DIMENSIONAL EFFECT OF MICRO CAPILLARY PUMPED LOOP

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ABSTRACT

This study discusses the components’ geometry and its effect on the capability of heat transmission and pressure drop because of its evident influence on the performance of micro capillary pumped loop (MCPL). On analyzing the dimensional effect on heat transmission and pressure gradient of MCPL device, some results were yielded and addressed as follows: The vapor line was the most important factor among the components of MCPL in heat transmission and pressure drop. Furthermore, the depth of vapor line was the main parameter because of its drastic effect. In addition, at depth of vapor line, \( h_v \), ranging from 20 \( \mu \)m to 150 \( \mu \)m, the amount of heat transferred for system will increase, but decrease the pressure drop. However, for \( h_v \) larger than 150 \( \mu \)m, the heat transfer and pressure drop both will reach a limit. A new family of geometrical dimensions of MCPL possessing an excellent heat flux of 178 W/cm\(^2\) would be obtained. These findings will be useful in designing a better MCPL.

Keywords : Micro capillary pumped loop, Dimensional effect, Heat transfer.

1. INTRODUCTION

Since personal communication devices, especially mobile device, have been toward the trend in miniaturization and slimness, the structural packaging density in the devices is increased. Therefore thermal dissipation problem will become an important issue. New technologies are needed to provide adequate cooling performance while meeting the size limitations of the system. Many potential configurations have been considered in developing micro-coolers. The micro capillary pumped loop developed by A. Hoelke [1] was built from a similar concept to the micro loop heat pipe [2,3]. Since then, micro capillary pumped loop was developed by Laura Meyer et al. [4]. All these days, a study for the micro capillary pumped loop has not been performed. A micro-cooler may exploit a micro-scale phase change or single-phase heat transfer to dissipate heat from electronic packages, including miniature heat pipes [5,6]. Regardless of the configuration, an effective micro-cooler will have to be able to fully control the motion of fluids throughout the system. The combination of micro-scale heat transfer and fluid dynamics along with high surface-to-volume ratios makes the development of an efficient micro-cooler challenging.

The conceptual design and fabrication of a micro-CPL was the first addressed by Kirshberg et al. [7]. The initial design involved a completely passive three-port micro-CPL, and is described schematically in their work. Herein, micro-devices could support extremely large gradients of pressure due to the small size.

Although, the MCPL system shown in Fig. 1 and developed by authors [8] starts up successfully and de-priming phenomenon does not occur until heat flux reaches 185.2 W/cm\(^2\). Thus, it can be regarded as an effective cooling device which meets current needs for electronic cooling. In this study, critical geometric parameters of MCPL device will be identified because these issues regarding with micro-scale heat transfer and pressure drop are important for better designing a MCPL.

2. ANALYTIC MODEL

In this study, geometrical parameters based on the theoretical model addressed by Dickey and Peterson [3] were analyzed for its effects on increasing the performance of a capillary pumped loop.
For energy analysis of capillary pumped loop, the energy equation shown in (1) was derived for the evaporator operating at a steady state.

\[ Q = Q_{\text{h}} + Q_{\text{Cp}} + Q_{\text{Cupal}} \]  

(1)

Where \( Q \) indicates the input heat energy, \( Q_{\text{h}} \) is the energy of latent heat for phase change from liquid to vapor. \( Q_{\text{Cp}} \) indicates the energy needed for temperature variations of liquid, and \( Q_{\text{Cupal}} \) is the energy needed for temperature variations of vapor.

Here, the amount of energy needed for temperature variations of vapor based on comparing with liquid can be neglected. In addition, \( Q_{\text{Cupal}} \) can be replaced by (2).

\[ Q_{\text{v}} \Delta T_{\text{liq}} = mc_p(T_v - T_l) \]  

(2)

Hence, Eq. (1) can be simplified to the following:

\[ Q = m h_f + mc_p(T_v - T_l) \]  

(3)

Concerning the system circulation, the driving forces resulting from the capillary pressure drop due to the groove structures and the pressure drop induced by the temperature gradient in the evaporator were the main sources for MCPL. Hence, to ensure system circulation, the total driving forces must be at least larger than all resistant forces resulting from those pressure losses for flow passing the vapor line (\( \Delta P_v \)), liquid line (\( \Delta P_l \)), and groove channel. The limitation shown in (4) must be held for ensuring circulation of system.

\[ \Delta P_v + \Delta P_l \geq \Delta P_c + \Delta P_f + \Delta P_s + \Delta P_g \]  

(4)

To calculate all kinds of pressure drop, the definition based on Young-Laplace Eq. [9] is addressed as follows:

First, the capillary pumping pressure, \( \Delta P_c \), is defined in (5).

\[ \Delta P_c = 2 \sigma \cos \theta \left( \frac{1}{h_g} + \frac{1}{w_g} \right) \]  

(5)

Here, \( \sigma \) is the surface tension, \( \theta \) is the contact angle, \( h_g \) is the depth of groove and \( w_g \) is the width of groove for evaporator. Also, the pressure drop, \( \Delta P_s \), resulting from the temperature variation of the groove at evaporator was obtained by the Clausius-Clapeyron equation and is shown in (6).

\[ \Delta P_s = h_{fg} \left( \frac{P_v}{RT} \right) \]  

(6)

Here, \( h_{fg} \) is enthalpy of latent heat of vaporization, \( P_v \) is saturated vapor pressure, \( R \) is the gas constant, \( T \) is saturated vapor temperature and \( T \) is the temperature gradient at the inside of channel.

According to Darcy’s law [10], the flow was set as laminar flow and the inertial force in microfluid was neglected. Factoring in the viscosity of liquid, permeability, and other parameters, the pressure drop of groove, \( \Delta P_c \), will be set as:

\[ \Delta P_c = \int_0^L \frac{\mu_i \dot{m}_i}{\rho_i A_i} K dx = \frac{\mu_i Q L_c (f_i Re_i) (2h_g + w_g)^2}{8\rho_i h_g N_g (h_g w_g)^3} \]  

(7)

Here, \( \mu_i \) is the viscosity coefficient, \( \dot{m}_i \) is the mass flow rate, \( \rho_i \) is the density of liquid, \( A_i \) is the cross-sectional area that flow passes through, and \( Q \) is the heat flux for input of the system. \( L \) is the length of evaporator, \( h_c \) is the convection coefficient, \( N_g \) is the number of groove, \( K \) is the permeability of the capillary groove and is defined in (8) as the ability of flow movement.

\[ K = \frac{8w_g (h_g w_g)^2 N_g}{f_i Re_i (2h_g + w_g)^2} \]  

(8)

Here (\( f_i Re_i \)) defined in (9) indicates the frictional relationship inside of the capillary channel.

\[ f_i Re_i = 24(1 - 1.3553\alpha + 1.9467\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2537\alpha^5) \]  

(9)

Where the groove aspect ratio, \( \alpha \), is defined as \( \alpha = h_g/w_g \).

As flow passes through a channel whose height is \( h \) and width is \( w \), the pressure drop and frictional relationship can be obtained by the Darcy-Weisbach Eq. [11] and is shown in (10).

\[ \Delta P_l = \frac{\rho \dot{V}^2}{2 D} \]  

(10)

When \( h/w \leq 1.0 \) and the hydraulic diameter \( D = 2hw/(h + w) \), the pressure drop, \( \Delta P_l \), for flow passing through the liquid line can be defined as (11).

\[ \Delta P_l = \rho \left[ \frac{64}{2 \frac{11}{3} 24 w} \frac{h}{h + w} \left( \frac{2 - h}{w} \right) \right] \left( \frac{L}{D} \right) \left( \frac{\rho \dot{V}^2}{2 D} \right) \]  

(11)

Where the Reynolds number of liquid is defined as \( Re_l = \rho \dot{V} D_l / \mu_l \), and the velocity of liquid is defined as \( \dot{V} = \dot{m}_l / \rho_l A_l \).

Similarly, the pressure drop of vapor line is obtained by (12).
\[ \Delta P = \rho \left( \frac{64}{\text{Re}} \right) \frac{2 + 11}{24} \left( \frac{h_v}{w_v} \right) \left( \frac{h_l}{w_l} \right) \left( \frac{v_v^2}{2} \right) \text{ (12)} \]

Where the Reynolds number of vapor is defined as \( \text{Re}_v = \frac{\rho_v \langle v_v \rangle D_v}{\mu_v} \), and the velocity of vapor is defined as \( \langle v_v \rangle = \frac{m_v}{\rho_v A_v} \).

Here, it should be noted that the pressure drop, \( \Delta P_v \), resulting from the force of gravity was neglected because of its lack of an obvious effect in MCPL.

According to the theoretical model, the mass flow rate can be obtained by Eq. (3) and is shown in (14).

\[ Q = \frac{\rho h_v}{h_{fg} + c_p} \Delta T \Rightarrow \frac{Q}{h_{fg} + c_p} = \rho A \nabla \text{ (14)} \]

Hence, the average velocity was found and is shown in (15).

\[ \nabla = \left( \frac{Q}{h_{fg} + c_p} \right) \left( \frac{1}{\rho A} \right) \left( \frac{1}{h_{fg}} \right) \left( \frac{1}{\rho h_w} \right) \text{ (15)} \]

The pressure drop for liquid and vapor has also been obtained by (16) and (17), respectively.

\[ \Delta P_l = \frac{8 \mu_l L_l Q}{\rho h_{fg} (h_l + w_l)^2} \left( \frac{2 + 11}{24} \left( \frac{h_l}{w_l} \right) \left( 2 - \frac{h_l}{w_l} \right) \right) \text{ (16)} \]

\[ \Delta P_v = \frac{8 \mu_v L_v Q}{\rho h_{fg} (h_v + w_v)^2} \left( \frac{2 + 11}{24} \left( \frac{h_v}{w_v} \right) \left( 2 - \frac{h_v}{w_v} \right) \right) \text{ (17)} \]

Therefore, when \( P_s, P_r, P_w, P_l \) and \( P_v \) have been incorporated in (4), the theoretical model for MCPL at equilibrium state can then be obtained and is shown in (18).

\[ 2 \sigma \cos \theta \left( \frac{1}{h_g} + \frac{1}{w_g} \right) + h_{fg} \Delta P / RT_v^2 = \left\{ \begin{array}{c} \frac{8 \mu_l L_l}{\rho h_{fg} (h_l + w_l)^2} \left( \frac{2 + 11}{24} \left( \frac{h_l}{w_l} \right) \left( 2 - \frac{h_l}{w_l} \right) \right) \\
\frac{8 \mu_v L_v}{\rho h_{fg} (h_v + w_v)^2} \left( \frac{2 + 11}{24} \left( \frac{h_v}{w_v} \right) \left( 2 - \frac{h_v}{w_v} \right) \right) \\
\frac{\mu_s L_s \left( \frac{R e_s}{h_{fg}} \right) \left( 2 \frac{h_g}{w_g} \right)^2}{8 \rho h_{fg} N_g \left( h_g w_g \right)^2} Q \end{array} \right\} \text{ (18)} \]

When the geometric dimensions of MCPL and fluidic properties have been decided, the heat transfer could then be determined by Eq. (18). To design a better MCPL, the numerical dimension data of MCPL and a theoretical model of (18) was collected and related to analysis’s of pressure drop and heat transfer for the vapor line, liquid line, capillary groove and evaporator in order to provide a reference in the fabrication of a MCPL device.

To simplify matching the real state of MCPL during the numerical simulation process, some assumptions were made: (1) The environmental pressure of MCPL was set as 1 atm. (2) The temperature gradient of the groove was set as 10°C. (3) The summation of \( (P_w + P_r) \) was assumed to be equal to the summation of \( (P_w + P_l + P_v) \) at the equivalent state of MCPL. (4) The working fluid was pure water and fluidic properties were constant. Some results and discussions related to dimensional effect of MCPL on the heat transmission and pressure drop would be addressed at next section.

3. RESULTS AND DISCUSSIONS

3.1 Dimensional Effect of Vapor Line

In this section, the geometrical influence related to those whose dimensions are length \( L_v \), width \( w_v \), and depth \( h_v \), for vapor line on heat transfer and pressure drop is investigated. Figure 2 shows that the pressure drop will have an observable increase when the length of vapor line increases from \( L_v = 2 \text{ mm} \) to \( 30 \text{ mm} \). In contrast, heat transfer will decrease simultaneously. Figure 3 indicates that increasing the width of vapor line from \( w_v = 1400 \text{ µm} \) will cause a slight increase in heat transfer but almost no effect on pressure drop. Figure 4 shows that changing the depth of the vapor line from \( h_v = 20 \text{ µm} \) to \( 150 \text{ µm} \) will increase the heat transfer and possess a minor effect on pressure drop simultaneously. In addition, the variations of pressure drop, \( P_v \), and heat transfer, \( Q \), will slowly appear in the range of \( h_v = 150 \text{ µm} \) to \( h_v = 270 \text{ µm} \).

3.2 Dimensional Effect of Liquid Line

Concerning the dimensional effect of liquid line related to \( L_l \times h_l \) and \( w_l \) on the heat transfer and pressure drop, some results shown in Figs. 5 to 7 are addressed. Figure 5 shows that increasing the length of the liquid line from \( L_l = 10 \text{ mm} \) to \( 50 \text{ mm} \) appears to cause a slight increase in pressure drop \( P_l \) but heat transfer decreased slightly. Figure 6 indicates that increasing the width of liquid line from \( w_l = 200 \text{ µm} \) to \( 1400 \text{ µm} \) will decrease the pressure drop but slightly opposed the heat transfer. Figure 7 shows that increasing the depth of the liquid line from \( h_l = 20 \text{ µm} \) to \( 160 \text{ µm} \) will decrease the pressure drop and cause a marginal increase in the heat transfer. To sum up these results, the dimensional effect of the liquid line seems unable to increase heat transfer effectively.

3.3 Dimensional Effect of Groove

Research in the heat transfer, pressure drop, and flow characteristics in small scale of groove have become
Fig. 2  Variations of length versus heat flux and pressure drop of closed looped system for vapor line

Fig. 3  Variations of width versus heat flux and pressure drop of closed looped system for vapor line

Fig. 4  Variations of depth versus heat flux and pressure drop of closed looped system for vapor line

Fig. 5  Variations of length versus heat flux and pressure drop of closed looped system for liquid line

Fig. 6  Variations of width versus heat flux and pressure drop of closed looped system for liquid line

Fig. 7  Variations of depth versus heat flux and pressure drop of closed looped system for liquid line
increasingly important for the continuation of technological advances in a wide variety of scientific and engineering fields. Hence, the groove structures whose dimensions are depth \( h_g \), width \( w_g \) and the number of groove \( N_g \) in the evaporator were studied and addressed. Figure 8 indicates that increasing the length of the groove from 20 \( \mu \)m to 40 \( \mu \)m both slightly decreases the pressure drop \( P_w \) and \( P_c \) but a little increase to heat transfer. Figure 9 shows that increasing the depth of groove from 10 \( \mu \)m to 40 \( \mu \)m will both slightly decrease the \( P_w \) and \( P_c \) but heat transfer slightly increased. Figure 10 indicates that increasing the number of groove in evaporator will decrease the pressure drop but produce less variation for heat transfer. To summarize these results, increasing the width and depth of the groove in evaporator will decrease both the pressure resistant \( P_w \) and pressure driving term \( P_c \). Thus, it cannot promote heat transfer effectively.

### 3.4 Dimensional Effect of Evaporator

Generally speaking, the highest temperature will often occur in the evaporator due to the existence of the hot spots. Hence, it is necessary to study the effect of the dimension of the evaporator on the heat transfer \( Q \) and pressure drop \( P_w \). Figure 11 shows that changing the length of the evaporator from \( L_e = 0.4 \) mm to 4.0 mm will create a slight increase in \( P_w \) but almost no effect on heat transfer. Similar results are shown in Fig. 12 for increasing the width of evaporator from \( w_e = 0.2 \) mm to 3.8 mm. Hence, the dimension of the evaporator does not seem to be a main factor for promoting heat transfer.

Finally, taking into consideration the discussions mentioned above, those dimensions of MCPL possessing an excellent heat flux of 178 \( \text{w/cm}^2 \) were obtained and listed on the Table 1.

### 4. CONCLUSIONS

This study discusses the components’ geometry and their effects on the capability of heat transmission and pressure drop for determining a better performance of MCPL.

Some results analyzed and yielded were addressed as following: First, the vapor line was the most dominant factor of influencing the heat transfer and pressure drop than other components of MCPL. Second, the depth of vapor line was the most important parameter because of its drastic effect. In addition, at \( h_v \) ranging from 20 \( \mu \)m to 150 \( \mu \)m, the amount of heat transfer energy for system will increase, but \( P_c \) will decrease instead. However, for \( h_v \) larger than 150 \( \mu \)m, the rate of heat transfer and \( P_c \) both arrived at a limit. Finally, a new family of geometrical dimensions of MCPL possessing an excellent heat flux of 178 \( \text{w/cm}^2 \) would be obtained. These findings will be useful in determining a better design for MCPL.
ACKNOWLEDGMENTS

The authors would like to thank the National Science Council of the Republic of China, Taiwan, for financially supporting this research under Contract No. NSC-96-2221-E-197-015. In addition, authors also like to thank Mr. C.K. Shaw, Department of Mechanical and Aerospace Engineering, University of California: Los Angeles, California, United States of America, for revising the manuscript.

NOMENCLATURE

\( A \)  cross-sectional area that flow passage  
\( C_p \)  Specific heat at constant pressure  
\( h_c \)  convection heat transfer coefficient  
\( h_{fg} \)  enthalpy of latent heat during vaporization  
\( h_g \)  depth of groove for evaporator  
\( h_l \)  depth of liquid line  
\( h_v \)  depth of vapor line  
\( h_w \)  height of groove for evaporator  
\( L_e \)  length of evaporator  
\( L_l \)  length of liquid line  
\( L_v \)  length of vapor line  
\( m_l \)  mass flow rate  
\( N_g \)  number of groove  
\( w_w \)  width of vapor line  
\( w_l \)  width of liquid line  
\( w_g \)  width of groove for evaporator  
\( P_v \)  saturated vapor pressure  
\( Q \)  input heat energy  
\( Q_{C_p \Delta T_{liq}} \)  energy need for temperature variations of liquid  
\( Q_{C_p \Delta T_{vap}} \)  energy need for temperature variations of vapor  
\( Q_{hg} \)  energy of latent heat from liquid phase to vapor phase  
\( R \)  gas constant  
\( Re_l \)  Reynolds number of liquid  
\( Re_v \)  Reynolds number of vapor  
\( T_s \)  saturated temperature of working fluid  
\( T_l \)  Temperature of liquid in evaporator  
\( T_v \)  saturated vapor temperature  
\( \Delta P_c \)  capillary pumping pressure  
\( \Delta P_g \)  pressure drop from the force of gravity  
\( \Delta P_t \)  temperature variation of the groove at evaporator  
\( \Delta P_l \)  pressure losses for flow passing the liquid line  
\( \Delta P_w \)  pressure drop of groove  
\( \Delta P_v \)  pressure losses for flow passing the vapor line  
\( \Delta T \)  temperature gradient at the inside of channel  
\( f_l Re_l \)  frictional relationship inside of the capillary channel

Table 1 Optimal dimensions of MCPL and estimative heat flux

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Optimal dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vapor Line Length, ( L_v )</td>
<td>32mm</td>
</tr>
<tr>
<td>Vapor Line Width, ( w_v )</td>
<td>800( \mu )m</td>
</tr>
<tr>
<td>Vapor Line Height, ( h_v )</td>
<td>150( \mu )m</td>
</tr>
<tr>
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<tr>
<td>Liquid Line Height, ( h_l )</td>
<td>150( \mu )m</td>
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<tr>
<td>Groove Width, ( w_g )</td>
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</tr>
<tr>
<td>Groove Height, ( h_g )</td>
<td>40( \mu )m</td>
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<tr>
<td>Number of Grooves, ( N_g )</td>
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</tr>
<tr>
<td>Evaporator Length, ( L_e )</td>
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<tr>
<td>Evaporator Width, ( W_e )</td>
<td>1mm</td>
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<tr>
<td>Estimative Heat Flux, ( Q )</td>
<td>178w/cm(^2)</td>
</tr>
</tbody>
</table>
σ: surface tension
θ: contact angle
μ: viscosity coefficient
ρ: density of liquid
K: permeability of the groove
α: aspect ratio, defined as $\alpha = h_g/w_g$

REFERENCES


(Manuscript received December 31, 2008, accepted for publication April 28, 2009.)