pipes

Among the grooved-sintered wicks, the hybrid mesh-groove wick is promising to attain a higher capillary heat transfer limit with lower fabrication cost, as verified by Dai et al. [64]. The critical heat flux of hybrid mesh-groove micromembrane was substantially enhanced up to 198% compared to mono-porous wick. Extensive researches
have been conducted to enhance the heat pipe thermal performance by hybrid mesh-groove wick. The technology of jointing mesh and groove includes physical compression [20] [65], electroplating [66, 67], and thermal diffusion by sintering [68]. Frédéric Lefèvre et al. [20] investigated a flat plate heat pipe with a composite mesh-groove wick, of which the parallel rectangular grooved were covered by physical plating rather than sintering. Unexpectedly, this type of hybrid wick structure worked not well in a tilted position. J.C. Hsieh et al. [65] designed a coronary-stent-like model to compact copper mesh on a micro-rectangular groove to form the hybrid wick. Experimental results showed that the heat transfer limit was enhanced by 25% using a single-layer mesh-groove hybrid wick compared to a pure groove wick. However, the evaporative heat transfer coefficient was degraded with increasing number of mesh layers. Oshman et al. [66] developed a hybrid mesh-micropillar wick to enhance a microscale flat polymer-based heat pipe. The maximum heat flux of this flexible heat pipe is as high as 11.94 W/cm². By reducing the micropillar gap from 200 µm to 31 µm, Oshman et al.[67] demonstrated that the hybrid wick with promoted capillarity could improve the thermal performance of a flat heat pipe under high adverse acceleration up to 10 g. S.-C. Wong and W.-S. Liao[68] operated visualization experiments on hybrid copper mesh-groove wicks of flat-plate heat pipes. One layer of 200-mesh was sintered on top of the semi-circular grooves. Results showed that the maximum heat load of the hybrid wick was 60 W, far overcoming those of 2 layers-mesh wick (25 W) and grooved wick (14 W).
In summary, all of the hybrid mesh-groove wicks in current researches fully covered both the evaporation and condensation region of the heat pipe.

However, hybrid mesh-groove wick presented large thermal resistance on the condensation section of a heat pipe, despite showing considerable enhancement on evaporation[64]. For example, for a hybrid one-layer mesh-groove wick, the heat transfer coefficient (HTC) as condenser was merely 1,000-2,600 W/m²·K [69], lagging far behind the HTC as an evaporator of 10,000-25,000 W/m²·K. In addition, for a flat heat pipe with hybrid two-layers mesh-groove wick, the thermal resistance in condensation ($R_c=0.7~1$ K/W) was more than 3.5 times that in evaporation ($R_e=\sim0.2$ K/W) [68]. A similar ratio of $R_c$ to $R_e$ was also reported to be more than 2.5 times in a screen mesh wicked heat pipe [70]. The large thermal resistance of condensation resulted from the confined liquid film induced by the high capillarity of the hybrid wick. However, hybrid mesh-groove wicks were still applied in the condensation region of heat pipes in the existing researches [20] [65] [66] [68].

Fully hybrid wicked heat pipes have been investigated, while partially hybrid wicked heat pipe has not been reported. In this study, an L-shaped partially hybrid mesh-groove wicked heat pipe (LPHHP) has been proposed to promote the condensate draining by the inclined grooved wick. The partially-hybrid mesh-groove wick needs to be
better studied in order to improve the heat pipe thermal performance in many extremely harsh conditions.

1.5 The optimizing study of heat pipe parameters

1.5.1 Study of wick thickness

A wick with the optimal thickness can well balance the flowing resistance and thermal resistance. On the one hand, a thicker wick can increase the throughput of the returning condensate liquid from the condensation section. At the same time, the return-flow pressure drop can be reduced. On the other hand, the thick saturated wick will induce extra thermal resistance of the condensation section between the wall and the vapor. Hwang et al. [71] investigated the mechanism of how modulation of wick thickness enhances extra evaporative surface area and axial liquid flow by capillary pressure. The uniform wick is covered by periodic stacks and grooves. Theoretically, the optimal thickness of the modulated wick is obtained in a close-form expressions with the condition of viscous-flow regime. Similar numerical results are also obtained for the viscous-flow regime, which indicates good consistence.

The number of mesh layers significantly affects the thermal performance of the heat pipe integrated with a hybrid mesh-grooved wick. The optimal values of the mesh-layers number of the partially hybrid wicked heat pipe (PHHP) are different from
those of fully hybrid wicked heat pipe (FHHP). The optimal mesh-layers number of the PHHP is primarily determined by the evaporative thermal resistance, while that of the FHHP is evaluated by the overall thermal resistance \([65]\), a sum of both evaporation and condensation thermal resistance. This study will investigate the optimal mesh-layers number of the PHHP since there are no reference data in existing studies.

1.5.2 Optimizing study of charging ratio

The charging ratio significantly affects the performance of the LPHHP. However, there are few studies on the charging ratio of the PHHPs. The optimal values of the charging ratio of the PHHPs are different from those of a fully hybrid wicked heat pipe. The optimal charging ratios of the FHHPs and grooved heat pipes were conflicting, and thus cannot be applied in a PHHP. For example, the optimal filling ratio of a flat heat pipe with fully hybrid sintered copper foam-mesh wick is reported to be 100\% by Zhou et al. [60]. Besides, the optimal charging ratios of other fully porous wicked heat pipes were reported to be 75\% [72], 84\% [73], and 150\% [74] of the wick volume. In contrast, the optimal charging ratio of a grooved heat pipe was reported to be as small as 25\% [75]. When combing the grooved wick and the porous hybrid wick to form a PHHP, the above charging ratio values are no longer applicable. It is necessary to re-examine the optimal charging ratio of the PHHP.
Except for the experimental method, the algorithm method can be applied to optimize the parameters of the heat pipe. Zhang et al. [30] conducted a multi-objective optimization on heat pipe design using a Pareto genetic algorithm. This artificial intelligence method avoids solving the Complicated internal relationships between the heat transfer performance and the heat pipe’s geometric structure. In this study, a machine learning algorithm will be applied to optimize the charging ratio of heat pipe based on experimental data.
CHAPTER 2 EFFECT OF HYBRID WICK COVERING LENGTH ON THE THERMAL PERFORMANCE OF THE L-SHAPED HEAT PIPES

This study firstly proposed a novel L-shaped partially hybrid mesh-groove wicked heat pipe (LPHHP). The evaporation region is covered with hybrid wick placed horizontally, while the condensation region is covered with grooved wick placed vertically. In this section, L-shaped heat pipes with three different mesh coverage lengths, i.e., zero (grooved wick), fully covered wick, and partially covered wick (only in evaporation), will be developed to evaluate the effectiveness of partially hybrid wick.

2.1 Development of L-shaped heat pipes with hybrid wicks

2.1.1 L-shaped heat pipes with three configurations and its mechanisms

To validate the effectiveness of partially hybrid wick in L-shaped heat pipes, three configurations with various coverage lengths were fabricated: groove (no coverage), fully hybrid wick (covering both the evaporation and the condensation region), and partially hybrid wick (only in the evaporation region). The schematics of the three configurations are shown in Figure 2.1.
Figure 2.1 Schematic of three configurations of wicks in L-shaped heat pipes: (a) grooved wick, (b) fully hybrid wick in both the evaporation and the condensation region, and (c) partially hybrid wick only in the evaporation region.

Figure 2.2 presents the liquid film distributions of the grooved wick and hybrid wick as evaporator and condenser, respectively. It illustrates that the hybrid wick effectively enhances the capillary evaporation, while the vertical grooved wick facilitates condensation heat transfer. High evaporation HTC can be achieved by capillary thin-film evaporation [64], and a high heat transfer limit ($Q_{\text{max}}$) can be achieved by liquid passage of hybrid wick [68].

Moreover, high condensation HTC can be realized by the vertical positioned grooved wick with highly efficient liquid draining [76]. Inclined grooved wick with gravity-assistance returning of working fluid presented a better performance for condensation than the porous hybrid mesh-groove wick. The condensation HTC of grooved wick placed in a vertical position, ranging from 5,000 to 9,000 W/m$^2$·K[77], was
at least three times larger than that of hybrid wick ranging from 1,000 to 2,600 W/m²·K [69]. On the grooved wick surface, gravity results in highly effective draining of condensed working fluid, rather than fluid retention induced by the capillarity of the porous hybrid wick. Besides, placing the condensation section of heat pipe with an inclined angle is actually feasible for many electronic devices, such as U shaped heat pipe[78, 79] and inverted T shaped heat pipe[80].

Table 2.1 Main parameters of the L-shaped heat pipe

<table>
<thead>
<tr>
<th>Items</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>The total length of the pipe</td>
<td>500 mm</td>
</tr>
<tr>
<td>Evaporation length</td>
<td>150 mm</td>
</tr>
<tr>
<td>Adiabatic length</td>
<td>190 mm</td>
</tr>
<tr>
<td>Condensation length</td>
<td>160 mm</td>
</tr>
<tr>
<td>Bending radius of L shape</td>
<td>60 mm</td>
</tr>
<tr>
<td>Length of the horizontal straight section</td>
<td>190 mm</td>
</tr>
<tr>
<td>Outer diameter</td>
<td>12.7 mm</td>
</tr>
<tr>
<td>Wall thickness</td>
<td>0.8 mm</td>
</tr>
<tr>
<td>Number of grooves</td>
<td>75</td>
</tr>
<tr>
<td>Working fluid</td>
<td>Ethanol</td>
</tr>
</tbody>
</table>

In this validating experiment, hybrid wicks were sintered with one-layer mesh, since one-layer was reported to show the minimum thermal resistance among one, two, and three layers [69]. The specific dimensions of the L-shaped heat pipes are listed in Table 2.1 and shown in Figure 2.1.
Figure 2.2 Schematic of liquid film distributions on the grooved wick and hybrid mesh-groove wick as evaporator and condenser, respectively. (a) The grooved wick with partial dry-out and the hybrid wick with fully saturated liquid in the horizontal evaporation region. (b) The highly efficient liquid draining of grooved wick and confined liquid film of hybrid wick in the vertical condensation region.

2.1.2 Fabrication of copper hybrid mesh-groove wicks

The micro-grooves were fabricated by a mechanical process named oil-filled high-speed spin forming [81]. The grooves have a trapezoidal cross-sectional shape with a side inclination angle of 65°. The specific dimensions of grooves are listed in Table 2.2. The hybrid mesh-groove wicks were sintered with 145-mesh (with a wire diameter of 56 μm) under compression provided by a set of semi-circular sliding-opening modules.
Table 2.2 Specific parameters of groove and mesh of hybrid wick

<table>
<thead>
<tr>
<th>Items</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Groove depth</td>
<td>0.25 mm</td>
</tr>
<tr>
<td>Groove width (bottom)</td>
<td>0.2 mm</td>
</tr>
<tr>
<td>Groove width (top)</td>
<td>0.4 mm</td>
</tr>
<tr>
<td>Fin width (bottom)</td>
<td>0.25 mm</td>
</tr>
<tr>
<td>Fin width (top)</td>
<td>0.05 mm</td>
</tr>
<tr>
<td>Inclination angle of fin side</td>
<td>65 °</td>
</tr>
<tr>
<td>Mesh count</td>
<td>145 inch⁻¹</td>
</tr>
<tr>
<td>Mesh wire diameter</td>
<td>56 μm</td>
</tr>
</tbody>
</table>

Before sintering, the inner-grooved tubes and the mesh screens were ultrasonically cleaned in 95% ethanol and 10% sulfuric acid for 15 minutes each. Then, they were rinsed by acetone, ethanol and DI water for 10 minutes each. The sintering temperature is 1010°C, with a hydrogen/argon protective atmosphere in an oven for 20 minutes. The top-view SEM images of the hybrid wicks are shown in Figure 2.3. As shown in Figure 2.3 (e), the meshes were bonded on the tip of the grooves with highly desired sintering necks [82], which effectively reduce the contact thermal resistance between meshes and grooves.
Figure 2.3 The top-view SEM image of the hybrid wicks developed in this study. (a) SEM image of microchannels. (b) Hybrid wick with one-layer mesh. (c) Hybrid wick with double-layer mesh. (d) Intermeshing between individual wires. (e) The formed sintered neck is connecting wire and fin tip. (f) Zooming in of double-layers mesh.

2.2 Experimental facility and methodology

2.2.1 Obtaining of the liquid amount required to saturate the hybrid wick

An imbibition test was conducted to evaluate the reference value of the optimal charging ratio. At an “ideal” status, the evaporator wicks are fully occupied by liquid [83], which is similar to the heating area sufficiently immersed in fluid in pool boiling. The wicks in the condensation and adiabatic section are also fully occupied by the returning working fluid. The rest of the space in the heat pipe is occupied by vapor. Based on this analysis, the optimal charging ratio is highly related to the amount of fluid required to saturate the wick ($V_{ws}$) in the heat pipe. In this study, the void volume of wick will be obtained by both experimental imbibition test and theoretical calculation using the porosity equations.
The imbibition test was operated on a hybrid sample wick with a dimension of 3×4 cm in an atmospheric environment. We used an injection syringe to drip 99% ethanol drops on the dry surface of the wick. The wick was vertically positioned with dripping droplets until a tiny drop began to fall down the edge of the wick. Then, the wick is sufficiently wet with all air well expelled out of the space. The dripping quantity was obtained by the calibration of the syringe.

The void volumes of wicks were theoretically calculated using empirical equations of porosity. The thicknesses of the hybrid wicks were measured in the SEM picture. The weighted average porosity of the hybrid wick is determined by the groove part and the mesh part. The porosity of the groove is calculated to be 0.66 according to the specific dimensions. The mesh part was predicted by the following equation proposed by Marcus [84]:

$$\varepsilon = 1 - \pi SMd / 4$$

(2.1)

where $S$ is assumed to be 1.05 as the crimping factor, $d$ is the wire diameter, and $M$ is the mesh number, which is defined as:

$$M = 1 / (d + w)$$

(2.2)

where $w$ is the opening width of wire mesh. The volumetric porosity of mesh can also be determined by a comprehensive model presented by Armour and Cannon[85]:

25
\[ \varepsilon = 1 - \frac{\pi AB}{2(A + B)} \sqrt{1 + \left( \frac{A}{1 + A} \right)^2} \]  

(2.3)

where \( A = d/w \), \( B = d/\delta \), and \( \delta \) is the mesh thickness.

The experimental and theoretical results of liquid volume needed to saturate the hybrid wick are presented in Figure 2.4. The porosity of six samples, i.e., hybrid wicks with mesh-layers number of 0 (groove), 1, 2, 3, 4, and 6, were tested and calculated. It showed that the experimental data are higher than the theoretical results, which can be attributed to the surface tension effect. Actually, the absorbed liquid amount is always a little larger than the void volume of the porous wick. However, this is not applicable for the groove wick, which was unable to make the interspace be fully filled due to poor capillarity.

![Figure 2.4](image)

**Figure 2.4** The experimental and theoretical liquid volumes of the saturated hybrid wick as a function of mesh-layers number.
2.2.2 Heat pipe fabrication process and charging ratios

The heat pipe fabrication process is described as follows. After the mesh was sintered to the half-inch inner grooved tube, one end of the tube was sealed by braze welding. Following an annealing process in the oven with hydrogen/argon atmosphere (500 ºC, four h), the other end of the heat pipe was connected to a 1/8-inch charging pipe by compression fitting. Then, the tube was bent to L shape with a manual pipe bender. A sealing check was executed by compressed air with a pressure of 8 bar. Ethanol (99%) was selected to be the working fluid owing to high wetting performance. The working fluid was degassed using the boiling method before charging. After the heat pipe was evacuated to a pressure of 0.08 Torr by a vacuum pump (Welch 8905), the working fluid was charged via a syringe. Afterward, the charging tube was sealed by cold welding using a pneumatic clamp. To further reduce the residual non-condensable gases, a second degassing process was utilized [86].

The charging ratios of the three-configurations heat pipes are based on the “ideally” loaded condition [74], at which the working fluid exactly fully saturate the wick. The study showed that the heat pipes exhibited little difference in thermal resistance when charging ratios were close to such an “ideally” loaded case. To obtain better performance for each configuration, three charging ratios (50%, 100%, and 150% of $V_{ws}$) were conducted, and the best one was selected as representative.
2.2.3 Experimental setup and testing procedures

Figure 2.5 presents the schematic of the test setup, which consists of a heating system, a circulatory cooling system, and a data acquisition system. Eight cartridge heaters were embedded in two aluminum blocks (Al 6061, 15 cm in length), which embraced the evaporation region to provide heat. A layer of thermally conductive paste (Omegatherm TM 201, the thermal conductivity of 3.5 W/m·K) was utilized on the interface between the heat pipe and heating block to reduce the contact thermal resistance. Constant electrical voltage was provided by a high precision digital DC power supply (BK-PRECISION XLN10014).

Figure 2.5 Schematic of the L-shaped heat pipe testing setup in this study.
The cooling system consisted of a circulating water bath (Neslab, RTE-4DD) and a cylindrical water jacket with a diameter of 11 cm. The heat was dissipated from the heat pipe to the circulating cooling water with a constant temperature of 35±0.1 °C and a constant flow rate of 90 L/h. Data was recorded by an Agilent digital multimeter (34970A) connected to a computer. During testing, the evaporation section and the condensation section of the L-shaped heat pipes were placed horizontally and vertically, respectively.

The thermocouple arrangement is schematically presented in Figure 2.6. Nine K-type thermocouples were applied to test the temperature distribution of the heat pipe. For each testing location of the evaporation region, two thermocouples were utilized to measure both the upper and the lower surface temperatures. Owing to uneven liquid film distribution induced by gravity, the upper surface temperature was always a little higher than the lower one. Two-points measurement effectively reduces testing errors. There was one testing point for each location in the condensation region due to uniform circumferential temperature distribution. The heat pipe and the heating block were altogether wrapped with two layers of thermal insulation.

The testing procedures are described as follows. Experiments were operated at an ambient temperature of 21±0.3 °C in a lab environment. The test began from 10 W heating power with an increment of 10 W. The temperature initially rose rapidly and finally became stable after approximately 40 minutes. The sampling frequency of the data record
was 0.2 Hz. Input power was increased step by step until the heat pipe failed to maintain at a steady state. Then, the maximum heat load ($Q_{\text{max}}$) was reached with the onset of fully dry out.

![Thermocouple arrangement on the L-shaped heat pipe surface for characterization.](image)

**Figure 2.6** Thermocouple arrangement on the L-shaped heat pipe surface for characterization.

### 2.3 Data reduction and uncertainty analysis

The real heat transferred ($Q_{\text{real}}$) by heat pipe can be estimated from:

$$Q_{\text{real}} = Q_{\text{inp}} - Q_{\text{loss}}$$  \hspace{1cm} (2.4)

where $Q_{\text{inp}}$ is the input power, and $Q_{\text{loss}}$ is the heat loss. The heat loss primarily comes from air convection of thermal insulation and can be evaluated by the following equation:
\[ Q_{\text{loss}} = h_{in}A_{in}\left(\frac{T_{s1} - T_{s2}}{2} - T_a\right) \]  

(2.5)

where \( T_a \) is the ambient air temperature, \( T_{s1} \) and \( T_{s2} \) are the surface temperatures of thermal insulations, \( A_{in} \) is the area of thermal insulations, and \( h_{in} \) is the convection heat transfer coefficient.

A preliminary calibration has been conducted to plot the \( h_{in} \) as a function of \((T_s - T_a)\), as shown in Figure 2.7. The heat loss calibration procedures are described in a previous study [87]. The \( h_{in} \) can be expressed by a non-linear fitting equation:

\[ h_{in} = 3.226 \ln(T_s - T_a) - 0.86 \]  

(2.6)

The average temperature of the evaporator section \((T_E)\) can be calculated as:

\[ T_E = \frac{T_{e1u} + T_{e1d} + T_{e2u} + T_{e2d}}{4} \]  

(2.7)

where \( T_{e1u}, T_{e1d}, T_{e2u}, \) and \( T_{e2d} \) are the upper and lower points of two locations in the evaporating section. The average temperature of the condenser section \((T_C)\) can be calculated as:

\[ T_C = \frac{(T_{C1} + T_{C2} + T_{C3})}{2} \]  

(2.8)

The thermal resistance \((R)\) of the heat pipe is defined as:

\[ R = \frac{T_E - T_C}{Q_{\text{real}}} \]  

(2.9)
The effective thermal conductivity \((k_{\text{eff}})\) of the heat pipe is defined as:

\[
k_{\text{eff}} = \frac{L_{\text{eff}}}{A_p R}
\]

(2.10)

where \(L_{\text{eff}}\) is the effective distance between the center of evaporator and condenser, and \(A_p\) is the cross-sectional area of the heat pipe.

![Figure 2.7](image)

Figure 2.7 The heat transfer coefficient \((h_{\text{in}})\) of heat loss as a function of the temperature difference between the thermal insulation surface temperature and ambient air temperature.

The absolute measurement uncertainty of temperature using K-type thermocouple was ± 0.5 °C. The uncertainty of thermocouple location along the axial direction of the heat pipe was within 0.5 mm. The input power (By BK-PRECISION XLN10014) yielded a relative measurement uncertainty of ± 0.5%. According to the standard error analysis
introduced by Taylor [88], the relative measurement uncertainty of thermal resistance could be expressed as:

$$\delta R = \frac{\Delta R}{R} \quad (2.11)$$

$$\Delta R = \sqrt{\left[ \frac{\partial R}{\partial (T_E - T_C)} \Delta (T_E - T_C) \right]^2 + \left( \frac{\partial R}{\partial Q_{rea\bar{l}} \Delta Q_{rea\bar{l}}} \right)^2} \quad (2.12)$$

The relative uncertainty of the thermal resistance in the present study was calculated to be in the range of ± 0.86%~4.8% with input powers rising from 30 W to 140 W. The effective thermal conductivity almost has the same relative uncertainty as to the thermal resistance.

**2.4 Result and discussion**

A comprehensive comparison of thermal performance has been made between the partially hybrid wick, grooved wick, and fully hybrid wick in L-shaped heat pipes. The charging ratios for heat pipes with grooved wick, fully hybrid wick, and partially hybrid wick are 100%, 100%, and 150%, respectively. Figure 2.8 shows the steady-state temperature distributions of the three types of tested heat pipes at three different heat loads of 60 W, 80 W, and 100 W, respectively. Figure 2.8 (d) presents the thermal resistance of the three different heat pipes. For heat load below 70 W, the grooved heat pipe shows the most uniform temperature profile. As the heat load exceeds 70 W, the
isothermal performance of the partially hybrid wick outperforms that of the grooved wick.

The grooved heat pipe performed well at the beginning heat loads but deteriorated rapidly, while the fully hybrid wicked heat pipe presented a poor performance in all range of heat loads. As illustrated in Figure 2.8 (d), the grooved heat pipe performed the lowest thermal resistance (ranging from 0.033 °C/W to 0.062 °C/W) at low heat loads within 60 W. This can be attributed to thin liquid film in evaporation region before onset of partial dry-out. However, the performance significantly deteriorated as the heat loads gradually increased. This was a result of a continuous liquid-front recession resulting from poor capillarity of groove, as indicated in the visualization experiment reported by Wong and Chen [89]. In particular, the fully hybrid wicked heat pipe presented the worst thermal resistance and encountered an early dry out at 80 W. In fact, the porous hybrid wick cannot be applied in the condensation section. As illustrated in the visualization study by S.-C. Wong et al.[90], the mesh wick will be submerged in the working fluid owing to the zero contact angle between ethanol and copper. The ineffective working fluid delivery out of the condenser will lead to quiescent ethanol condensation with high thermal resistance. The thermal resistance of mesh wick as the condenser is reported to be more than 2.5 times[70] and 3.5 times[68] of that as an evaporator. Moreover, this also results in poor heat transfer limits of the fully hybrid wicked heat pipe. The early dry out can be
attributed to working fluid accumulated in the porous hybrid wick of condensation section resulting from strong capillary filling.

Figure 2.8 The steady-state wall temperature distributions of three heat pipes with various coverage lengths of hybrid wick at the heat loads of (a) 60 W, (b) 80 W, and (c) 100 W. (d) Comparison of thermal resistances between the hybrid wick heat pipes and grooved heat pipe.

The thermal performance of the LPHHP substantially outperforms the other two heat pipes for heat load higher than 70 W, as shown in Figure 2.8 (d). The thermal resistance of the LPHHP is relatively small (ranging from 0.06 °C/W to 0.10 °C/W) at all
heat loads. The partially hybrid wick reduces the overall thermal resistance up to 57.4% compared to that of the grooved wick for heat loads higher than 70 W. The underlying enhancement mechanism is the even liquid film distribution and high-efficient capillary evaporation [64] enabled by the mesh-groove wick in the evaporation region, as schematically presented in Figure 2.2 (a). Capillary flow induced by receding menisci of hybrid wick promotes rewetting to prevent local dry-out [91]. Heat transfer is enhanced by inducing advection and improving thin-film evaporation. Besides, the capillary limit of the LPHHP is increased by ~40% (from 100 W to 140 W) compared to that of the grooved heat pipe, because of the separation of the fluid transport process and capillary pressure generation enabled by the hybrid wick [64].

The LPHHP also presents much better thermal performance than the fully hybrid wicked heat pipe. The reductions of the overall thermal resistance by the partially hybrid wick range from 32.7% to 69.4% compared to those of fully hybrid wick. This can be attributed to the high condensation HTC of the LPHHP enabled by the thin liquid film resulting from the high-efficient condensate draining of vertically placed grooved wick [76], as shown in Figure 2.2 (b). In particular, the condensation HTCs of an inclined groove in heat pipe range from 5,000 to 9,000 W/m²·K [77], which are far higher than those of hybrid mesh-groove wick ranging from 1,000 to 2,600 W/m²·K [69]. Thus, the partially hybrid wick configuration, i.e., hybrid wick in the evaporation region and
vertically placed grooved wick in the condensation region, is the optimal strategy for an L-shaped heat pipe.

2.5 Conclusion

This study proposed a partially hybrid mesh-groove wick for an L-shaped heat pipe. Contrast experiments were conducted on a fully hybrid wick and a non-hybrid wick (groove) to validate the effectiveness of partially hybrid wick. The main conclusions are drawn as below:

(1) The partially hybrid mesh-groove wick (only in the evaporation section) significantly outperforms the grooved wick (non-coverage) and fully hybrid wick in L-shaped heat pipes. The underlying mechanism is the highly efficient capillary evaporation enabled by the hybrid wick.

(2) The L-shaped heat pipe performance is not always enhanced using the hybrid mesh-groove wick. Liquid film confined in the hybrid wick of the condensation section will lead to additional thermal resistance.

(3) The strategy of separately optimizing the evaporation section and the condensation section is more effective in maximizing the performance of the heat pipe.
CHAPTER 3 THE OPTIMAL MESH-LAYERS NUMBER AND CHARGING RATIO OF THE L-SHAPED HEAT PIPES WITH PARTIALLY-HYBRID MESH-GROOVE WICKING STRUCTURE

The present study aims to systematically investigate the optimal mesh-layers number and optimal charging ratio of the L-shaped copper/ethanol heat pipes with sintered partially hybrid mesh-groove wicks. Four numbers of mesh layers will be evaluated by comparing the overall thermal resistance and maximum heat load. Five charging ratios will be conducted for each number of mesh layers to find out the optimal charging ratio region. Finally, the optimal charging ratio region will be obtained quantitatively using a regression analysis method named Random Forest Algorithm of machine learning.

3.1 Experimental methodology and data analysis

3.1.1 Hybrid mesh-groove wicks with various mesh-layers numbers

In present work, hybrid mesh-groove wicks with four different number of mesh layers, i.e., one layer, two layers, three layers, and six layers, were fabricated and integrated into L-shaped heat pipes. The numbers of mesh layers were selected, referring
to R. Kempers et al.’s study [74]. The cross-sectional views of the four hybrid wicks are shown in Figure 3.1 by SEM (scanning electron microscope) pictures. After sintering, the total thicknesses of compressed meshes were about 0.12 mm, 0.17 mm, 0.25 mm, and 0.42 mm for one layer, two layers, three layers, and six layers of mesh, respectively.

![Figure 3.1 The SEM images of the hybrid mesh-groove wicks with (a) one, (b) two, (c) three, and (d) six layers of mesh in cross-sectional views.](image)

### 3.1.2 The charging ratios of the L-shaped heat pipes

The effects charging ratio were investigated in the LPHHPs. Five charging ratios, i.e., 50%, 100%, 150%, 200%, and 300%, were evaluated. The charging ratio \( \phi \) is defined as:

\[
\phi = \frac{v_{wf}}{v_{ws}}
\]  
(3.1)
where \( v_{wf} \) is the volume of working fluid and \( v_{ws} \) is the volume of fluid required to saturate the wick of a heat pipe. Here, the \( v_{ws} \) is the experimental result in Figure 2.4 rather than the theoretical result. The corresponding charging amounts of each mesh-layers number are listed in Table 3.1 with an uncertainty of 0.05 ml. The presented results of studying the effect of the mesh-layers number were the best one out of all charging ratios conducted in Table 3.1. Since the heat pipe with 6-layers mesh showed poor performance, only three charging ratios (50%, 100%, and 150%) were investigated.

The heat pipe fabrication process, the heat pipe performance testing setup, the data reduction process, and uncertainty analysis have been described in chapter 2.

Table 3.1 The charging amounts of the L-shaped partially hybrid wicked heat pipes

<table>
<thead>
<tr>
<th>Mesh layers</th>
<th>50%</th>
<th>100%</th>
<th>150%</th>
<th>200%</th>
<th>300%</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 layer</td>
<td>1.22 ml</td>
<td>2.43 ml</td>
<td>3.65 ml</td>
<td>4.86 ml</td>
<td>7.29 ml</td>
</tr>
<tr>
<td>2 layers</td>
<td>1.39 ml</td>
<td>2.78 ml</td>
<td>4.17 ml</td>
<td>5.56 ml</td>
<td>8.34 ml</td>
</tr>
<tr>
<td>3 layers</td>
<td>1.59 ml</td>
<td>3.19 ml</td>
<td>4.79 ml</td>
<td>6.38 ml</td>
<td>9.57 ml</td>
</tr>
<tr>
<td>6 layers</td>
<td>2.17 ml</td>
<td>4.34 ml</td>
<td>6.51 ml</td>
<td>Non</td>
<td>Non</td>
</tr>
</tbody>
</table>

### 3.1.3 Regression analysis to calculate the \( k_{eff} \) of the heat pipe

A non-linear regression analysis of the \( k_{eff} \) of heat pipes has been conducted using a machine learning method, which can find the optimal charging ratio range as a function of heat load. The regression analysis was operated by the multiple decision trees of the machine learning method, which naturally learn non-linear relationships from multiple
variables. Specifically, an ensemble learning algorithm, i.e., the Random Forest Algorithm, was selected for its high accuracy for multiple input variables and high efficiency on an extensive database.

The regression code was implemented in Python, and the steps are listed as follows: (1) Imported the related libraries in Python, such as Scikit-Learn. (2) Imported the data sets with heat load and charging ratio as input variables ($X$) and effective thermal conductivity as target variable ($Y$). (3) Standardization of data set by preprocessing and dividing the data sets to the training group and the validated group. (4) Selected the Random Forest Regressor, set the initial hyperparameters, and found out the optimal hyperparameters using the validated data set. (5) Fitted the Random Forest Regressor to the training datasets using the optimal hyperparameters. (6) Generated the predicted targeted values using the trained regressor.

3.2 Results and discussion

3.2.1 Effect of number of mesh layers on the thermal performance of the LPHHP

The effect of mesh-layers number was investigated by testing the LPHHPs with 1, 2, 3, and 6 layers of wire mesh. Figure 3.2 presented the axial temperature distributions and effective thermal conductivity of the LPHHPs with four various numbers of mesh layers at low heat loads and high heat loads, respectively. It is noted that the charging
ratios are not uniform for each mesh-layers number. The presented results in given mesh-layers numbers were the best out of all charging ratios conducted in the experiments. It is obvious that the mesh-layers number had a great effect on the thermal performance of the LPHHPs as expected. The surface temperatures were evenly distributed for the 1 and 2 layers of mesh but uneven for the 3 and 6 layers of mesh.

There exists an optimal mesh-layers number for the LPHHPs to the minimum the thermal resistance. Insufficient mesh layers lead to the inadequate area for thin-film evaporation, while excessive mesh layers result in an increased thermal resistance of evaporator [92] induced by the thick saturated wick. The optimal mesh-layers number is affected by the bonding strength of intermeshing. On the one hand, the optimal mesh-layers number will be less for poor jointing strength with large contact resistance. For example, the optimal mesh-layers number was reported to be only one layer [20] for a composite mesh-groove wick with a relaxing connection method of “plating” rather than “sintering”. A similar conclusion was reported by J.C. Hsieh et al. [65] that the evaporating heat transfer coefficient was reduced as the number of mesh layers increased in a hybrid wick supported by a coronary stent-like structure. There could be high contact thermal resistances with more mesh layers and deteriorated capillarity induced by possible local detachments. On the other hand, for mesh wick with better bonding strength induced by sintering, the optimal mesh layers number was reported to be a larger value of three [74].
In the present study, the optimal mesh-layers numbers are two layers and one layer for heat loads ranging from 10 W to 70 W and ranging from 80 W to 140 W, respectively. These numbers are owing to a moderated bonding strength between groove tip and mesh, as shown in Figure 2.3. At low heat loads, thin-film evaporation dominates the heat transfer mechanism. The LPHHP with two layers of mesh performed the best by balancing between evaporate area and saturated wick thickness. The $k_{eff}$ of the LPHHP
with 2-layers mesh was the highest (ranging from 32,819 to 58,776 W/m·k) at heat loads within 70 W. At high heat loads, the dominated mechanism turned to be suppressing the onset of partial dry out [93] by capillary liquid transport. The PHLPH with one layer of mesh and 150% charging ratio performed the most rapid axial capillary spreading by avoiding unnecessary radial capillary filling occurring in the thick porous wick. Thus, the maximum $k_{eff}$ of around 30,000 W/m·k was achieved by the one layer of mesh at heat loads exceeding 80 W.

### 3.2.2 Effect of charging ratio on the thermal resistance of the LPHHPs

This section systematically investigated the charging ratio effect on the thermal resistance of the LPHHPs with one, two, and three layers of mesh, except for the six layers with poor performance, as presented in Figure 3.3. It was obvious that the charging ratio of 50% was severely insufficient for all tested samples. Sharply increased thermal resistance induced by the burning of evaporator initiated early from a low heat load. The LPHHPs with a higher charging ratio performed lower thermal resistance at high heat loads exceeding 100 W. Drastic jumps of thermal resistance due to the onset of partial dry-out stage [68] were observed in some curves. The capillary heat transfer limit presented an apparent non-linearity versus the charging ratio. As illustrated in Figure 3.3, under-loading resulted in a significant reduction of heat transfer limit while overloading lead to a marginally higher heat transfer limit at all numbers of mesh layers. This
non-linear trend is consistent with the experimental results on a screen mesh wicked heat pipe reported by Kempers R. et al. [74].

![Graphs showing thermal resistance vs. input power for LPHHPs with different layer meshes and charging ratios.](image)

**Figure 3.3** Effect of charging ratio on the thermal resistance of the LPHHPs with (a) one, (b) two, and (c) three layers of mesh. The charging ratio ranged from 50% to 300% of the void volume of the wick.

Excessive or insufficient charging of liquid would substantially deteriorate the thermal performance of the LPHHPs. On the one hand, an excess charging ratio for a particular heat load would result in a reduction of heat transport capability [75].
Excessive liquid deteriorates heat transfer by inducing pool in evaporator [74] or generating a gas film in the porous wick to separate the fluids and pipe wall [73]. On the other hand, an insufficient charging ratio would lead to an increase in thermal resistance in the evaporation section [73]. This can be attributed to working fluid accumulating in the wick of the condensation section, causing inadequate condensate and onset of dry-out in evaporator [75]. Hence, there exists an optimal charging ratio range for a particular heating condition.

In the present study, the optimal charging ratio range of the LPHHPs was recommended to be 100%-200% without a particular range of heat load. This range is consistent with the one of a heat pipe with partially porous wick [83], which was reported to be 200% (3.6 ml) of the void volume of the wick (1.8 ml). The optimal charging ratio range (100%-200%) in the present study was larger than those of fully covered mesh heat pipe, i.e., 50%-150% [74] for a copper mesh heat pipe and 75% [72] for stainless steel mesh heat pipe. This can be attributed to the high draining efficiency of grooved wick in the condensation section. On the one hand, for the fully covered mesh heat pipe, a large charging amount may lead to an increased thickness of saturated porous wick, resulting in sharply increased condensation thermal resistance. On the other hand, for partially hybrid wicked heat pipe in present study, more charging amount is needed for
the highly efficient draining of condensed film in a vertically positioned grooved wick [76].

### 3.2.3 The optimal charging ratio region obtained by regression analysis

This study has applied a regression analysis using machine learning to find the optimal charging ratio region as a function of heat load. Figure 3.4 presents the $k_{eff}$ of the LPHHPs as a function of charging ratio and heat load in 3D-surface plots. The achieved maximum $k_{eff}$ is shown in Figure 3.4 (d). In overview, the optimal charging ratio indicated by the ridge of the ‘mountain’, increases slightly with the increase of heat load. This trend is consistent with the conclusions on an aluminum flat plate heat pipe reported by J.-S. Chen and J.-H. Chou [75]. In particular, for the heat pipe with three layers of mesh, the optimal charging ratios in the heat load ranges of 10 W-60 W, 70 W-90 W, 100 W-120 W, and 130 W-180 W, were 100%, 150%, 200%, and 300%, respectively.

To obtain a more visible map to present the optimal charging ratio region, non-linear regression analysis has been conducted by gathering all of the $k_{eff}$ data in Figure 3.4. Figure 3.5 (a) presents results obtained from the Random Forest Algorithm of machine learning, of which the implementation steps are shown in section 4.2.2. After the regression analysis, the ‘mountain’ is smoother without sudden slope changes. Figure 3.5 (b) is the top view of Figure 3.5 (a) as a map quantitatively showing the optimal charging ratio region in a more apparent 2-D view, where the gradation of color indicated the value
of the $k_{eff}$. Figure 3.5 (b) is divided into three sub-regions with various color-bar ranges to increase the contrast.

Figure 3.4 Coupling effect of charging ratio and heat load on the effective thermal conductivity of the LPHHPs with (a) one, (b) two, and (c) three layers of mesh. (d) The maximum $k_{eff}$ achieved by the LPHHPs as a function of charging ratio and heat load.

The optimal charging ratio is in a region as a function of heat load, rather than a constant point. The tolerance range of the region is around 100% at heat loads within 120 W, and increases to ~150% at heat loads ranging from 120 W to 150 W. The optimal charging ration ranges increase from 80%-180% to 100%-200%, 150%-300%, and 230%-300% as the heat load ranges increase from 10 W-70 W to 70 W-120 W, 120
W-150 W and 150 W-180 W, respectively. This result well demonstrates the strategy that the applied heat load range should be considered in the determination of the charging ratio of the heat pipe.

Figure 3.5 (a) The non-linear regression analysis of the $k_{eff}$ using the Random Forest Algorithm of machine learning. (b) A map of the optimal charging ratio region versus heat load.

3.3 Conclusion

The coupling effect of mesh-layers number (1, 2, 3, and 6 layers) and charging ratio were experimentally investigated in the present study. The optimal charging ratio region was obtained as a map using a regression analysis of the machine learning algorithm. The main conclusions are drawn as below:

(1) The optimal mesh-layers number increases as the intermeshing bonding strength increases. The optimal mesh layers number is found to be 1-2 layers in the present study.
(2) Insufficient or excessive charging ratio would deteriorate the heat pipe performance owing to the early onset of dry-out or liquid pool in the evaporation region. The universal optimal charging ratio of the LPHHPs is recommended to be 100% -200%.

(3) More precisely, the optimal charging ratio is in a region depending on the heat load, rather than a point with a constant value.
CHAPTER 4 THE OPTIMAL DESIGN OF GROOVE FOR CAPILLARY FLOWING AND ITS MECHAMISM ANALYSIS BY SIMULATION METHOD

To enhance the capillary limit of a heat pipe, it is significant to optimize the design of the wicking structuring. Groove wick is the most common and typical wick with low cost. Besides, a groove is an important component of a hybrid wick. Thus, it is significant to optimize the design of the groove and reveal the capillary flowing mechanism of the groove [94].

Sometimes heat pipes need to work in different gravitational conditions. For example, an aircraft with heat pipes may speed up or slow down. The gravity may be in the same direction or the opposite direction of the flowing liquid. To investigate the gravitational effect of the wicking performance provides a guide of heat pipe used in space. In this section, the effect of contact angles, gravity, and groove shape on the capillary filling performance will be investigated. This helps optimize the groove parameter (groove shape and surface contact angle) and understand the flowing mechanism of the gravitational effect. Specifically, the simulation result presents the pressure field, the velocity field, and the shape and position of the water surface.
4.1 The introduction of theory for capillary-flowing simulation in literature

During capillary flowing inside microchannels, capillary force induced by surface tension and wall adhesive forces is two major forces to be considered. There are other examples where surface tension and wall adhesion significantly affect the flowing dynamic, such as droplets on wall surface and multiphase flow through porous mediums. The analysis of forces can also be applied in measuring the Position and transportation of a small amount of liquid using micropipettes or the capillary transport dynamic in MEMS devices.

4.1.1 Moving interface method

Moving interfaces setting method is an effective methods to simulate two phase flow. A diffuse interface model is an effective tool for modeling the fluids dynamics with moving interfaces and it is solved by the lattice Boltzmann algorithm [95-97]. In the study of Pooley et al. [98], it is demonstrated that the moving interface approach is able to be applied to simulate capillary filling in microchannels with smooth surface. In addition, mesoscale modeling approach is appropriate to be applied to describe the system with micron length scales [99].
4.1.2 The VOF method

The simulation of capillary flow can be performed using the VOF method [100]. In addition to the momentum equation and continuity equation, a volume fraction transport equation is also applied in this method. The gas-liquid two phases are treated as a homogeneous mixture in this method. Therefore, the following uniform incompressible and non-gravity Navier-Stokes equations to describe the mixture can be employed in the calculation.

\[ \nabla \cdot \mathbf{V} = 0 \]  
\[ \frac{\partial \rho \mathbf{V}}{\partial t} + \nabla \cdot (\rho \mathbf{V} \mathbf{V}) = -\nabla p + \mathbf{F}_s + \nabla [\mu (\nabla \mathbf{V} + \nabla \mathbf{V}^T)] \]  

where \( t \) is the time, \( p \) is the pressure, \( \mathbf{V} \) is the velocity of the mixture, \( \rho \) is the density, \( \mu \) is the viscosity, and \( \mathbf{F}_s \) is the volumetric force induced by the surface tension.

The mixture’s physical properties are derived from that of the two phases through a volume fraction function:

\[ \rho = \alpha_L \rho_L + \alpha_G \rho_G \]  
\[ \mu = \alpha_L \mu_L + \alpha_G \mu_G \]

where the subscripts \( L \) and \( G \) represent the liquid phase and the gas phase, respectively, \( \alpha \) represents the volume fraction of a phase in a computational cell with \( \alpha_L + \alpha_G = 1 \). When \( \alpha_L = 0 \), the cell contains no liquid. When \( \alpha_L = 1 \), the cell is full of liquid.
Surface tension and wall adhesion are needed here. The surface tension is specified as a source term $F_S$ in Eq. (4.5) according to the continuum surface (CSF) model [101]:

$$F_S = \frac{\rho \sigma k \nabla \alpha_L}{(\rho_L + \rho_g)/2}$$

(4.5)

$$k = -\nabla \cdot \hat{n} = [\left( \frac{1}{|n|} \cdot \nabla \right) |n| - \nabla \cdot n]$$

(4.6)

where $k$ represents the curvature of the surface, $\hat{n}$ is the unit vector normal to the surface.

Wall adhesion is included in the model through the contact angle:

$$\hat{n} = n_w \cos \theta_w + t_w \sin \theta_w$$

(4.7)

where $\hat{n}_w$ and $\hat{t}_w$ represents the unit vector normal and tangent to the wall, respectively, and Youngs’ scheme is adopted for the interface reconstruction.

4.1.3 The surface energy model

From the point of view of energy, the capillary phenomenon is caused by the minimization of the total system surface energy. The capillary force can be derived from the surface energy model. The surface energy is equal to the surface area multiplied by the surface tension of the interface between two materials or phases.

Figure 4.1 shows the configuration of a microchannel. Then the total surface energy of the system is:
\[ E_s = E_R + S_0 \sigma + S_L \sigma_{SL} + (S_T - S_L) \sigma_{SG} \] (4.8)

where \( \sigma_{SL} \), \( \sigma_{SG} \), and \( \sigma \) were used to represent the surface tensions of solid-liquid, solid-gas, and liquid-gas interfaces, respectively. \( S_T \) represents the total area, \( S_L \) is the wetting area, and \( S_0 \) is the gas-liquid interface area. \( E_R \) is the surface energy of the reservoir.

For a homogeneous-surface microchannel having a rectangular cross-section with a width \( W \) and a height \( H \), the total surface energy can be expressed as:

\[ E_s = E_R + S_0 \sigma + 2(H + W)x\sigma_{SL} + 2(H + W)(x_T - x)\sigma_{SG} \] (4.9)

where \( x_T \) represents the total length and \( x \) the wetting length. The wetting length is usually defined as the distance between the inlet and one typical point on the gas-liquid interface for simple liquid front shapes.

Under the static contact angle assumption, the deformation of the liquid front shape can reasonably be neglected. Thus, \( S_0 \) can be considered a constant. Then the capillary force \( F_C \) can be derived as:
\[ F_c = -\frac{dE_s}{dx} = 2(H + W)(\sigma_{SL} - \sigma_{SG}) = \Delta \rho_{LG} WH \]  

(4.10)

where \( \Delta \rho_{LG} \) is the pressure drop across the liquid-gas interface.

4.2 The selected models for capillary filling in this study

We used Comsol FEM software in this study to simulate the capillary filling in a groove or hybrid wick. For capillary fill in porous media or channel, Comsol provides either the TwoPhase Flow, Level Set, or the Two-Phase Flow, Phase Field multiphysics coupling feature. The Level Set interface uses a reinitialized level set method to represent the fluid interface between the air and the water. The Phase Field interface, on the other hand, uses a Cahn-Hilliard equation, including a chemical potential to represent a diffuse interface separating the two phases. Here, we selected the Level Set and TwoPhase Flow model in this study for better accuracy as reported in the literature.

4.2.1 The Level set Method for the transport of fluid interface

The Level Set interface automatically sets up the equations for the convection of the interface. The fluid interface is represented by the 0.5 contours of the level set function. In the air and water, the level set function can thus be thought of as the volume fraction of water. The transport of the fluid interface separating the two phases is given by:
\[ \frac{\partial \phi}{\partial t} + u \cdot \nabla \phi = \gamma \nabla \cdot \left( \varepsilon \nabla \phi - \phi (1 - \phi) \frac{\nabla \phi}{|\nabla \phi|} \right) \]  

(4.11)

The \( \varepsilon \) parameter determines the thickness of the interface. When stabilization is used for the level set equation, you can typically use an interface thickness of \( \varepsilon = h_c/2 \), where \( h_c \) is the characteristic mesh size in the region passed by the interface. The \( \gamma \) parameter determines the amount of reinitialization. A suitable value for \( \gamma \) is the maximum velocity magnitude occurring in the model. The multiphysics coupling feature defines the density and viscosity according to:

\[ \rho = \rho_{air} + (\rho_{water} - \rho_{air}) \phi \]  

(4.12)

\[ \mu = \mu_{air} + (\mu_{water} - \mu_{air}) \phi \]  

(4.13)

Due to these definitions, the density and viscosity vary smoothly across the fluid interface. The delta function is approximated by:

\[ \delta = 6 |\phi(1 - \phi)| |\nabla \phi| \]  

(4.14)

and the interface normal is calculated from:

\[ n = \frac{\nabla \phi}{|\nabla \phi|} \]  

(4.15)
4.2.2 The mass and momentum transport

The Navier-Stokes equations are used to describe the momentum transport and the conservation of mass. To account for capillary effects, it is crucial to include surface tension in the model. The Navier-Stokes equations are then:

\[ \frac{\partial \mathbf{u}}{\partial t} + \rho (\mathbf{u} \cdot \nabla) \mathbf{u} = \nabla \cdot [-p \mathbf{I} + \mu (\nabla \mathbf{u} + (\nabla \mathbf{u})^T)] + \mathbf{F}_{st} + \rho \mathbf{g} \]  
\[ \nabla \cdot \mathbf{u} = 0 \]

Here, \( \rho \) denotes the density (kg/m\(^3\)), \( \mu \) equals the dynamic viscosity (N.s/m\(^2\)), \( \mathbf{u} \) represents the velocity (m/s), \( p \) denotes the pressure (Pa), and \( \mathbf{g} \) is the gravity vector (m/s\(^2\)). \( \mathbf{F}_{st} \) is the surface tension force acting at the air/water interface.

4.2.3 The surface tension force

In the Level Set interface, the surface tension force is computed as:

\[ \mathbf{F}_{st} = \sigma \delta \kappa \mathbf{n} \]

Here, \( \mathbf{n} \) is the interface normal, \( \sigma \) is the surface tension coefficient (N/m), \( \kappa \) is the curvature, and \( \delta \) equals a Dirac delta function that is nonzero only at the fluid interface. The following boundary force is added to enforce the contact angle:

\[ \mathbf{F}_{\theta} = \sigma \delta (\mathbf{n}_{wall} \cdot \mathbf{n} - \cos \theta_{w}) \mathbf{n} \]

where \( \theta \) is the contact angle. If you apply a no-slip boundary condition, the velocity vanishes on that boundary, and you cannot specify the contact angle. Instead, the
interface remains fixed on the wall. However, if the assumption allows a small amount of slip, it is possible to specify the contact angle. The Wetted Wall coupling feature adds the term given by Eq. (4.19) and consequently makes it possible to set the contact angle.

### 4.2.4 The Bond number and Capillary number

Capillary phenomena become obvious when the capillary force plays a dominant role. The dimensionless number $Bo$ (Bond number) is usually used for measuring the importance of gravity relative to capillary force. When the microchannels are set horizontally, the Bond Number can be considered as the ratio between two pressure drops: one caused by gravity and the other by capillary force. That is:

$$Bo = \frac{\rho g H^2}{\sigma} = \frac{\rho g H}{\sigma / H}$$  \hspace{1cm} (4.20)

where $\rho$ represents the density, $g$ is the gravitational acceleration, $H$ is the channel height, $\sigma$ is the surface tension, $\Delta p_g$ the gravitational pressure drop, and $\Delta p_\sigma$ is the capillary pressure drop.

In this study, 250 µm-scale horizontal microchannels using water as the working liquid are employed. The Bond number is:

$$Bo = \frac{1000 \times 9.8 \times (0.25/1000)^2}{0.0728} = 0.0084 \ll 1$$  \hspace{1cm} (4.21)
It indicates the gravitational pressure drop is far smaller than the capillary pressure drop if the microchannel is placed horizontally. The capillary force plays a dominant role in capillary filling in our study.

Another important dimensionless number in capillary flows is the capillary number:

\[ Ca = \frac{u\mu}{\sigma} \]  

(4.22)

where \( u \) is the flow velocity. In microchannel flows the capillary number is usually very small. The smaller \( Ca \) is, the slighter the dynamic effect of the contact angle will be. Thus, the contact-angle dynamics are not taken into account in this study and a static contact angle approach is used.

4.3 Validation of simulation by analytical modeling

It is significant to validate the simulation case with classical models or experimental results. Improper setting of parameters or physical modeling may lead to large simulated errors. In this section, a simulated capillary filling case operated by Comsol 5.5 has been validated by the famous Known Washburn Modeling.

Figure 4.2 presents the basic boundary setting of the validation case in Comsol 5.5. The model consists of a vertical channel with a radius of 0.5 mm attached to a liquid reservoir. The liquid can flow freely into the reservoir. Both the reservoir and the channel
are set in a 2D region. Initially, the channel is fully filled with air. Wall adhesion leads water to creep up along the channel boundaries. The deformation of the air/water interface induces surface tension and creates a pressure jump across the liquid/air interface. The pressure difference causes the liquid to move forward. The water continues to fill the channel until the capillary forces are balanced by the gravity force as a function of the water height. In this case, with a channel length of 7 mm, the capillary force is always larger than the gravity. Thus, the interface kept moves upward throughout the entire simulation. The meshes number is 32,865 with a time step of $5 \times 10^{-6}$ seconds.

Figure 4.2 The physical model and boundary setting for validation of 2D CFD simulation with an analytical model. The applied CFD software is Comsol 5.5.

After the simulation, we validated the result with the well-known Lucas Washburn equation. Washburn’s equation describes the capillary flow in a cylindrical
tube. It describes the penetration length of liquid into a tube as a function of time. The first-order differential equation for the length of the fluid can be expressed by:

\[
\frac{\delta l}{\delta t} = \frac{[P_A + g\rho(h - l\sin\psi) + \frac{2\sigma}{r}\cos\theta](r^4 + 4\epsilon r^3)}{8r^2 \mu l}
\] (4.23)

where \(l\) is the length of liquid, \(P_A\) is the atmosphere pressure, \(\rho\) is the density, \(\psi\) is the angle of the tube to the horizontal line, \(\sigma\) is the surface tension, \(\theta\) is the contact angle, \(\mu\) is the viscosity of the liquid, \(r\) is the capillary radius.

![Figure 4.3 The validation of the CFD result using the Washburn model.](image)

To validate the proposed numerical system, the capillary rising length-time is calculated using the Washburn model. Figure 4.3 presents the comparisons of the simulation result and the Washburn model result. The simulated results are in good agreement with the analytical data. At the initial stage, the simulated results are slightly
smaller than the modeling data. The two results are very close to each other after 2 seconds. The results indicate that the numerical case successfully simulated the three typical periods, i.e., $l\sim r^2$, $l\sim t$, $l\sim t^{1/2}$, in the process of capillary flow. In all, the simulation result has been well validated by the analytical result.

4.4 Model setup for capillary filling of grooves in this study

This section introduces the model setup, initial condition, boundary condition, and mesh information. Figure 4.4 presents the physical model setting of capillary filling in a grooved wick and a mesh-groove hybrid wick. The groove shape is a trapezoid, with a height of 0.25 mm, an upper base of 0.2 mm, and a lower base of 0.4 mm. For the mesh-groove hybrid wick, the groove is covered with one layer of mesh with a wire width of 0.056 mm and a wire gap of 0.12 mm. The selected physical model includes Level set, Two phase flow, and Laminar Flow. Since we study the filling length as a function of time, the study type of time-dependent was applied. Initially, the reservoir is filled with water and the capillary channel is filled with gas. In all of the cases, the gas is set to be air at room temperature of 20 °C with properties of $\rho=1.22$ kg m$^{-3}$ and $\mu=1.78\times10^{-5}$ kg.m$^{-1}$s$^{-1}$.

The groove and hybrid wick investigated in this study have three different kinds of boundaries: flowing inlet, flowing outlet, and the wall. Firstly, the inflow boundary is modeled as a hydrostatic pressure inlet with values as a function of liquid depth ($\rho gy$, y is
the liquid depth). The pressure boundary equation automatically compensates for the hydrostatic pressure induced by gravity, so the actual pressure value is set to zero. The volume fraction of water is 100% at the inlet because only water enters the inlet boundary. Secondly, the outflow boundaries are modeled as constant-pressure surfaces with a pressure value of zero, which equals to pressure value at the top of the inlet. Thirdly, various contact angles were set for the wetted wall as needed. The contact angle is defined as the angle between the wall and the fluid interface. The slip length on the wetted wall equals the basic element size.

![Figure 4.4 The physical model setting of capillary filling in (a) Groove wick and (b) mesh-groove hybrid wick.](image)

The created meshes of groove wick and mesh-groove hybrid wick are shown in Figure 4.5. The operation tool for dividing the grid is Comsol 5.5. The basic mesh shape is tetrahedra shape for both groove wick and hybrid wick. For the groove meshes, the total mesh number is 106,569 with an average mesh quality of 0.664. For the hybrid wick meshes, the mesh number is 126,458 with a mesh quality of 0.657.
4.5 The effect of contact angle on the capillary filling of groove

The contact angle of the wall surface plays an important role in the capillary filling of a microchannel. The copper surface can have various contact angles with water by surface coating technology at a low cost. By understanding the contact angle effect and mechanism, we can better utilize such a strategy to enhance the capillary flowing. However, the flowing process and mechanism of trapezoidal grooves with various contact angles have not been studied. The highly developed CFD computation technology makes this possible. In this section, the contact angle effect on the capillary filling length, pressure field, and velocity field will be investigated.

Figure 4.6 presents the liquid front shape and filling length of the groove with contact angles (CA) of 0°, 30°, 60°, and 90°. The selected times for presentation are 0.5 ms, 1 ms, and 1.5 ms with an interval of 0.5 ms. The figure indicates that the capillary
filling velocity is significantly affected by the contact angle. Specifically, the filling time is \(~0.9\) ms, \(1.3\) ms, and \(>5\) ms for the CAs of \(0^\circ\), \(30^\circ\), and \(60^\circ\), while the one with \(90^\circ\) CA fail to fill the channel due to lacking Laplace pressure. Besides, the liquid front shape is choppy during the filling process. The forefront liquid/air interface is at the corner with the smallest meniscus. Then, the side liquid/air interface pulls the central interface forward.

![Figure 4.6](image.png)

Figure 4.6 The filling length and liquid front shape comparison of grooves with various contact angles \((0^\circ, 30^\circ, 60^\circ,\) and \(90^\circ)\) at the time of (a) \(0.5\) ms, (b) \(1\) ms, and (c) \(1.5\) ms.

Figure 4.7 presents the analysis of the pressure field and velocity field of flowing liquid on grooves with various contact angles. The minimum pressure occurs in the curved liquid front due to the Laplace pressure induced by a concave meniscus. Besides, the contact angle significantly affects both the pressure field and the velocity field. A smaller contact angle induces a larger pressure drop and velocity for the same groove.
The groove with a CA of 0° presents the minimum pressure of ~650 Pa and the fastest velocity of ~1.8 m/s among all.

Figure 4.7 The (a) pressure and (b) velocity distributions of capillary filling on grooves with various contact angles of 0°, 30°, and 60°. The cut plane is at the bottom and the results are at the moment when the liquid front reaches 0.5 mm.

4.6 The gravitational effect on the capillary filling of groove

The aircraft in space is always working in a micro-gravity environment, where gravity can be neglected. However, gravity has a great effect on the capillary flowing of the wick, especially when the aircraft is speeding up with an accelerated speed larger than 10 g. Besides, when the wick is anti-gravity with a length longer than 20 cm, the gravity is difficult to be overcome. Nonetheless, the mechanism of gravity on 3D capillary flowing is still not clear. In this study, the capillary flowing mechanism of gravities ranging from -10 g to 10 g will be studied. The research significance of this study is to
better understand the gravity effect so that we can utilize gravity or prevent failure ofwick under anti-gravity conditions.

The capillary phenomena will be greatly affected when the gravity force plays a dominant role. The Bond number \((Bo, \text{ dimensionless})\) is usually used to evaluate the importance of gravity force relative to capillary force. The Bond number could be considered as the ratio of the gravity force and capillary force, which is expressed by:

\[
Bo = \frac{\rho g H^2}{\sigma} \alpha \frac{\Delta p_g}{\Delta p_c}
\]

(4.24)

where \(\rho\) is the density, \(g\) is the gravitational acceleration, \(\sigma\) is the surface tension, \(H\) is the height of the liquid in the direction of gravity, \(\Delta p_g\) is the gravitational pressure drop, and \(\Delta p_c\) is the capillary pressure drop.

When the groove is placed horizontally, \(H\) is the height of the groove (0.25 mm) with a \(Bo\) number of 0.008<<1. Then, gravity can be neglected. However, when the gravity is in the direction of the groove axis and the gravity is increased to 5 g to 10 g, the gravity greatly affects the capillary flowing process. The bond numbers with \(H=1\) mm, 10 mm, 100 mm, and \(G=1\) g, 5 g, 10 g, -10 g are calculated and listed in Table 4.1.

In our study, the \(H\) is equal to 1 mm. When the gravity is 1 g, the Bond number is shown to be 0.134 < 1, which indicates that gravity doesn’t play an important role. When the gravity is larger than 5 g, the Bond number ranges from 0.673 to 1.34, which means
that the gravity greatly affects the capillary process. Besides, when $H$ is larger than 10 mm, the Bond number is far larger than 1 (13.46-13461). It indicates that gravity dominants in this case.

Table 4.1 The Bond numbers with various liquid heights and gravities

<table>
<thead>
<tr>
<th>Height of liquid</th>
<th>$G=-10$ g</th>
<th>$G=1$ g</th>
<th>$G=5$ g</th>
<th>$G=10$ g</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 mm</td>
<td>0.134</td>
<td>0.673</td>
<td>1.34</td>
<td>1.34</td>
</tr>
<tr>
<td>10 mm</td>
<td>13.46</td>
<td>67.3</td>
<td>134.6</td>
<td>134.6</td>
</tr>
<tr>
<td>100 mm</td>
<td>1346.1</td>
<td>6730.7</td>
<td>13461</td>
<td>13461</td>
</tr>
</tbody>
</table>

Figure 4.8 shows the liquid front shape and filling length of liquid in grooves with gravities of -10 g, 0 g, 5 g, and 10 g, respectively. Firstly, the results indicate that gravity greatly affects the capillary filling process of a groove. The capillary filling velocity increases as the positive gravity increases. Secondly, the case with 10 g gravity presented the fastest capillary filling velocity, which is around 2.5 times of the one with -10 g. Lastly, filling length differences induced by various gravities are more obvious in the later stage than the earlier stage. For example, the filling length of -10 g is only ~3.5 mm at 5 ms, while the one of 10 g is ~8.8 mm at the same time.

Figure 4.9 presents the pressure and velocity analysis of the capillary flowing process on grooves with gravities of -10 g, 0 g, 5 g, and 10 g. This is the result of cut planes at the bottom with the Z-axis to the upside. The distribution images are captured
when the liquid front arrives at approximately 0.3 mm. To begin with, gravity didn’t make an obvious difference in the pressure distribution.

Figure 4.8 The filling length and liquid front shape of grooves under various gravities of -10g, 0g, 5g, and 10g. The selected times are (a) 1 ms, (b) 3 ms, and (c) 5 ms.

Figure 4.9 The (a) pressure and (b) velocity distributions of capillary filling on grooves with various gravities of -10g, 0g, 5g, and 10g. The cut plane is at the bottom and the results are at the moment when the liquid front reaches 0.3 mm.
It is noted that a low-pressure region in the entire cross-section was formed in the case of \( G = -10 \, \text{g} \) because the gravity and capillary force pull the liquid front in two opposite directions. Besides, gravity significantly affects the velocity field. A larger positive gravity leads to a larger velocity distribution. The velocity of the liquid fronts is in ranges of 0.6~0.7 m/s for the 10 g case, which is \(~3\) times larger than those of the -10 g case (0.2~0.3 m/s).

### 4.7 The optimal groove shape for the fastest capillary filling

In this section, the grooved shape will be optimized for the fastest capillary filling process. Equally, the mechanism and effect of groove shape on the pressure and velocity field will be investigated. The significance of this study is for designing the grooved wick or groove- mesh hybrid wick with the maximum capillary limit. Four typical groove shapes, i.e., triangle, trapezoid, rectangle, and dovetail slot, have been selected and investigated.

The constraint for the groove is that they all have the same dimension of depth and maximum side width. This ensures that the depth/width ratio is the same for different grooved shapes. Normally, the depth/width ratio is the determining factor for the fabrication cost. The dimensions of each groove with various shapes are presented in Figure 4.10. The depths of the grooves are all 0.25 mm and the width is 0.4 mm. The hydraulic diameter \( (D_h) \) can be calculated by:
\[ D_h = \frac{4A}{P} \]  

where \( A \) is the cross-sectional area and \( P \) is the perimeter. The hydraulic diameters for the triangle, trapezoid, rectangle, and dovetail slot are 0.31 mm, 0.44 mm, 0.41 mm, and 0.32 mm, respectively.

![Diagram showing dimensions and hydraulic diameters](image)

Figure 4.10 The dimensions and the hydraulic diameter of the grooves with various shapes.

It is necessary to simulate the capillary filling process using the CFD method, rather than the empirical formula. Firstly, although there is an empirical formula for predicting the capillary filling in a groove, the CFD simulation can reveal more mechanisms and details in a 3D dynamic view. Besides, the empirical formula only covers triangle and rectangle shapes, and the trapezoid or dovetail slot shapes in arbitrary angles are difficult to be solved. Secondly, the analytical results only characterize the average length of filling liquid, but the liquid front shape or fluctuation surface of the
liquid. However, the thin-film region in the liquid front is an important region with high heat flux. Thirdly, the corner effect is difficult to be presented by an analytical solution. Especially, when the contact angle is very small, the error of the analytical equation will be increased. In summary, with the development of CFD technology and the sharped increased computation speed, calculating the 3D liquid surface shape with refined mesh becomes possible. The CFD simulation of 3D capillary filling expands our view on the capillary mechanism.

Figure 4.11 presents the liquid front shape and filling length of grooves with shapes of triangle, trapezoid, rectangle, and dovetail slot. The presented flowing duration is from 0.3 ms to 0.9 ms. It illustrates that the groove shape significantly affects the capillary filling velocity. For example, the filling time for the dovetail slot groove is the shortest (0.6 ms), while the trapezoid groove takes the longest time of 1.7 ms. Besides, the dovetail slot shape is proved to be the optimal groove shape for capillary filling. The capillary flowing velocity strongly depends on the included angle between the walls. A smaller angle of the dovetail slot induces a larger capillary pressure and a larger flowing velocity.

Figure 4.12 and Figure 4.13 present the analysis of pressure field and velocity field of grooves with various shapes in both a cross-section view and an axial-section view, respectively. In the cross-section view, the effect of boundary shape on the velocity and
pressure distribution can be compared. In the view of an axial-section view, the velocity and pressure distribution along the flowing direction can be observed and studied. The cut plane of the axial-section view overlaps with the sidewall and the selected time for presented is 0.6 ms.

Figure 4.11 The filling length and liquid front shape of grooves with various shapes of triangle, trapezoid, rectangle, and dovetail slot. The selected times are (a) 0.3 ms, (b) 0.6 ms, and (c) 0.9 ms.

In the view of cross-section, the dovetail slot groove induces the largest pressure difference of 550 Pa with the fastest velocity of 1.3 m/s, while the trapezoid groove only has the smallest pressure difference of 310 Pa and the smallest velocity of 0.95 m/s. In view of the axial-section, different flowing mechanisms of grooved shape can be revealed. The lowest pressure region is always on the liquid front under the thin liquid surface. The highest velocity region is also the same region with the lowest pressure. Besides, the
liquid front of the dovetail slot extends to a long distance with a thin thickness, while one of the trapezoidal grooves is short and thick.

Figure 4.12 The (a) pressure distribution and (b) velocity distribution of flowing liquid in grooves with various shapes of triangle, trapezoid, rectangle, and dovetail slot.

Figure 4.13 The pressure and velocity distribution of grooves with various shapes at the filling time of 0.6 ms.
Specifically, in the flowing direction, the maximum flowing velocity achieved by the dovetail slot is 1.95 m/s, which is 77.3% higher than that of the trapezoid (only 1.1 m/s). In all, the dovetail slot is illustrated to be the best optimal grooved shape for the fastest capillary filling. The enhanced mechanism is the smallest meniscus induced by the dovetail slot compared to other shapes.

4.8 Conclusion

The conclusions of this chapter can be drawn as follows:

(1) The capillary filling of a straight channel using COMSOL software is well validated by the analytical Washburn model result.

(2) The wall contact angle (CA) significantly affect the capillary filling velocity. The grooves with CA of 0° induce a filling velocity of 1.8 m/s, far larger than 0.58 m/s of the 60 ° CA case.

(3) The gravity greatly affects the capillary filling of a groove. For example, the velocity of 10 g gravity is around 2.5 times of that of the -10 g gravity.

(4) The groove shape plays an important role in the capillary filling process. The dovetail slot groove is believed to be the optimal shape for the fastest capillary filling.
CHAPTER 5 THE ENHANCED CAPILLARY PERFORMANCE OF
MESH-GROOVE HYBRID WICK AND THE OPTIMIZATION OF
CONTRACTING HYBRID WICK

The mesh-groove hybrid wick is experimentally proved to enhance the capillary filling in the previous chapter. However, the mechanisms of how the hybrid wick enhances the capillary filling process are still not clear, especially in a 3D-free surface view. Understanding the mechanism contributes to further improve the design of the hybrid wick. In this section, we firstly used the simulation method to demonstrate that the capillary filling can be enhanced by the mesh-groove hybrid wick, compared to the grooved wick. Besides, a leaf-structure-inspired contracting hybrid wick has been proposed for the fastest capillary filling. The optimal width ratio between the two channels has been investigated.

5.1 The enhanced capillary filling of mesh-groove hybrid wick

5.1.1 Model setup

The model setup and the meshes for the mesh-groove hybrid wick is presented in Figure 5.1. Some detail for the capillary flowing setting of simulation has been introduced
in section 4.2. Figure 5.1 (a) present the hybrid wick geometry used in this paper. The hybrid wick is connected to a reservoir of liquid, running along Z-axis. The important hybrid wick dimensions are wire width (0.056 mm), wire gap (0.12 mm), meshes layer number (1 layer), groove height (0.25 mm), groove length (1 mm), and groove width (0.2-0.4 mm).

Figure 5.1 The (a) three-dimensional view, boundary conditions setting, and (b) generated mesh of the mesh-groove hybrid wick. The structure and significant parameters are presented. The fluid flows from left to right along the Z-axis.

The simulation parameters are stated as follows. The contact angle of the wetted wall is 60°. The physical models are Laminar Flow, Level Set, and Multiphysics. The Multiphysics couples the two-phase flow and the wetted walls. Initially, the velocity field is set to be \( \mathbf{u} = 0 \) in the x, y, and z direction. The gravity is in a direction of the y-axis, which is to the downside of the schematic. For the level set model, the parameter controlling interface thickness \( \varepsilon_{IS} \) is \( 5 \times 10^{-6} \) m. The mesh type is set to be free tetrahedral with a finer mesh size. The calculation time is 5 ms with a time interval of 0.025 ms. The initial time-step fraction is \( 1 \times 10^{-3} \) with an initial step-growth rate of 1.5.
5.1.2 Results and discussion

Figure 5.2 presents the enhanced capillary filling by the mesh-groove hybrid wick compared to the grooved wick at the contact angle of 60°. The filling processes with contact angles of 0° and 30° are also presented to study the contact angle effect on the capillary filling of hybrid wicks. Firstly, the mesh-groove hybrid wick significantly enhances the capillary flow compared to the groove wick. The filling time of hybrid wick (2.0 ms-60 °) is lower than 40% of that of the groove wick (>>5 ms-60 °). Secondly, as the contact angle decreases, the filling velocity increases. Then, the overall time needed to fill the entire channel will be reduced. Specifically, the filling time of the hybrid wick with 0 ° contact angle (0.9 ms) is 45% of that with 60 ° contact angle (2 ms).

Figure 5.2 The enhanced capillary flow of hybrid wicks (with contact angles of 0°, 30°, and 60°) compared to the grooved wick (with contact angles of 60°). The selected times are (a) 0.5 ms, (b) 1 ms, and (c) 1.5 ms.
Figure 5.3 and Figure 5.4 presents the enhanced mechanisms of the hybrid wick, in view of the liquid front shape and the pressure distribution. The liquid front shapes between the hybrid wick and the groove wick have big differences. The meniscus of the groove is located in the bottom wall, while the meniscus of the hybrid wick is in both the bottom wall and the upper wire surface. The free surface of the groove is long in the Z-axis direction. But the distance (in the Z-axis direction) between the upper and the bottom free surfaces of the hybrid wick is short, inducing a smaller capillary radius.

Figure 5.3 The liquid front shape of (a) hybrid wick and (b) groove at the contact angle of 60 °. Extra meniscuses are induced by the wires, resulting in improved driven pressure.

The pressure distribution between the two wicks also makes a big difference. The lowest pressure region of the groove wick is in the bottom liquid front, while that of the hybrid wick is in both the bottom and upper sides of the liquid front. The average liquid
pressure of the hybrid wick is about -200 Pa, much lower than that of the groove wick (about -30 Pa). In all, the enhanced mechanism of the mesh-groove hybrid wick is the extra meniscus induced by wire, resulting in a larger capillary pressure drop. Specifically, the negative driven pressure was improved from -50 Pa (groove) to -350 Pa after applying mesh wire.

![Pressure distribution of (a) hybrid wick and (b) groove wick.](image)

Figure 5.4 Pressure distribution of (a) hybrid wick and (b) groove wick. The cross-section is parallel to the Z-axis and perpendicular to the bottom wall.

5.2 Optimizing the hybrid wick with a bioinspired contracting channel for the fastest capillary filling

5.2.1 The bioinspired contracting configuration

In nature, the water for capillary-evaporation of leaves is transferred from the trunk to the branches and finally to the leaves. Such structure is defined as contracting branch structures, as presented in Figure 5.5 (a). A reducing pore radius can achieve faster capillary transportation [102], as presented in Figure 5.5 (b). The bioinspired contracting channel can be applied in a hybrid wick for the fastest capillary filling. The investigation
of the hybrid wick with contracting design can efficiently enhance the capillary limit of a heat pipe. In this section, the width ratio of the contracting hybrid wick will be optimized. The enhanced mechanism of the contracting structure will be analyzed in view of the pressure and velocity field.

![Image](image_url)

Figure 5.5 The contracting capillary-radius structure inspired by nature. (a) The contracting branches structure for capillary-evaporation of leaves in nature. (b) The fastest capillary transportation is achieved by reducing pore radius [102].

Figure 5.6 presents the 3D structures of the hybrid wick with different contracting width ratios (ranging from 0.1 to 2). The constraint condition is the same volume of groove, ensuring that the theoretical charging ratio is the same amount. The width ratio (m) is defined as:

\[
m = \frac{w_{out}}{w_{ini}}
\]

(5.1)
where $w_{out}$ is the width of the half channel near the outlet, and $w_{in}$ is the one near the inlet.

With $m=1$, the channel is straight for comparison. The contracting structures have $m$ ranges from 0.1, 0.25, 0.5 to 0.75. The $m$ values for expanding structures are 1.5 and 2.

![Diagram showing 3D structures of contracting hybrid wicks with various width ratios ranging from 0.1 to 2.](image)

**Figure 5.6** The 3D structures of contracting hybrid wicks with various width ratios ranging from 0.1 to 2.

5.2.2 **The optimal width ratio of the contracting hybrid wick for the fastest capillary filling**

Figure 5.7 presents the liquid front shape and filling length of the hybrid wick with various $m$ values. The presented times range from 0.6 ms to 2.6 ms. Firstly, the $m$ value determines the liquid front shape in the half section near the back. The liquid fronts are concave with $m<1$ and convex with $m>1$. Besides, the liquid fronts of hybrid wicks are not smooth curves due to the effect of mesh wire. Secondly, the one with the fastest filling velocity is $m=0.5$, while the one with the lowest filling velocity is $m=2$. On one
hand, the one with $m=0.5$ presents the shortest filling time of 1.8 ms. On the other hand, the one with $m=2$ shows the longest filling time of 4.5 ms. However, the wick with a width ratio of $m=2$ is the first to reaches half of the length. Lastly, achieving an $m$ value smaller than one cannot always enhance the capillary flowing compared to $m=1$. Largely insufficient $m$ will lead to longer filling time. For example, the case with $m=0.1$ takes approximately 3.05 ms to fill the channel, which is 32% longer than that of $m=1$ (2.3 ms). The reason is that this case takes too much time at the front half section of the channel, which has a very large radius and insufficient capillary radius.

Figure 5.8 presents the capillary filling length of hybrid wicks versus time. The black solid line is for $m=1$ as a based line for comparison. The filling velocity is strongly affected by the $m$ value. For $m=1.5$ and 2, the filling lengths are higher than that of $m=1$ at $L<0.5$ mm section. However, they rose slowly at $L$ ranging from 0.5 to 1 mm, and become the slowest one. The curves of $m=0.5$ and $m=0.75$ are the fastest ones to reach $L=1$ mm, although they are slower than the $m=1$ curve at $L<0.5$ mm.

The slopes of the rising curves can be divided into two sections due to different channel radius in the front and back section. For the curves with $m<1$, the slopes transferred from low to high. For the curves with $m>1$, the slopes transferred high to low. In each section, the length is not linear to $t^{0.5}$ due to a micro-scale. Besides, fluctuations are observed in the $L$-$t$ curves, indicating that the capillary flowing velocity is not
uniform. The fluctuations are obvious in hybrid wick, which is caused by the micro-holes of mesh. When the mesh hole is absorbing the liquid with strong capillary pressure, a shortstop is observed on the liquid front, as presented in Figure 5.9.

Figure 5.7 The filling length and liquid front comparison of hybrid wicks at various width ratios (m) ranging from 0.1 to 2. The selected times are (a) 0.6 ms, (b) 1 ms, (c) 1.4 ms, (d) 1.8 ms, (e) 2.2 ms, and (f) 2.6 ms.
Figure 5.8 The filling length-time curves of hybrid wicks with various m value.

Figure 5.10 presents the normalized filling times of the mesh-groove hybrid wick as a function of the width ratio (m). The normalized time ($t_n$) is defined as:

$$t_n = \frac{t_{m=x}}{t_{m=1}}$$  \hspace{1cm} (5.2)

where $t_{m=x}$ is the time needed to fully fill the hybrid wick at $m=x$ and $t_{m=1}$ is the filling time of hybrid wick with $m=1$.

The liquid is filling the mesh hole.

Figure 5.9 The moving-stop of liquid at capillary filling of hybrid wick due to absorption of liquid into micro-hole.
Figure 5.10 The normalized filling times of hybrid wick versus the width ratio (m).

The optimal width ratio ($m$) is 0.5, with a normalized time of 78.2%. The structure with the optimal width ratio well balances the high capillary pressure in the back section and the low flowing resistance in the front section. Besides, the normalized time ($t_n$) is always larger than 1 with $m>1$. The convex meniscus in expanding channel forms a positive pressure region and hinders the movement of liquid. However, the $t_n$ is not always smaller than 1 with $m<1$. The $t_n$ is 132% with $m=0.1$ due to a very slow filling velocity in the front section resulting from a large capillary radius.

5.2.3 Analysis of pressure and velocity field of the contracting hybrid wicks with various width ratios

Figure 5.11 and Figure 5.12 present the pressure distribution at the filling length of 0.5 mm and 0.7 mm, respectively. The results at 0.5 mm present the liquid in the front section, while the results at 0.7 mm show the filling liquid in the back section. The
uniform scale and non-uniform scale are presented for each figure. The uniform scale figures are applied to compare the pressure distribution difference between various \( m \) values. The non-uniform scale is used to present the pressure difference within one hybrid wick configuration.

Figure 5.11 Comparison of pressure field of hybrid wicks with various width ratios at the filling length of 0.5 mm.

At the front section of the channel, the liquid/water interface is not always the region with the lowest pressure. There exists some convex high-pressure regions induced by the mesh. The pressure drop of \( m=2 \) is the maximum due to the smallest capillary radius. At the back section, the liquid surfaces of \( m<1 \) are concave with negative pressure. At this status, the flowing liquid was speeding up. On the contrary, the liquid surfaces of \( m>1 \) are concave with positive pressure, indicating that the filling velocity was slowing
down. At $m=0.1$, the minimum negative pressure is as low as $-1100$ Pa, while at $m=2$, the maximum positive pressure is as high as $+450$ Pa.

Figure 5.12 Comparison of pressure field of hybrid wicks with various width ratios at the filling length of 0.7 mm.

Figure 5.13 Comparison of the velocity field of hybrid wicks with various width ratios at (a) uniform scale and (b) non-uniform scale.
Figure 5.13 presents the velocity field of the mesh-groove hybrid wicks with various m values. The distributions are presented at the moment when the channels are just about to fully filled in both uniform scale and non-uniform scale. When \( m < 1 \), the region with the highest velocity is located in the front of the outlet section. When \( m > 1 \), such region occurs in the outlet of the front section. The maximum local filling velocity for all cases is 1.8 m/s with an m value of 0.1. The most uniform velocity field is induced by \( m = 1 \), with the maximum velocity of 0.58 m/s at the central line of the groove.

5.3 Conclusion

In this chapter, we investigate the mesh-groove hybrid wick for its enhanced capillary performance. Besides, the bio-inspired contracting hybrid wick for the fastest capillary filling performance has been optimized. The main conclusions can be drawn as follows:

(1) The capillary filling process can be significantly enhanced by the mesh-groove hybrid wick in comparison with the groove wick. The capillary filling time (2.0 ms-60 °) of the mesh-groove hybrid wick is less than 40% of the time of the traditional groove wick (>>5 ms-at 60 °).

(2) The enhanced mechanism of the hybrid wick is an extra meniscus on the upper wire surface, resulting in higher capillary pressure. Specifically, the mesh wire can improve the pressure drop from -50 Pa (groove) to -350 Pa (hybrid wick).
(3) The capillary filling process can be enhanced by the bioinspired contracting hybrid wicking structure. The filling velocity is significantly affected by the width ratio (m) of the two sections.

(4) The optimal width ratio (m) is demonstrated to be 0.5, with a normalized time of 78.2% compared to the $m=1$ case.

(5) The configuration with the optimal width ratio well balance a low flowing resistance in the front section and high capillary pressure in the back section.
CHAPTER 6 CONCLUSIONS AND FUTURE WORKS

This study proposed a partially hybrid mesh-groove wick for an L-shaped heat pipe. Contrast experiments were conducted on a fully hybrid wick and a non-hybrid wick (groove) to validate the effectiveness of partially hybrid wick. The coupling effect of mesh-layers number (1, 2, 3, and 6 layers) and charging ratio were experimentally investigated in the present study. The optimal charging ratio region was obtained as a map using a regression analysis of the machine learning algorithm.

Besides, the mechanism of capillary filling of groove is simulated using COMSOL software. The effect of contact angle, gravity, and groove shape on the capillary flowing of groove have been studied in 3D-free surface view. Furthermore, we investigate the mesh-groove hybrid wick for its enhanced capillary performance. Besides, the bio-inspired contracting hybrid wick for the fastest capillary filling performance has been optimized. The main conclusions can be drawn as follows:

(1) The partially hybrid mesh-groove wick (only in the evaporation section) significantly outperforms the grooved wick (non-coverage) and fully hybrid wick in L-shaped heat pipes. The underlying mechanism is the highly efficient capillary evaporation enabled by the hybrid wick. The L-shaped heat pipe performance is not always enhanced using the
hybrid mesh-groove wick. Liquid film confined in the hybrid wick of the condensation section will lead to additional thermal resistance. The strategy of separately optimizing the evaporation section and the condensation section is more effective in maximizing the performance of the heat pipe.

(2) The optimal mesh-layers number increases as the intermeshing bonding strength increases. The optimal mesh layers number is found to be 1-2 layers in the present study. Insufficient or excessive charging ratio would deteriorate the heat pipe performance owing to the early onset of dry-out or liquid pool in the evaporation region. The universal optimal charging ratio of the LPHHPs is recommended to be 100% -200%. More precisely, the optimal charging ratio is in a region depending on the heat load, rather than a point with a constant value.

(3) The capillary filling of a straight channel using COMSOL software is well validated by the analytical Washburn model result. The wall contact angle (CA) significantly affect the capillary filling velocity. The grooves with CA of 0° induce a filling velocity of 1.8 m/s, far larger than 0.58 m/s of the 60 ° CA case. The gravity greatly affects the capillary filling of a groove. For example, the velocity of 10 g gravity is around 2.5 times of that of the -10 g gravity. The groove shape plays an important role in the capillary filling process. The dovetail slot groove is believed to be the optimal shape for the fastest capillary filling.
(4) The capillary filling process can be significantly enhanced by the mesh-groove hybrid wick in comparison with the groove wick. The capillary filling time (2.0 ms-60 °) of the mesh-groove hybrid wick is less than 40% of the time of the traditional groove wick (>>5 ms-at 60 °). The enhanced mechanism of the hybrid wick is an extra meniscus on the upper wire surface, resulting in higher capillary pressure. Specifically, the mesh wire can improve the pressure drop from -50 Pa (groove) to -350 Pa (hybrid wick).

(5) The capillary filling process can be enhanced by the bioinspired contracting hybrid wicking structure. The filling velocity is significantly affected by the width ratio (m) of the two sections. The optimal width ratio (m) is demonstrated to be 0.5, with a normalized time of 78.2% compared to the m=1 case. The configuration with the optimal width ratio well balance a low flowing resistance in the front section and high capillary pressure in the back section.

Future plans include the following two steps:

1. Further study the capillary flowing and evaporation mechanism of hybrid mesh-groove wick.

Although experiments have been fully conducted to study the hybrid mesh-groove wick, the capillary evaporation mechanisms of the hybrid wick are still not very clear. Specifically, the liquid film thickness distribution on the wick, the flowing velocity field,
and how phase change occurs in the liquid-vapor interface need further study. It is also very interesting to investigate how the liquid-front move backward when dry-out occurs. Such mechanisms are important to promote the design of hybrid wick. In the future work, simulation work by Comsol CFD software will be operated for deep understanding.

(2) An optimizing study of hybrid inter-connected groove/mesh wick for capillary evaporation/boiling under point heat sources.

Although the current partially hybrid mesh-groove wick presents highly efficient heat transfer, it is only applicable for uniform face heating conditions. Since the liquid is limited to flow through the neighboring groove, heat can only be transferred along the axial direction of the pipe. Thus, the present hybrid wick can not be applied for point heating conditions. However, an electronic chip is considered as point heating source due to the small dimension and high-power density. Therefore, we should improve the current hybrid wick design to fit the application of point heating source.

Here, we proposed a hybrid inter-connected groove/mesh wick, which allows liquid flow transversely between the axial groove, as shown in Figure 6.1. Further studies are needed to investigate the optimal parameters of this wick, including the main channel width, slot width, and slot density. CFD method will be utilized to conduct the optimal design of the proposed wick.
Figure 6.1 The hybrid inter-connected groove/mesh wick for capillary evaporation under point heat sources.
REFERENCES


