International Journal of Heat and Mass Transfer 81 (2015) 737–749

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Analysis of overloaded micro heat pipes: Effects of solid thermal conductivity

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ARTICLE INFO

Article history:
Received 14 July 2014
Received in revised form 12 September 2014
Accepted 24 October 2014
Available online 20 November 2014

Keywords:
Overloaded micro heat pipe

ABSTRACT

Starting from the principles of mass, momentum, and energy conservations as well as the Young–Laplace capillary equation, a mathematical model of triangular MHPs has been developed, primarily to investigate the effects of the thermal conductivity of the solid wall on their performance under overloaded conditions. For this purpose, two solids of significantly different thermal conductivities, copper and nickel, have been selected for the wall material. Using the model, a map encompassing the various possible operation zones of an MHP was constructed, to provide insight into the modes of operation under a given operating temperature but with varying heat loads and charge levels of the working fluid. The model predicts that in the overloaded zones, the dryout and flooded lengths increase with the applied heat load, resulting in a decrease in the effective length of the MHP. Moreover, it is also observed that the existence of dryout is accompanied by a large temperature rise over the dry region and, thus, a large total axial temperature drop, which increases rapidly with the applied heat load. Finally, comparison between copper and nickel MHPs on how the key performance indicators, such as the dryout and flooded lengths, the effective length, and the total axial temperature drop, response to the applied heat load shows that an MHP of solid wall of higher thermal conductivity out-performs one with lower solid thermal conductivity.

1. Introduction

Owing to their ability to deliver high heat fluxes, micro heat pipes (MHPs) are efficient heat transfer devices, especially in applications where space constraint is a major consideration. However, for an MHP to operate effectively, it must not be overloaded. Otherwise, adverse operating conditions such as dryout, flooding, or the combination of both may set in [1]. In the existing literature, most analytical studies do not include overloaded MHPs. While the relatively small number of studies which deal with overloaded MHPs have provided useful insights, many questions remain unanswered. It is, therefore, the purpose of this paper to address some of those issues. Specifically, we shall investigate the effects of the thermal conductivity of the solid wall of overloaded MHPs.

One of the most important problems associated with an overloaded MHP is the possible existence of a dryout region at its evaporator section, resulting in a high solid wall temperature there. Peterson and co-workers [2,3] and Suman and co-workers [4,5] have investigated overloaded MHPs with a focus on dryout. However, the occurrence of dryout at the evaporator is not the only possible manifestation of overload, because flooding may also exist in an overloaded MHP. In fact, flooding at the condenser section may take place in the absence of or simultaneously with the occurrence of dryout at the evaporator. Kim and co-workers [6,7] and Launay et al. [8], respectively, have conducted analytical studies pertaining to these two cases. In this paper, we shall cover all the three cases and elaborate on the conditions for the prevalence of each of them.

Since MHP was first conceptualized in the 1980s, it has been a common practice by MHP investigators, in earlier studies [9,10] as well as the more recent ones [11,12], to exclude from their analyses axial heat conduction in the solid wall. This practice is justified on the ground that heat transport by axial solid conduction is negligible compared to that by phase change of the working fluid. The absence of axial solid conduction then implies that the temperature of the solid wall does not vary in the axial direction; in other words, an MHP may be regarded as an isothermal device. However, a few recent analytical studies [7,13–15] show that the temperature of the solid wall varies continuously along the axis of an MHP. Moreover, while the heat transport by phase change usually dominates that by axial solid conduction, the exclusion of

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.10.060
0017-9310/

Nomenclature

A	cross-sectional area; cross-sectional area of micro heat pipe, m^2
C	constant; geometrical parameter, Eqs. (8)–(12), (45), (46)
Ca	capillary number, defined in Eq. (28)
D_H	hydraulic diameter, Eqs. (47), (48), m
F	function defined in Eq. (30)
f	friction factor
G	function defined in Eq. (31)
h	heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
h_{fg}	latent heat of evaporation, $J\,kg^{-1}$
K	constant, Eqs. (25), (26)
k	thermal conductivity, $W\,m^{-1}\,K^{-1}$
L	length; length of micro heat pipe, m
L^*	effective length; effective length of micro heat pipe, Eq. (33), m
L_0	axial distance from evaporator end where solid and liquid temperatures are equal, m
M	mass of working fluid, kg
$\widehat{M}$	charge level
$\widehat{M}_{opt}$	optimal charge level
$\dot{m}$	mass flow rate, $kg\,s^{-1}$
$\widehat{m}$	dimensionless mass flow rate, defined in Eq. (21)
N	number of corners/vertices of a polygon
Nu	Nusselt number, defined in Eq. (2)
P	length of interface, m
p	pressure, $N\,m^{-2}$
$\dot{Q}$	rate of heat transport; applied heat load, W
$\dot{Q}_c$	rate of heat transport by axial conduction in solid wall, W
$\dot{Q}_{cap}$	heat transport capacity, W
$\dot{Q}_p$	rate of heat transport by phase change of working fluid, W
$\dot{q}$	rate of heat transfer per unit axial length, $W\,m^{-1}$
r	meniscus radius of curvature, m
Re	Reynolds number, defined in Eq. (19)
s	volume fraction occupied by liquid phase
T	temperature, °C
$\bar{T}$	average temperature, °C
T_0	solid temperature at evaporator end, °C
T_1	solid temperature at condenser end, °C
T_{op}	operating temperature, °C
ΔT	total axial temperature drop, °C
u	velocity, $m\,s^{-1}$
w	side width of cross section of flow channel, m
We	Weber number, defined in Eq. (29)
x	axial distance from evaporator end, m
$\widehat{x}$	dimensionless x, in units of L

Greek symbols

γ	vapor-to-liquid kinematic viscosity ratio
ε	vapor-to-liquid density ratio
η	coefficient defined in Eq. (23), K^{-1}
Θ	quantity defined in Eq. (5), K
θ	contact angle, rad
λ	coefficient defined in Eq. (4)
μ	dynamic viscosity, $kg\,s^{-1}\,m^{-1}$
ξ_c,ξ_p	fractions of applied heat load, defined in Eq. (36)
$\widetilde{\omega}$	angular parameter, defined in Eq. (41)
ρ	density, $kg\,m^{-3}$
σ	coefficient of surface tension, $N\,m^{-1}$
τ	shear stress, $N\,m^{-2}$
ϕ	half corner angle at vertices of polygon, rad

Subscripts

a	of adiabatic section
c	of condenser section
cl	of capillary limit
d	of dried out segment of solid wall
e	of evaporator section
f	of flooded segment of solid wall
fl	of onset of flooding
l	of liquid phase of working fluid
lv	of liquid–vapor interface
s	of solid wall
sl	of solid–liquid interface
sv	of solid–vapor interface
v	of vapor phase of working fluid

the latter may incur significant errors in the calculation of the heat transport capacity of the MHP [13]. To properly model an MHP, it is therefore necessary to include axial solid conduction and the associated axial temperature variation of the solid wall. This is especially true for an overloaded MHP, since large axial temperature drops may be involved. In fact, Lin et al. [16] have experimentally observed axial temperature drops as large as 50 °C for MHPs operating with, presumably, dryout. For the liquid and vapor phases of an MHP which is not overloaded, uniformity in temperature can be assumed, since their total axial temperature drops are much smaller than that of the solid wall [15]. This isothermal assumption may still be valid for overloaded MHPs, since the liquid and vapor phases which participate in phase change and circulation are both saturated.

Fig. 1 is a schematic illustration of the heat and fluid flows together with the axial volume distribution of the liquid phase inside an overloaded MHP. In this paper, the incoming heat flux at the evaporator and the outgoing heat flux at the condenser are both assumed to be uniform, but their respective values may be different from each other. However, the model developed based on this assumption can be very easily modified for cases of non-uniform fluxes. In Fig. 1, the evaporator length L_e, condenser length L_c, and the length of the adiabatic section, L_a, are all prescribed geometric parameters. On the other hand, the dryout length L_d and the length of the flooded region, L_f, are not pre-set constants; rather, they are obtained by solving the governing equations subject to the heat load and, therefore, are functions of the heat input to the MHP. The three cross sections depicted in Fig. 1 illustrate the changes in the respective areas occupied by the liquid and vapor phases as we proceed from the evaporator toward the condenser, the third being an illustration of the onset of flooding. This cross section corresponds to the furthest point of the vapor flow, since the flow channel beyond is completely flooded by stagnant liquid [6–9]. The scenario depicted in Fig. 1 is the most general case, in the sense that both dryout and flooding occur. However, as we shall see later, either dryout or flooding can take place without the other.

The MHP model developed in this paper is a steady-state, one-dimensional construction. All the pertinent variables, such as the temperature of the solid wall or the liquid and vapor velocities and pressures, are assumed to have been properly averaged over the cross-section and, therefore, are functions of the axial position only. As remarked previously, we shall assume a uniform temperature for both the liquid and vapor phases, with the understanding that the stagnant liquid occupying the flooded region, if any, may have a slightly lower temperature than the circulating fluid. This