An overall numerical investigation on heat and mass transfer for miniature flat plate capillary pumped loop evaporator

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ABSTRACT

Two-dimensional mathematical model of the miniature flat plate capillary pumped loop (CPL) evaporator is presented to simulate heat and mass transfer in the capillary porous structure and heat transfer in the vapor grooves and metallic wall. The overall evaporator is solved with SIMPLE algorithm as a conjugate problem. The shape and location of vapor–liquid interface inside the wick are calculated and the influences of applied heat fluxes, liquid subcooling and metallic wall materials on the evaporator performances are discussed in detail. The effect of heat conduction of side metallic wall on the performance of miniature flat plate CPL evaporator is also analyzed, and side wall effect heat transfer limit is introduced to estimate the heat transport capability of capillary evaporator. The conjugate model offers a numerical investigation in the explanation of the robustness of the flat plate CPL operation.

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1. Introduction

In the developments of the small spacecraft in the industry, thermal control requirements to such systems have certainly outgrown the capability of conventional single-phase systems and heat pipes in terms of heat transport, heat density, transfer distance and temperature control [1,2]. The capillary pumped loop (CPL) is a two-phase thermal control device, which has been advanced in recent twenty years. As a derivative of the heat pipe, CPL is capable of transporting large heat density and passively transporting heat over large distances with minimal temperature differences, and uses capillary action for fluid transport and contains no moving parts. Moreover heat is transferred along with evaporation and condensation, a CPL is much more economical in terms of weight than conventional heat systems [3,4]. Therefore, miniature CPL is especially well suited for thermal management in small spacecraft [5], and because flat plate evaporator has the advantages of good contact condition, low thermal resistance and isothermal heated surface, our primary interest in this investigation is miniature flat plate CPL.

As shown in Fig. 1, a generic miniature flat plate CPL consists of an evaporator, a condenser, a reservoir, vapor and liquid transport lines. Detailed descriptions of working principle of CPL can be found in Nakayama et al. [6]. The capillary evaporator is the most important part in a CPL system because it is the heat absorbing element and provides the capillary force of fluid through the loop. Consequently, the optimal design of CPL requires a thorough understanding of physical behaviors occurring inside the evaporator.

Cao and Faghri [7] developed an analytical solution for heat and mass transfer processes during evaporation in the wick of a CPL evaporator. In this study, it was assumed that the entire porous structure was saturated with liquid and the liquid–vapor interface was located on the unheated portion of the upper surface. Demidov and Yatsenko [8] presented a numerical study showing that vapor zones can take place within the wick in the capillary evaporator under the fins. Figus et al. [9] have also presented a numerical solution for heat and mass transfer in the cylindrical evaporator wick by using the Darcy model and a two-dimensional pore network model. Yan and Ochterbeck [10] provided a numerical study on the flow and heat transfer in the wick of evaporator based on two-phase mixture model. In these works mentioned above, the computational domain is a single segment of wick structure in the evaporator. However, as shown in Fig. 2, the evaporator consists of wick, metallic wall, vapor and liquid grooves. The computational domain mentioned above is only a very small part of wick, and it leads to some shortages in evaluating the overall evaporator performance by these models. Firstly, these models can not predict the influences of metallic wall, vapor grooves and liquid grooves on heat and mass transfer of the evaporator, the heated surface temperature which is very important in estimating the performance of thermal management system can not obtained, too. Secondly, the heat conduction of metallic wall is not considered, especially for a miniature flat plate evaporator. Due to the high thermal conductivity of metallic wall, the liquid inside the bottom of wick and the
Liquid grooves may possibly heat up to a temperature that is greater than the saturation temperature corresponding to local saturation pressure by heat conduction of metallic side wall, vapor potentially may generate there as well, and vapor presence in there has been found to fully or partially block liquid flow to the vapor–liquid interface [11], which may lead eventually to dry-out of an evaporator and failure of CPL system. Therefore, the entire model is indispensable for complete understanding of the behavior of the capillary evaporator.

2. Mathematical model

As the evaporator is taken to be symmetric, a half of evaporator cross section is selected as the computational domain, as shown in Fig. 3.

To develop the mathematical model, the main assumptions are made as follows:

1. For simplicity, the vapor flow in the vapor grooves is neglected, and heat transfer is mainly by conduction.
2. The porous media is rigid, homogenous, isotropic and fully saturated with fluid. There is local thermodynamic equilibrium between solid phase and fluid phase, and the fluid is incompressible and has constant property.
3. The liquid in the liquid grooves comes from the condenser directly, and the temperatures in these regions are equal to liquid inlet temperature.
For the space environment, the effect of gravity is neglected. Figus et al. [9] have developed the continuum model and discrete model of porous media to investigate the heat and mass transfer in the porous media and the assumption of zero thickness for the two-phase zone were given, and their theoretical results are credible. In the present work, the assumption of zero thickness for the two-phase zone is used to simplify model of the wick. The heat and mass transfer for the liquid and vapor regions in the wick structure are based on the volume-averaged technique and Brinkman–Darcy–Forchheimer model of porous media.

Governing equations:

1. Metallic wall:

\[
\left(\rho c_p\right)_m \frac{\partial T}{\partial t} = k_{mw} \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right)
\]

(1)

2. Vapor grooves:

\[
\left(\rho c_p\right)_v \frac{\partial T}{\partial t} = k_v \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right)
\]

(2)

3. Wick structure

Continuum equation:

\[
\frac{\partial (\rho i \dot{V}_i)}{\partial t} + \nabla \cdot (\rho \dot{V}_i) = 0
\]

Momentum equation:

\[
\frac{\rho_i}{\epsilon_i} \frac{\partial \dot{V}_i}{\partial t} + \frac{\rho_i}{\epsilon_i^2} \nabla (\rho \dot{V}_i) \dot{V}_i = -\nabla p - \left( \mu_i \frac{\nabla \dot{V}_i}{K} + \frac{C_i |\dot{V}_i|}{\sqrt{K}} \right) \dot{V}_i
\]

\[+ \frac{\mu_i}{\epsilon_i} \nabla^2 \dot{V}_i\]

Energy equation:

\[
\left(\rho c_p\right)_i \frac{\partial T}{\partial t} + \rho_i c_i (\dot{V}_i \cdot \nabla) T = (k_{effi}) \nabla^2 T
\]

(3)

4. Liquid grooves

\[
T = T_m
\]

(4)

where \(i = l, v; C\) is the Forchheimer's constant; \(n\) is the normal unit vector at the vapor–liquid interface; \((\epsilon_t)\) is the effective thermal conductivity, \((\frac{\partial \rho}{\partial x})\) is the density–capacity heat product defined in the energy equations, \((\frac{\partial \rho}{\partial x}) = \epsilon \rho \dot{V}_i + (1 - \epsilon) \rho_s C_s\).

The mathematical model includes several parts, which consist of metallic wall, vapor grooves, wick structure and liquid grooves. The geometrical configuration between parts is shown in Fig. 2, and flow and heat transfer between two parts are coupling. The conjugate boundary conditions exist at the interface of two regions. The heat flux and temperatures should be continuous at the interface on both sides.

Boundary conditions:

At \(x = 0:\)

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

(7)

At \(x = L_x:\)

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

(8)

At \(y = 0:\)

\[
k_{mw} \frac{\partial T}{\partial y} = 0
\]

(9)

At \(y = L_y:\)

\[
k_{mw} \frac{\partial T}{\partial y} = q
\]

(10)

Due to fluid flow occurring inside the wick, the boundary conditions of momentum equations are as follows:

At \(y = L_{y1}\):

\[
u_1 = 0, \eta_1 = 0 \quad \text{wick-bottom wall border}
\]

(11)

At \(y = L_{y2}\):

\[
u_1 = 0, \eta_1 = \nu_m \quad \text{wick-liquid grooves border}
\]

(12)

The inlet liquid velocity \(\nu_m\) in Eq. (12) at the wick–liquid grooves border can be obtained by performing an overall energy balance on the capillary evaporator.

\[
n_m = \frac{\Delta \rho c_v}{\Delta T} \left( h_f + c(T_{sat} - T_{in}) + c(T_{sat} - T_{sub}) + c(T_{sat} - T_{sup}) \right)
\]

(17)

\[\text{Shallow discontinuities of the fluid properties appear across vapor–liquid interface, but continuities of the mass and heat flux should be maintained there.}
\]

\[\text{Mass continuity condition:}
\]

\[
\rho_i \dot{V}_i = \rho_v \dot{V}_v
\]

(18)

\[\text{Energy conservation condition:}
\]

\[
(k_{effi}) \nabla T_i \cdot n - (k_{effi}) \nabla T_i \cdot n = \rho_i \dot{V}_i |h_f|
\]

(19)

\[\text{Temperature continuity condition:}
\]

\[
T_1 = T_v = T_{sat}
\]

(20)

As evaporation takes place inside the wick structure, the capillary menisci is established at the vapor–liquid interface and capillary pressure is developed, which circulates the fluid throughout the CPL.

\[
\Delta P_c = P_v - P_i = \frac{2\pi \cos \theta}{\tau_e}
\]

(21)

Eq. (20) presents the local thermal equilibrium at the vapor–liquid interface inside the wick structure, and the saturation temperature \(T_{sat}\) is equal to the setpoint temperature of two-phase reservoir of CPL. This condition is used to determine the location of vapor–liquid interface. Because the maintenance of a continuous flow of fluid is assured by the capillary pressure of wick, \(\Delta P_c\) in Eq. (21) should be equal to the total pressure loss in the CPL loop.

3. Numerical procedure

The finite difference method is used in the numerical procedure and the governing equations together with the boundary conditions in the different regions of the evaporator are solved as a conjugate problem with the SIMPLE algorithm [12]. The mathematic model can predict the transient characteristics of evaporator,
Numerical results are determined with methanol as the working fluid. The thermal properties taken in the computation are: \( k_l = 0.202 \text{ W m}^{-1} \text{ K}^{-1} \), \( k_v = 0.0139 \text{ W m}^{-1} \text{ K}^{-1} \), \( \mu_l = 5.2 \times 10^{-4} \text{ N s}^{-1} \text{ m}^{-2} \), \( \mu_v = 1.07 \times 10^{-5} \text{ N s}^{-1} \text{ m}^{-2} \), and \( h_l = 1.12 \times 10^5 \text{ J kg}^{-1} \). The sintered stainless steel wick is used as the wick structure, the properties for the wick structure are: \( k_w = 15.2 \text{ W m}^{-1} \text{ K}^{-1} \), \( \varepsilon = 0.611 \), and \( K = 6.616 \times 10^{-13} \text{ m}^2 \). Here copper and stainless steel are chosen as the material of metallic wall, since they are compatible with methanol. The geometric parameters of the evaporator are: \( L_x = 21.5 \times 10^{-3} \text{ m} \), \( L_y = 3 \times 10^{-3} \text{ m} \), \( L_v = 10 \times 10^{-3} \text{ m} \), \( L_g = 3 \times 10^{-3} \text{ m} \), and \( L_{v2} = 7 \times 10^{-3} \text{ m} \). Both length and height of vapor and liquid grooves are \( 0.5 \times 10^{-3} \text{ m} \). The two-phase reservoir of CPL system is used to control the temperature of the loop and accommodates fluid inventory shifts during changes in operating conditions, and the setpoint temperature of reservoir is the liquid saturated temperature \( T_{sat} \), which is the temperature of vapor–liquid interface. The setpoint temperature is \( 35 \text{ °C} \), which is the liquid saturated temperature \( T_{sat} \). The grid schemes of global evaporator are \( 237 \times 82 \).

**Fig. 4.** Liquid velocity vectors in the wick structure (copper wall, \( q = 3 \times 10^4 \text{ W m}^{-2}, \Delta T_{sub} = 3 \text{ °C} \)).
Fig. 5. Temperature fields in the evaporator (copper wall, $\Delta T_{\text{sub}} = 3^\circ$ C). (a) $q = 3 \times 10^4 \text{ Wm}^{-2}$, (b) $q = 5 \times 10^4 \text{ Wm}^{-2}$ and (c) $q = 8 \times 10^4 \text{ Wm}^{-2}$.

Fig. 6. Temperature fields in the evaporator (copper wall, $q = 8 \times 10^4 \text{ Wm}^{-2}$). (a) $\Delta T_{\text{sub}} = 6^\circ$ C, and (b) $\Delta T_{\text{sub}} = 9^\circ$ C.
temperature difference aids in preventing vapor formation in the liquid grooves and inside the bottom wick, and it helps to increase the safety of CPL. Furthermore, as seen from Figs. 5(c) and 6, at the low liquid subcooling of 3 °C, the left vapor–liquid interface has invaded deeply into the liquid groove, and it indicates that evaporator has nearly reached its side wall effect heat transfer limit. But at the high liquid subcooling of 6 °C and 9 °C, the vapor–liquid interface locates near the wick-side wall border, not goes deep into liquid groove, and the CPL can operate safely. As a result, increasing the liquid subcooling can enhance the side wall effect heat transfer limit of the evaporator.

In order to investigate the effects of wall material on the performance of evaporator, stainless steel wall and combined wall (upper wall material is copper, side and bottom wall material is stainless steel) are also used in the computation. Fig. 7 shows the temperature distributions in the evaporator with stainless steel wall and combined wall at $\Delta T_{sub} = 3^\circ C$ and $q = 8 \times 10^4$ W m$^{-2}$. Because of low thermal conductivity for stainless steel wall, the temperature gradient is very large in the upper wall zone, and decreases in the wick and bottom wall zone. It is very clear to find that most of applied heat load has been used for the liquid evaporation near the upper and left surface of wick. For the cases of copper wall and combined wall, because of high thermal conductivity, the temperature gradient is small in the upper wall zone. From Fig. 5(c), it can see that the vapor–liquid interface has gone deeply into the liquid groove, naturally, the evaporator has almost reached its side wall effect heat transfer limit. But as seen from Fig. 9, the vapor–liquid interfaces have not invaded deeply into the grooves at the same heat flux, which indicated that the evaporator with stainless steel or combined wall has higher side wall effect heat transfer limit.

The heated surface of the evaporator is attached to the apparatus which needs cooling, and high temperatures of heated surface may result in abnormality of the apparatus. Consequently, the temperature level of heated surface is a very important parameter to estimate the performance of CPL evaporator.

Fig. 8 shows the heated surface temperatures along $x$ direction for different wall materials at $q = 8 \times 10^4$ W m$^{-2}$. As seen, the heated surface temperature of evaporator with stainless steel wall is more than 65 °C, and it is dangerous to the normal operation of cooled apparatus. But the heated surface temperatures with copper wall and combined wall are less than 45 °C, it is moderate for the cooled apparatus, and the cooled apparatus can operate normally under...
these conditions. Thereby, the thermal conductivity of wall has a significant influence on the heated temperature, and the upper metallic wall with high thermal conductivity results in low heated surface temperature.

To decrease the temperatures of the heated surface and increase the side wall effect heat transfer limit of miniature flat plate CPL evaporator, the combined wall should be chosen for dissipating high heat fluxes. Fig. 9 shows the temperature distribution in the evaporator at \( q = 10 \times 10^4 \text{ W m}^{-2} \), as seen, the vapor–liquid interface has not gone deeply into the liquid grooves, and thus the evaporator has not reached the side wall effect heat transfer limit. At the same time, the maximal temperature of heated surface is about 47.5 °C, which is moderate for cooling of electronic devices. Therefore, the miniature flat plate evaporator with combined wall can improve the heat transport capacity and maintain an appropriate temperature level of the heated surface.

5. Conclusions

A two-dimensional model for the global evaporator of miniature flat CPL evaporator is developed, and the governing equations for different zones are solved numerically as a conjugate problem using the SIMPLE algorithm.

The liquid evaporation occurs in the vicinity of upper and left surface of wick structure, and the location and shape of vapor–liquid interface are mainly affected by the heat fluxes and geometric parameters of fins and vapor grooves. With the increases of the applied heat fluxes, the vapor–liquid interface inside the wick moves away from the fins and the size of the vapor zone enlarges. The side wall effect heat transfer limit is introduced to estimate the heat transport capability of the miniature flat plate evaporator, and it is a very important heat transfer limit compared to others. Increasing the inlet liquid subcooling can increase the temperature difference between the upper and bottom surface of wick, and thus helps to prevent the vapor formation in the liquid grooves and the bottom wick, as well as increase heat transport capacity of the evaporator. The evaporator with single aluminum wall results in low side wall effect heat transfer limit, but leads to low temperature level of the heated surface. On the other hand, the evaporator with single stainless steel wall leads to higher heat transport capacity, but to higher temperature level of the heated surface. The evaporator with combined wall can greatly increase side wall effect heat transfer limit and maintain an appropriate temperature level on the heated surface, which imply that the CPL can operate safely and cooled apparatus can also work effectively under higher heat fluxed.

The present overall numerical model of miniature flat plate CPL evaporator can effectively estimate the heat transfer capacity of evaporator and evaluate the temperature level of the heated surface, and it is useful for the evaporator design and performance optimization of miniature flat plate CPL.

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