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Analysis of non-uniform heat loads on evaporators with loop heat pipes

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ABSTRACT

Loop heat pipes (LHPs) are passive two-phase heat transfer devices which are widely used in cooling applications such as spacecraft thermal control and electronics cooling. As these devices often encounter non-uniform heat load during operation, a numerical analysis based on a 2-dimensional dynamic mesh model is conducted to gauge the influence of non-uniform load on the performance of a flat-plate evaporator used in LHPs. The variations of evaporation heat transfer coefficient, outflow working fluid temperature, vapor and liquid interface position as well as surface temperature with heat load concentration are analyzed. The results reveal that when the heat load is not highly concentrated and relatively low, non-uniformity facilitates the heat transfer. On the other hand, highly non-uniform heat load will deteriorate heat transfer. Furthermore, the maximum surface temperature will increase significantly under non-uniform heat flux, although the phenomenon can be mitigated by using shell materials with higher heat conductivity.

1. Introduction

Capillary pumped loops (CPLs) and loop heat pipes (LHPs) are similar two-phase heat transfer devices that operate on closed evaporation and condensation cycles, relying on phase transition and capillary pressure to maintain the operation. The first CPL was developed by Stenger in the 1960s [1], while LHPs were invented in Russia in the 1970s [2]. Compared with traditional heat transfer devices such as conduction or air convection cooling systems, CPLs and LHPs are superior in that much higher heat transfer coefficient are obtainable while providing lower surface temperature and temperature differences. Secondly, longer distance and high amount heat transfer are achievable. Therefore, they are wildly used in thermal management systems such as those employed in spacecraft [3] and laptop computers [4].

An evaporator which connects with heating surface directly and provides capillary pressure for the cycle is an important component of LHPs. Some experimental results had been reported in evaporator optimal design [5–8], while Udell [9] showed that within the porous medium there are three different zone, vapor, liquid and two-phase, and the size of two-phase zone could be a function of working fluid and porous media characteristics. Zhao and Liao et al. [10] investigated experimentally the evaporation mechanism within the porous wick and found that under minor heat flux, the porous medium region is saturated, except a few regions near the heating surface. With increasing heat flux, an increasing two-phase zone appears in the top region of the wick. As long as the heat flux exceeds a threshold value, a vapor zone will also appear.

Unfortunately, numerical simulations of the phase transition and heat transfer in evaporator are rather complicated. In the actual situation, LHPs will display an oscillating behavior [11–13] rather than a steady-state one, so that it is not easy to present a conjugate model. Cao and Faghri [14] developed an analytical solution based on a simple model of heat and mass transfer with a liquid-saturated porous wick. Later the same authors [15] used an improved model that coupled vapor groove and porous wick, but assumed that evaporation only occurs in the interface between groove and capillary wick. Demidov and Yatsenko [16] simulated the evaporation process in a capillary structure, and found that at low load the evaporation occurs at the capillary wick and heating wall interface. At higher heat loads, the liquid–vapor interface moves into the capillary structure. Figus et al. [17] also found that the size of the vapor zone is not a simple function of the heat load. Thus, we can expect significant and interesting evaporative heat transfer coefficient variation with increasing heat load.

Recent investigations concentrated in three aspects. The first one is studying new type porous wicks [18–22], Xu et al. [18]

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Nomenclature

Latin symbols

A_t	the area of heating surface, m^2
c	specific heat, $J\ K^{-1}\ kg^{-1}$
C_2	inertial resistance factor, m^{-1}
C_{evap}	thermal conductivity from heating surface to the interface of wick and shell, $W\ m^{-1}\ K^{-1}$
D_p	the porous radius of capillary wick, m
e	natural constants
H_{evap}	evaporative heat transfer coefficient, $W\ m^2\ K^{-1}$
h_f	convection heat transfer coefficient, $W\ m^2\ K^{-1}$
$\underline{V}$	surface normal vector
p	pressure, Pa
p_c	capillary pressure, Pa
$q(x)$	head load function
q_{load}	average heat flux on heating surface, $W\ m^{-2}$
Q_{load}	heat load, W
S	additional momentum source, $kg\ m^{-1}$
t	time, s
T	temperature, K
$\vec{V}$	the vector of velocity, $m\ s^{-1}$
x, y	Cartesian space coordinates

Greek symbols

α	permeability, m^2
γ	heat concentration degree
ε	wick porosity
θ	contact angle, rad
ρ	density, $kg\ m^{-3}$
λ	thermal conductivity, $W\ m^{-1}\ K^{-1}$
μ	dynamic viscosity, $N\ m^{-2}\ s$
σ	mean square deviation
π	Pi
τ	surface tension of working fluid, $N\ m^{-1}$

Subscripts

eff	effective value
evap	evaporation
in	inlet
i	interface between liquid and gas
l	liquid phase
max	maximum value
min	minimum value
outf	outflow
out	outlet
sat	saturated
sh	shell
v	gas phase
wick	porous wick

and Xin et al. [21] studied sintered wicks. Wu et al. [19] and Deng et al. [22] investigated composite wicks. And the work by Lin et al. [20] mainly focused on bidisperse wick. The second is the performance of variety of new structure design evaporator [23–27]. The last is LHP's performance under low temperature [28–30]. These researches are conducted mainly by Bai et al. [29] and Zhao et al. [30].

However, almost all of the experiments and simulations surveyed were conducted with uniform heat fluxes being imposed upon the heating surfaces. While in applications, especially for small-scale flat-plate evaporators, LHPs need to confront with non-uniform heat flux. For example, it is common to use a single evaporator for the cooling of a few close electronic components; and even for a single component, the heat flux will be non-uniform due to the indispensable encapsulation shell which could be much larger than its inner component. Therefore, we report here an investigation on the performance of an evaporator under non-uniform heat flux with numerical simulation method of a dynamic mesh model by using the software FLUENT 14.0, within the commercial software ANSYS 14.0. The model coupled liquid–gas interface moving, phase change, heat and mass transfer rather than merely based on given interface position.

2. The evaporator

2.1. Geometric model

Based on the models of Muraoka et al. [30] and Wan et al. [31,32], a flat-plate type loop heat pipe evaporator is adopted for the numerical simulation, in this paper. In order to limit side metallic wall evaporation, which is unfavorable to the steady operation of the evaporator [31,32], the vapor groove position was modified. As shown in Fig. 1, the size of the evaporator is 43 mm × 10 mm; the wick inside is 37 mm × 4 mm; as for vapor and liquid grooves, it is 1 mm × 1 mm. There are 19 vapor grooves and 18 liquid grooves at the top and the bottom of the porous wick, respectively. In the simulation, the x-axis is aligned in the length direction which ranges from −21.5 mm to 21.5 mm; while the y-axis is in along the width ranging from −5 mm to 5 mm. Regions 1, 2, 3, 4 are evaporator shell, vapor groove, porous wick and liquid groove, respectively. The heating surface is aligned at the top of evaporator and ranges from −18.5 mm to 18.5 mm: the same as the length of porous wick. We assume the surrounding walls as adiabatic.

In the 2-D simulation of inflow and outflow, the inlet and outlet are set as squares, with size 0.4 mm by 0.4 mm, in the center of every groove. The material of both the shell and wick is steel. The pore diameter of the steel wire for the evaporator porous wick is 25 μm. Methyl-alcohol is adopted as the working fluid. The geometric parameters are summarized in Table 1.

2.2. Mathematical model

The mathematical model is based on the following main assumptions: (1) zero gravity effect in space applications, and negligible gravity effect when compared with capillary pressure; (2) the capillary wick is isotropic; (3) the fluid, both vapor and liquid, in wick, grooves and evaporator shell are in thermal equilibrium; (4) the variance of saturation temperature is negligible [33] because the pressure drop in the evaporator is much smaller than the absolute pressure; (5) radiation and convection heat transfers are neglected when compared with conductive and latent heat transfers; (6) fluid flow within the evaporator is laminar, as flow velocity and viscosity influence are small; (7) both liquid and gas phases are incompressible, since incompressibility assumption has borne up well with reported work in this area [14–17]. Based on the above assumptions, the mathematical model under steady state condition in each part of the evaporator are as follows.

2.2.1. Evaporator shell

Energy equation:

$$\lambda_{sh}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)=0. \tag{1}$$