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Topology optimization of the wick geometry in a flat plate heat pipe

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ABSTRACT

In the present study, a topology optimization approach is proposed to determine an optimal geometry of a wick sintered inside a flat plate heat pipe. Total thickness of the heat pipe is assumed as fixed, and thus the task involves redistributing the wick material and selecting the internal shape of the vapor core to provide a minimum power dissipation in the liquid flow and minimum total pressure drop in the entire heat pipe. Constraints are used for the maximum volume of the wick and maximum allowable temperature of the heat pipe wall. With respect to the preliminary assessments, the simulations are realized based on simplified 2D steady state thermal and hydrodynamic models assuming constant temperature and laminar flow in the vapor core and Darcy's law for the liquid flow through the porous wick. Optimization results are presented for rectangular flat heat pipes with different lengths and thicknesses. Optimal placement of the wick columns and grooves are obtained by using the proposed topology optimization scheme. The results indicate that the use of the wicks of optimal shape increases the operating performance of flat heat pipes and especially increases their heat transfer capability up to twice that of heat pipes with flat wicks of constant thickness.

1. Introduction

Flat heat pipes (FHP) and vapor chambers are widely used in heat transport and heat spreading applications [1–3]. They are used in the cooling systems of high power electronic devices under limited space conditions due to high thermal conductivity and small thickness. They can be used at different tilts angles including anti-gravitation/acceleration regimes due to the high capillary limits of FHP with porous wicks. Superhydrophilic, bi-porous, and different composite wicks are recently proposed to improve the operating performance of FHP [4–8]. The results indicated that optimization of the wick's material composition and microstructure significantly increase the capillary pressure, reduce thermal resistance of FHP, and prevent dryout at high heat fluxes.

A potential method to improve the FHP performance without changes in the wick material involves using the wick columns sintered inside the heat pipe in addition to the thin wick layer that lines the heat pipe walls. The aforementioned type of FHP were recently designed and examined via experimental [9–11] and theoretical [12–15] approaches. Wicks with uniform radial grooves were used in [10,9]; biomorphic leaf-type structures were proposed and examined in [13,11,16]; and multi-artery heat-pipes were developed in [17,8]. The results indicated that the use of the wick columns of the aforementioned special geometry decreased the hydraulic resistance of the wick and provided fast liquid backflow to the evaporator area.

In the present study, we propose an FEM-based topology optimization approach that aids in determining an optimum geometry of the wick to provide high thermal/hydraulic performance of an FHP. Given the simplified variant of the steady state model proposed in [18], we solve an optimization problem using power dissipation in the liquid flow and total pressure drop as the objective functions. The maximum allowable volume of the wick and the maximum temperature of the FHP are used as the constraints. The model under study bases on the hypotheses of two-dimensional laminar isothermal flow of the vapor and Darcy's law for the liquid in the wick. The model appears as sufficiently simple and useful to realize fast numerical simulations during the preliminary design of the FHP. Simultaneously, the model is sufficiently rigorous to provide clarified predictions [19,20]. More general models consider the transient and three-dimensional effects, convection heat transfer, compressibility of the vapor flow, and Brinkman–Forchheimer effects in the wick [21,20,22,23] can be also implemented in a similar manner in the topology optimization scheme. However, this involves more computational time and is a task for a future study.

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We note the peculiarity of the considered optimization problem. Parameters of the fluid flow in the wick and in vapor core significantly depend on its geometry. Increases in the wick thickness leads to decreases in the vapor core volume, and thereby increase the pressure drop in the vapor phase. Conversely, decreases in the wick volume increase the pressure drop in the liquid phase inside the wick. Both effects reduce the capillary limit of the FHP. Additionally, the heat transfer parameters of the thermal conduction problem that are solved in the solid wall are also evaluated through the wick thickness in the considered model. Thus, the goal of the present study involves determining optimum pathways for oppositely directed liquid and vapor flows by considering limitations and requirements for the FHP temperature. To the best of the authors' knowledge, the aforementioned topology optimization problems for the FHP are not considered to date [24–27].

2. Models and methods

2.1. Geometry

We consider that the FHP consists of solid walls, porous wick, and vapor core (Fig. 1). The FHP has rectangular shape with dimensions $a \times b$ in XY plane, and its total thickness in z direction is H. The domain occupied by the FHP in XY plane is denoted as Ω, and its boundary is $\partial\Omega$. Solid walls have constant thickness H_s. Thicknesses of the wick and the vapor core depend on the coordinates $\mathbf{r} = \{x, y\}$ and denoted as $H_l(\mathbf{r})$ and $H_v(\mathbf{r})$, respectively. The following relations are obtained for the geometric parameters:

$$\begin{aligned} &H_v = H - 2H_s - H_l, \\ &H_0 \leqslant H_l \leqslant H, \end{aligned} \tag{1}$$

where H_0 denotes the minimum thickness of the wick layer that fully covers the upper internal surface of FHP (Fig. 1).

2.2. Assumptions and simplifications

The operating processes in the considered heat pipe are as follows. Heat is applied at the evaporator region and removed at the condenser. The fluid is vaporized around the evaporator region due to input heat. The resultant vapor pressure drives the vapor from the evaporator to the condenser where the vapor condenses to liquid. The capillary pressure in the wick provides liquid backflow to the evaporator.

The following assumptions are adopted in the considered model [18]:

- Processes correspond to steady state.
- Evaporator and condenser regions with prescribed uniform heat fluxes are placed on the upper wall of heat pipe. All other external surfaces are adiabatic.
- Uniform temperature (equal to the saturation temperature T_{sat}) and laminar flow are assumed for the vapor core. Temperature of the lower wall of the FHP is also equals to T_{sat}.
- Wick is isotropic and completely saturated with the liquid.
- Darcy model is used for the liquid flow in the wick.
- Liquid and vapor flow are assumed as incompressible.
- Vapor condenses and liquid evaporates only at the interface between the vapor and liquid-saturated wick.
- All material properties are constant and calculated at the saturation temperature.
- Gravitational forces are neglected

For the preliminary design, the simplified 2D thermal and hydrodynamic models are used. It should be noted that in contrast to the initial approach [18], we assume that the temperature of the solid wall does not vary across the z direction due to the small thickness and high conductivity. The heat transfer boundary condition on the internal surface of the wall that was used in [18] is included in the 2D heat equation as the distributed linear heat source (see (2) below).

2.3. Thermal model for the wall

Energy balance for the upper solid wall of the FHP defines the following heat conduction equation:

$$\begin{aligned} &k_s H_s \nabla^2 T = \phi + h(T - T_{sat}) \quad in \quad \Omega \\ &\nabla T \cdot \mathbf{n} = 0 \quad on \quad \partial\Omega \end{aligned} \tag{2}$$

where T denotes the temperature of the wall; ∇ denotes the 2D nabla operator; k_s denotes the thermal conductivity of the wall; $\mathbf{n}$ denotes the unit normal vector to the domain's boundary; $\phi = \phi_0$ at the evaporator region and $\phi = -\eta\phi_0$ at condenser region, ϕ_0 denotes the uniform heat flux, and η denotes the ratio between the evaporator and condenser area.

Heat transfer coefficient h between the FHP wall and fluid in (2) is obtained based on the thermal conductivity of the wick k_l, its thickness H_l, and local inclination angle of the wick surface θ as follows:

$$h(x, y) = \frac{k_l}{H_l \cos\theta}, \quad \cos\theta = 1/\sqrt{1 + |\nabla H_l|^2} \tag{3}$$

It should be noted that lateral heat flux from the wick columns to the vapor space is considered in (3). Specifically, if the surface of the wick is flat and parallel to the walls, then $|\nabla H_l| = 0$ and $h(\mathbf{r}) = h_{flat} = k_l/H_l$ [18], thereby indicating that the wick/vapor interface area corresponds to the area of its corresponding projection on the solid/wick interface. However, if there is a wick with non-uniform thickness, and the surface exhibits an inclination angle, then the wick/vapor interface area exceeds the corresponding solid/wick interface area, and we consider it by dividing the

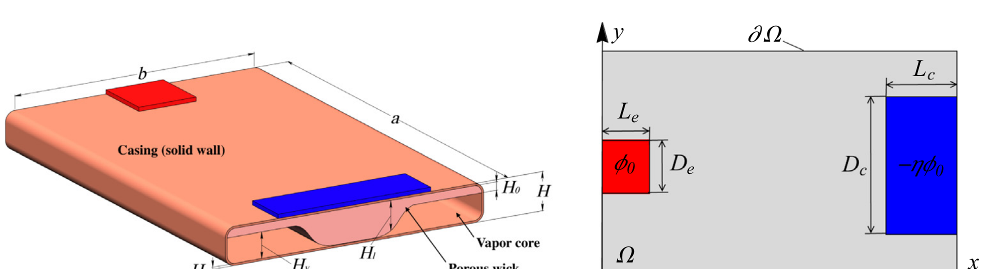

Fig. 1. Geometrical parameters and 2D model of the flat heat pipe. Evaporator area – red, condenser area – blue, adiabatic surface – gray (in 2D model). (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)