Numerical exploration of optimal microgroove shape in loop-heat-pipe evaporator

Yuya YAMADA*,**, Masahito NISHIKAWARA*** and Hideki YANADA*

*Department of Mechanical Engineering, Toyohashi Univ. of Tech.
1-1, Hibiyaoka, Tempaku-cho, Toyohashi, Aichi 441-8580, Japan
**currently at Panasonic Co.
1006, Oaza Kadoma, Kadoma-shi, Osaka 571-8501, Japan
***Department of Mechanical Engineering, Worcester Polytechnic Inst.
100 Institute Road, Worcester, MA, 01609, USA
E-mail: nishikawara@me.tut.ac.jp

Received: 12 July 2020; Revised: 22 September 2020; Accepted: 7 October 2020

Abstract
The heat and mass transfer in a loop heat pipe evaporator with microgrooves are investigated in a three-dimensional numerical model. The design of the microgroove wick must consider the pressure loss of the vapor. The simulation obtains the detailed pressure distribution of the vapor flow in the grooves. The simulated heat flux distribution in the evaporator revealed that the applied heat flux concentrates along the three-phase contact line (TPCL) within the casing, wick, and grooves. The effect of TPCL length was investigated in three microgroove wicks with circumferential and axial grooves and a classical wick with only axial grooves. The heat-transfer coefficient initially increased with TPCL length increasing, but thereafter decreased because when the TPCL becomes too long, a large pressure loss occurs in the grooves. The developed model was validated experimentally. In both the model and experiment, the heat-transfer coefficient was locally maximized at a certain TPCL length. The proposed method is expected to provide a simple approach for wick design; especially, the wick shape can be optimized merely by varying the TPCL length. The effect of thermal conductivity of the wick material is also discussed.

Keywords : Capillary evaporator, Heat-transfer coefficient, Loop heat pipe, Microgrooves, Numerical simulation, Three-phase contact line

1. Introduction

A loop heat pipe (LHP) is a heat transport device driven by capillary pressure in porous media and the vapor–liquid phase change of a working fluid. LHPs are widely used in the electronic cooling systems of spacecraft (Okamoto et al., 2016), insulated gate bipolar transistors (Dupont et al., 2013), laptops (Lin et al., 2013), and other thermal engineering applications. As shown in Fig. 1, an LHP consists of an evaporator, a condenser, a vapor line, a liquid line, and a compensation chamber (CC). Unlike conventional heat pipes, the wick is enclosed in the evaporator, enabling long transport lengths and large radiation areas. The operating characteristics and working mechanisms of LHPs are reviewed in Maydanik (2005), Ku (1999), and Launay et al. (2007).

The wick shape as well as porous materials and capillary structures critically affects the evaporator performance, but the design of the wick shape is complicated, which is focused in this paper. Some papers have attempted enhancement of the evaporator heat-transfer coefficient by parametric studies with respect to the shape (e.g., contact surface area between casing and wick, groove number, groove width, groove pitch, groove cross-sectional shape, groove location (in a wick or casing) and wick thickness) with a cylindrical or flat evaporator (North et al., 1997; Riehl et al., 2008; Kiseev et al., 2010; Yakomaskin et al., 2013; Yakomaskin, 2016; Kuroi and Nagano, 2012; Hodot et al., 2013; Wu et al., 2014; Uchida, 2014; Zhang et al., 2012). Especially, the heat-transfer coefficient was significantly improved by forming microgrooves on the wick surface (North et al., 1997; Riehl et al., 2008; Kiseev et al., 2010; Yakomaskin et al., 2013; Yakomaskin,
Microscale grooves regardless of the direction are called microgroove in this paper. North et al. (1997) extended the limit of the applied heat flux to the evaporator by two approaches: increasing the number and size of the grooves circumferentially formed (called circumferential grooves) on the wick and filling the circumferential grooves with a biporous material. The resulting heat-flux limit was boosted tenfold, from 7 W/cm² in a classical evaporator to 70 W/cm² in the modified evaporator. Riehl et al. (2008) manufactured 1,500 circumferential microgrooves per meter, and experimentally and numerically investigated another evaporator with 2,500 microgrooves per meter. The heat-transfer coefficient was 60% higher in the latter than in the former evaporator. Kiseev et al. (2010) conducted parametric experiments on the wick shape. They determined the optimum contact-area ratio between the wick and casing as 0.4–0.5. They also increased the heat-transfer coefficient by reducing the size and separation distance of the grooves. Yakomaskin et al. (2013 and 2016) created fine microgrooves of widths 0.1 mm and 0.3 mm by the deformational cutting method. The 0.1 mm-wide and 0.3 mm-wide grooves delivered their best performances under low and high heat fluxes, respectively. In LHP tests, the 0.1 mm-wide grooves achieved a heat-transfer coefficient of 12,500 W/m²K under a heat flux of 10 W/cm².

Although the above investigations related the evaporator performance to wick shape, the universal optimization of wick shape is extremely difficult because it involves many variables. Circumferential and micro grooves are known to enhance the heat-transfer coefficient, but their design methods are complicated. A simple but comprehensive approach for optimizing the wick shape would realize the full benefit of circumferential grooves and microgrooves.

The heat and mass transfers in an evaporator have been analyzed in a three-dimensional numerical model developed for that purpose (Nishikawara et al., 2015 and 2017a). The developed model considers both fully saturated (i.e., all liquid) and unsaturated (i.e., liquid/vapor) wick states, and simulates the capillary structure in a pore network model. In both wick states, the applied heat flux was concentrated along the three-phase contact line (TPCL) within the casing, wick, and grooves, and the heat was transferred from the TPCL to the grooves with low thermal resistance. Extending the TPCL increases the number of circumferential and axial grooves, thereby enhancing the heat-transfer coefficient (Nishikawara et al., 2017a). However, when the TPCL is excessively extended, a large pressure loss occurs in the excessively small grooves. For this reason, the authors of ref. (Nishikawara and Nagano, 2017b) proposed a design method based on TPCL length alone. They found an optimum TPCL length that maximizes the evaporator heat-transfer coefficient. However, in their study, the pressure loss in the groove was estimated by a simplified one-dimensional model. As pressure loss in the wick with long TPCL is high, accurate prediction is needed. Therefore, this study develops a three-dimensional numerical model that considers the vapor flow and assesses the effect of pressure loss in the grooves on the evaporator heat-transfer coefficient to ensure the design method based on TPCL length alone. Microgroove wicks with different groove configuration are simulated. The method raises the possibility of optimizing the wick shape using one parameter (the TPCL length) instead of conducting parametric simulation with many variables. The calculated results are validated experimentally. Effect of wick material with respect to thermal conductivity is also investigated.

2. Design of wick shape

When designing the wick shape, we noted that both the evaporator heat-transfer coefficient and the pressure loss in the grooves are increasing functions of TPCL length. When the TPCL is too long, the pressure loss in the grooves generates a large distribution of the saturation temperature, and the evaporator heat-transfer coefficient decreases. Under these competing effects, the heat-transfer coefficient is locally maximized at a certain TPCL length. Note that the
The proposed design method enhances the evaporator heat-transfer coefficient rather than the maximum heat transport (i.e., the capillary limit).

The TPCL can be extended by narrowing the grooves and increasing their number. The minimum width of a manufactured groove is obviously limited. On our cylindrical wick, the minimum machined widths of the circumferential and axial grooves were 0.3 mm and 0.4 mm, respectively. Under these limitations, the TPCL length was calculated for different numbers of both types of grooves. The ratio of the casing-wick contact surface area to the inner surface area of the casing was fixed at 0.5. With this setting, the cross-sectional area and resistance of the liquid flow in the wick were independent of wick shape. The maximum heat transport was dictated by the capillary limit and was also invariant, so was excluded from the analysis. When the area of the casing-wick contact surface is fixed, the numbers of the circumferential and axial grooves are determined uniquely. Figure 2 plots the number of circumferential grooves and the TPCL length as functions of number of axial grooves. Here, $L_{thr}$ represents the density of the boundary line between the wick and groove on the area of the casing inner surface, so its unit is $m^{-1}$. $L_{thr}$ is calculated by

$$L_{thr} = \left[ 2L_{ax}N_{ax} + \pi D_{wick} \left( 2N_{ci} + 1 \right) - 2N_{ax}N_{ci} \left( W_{ax} + W_{ci} \right) - W_{ax}N_{ax} \right] / \left( \pi D_{wick}L_{ax} \right)$$

where $L_{ax}$ is length of axial groove, $N_{ax}$ is number of axial grooves, $D_{wick}$ is diameter of wick, $N_{ci}$ is number of circumferential grooves, $W_{ax}$ is width of axial groove and $W_{ci}$ is width of circumferential groove. Four wicks were simulated: wick 1 with 16 axial grooves, 56 circumferential grooves and $L_{thr} = 2630 \, m^{-1}$, wick 2 with four axial grooves, 71 circumferential grooves and $L_{thr} = 3150 \, m^{-1}$, wick 3 with one axial groove, 73 circumferential grooves and $L_{thr} = 3300 \, m^{-1}$, and a classical wick with 16 axial grooves, each of width 1.0 mm. Those were selected to compare with the experiments. The design of wick 1 is shown in Fig. 3. Three wicks (wick 1, wick 2, and the classical wick) were also

![Figure 2](image-url)

**Fig. 2** Number of circumferential grooves (black plot) and TPCL length (red plot) versus number of axial grooves.

![Figure 3](image-url)

**Fig. 3** Design of wick 1 with 16 axial grooves, 56 circumferential grooves, and $L_{thr}=2630 \, m^{-1}$. 
Table 1 Configuration of wicks used in this study.

|                  | Wick 1 | Wick 2 | Wick 3 | Classical |
|------------------|--------|--------|--------|-----------|
| Axial groove number, $N_{ax}$ | 16     | 4      | 1      | 16        |
| Axial groove width, $W_{ax}$    | 0.4    | 0.4    | 0.4    | 1.0       |
| Circumferential groove number, $N_{ci}$ | 56     | 71     | 73     | N/A       |
| Circumferential groove width, $W_{ci}$ | 0.3    | 0.3    | 0.3    | N/A       |
| Length of TPCL, $L_{thr}$       | 2,630  | 3,150  | 3,300  | 1,020     |

Fig. 4 Photos of classical wick, wick 1 and wick 2 made of polytetrafluoroethylene (PTFE).

manufactured for the experiment. Configuration and photos of the wicks are shown in Table 1 and Fig 4.

3. Numerical model

This section describes our three-dimensional numerical model of the evaporator. The model was implemented in the ChtMultiRegionSimpleFoam solver of the OpenFOAM CFD package, which solves steady fluid flow and solid thermal conduction with conjugate heat transfer between multiple regions.

3.1 Computational domain

The computational domain, representing a portion of the periodic structure of the cylindrical evaporator, is shown in Fig. 5. The calculation range of the domain is indicated in Fig. 3. The domain is the half-width portion often considered in the literature (Yiding and Faghri, 1994; Ji and Peterson, 2011; Demidov and Yatsenko, 1994; Kaya and Goldak, 2006; Chernysheva and Maydanik, 2009), and consists of the casing, the wick, and the grooves. Unlike the earlier works, our computational domain includes the sealing section ($0 < x < 5$ mm) in which grooves are not machined.

3.2 Governing equations

The governing equations of the model were constructed under the following assumptions:

(a) The process is steady state.
(b) The porous structure of the wick is homogeneous and isotropic.
(c) The wick is fully saturated with the liquid. Meniscus is developed at the interface between the wick and groove.

Fig. 5. Computational domain of an evaporator with a microgroove wick.
(d) The liquid in the wick is in local thermal equilibrium with the bulk solid.
(e) The vapor flow in the grooves is compressible and laminar, and the vapor is an ideal gas.
(f) The effects of gravity and thermal radiation are negligible.

In addition to the assumption (c), because the simulation doesn’t predict capillary limit, capillary pressure isn’t considered. Under the above assumptions, which are commonly made in the literature (Yiding and Faghri, 1994; Ji and Peterson, 2011; Kaya and Goldak, 2006; Liao et al., 2018), the governing equations of the casing, groove, and wick are given below.

### 3.2.1 Casing

The following energy equation is solved in the casing:

$$k_{\text{case}} \nabla^2 T = 0.$$  \hspace{1cm} (2)

### 3.2.2 Wick

The liquid flow in the porous media is governed by Darcy’s law. The continuity, momentum, and energy equations are respectively given by

$$\nabla \cdot \mathbf{u} = 0,$$  \hspace{1cm} (3)

$$\rho_l (\mathbf{u} \cdot \nabla) \mathbf{u} = -\nabla p + \frac{1}{3} \mu_l \nabla (\nabla \cdot \mathbf{u}) + \frac{\mu_l}{K} \mathbf{u},$$  \hspace{1cm} (4)

$$\nabla \cdot (\rho_l c_p T \mathbf{u}) = k_{\text{eff}} \nabla^2 T,$$  \hspace{1cm} (5)

where $K$ is the permeability and $k_{\text{eff}}$ is the effective thermal conductivity of the wick. Equation (4) is referred to as the Darcy–Brinkman equation (Bars and Worster, 2006). The last term of the right-hand represents the viscous drag forces imposed by the pore walls on the liquid. The bulk solid and the liquid share the same energy equation.

### 3.2.3 Groove

Under assumption (e), the following continuity, momentum, and energy equations are solved in the grooves:

$$\nabla \cdot \rho_v \mathbf{u} = 0,$$  \hspace{1cm} (6)

$$\rho_v (\mathbf{u} \cdot \nabla) \mathbf{u} = -\nabla p + \frac{1}{3} \mu_v \nabla (\nabla \cdot \mathbf{u}) + \mu_v \nabla^2 \mathbf{u},$$  \hspace{1cm} (7)

$$\nabla \cdot (\rho_v c_p T \mathbf{u}) = k_v \nabla^2 T.$$  \hspace{1cm} (8)

The vapor density is calculated by the ideal gas law:

$$\rho_v = \frac{p}{RT}.$$  \hspace{1cm} (9)

### 3.3 Boundary conditions

Heat transfer among the casing, wick and groove is coupled as a conjugate problem. The boundary conditions of the momentum and energy depend on the type of boundary as described below.

(i) At $y = h_{\text{wick}} + h_{\text{case}}$, the heat flux is constant and applied to whole upper surface of the casing:

$$\dot{q}_{\text{apply}} = -k_{\text{case}} \frac{\partial T}{\partial y}.$$  \hspace{1cm} (10)
(ii) At \( y = 0 \) on the wick–CC interface (inlet for the working liquid), the heat leak to the CC and the constant-pressure condition are respectively given as

\[
-h_{\text{wick-cc}} (T - T_{\text{cc}}) = -k_{\text{eff}} \frac{\partial T}{\partial y},
\]

\[
p = p_{\text{cc}} = p_{\text{evap}}.
\]

(iii) At \( x = 0 \) on the casing, the heat leak through the casing to the CC is given by

\[
-h_{\text{case-cc}} (T - T_{\text{cc}}) = -k_{\text{case}} \frac{\partial T}{\partial x}.
\]

(iv) To obtain a pressure distribution in the groove, the constant-pressure condition is imposed at \( x = L_{\text{evap}} \) of the groove outlet (vapor line inlet):

\[
p = p_{\text{out}} = p_{\text{dis}}.
\]

As the LHP is a closed system, the pressure distribution in the groove depends on the heat flux applied to the casing, and on the elevation heads and pressure losses of the vapor and liquid lines (including the condenser section). Therefore, \( p_{\text{out}} \) is difficult to acquire and was set to an arbitrary value in this study.

(v) At the impervious walls (\( x = 0 \) and \( x = L_{\text{evap}} \)) except (iii) and (iv), the insulation conditions hold:

\[
\mathbf{u}_v = \mathbf{0}, \quad \mathbf{u}_l = \mathbf{0},
\]

\[
\frac{\partial T}{\partial x} = 0
\]

(vi) the symmetric boundary conditions are imposed at \( z = 0 \) and \( z = p_{\text{cav}}/2 \).

(vii) At the casing-wick and casing–groove interfaces, the non-slip conditions are given by Eq. (17), and the temperature equilibrium and energy balance requirements are given by Eq. (18):

\[
\mathbf{u}_v = \mathbf{0}, \quad \mathbf{u}_l = \mathbf{0},
\]

\[
-k_{\text{case}} \frac{\partial T}{\partial y} = -k_{\text{eff}} \frac{\partial T}{\partial y},
\]

\[
-k_{\text{case}} \frac{\partial T}{\partial y} = -k_{\text{eff}} \frac{\partial T}{\partial y}.
\]

(vi) At the vapor–liquid interface \( \Gamma \) between the groove and wick, the mass and energy are respectively conserved as follows:

\[
\mathbf{u}_v \rho_v = \mathbf{u}_l \rho_l = \mathbf{m}_{\text{evap}},
\]

\[
\mathbf{m}_{\text{evap}} = \left( -k_{\text{eff}} \frac{\partial T}{\partial n_{\Gamma}} - k_{\text{eff}} \frac{\partial T}{\partial n_{\Gamma}} \right) h_{fg},
\]

where \( n \) is the coordinate in the direction normal to the \( \Gamma \) interface, and \( h_{fg} \) is the latent heat of the working fluid. The first term in the numerator of Eq. (20) describes the thermal conduction of the liquid at the wick side, while the second
term represents the thermal conduction of the vapor at the groove side. The temperature $T_{\text{int}}$ at the vapor–liquid interface is the saturation temperature corresponding to the pressure of the vapor adjacent to the interface.

$$T_{\text{int}} = T_{\text{sat}}[p_v].$$ (21)

The computational domain was discretized by the finite volume method. The computational grid was a regular hexahedral lattice with a grid size $\Delta$ of 0.05 mm. Wick 3 was described by the maximum number of cells (16 million). The above governing equations were solved by the SIMPLE algorithm (Patanker, 1980). After convergence, the energy conservation of the computational domain was satisfied within 0.1%.

4. Results and discussion

The calculation conditions are shown in Table 2. The heat flux applied to the casing was 1.3 W/cm$^2$. The wick material was PTFE, the casing was constructed from stainless steel, and the working fluid was ethanol. The effective thermal conductivity of the wick with 0.34 of porosity was obtained by solving the three-dimensional thermal conduction in the reconstructed porous media saturated with ethanol liquid (Nishikawara et al., 2018). The permeability was obtained by measuring pressure difference of a disk porous sample under a constant flow rate. $h_{\text{wick-cc}}$ is convective heat-transfer coefficient referred from the Ref (Chernysheva and Maydanik, 2012). $h_{\text{case-cc}}$ was calculated based on thermal conduction of shape of the casing and the CC. The fluid properties were calculated using the Reference Fluid Thermodynamic and Transport Properties (REFPROP) program (Lemmon et al., 2013).

Table 2 Calculation conditions of the present study (Chernysheva and Maydanik, 2012; Nishikawara and Nagano, 2017b; Nishikawara et al., 2018; Lemmon et al., 2013).

| Properties of working fluid: Ethanol at 330 K |  |
|---------------------------------------------|-----------------|
| Latent heat, $h_{fg}$ | J/kg | 882,000 |
| Saturation temperature, $T_{\text{sat}}[p_v]$ | K | $21.705 \times \ln(p_v)+100.38$ |
| Specific heat at constant pressure of liquid, $c_{pl}$ | J/kg K | 2,710 |
| Specific heat at constant pressure of vapor, $c_{pv}$ | J/kg K | 1,590 |

| Properties of casing material: Stainless steel (SUS304) |  |
|--------------------------------------------------------|-----------------|
| Thermal conductivity, $k_{\text{case}}$ | W/m-K | 16.7 |

| Properties of wick material: PTFE porous media |  |
|----------------------------------------------|-----------------|
| Effective thermal conductivity, $k_{\text{eff}}$ | W/m-K | 0.215 |
| Permeability, $K$ | m$^2$ | $2.0 \times 10^{-14}$ |

| Boundary conditions |  |
|---------------------|-----------------|
| Applied heat flux to the evaporator, $\dot{q}_{\text{apply}}$ | W/cm$^2$ | 1.3 |
| Pressure of groove outlet, $p_{\text{out}}$ | Pa | 31,500 |
| Pressure of wick inlet, $p_{\text{in}}$ | Pa | 27,100 |
| CC temperature, $T_{cc}$ | K | 308 |
| Heat-transfer coefficient between wick and CC, $h_{\text{wick-cc}}$ | W/m$^2$ K | 100 |
| Heat-transfer coefficient between casing and CC, $h_{\text{case-cc}}$ | W/m$^2$ K | 700 |
Figure 6 displays (a) the streamlines, (b) velocity contours, and (c) pressure contours of the working liquid in the classical wick, along with (d) the temperature contours in the $y$–$z$ cross-section at $x = L_{\text{evap}}/2$ of the classical wick (d). Similar trends were observed for the other wicks. The liquid flowed upward from the bottom surface of the wick ($y = 0$) and evaporated at the vapor–liquid interface between the groove and the wick (Fig. 6(a)). Observe that most of the working liquid flowed to the vapor–liquid interface close to the TPCL. That is, the evaporation concentrated on the TPCL, which explains why the heat-transfer coefficient increases by extending the TPCL. The evaporation concentration maximized the velocity and minimized the pressure at the TPCL, as shown in Fig. 6 (b) and (c), respectively. The maximum pressure difference at the vapor–liquid interface which was calculated by subtracting liquid phase pressure in the wick adjacent to the vapor–liquid interface from vapor pressure in the groove adjacent to the same interface was 4.6 kPa, smaller than the capillary pressure (19.2 kPa) corresponding to the maximum pore radius ($1.7 \mu$m). Owing to the differences in the cross-sectional area, the pressure gradient was larger in the fin section ($y > 0.5$ mm) than in the base section ($y < 0.5$ mm) of the wick. The casing was uniformly heated by the heat load and the heat was transferred to the working fluid in the groove and the wick (Fig. 6(d)). The upward- flowing working liquid in the wick was rapidly heated close to the casing, especially at the vapor–liquid interface side. This indicates a large heat flux along the $y$-direction, which evaporates the working liquid. This result is consistent with the streamlines and the velocity contours shown in Fig. 6 (a) and (b), respectively.

For a quantitative analysis, the dimensionless heat flux was calculated as

$$
\dot{q}^* = \frac{-k_{\text{eff}} \nabla T}{-\dot{q}_{\text{apply}}}.
$$

Figure 7 presents the analytical heat fluxes in the classical wick and wick 1. In both wicks, the heat flux concentrated at the TPCL along circumferential grooves as well as axial grooves. The maximum dimensionless heat fluxes in the classical wick, wick 1, wick 2, and wick 3 were 8.0, 6.8, 6.2, and 5.9, respectively. Lengthening the TPCL dispersed the heat flux, thereby lowering the contribution of the TPCL to the heat transport process. This result demonstrates the literature assertion (Nishikawara and Nagano, 2017b).

Figure 8 shows the temperature contours in the $x$–$y$ cross sections of the classical wick and wick 1 at $z = p_{\text{axi}}/2$. The $x$–$y$ plane at $z = p_{\text{axi}}/2$ is the central cross-section of the axial groove, which represents a symmetric boundary. No temperature gradient developed in the $y$- and $z$-directions of the casing (see Fig. 6 (d)). However, the temperature gradient was large in the $x$-direction of the casing, close to the sealing section ($0 < x < 5.0$ mm). In the sealing section, where grooves were not formed, the vapor–liquid interface was absent, so the casing temperature increased due to the thermal-conduction resistance of the wick. For this reason, a wastefully long sealing section should be avoided. The maximum temperature was lower in the evaporator with the microgroove wick than in the evaporator with the classical wick, because the heat flux concentrated at the longer TPCL in the former case.

![Fig. 6 Physical and thermal behaviors in the classical wick: (a) streamlines, (b) velocity contours, (c) pressure contours of the working liquid in the wick and (d) temperature contours in the $y$–$z$ cross-section of the wick at $x = L_{\text{evap}}/2$.](image-url)
Fig. 7 Distribution of dimensionless heat flux $q^\ast$ on the surface where the groove and wick contact the casing ($x$–$z$ plane, $y = 1.5$ mm). The horizontal bottom line represents one half-width of an axial groove, and several circumferential grooves are positioned in the orthogonal direction.

Fig. 8 Comparison of the temperature contours in the $x$–$y$ cross sections of the classical and microgroove wicks at $z = p_{axi}/2$.

Fig. 9 Comparison of the vapor-velocity contours in the $x$–$y$ cross sections of the classical and microgroove wicks at $z = p_{axi}/2$.

Fig. 10 Comparison of the vapor-pressure contours in the $x$–$y$ cross sections of the classical and microgroove wicks at $z = p_{axi}/2$.

Figures 9 and 10 show the velocity and pressure contours, respectively, in the $x$–$y$ cross sections of the classical wick and wick 1 at $z = p_{axi}/2$. The vapor velocity increased along the flow direction ($x$-direction), driven by accumulation of the working fluid evaporated from the vapor–liquid interface. As the areas of the groove outlets of the microgroove and classical wicks differed by a factor of 2.5, the vapor velocity and pressure loss were increased in the microgroove wick. In the microgroove wick, the velocity increases at the intersection of axial and circumferential grooves. Therefore, the pressure loss of the vapor through the grooves must be considered when designing the microgroove wick.

Figure 11 plots the casing temperature and pressure loss of the vapor through the grooves as functions of TPCL length. The casing temperature, defined as the average temperature on the heating surface, decreased with increasing TPCL length except in wick 3 with the highest TPCL length. As mentioned above, this behavior was affected by the distribution of the saturation temperature caused by pressure loss of the vapor in the grooves. The maximum temperature appears above the sealing section (left side in Fig. 8) in all cases.

The pressure loss increased with TPCL length. A wick with a long TPCL has fewer axial grooves than one with a
shorter TPCL, so the vapor velocity in these grooves increases. In addition, the pressure loss is larger through the circumferential grooves than through the axial grooves, as the latter are spaced farther apart. These two observations explain why the pressure loss increased with TPCL length. The pressure loss predicted by the three-dimensional model is larger than that by one-dimensional model shown in Ref. (Nishikawara and Nagano, 2017b). The classical wick with negligible pressure loss (18 Pa) can be modeled as a simple two-dimensional problem. The maximum vapor–liquid interface temperature was calculated by Eq. (21). Note that when the pressure loss increased the vapor pressure at the upstream side of the grooves, the vapor–liquid interface temperature increased concordantly. In wick 3, where the pressure loss was maximized, the maximum and minimum saturation temperatures differed by 4.1 °C. A higher saturation temperature hinders evaporation of the working fluid and raises the casing temperature. The distribution of the saturation temperature depends on the gradient of the saturation curve \( \frac{dT}{dP_{sat}} \), which is peculiar to the working fluid. The large gradient leads to the large temperature distribution.

Figure 12 compares the simulated evaporator heat-transfer coefficients and their equivalent experimental values obtained at the heat flux of 1.3 W/cm². The LHP with a 50 mm-long and 12 mm-diameter evaporator, a 1.15 m-long vapor line, a 0.5 m-long condenser, a 1.56 m-long liquid line was used in the experiment. An aluminum heater block with four cartridge heaters was attached to the evaporator. The condenser was mounted on a cold plate with the pipe where coolant of an ethylene glycol/water mixture was circulated by a chiller unit. During the test, the entire LHP was covered
with thermal insulators but heat leak from the evaporator to ambient was measured and considered in calculating heat flux applied to the evaporator. Two absolute pressure sensors were located at the inlet of the vapor line and the outlet of the liquid line to measure the saturation temperature of the fluids. The casing temperatures were measured by T-type thermocouples. The experimental apparatus is detailed in (Nishikawara and Nagano, 2014). The evaporator heat-transfer coefficient \( h_{evap} \) was calculated as follows:

\[
 h_{evap} = \frac{\dot{q}_{apply}}{T_{case} - T_{sat}[p_{vl}]},
\]

with

\[
\dot{q}_{apply} = \frac{\dot{Q}_{apply}}{A},
\]

\[
A = \pi D_{wick} L_{wick}.
\]

In these expressions, \( T_{case} \) is the average temperature of the heating surface on the casing, \( T_{sat[p_{vl}]} \) is the saturation temperature at the inlet of the vapor line, \( A \) is the area of the casing inner wall, \( D_{wick} \) is wick diameter, and \( L_{wick} \) is wick length. In both the simulation and experiment, the heat-transfer coefficient was locally maximized at a specific TPCL length. The evaporator heat-transfer coefficient increases with the TPCL length where applied heat flux concentrates and efficient heat transfer takes place (positive effect). However, the pressure loss in the grooves increases at the same time, thereby leading to a large distribution of saturation temperature as mentioned above paragraph. Consequently, the evaporator heat-transfer coefficient decreases as the TPCL length increases (negative effect). Because of these competing effects, there is a local maximum in the heat-transfer coefficient at a certain TPCL length. The characteristics of the heat-transfer coefficient as a function of TPCL length is predicted with not only the simplified one-dimensional model shown in the Ref. (Nishikawara and Nagano, 2017b), but also the three-dimensional simulation to predict pressure loss through microgroove, ensuring validity of the design method of the evaporator with the TPCL length alone. The different heat-transfer coefficients obtained in the simulation and experiment can be attributed to heat transfer through the meniscus at the TPCL, as shown in Fig. 13, which was ignored in the simulation. Because of wettability, the menisci form at all corners of the grooves unless the wick is two-phase state. The effect of the meniscus which is affected by pressure difference at the liquid-vapor interface and two contact angles with the wick and the casing might be significant, especially at low heat flux (Yamada et al., 2019), as the meniscus can direct the applied heat to the groove. It is considered that the simulation underestimated the heat transfer at the TPCL. For a precise determination of the heat-transfer coefficient, the meniscus on the TPCL with thin liquid film must be included in the heat and mass transfer model. The heat and mass transfer on the meniscus can be integrated by imposing the heat transfer coefficient predicted by microscopic model of meniscus at the boundary between the casing and groove.

Local maxima in the simulated and experimental curves appeared even at heat fluxes of 0.63 W/cm\(^2\) and 2.0 W/cm\(^2\) and wick 1 achieved the maximum heat flux of 3.4 W/cm\(^2\) and heat-transfer coefficient of 2,400 W/m\(^2\)K, although those results are not shown. The maximum temperature of the casing is important rather than the average temperature in some cases. Although only the heat-transfer coefficient calculated with the average temperature of the casing is shown, the maximum temperature of the casing has the same tendency to the TPCL length as well.

To investigate the effect of thermal conductivity of the wick material, the wick material in the simulation was changed to stainless steel (SS). The SS–ethanol LHP has a higher heat-transfer coefficient than the PTFE–ethanol LHP (Fig. 14). Because the evaporator heat-transfer coefficient is dominated by thermal conduction from the heating surface to the liquid–vapor interface on the wick, increasing the interface area is more beneficial to a highly conductive wick than to a lowly conducting wick. Comparing the results of the PTFE–ethanol and SS–ethanol LHPs, the bulk thermal conductivities of PTFE and SS were determined as 0.25 and 16.7 W/m K, respectively, meaning that SS was 67 times more conductive than PTFE.

When the wick material changed to SS, the peak of the heat-transfer coefficient shifted to a shorter TPCL length. As the working fluid was unchanged, the pressure losses of vapor and distributions of the saturation temperature were very similar in the two LHPs. However, in the SS–ethanol LHP, the temperature difference caused by thermal conduction
The resistance in the wick was small, thereby enhancing the negative effect of the saturation temperature distribution. The heat-transfer coefficient with SS wick is more sensitive to the effect of pressure loss through groove that decreases heat-transfer coefficient.

From the above results, it was concluded that the pressure losses of vapor and distribution of the saturation temperature should be considered in the microgroove design, especially when the wick is formed from a thermally conductive material. In addition, the shape of the wick can be optimized by establishing a tradeoff between increasing the heat-transfer coefficient and reducing the pressure loss in the grooves. Balancing this tradeoff requires careful selection of the TPCL length. A detailed heat-transfer simulation at the TPCL should be performed in future work.

5. Conclusions

This paper presented the results of thermo-fluid simulations in an LHP evaporator with microgrooves. The LHP was simulated in a three-dimensional model considering the vapor flow through the grooves. The effect of the TPCL length was examined in three microgroove wicks with both circumferential and axial grooves, and in a classical wick with axial grooves only.

The heat flux was found to concentrate on the TPCL within the casing, wick, and grooves. The evaporator heat-transfer coefficients of LHPs with two wick materials (polytetrafluoroethylene and stainless steel) were locally maximized at a certain TPCL length at less than 2.0 W/cm² of heat flux. The saturation-temperature distribution caused by pressure loss of the vapor through the grooves reduced the heat-transfer coefficient. The comparison with experiments ensures validity of design method with the TPCL length alone. The shape of the wick could be optimized by adjusting...
the TPCL length in the case that meniscus at the TPCL is ignored. To more accurately obtain the heat-transfer coefficients, the simulation model must consider the meniscus effects on the TPCL.

Nomenclature

\[ A \quad \text{area, m}^2 \]
\[ c_p \quad \text{specific heat at constant pressure, J/(kg\cdot K)} \]
\[ dT/dP_{sat} \quad \text{gradient of the saturation curve, K/Pa} \]
\[ h \quad \text{height, m} \]
\[ h_t \quad \text{heat-transfer coefficient, W/(m}^2\cdot\text{K)} \]
\[ h_{fg} \quad \text{latent heat, J/kg} \]
\[ k \quad \text{thermal conductivity, W/(m\cdot K)} \]
\[ K \quad \text{permeability, m}^2 \]
\[ L \quad \text{length, m} \]
\[ m_{\text{evap}} \quad \text{evaporative mass flux, kg/(m}^2\cdot\text{s)} \]
\[ p \quad \text{pressure, Pa} \]
\[ p \quad \text{pitch, m} \]
\[ q_{\text{apply}} \quad \text{heat flux applied to the evaporator, W/m}^2 \]
\[ \dot{Q}_{\text{apply}} \quad \text{amount of heat applied to the evaporator, W} \]
\[ R \quad \text{gas constant, J/(kg\cdot K)} \]
\[ T \quad \text{temperature, K} \]
\[ u \quad \text{velocity, m/s} \]
\[ w \quad \text{width, m} \]

Greek

\[ \mu \quad \text{viscosity, Pa\cdot s} \]
\[ \rho \quad \text{density, kg/m}^3 \]

Subscripts

\[ \text{axi} \quad \text{axial} \]
\[ \text{cap} \quad \text{capillary} \]
\[ \text{cc} \quad \text{compensation chamber} \]
\[ \text{cir} \quad \text{circumferential} \]
\[ \text{evap} \quad \text{evaporator} \]
\[ \text{eff} \quad \text{effective} \]
\[ \text{int} \quad \text{vapor-liquid interface} \]
\[ \text{sat} \quad \text{saturation} \]
\[ \text{v} \quad \text{vapor} \]
\[ \text{vl} \quad \text{vapor line} \]

References

Bars, M., and Worster, M., Interfacial conditions between a pure fluid and a porous medium: implications for binary alloy solidification, Journal of Fluid Mechanics, Vol.550 (2006) DOI:10.1017/S0022112005007998
Chernysheva, M., and Maydanik, Y., Heat and mass transfer in evaporator of loop heat pipe, Journal of Thermophysics Heat Transfer, Vol.23 (2009) DOI: 10.2514/1.43244
Chernysheva, M., and Maydanik, Y., 3D-model for heat and mass transfer simulation in flat evaporator of copper-water loop heat pipe, Applied Thermal Engineering, Vol. 33 (2012) DOI: 10.1016/j.applthermaleng.2011.09.025
Demidov, A., and Yatsenko, E., Investigation of heat and mass transfer in the evaporation zone of a heat pipe operating by the ‘inverted meniscus’ principle, International Journal of Heat and Mass Transfer, Vol.37 (1994) DOI:
Dupont, V., Vanoolst, S., and Barremaecker, L., Railways qualification tests of capillary pumped loop on a train, Proc. of 17th International Heat Pipe Conference, (2013).

Hodot, R., Sartre, V., Lefevre, F., Sarno, C., 3D modeling and optimization of a loop heat pipe evaporator, in: 17th International Heat Pipe Conference. (2013) pp.1–6.

Ji, L., and Peterson, G., 3D heat transfer analysis in a loop heat pipe evaporator with a fully saturated wick, International Journal of Heat and Mass Transfer, Vol.54 (2011) DOI: 10.1016/j.ijheatmasstransfer.2010.09.014

Kaya, T., and Goldak, J., Numerical analysis of heat and mass transfer in the capillary structure of a loop heat pipe, International Journal of Heat and Mass Transfer, Vol.49 (2006) DOI: 10.1016/j.ijheatmasstransfer.2006.01.028

Kiseev, V., Vlassov, V., and Muraoka, I., Experimental optimization of capillary structures for loop heat pipes and heat switches, Applied Thermal Engineering, Vol.30 (2010) DOI: 10.1016/j.applthermaleng.2010.02.010

Kuroi, M., Nagano, H., The influence of groove shape on loop heat pipe performance, Heat Pipe Sci. Technol., Int. J. Vol.3 (2012) DOI: 10.1615/HeatPipeSciTech.2013006554

Launay, S., Sartre, V., and Bonjour, J., Parametric analysis of loop heat pipes operation: a literature review, International Journal of Thermal Science, Vol.46 (2007) DOI: 10.1016/j.ijthermalsci.2006.11.007

Lemmon, E., Huber, M., and McLinden, M., NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties: REFPROP, Version 9.1, National Institute of Standards and Technology, Standard Reference Data Program, (2013).

Liao, Z., Xu, C., Ren, Y., Gao, F., Ju, X. and Du, X., Thermal analysis of a conceptual loop heat pipe for solar central receivers, Energy, Vol.158 (2018) DOI: 10.1016/j.energy.2018.06.069

Lin, Z., Lin, W., Zhang, L., and Wang, S., An experimental study on applying miniature loop heat pipes for laptop PC cooling, Proc. of Semiconductor Thermal Measurement and Management Symposium, (2013).

Maydanik, Y.F., Loop heat pipes, Applied Thermal Engineering, Vol. 25 (2005) DOI: 10.1016/j.applthermaleng.2004.07.010

Nishikawara, M., and Nagano, H., Parametric experiments on a miniature loop heat pipe with PTFE wicks, International Journal of Thermal Sciences, Vol.85 (2014) DOI: 10.1016/j.ijthermalsci.2014.05.016

Nishikawara, M., Nagano, H., Mottet, L., and Prat, M., Formation of unsaturated regions in the porous wick of a capillary evaporator, International Journal of Heat and Mass Transfer, Vol.89 (2015) DOI: 10.1016/j.ijheatmasstransfer.2015.05.054

Nishikawara, M., Nagano, H., and Prat, M., Numerical Study on Heat Transfer Characteristics of Loop Heat Pipe Evaporator Using Three-Dimensional Pore Network Model, Applied Thermal Engineering, Vol. 126, No. 5 (2017a) DOI: 10.1016/j.applthermaleng.2017.02.050

Nishikawara, M., and Nagano, H., Optimization of wick shape in a loop heat pipe for high heat transfer, International Journal of Heat and Mass Transfer, Vol. 104 (2017b) DOI: 10.1016/j.ijheatmasstransfer.2016.09.027

Nishikawara, M., Ueda, Y., and Yanada, H., Liquid-vapor phase displacement in the evaporator of a loop heat pipe at the start-up involving nucleate boiling, Transactions of the JSME (in Japanese), Vol. 84, No. 860 (2018) DOI:10.1299/transjsme.17–00576

North, M., Sarraf, D., Rosenfeld, J., Maidanik, Y., and Vershinin, S., High heat flux loop heat pipes, AIP Conference Proc., Vol.387 (1997) pp.371–376.

Okamoto, A., Hatakenaka, R., Miyakita, T., and Sugita, H., Development of 100W-class Loop Heat Pipes for Space Use and On-orbit Experiment Test Plan, Proc. of Joint 18th International Heat Pipe Conference and 12th International Heat Pipe Symposium, (2016).

Patankar, S., Numerical heat transfer and fluid flow, Taylor&Francis, ISBN-13: 978-0891165224, (1980).

Riehl, R., Santos, N., and Santos, N., Loop heat pipe performance enhancement using primary wick with circumferential grooves, Applied Thermal Engineering, Vol.28 (2008) DOI: 10.1016/j.applthermaleng.2007.11.005

Uchida, H., Thermal performance of a loop heat pipe containing an evaporator with a pin array conduction structure for electronic devices, Therm. Sci. Eng. Vol.22 (2014) pp.85–95 (in Japanese).

Wu, S.C., Wang, D., Gao, J.H., Huang, Z.Y., and Chen, Y.M., Effect of the number of grooves on a wick’s surface on the heat transfer performance of loop heat pipe, Appl. Therm. Eng. Vol.71 (2014) DOI:
Yakomaskin, A. A., Afanasiev, V., Zubkov, N., and Morskoy, D., Investigation of heat transfer in evaporator of microchannel loop heat pipe, Journal of Heat Transfer, Vol.135 (2013) DOI:10.1115/HT2012-58503
Yakomaskin, A.A., Prototype of thin loop heat pipe with glass fiber wick for compact electronics cooling, Proc. of Joint 18th International Heat Pipe Conference and 12th International Heat Pipe Symposium, (2016) pp. 636–641.
Yamada, Y., Nishikawara, M., Yanada, H., and Ueda, Y., Predicting the performance of a loop heat pipe considering evaporation from the meniscus at the three-phase contact line, Thermal Science and Engineering Progress, Vol. 11 (2019) DOI: 10.1016/j.tsep.2019.03.022
Yiding, C., and Faghri, A., Conjugate analysis of a flat-plate type evaporator for capillary pumped loops with three-dimensional vapor flow in the groove, International Journal of Heat and Mass Transfer, Vol.37 (1994) DOI: 10.1016/0017-9310(94)90074-4
Zhang, X., Li, X., and Wang, S., Three-dimensional simulation on heat transfer in the flat evaporator of miniature loop heat pipe, Int. J. Therm. Sci. Vol.54 (2012) DOI: 10.1016/j.ijthermalsci.2011.12.002