International Journal of Heat and Mass Transfer 94 (2016) 366–372

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Operation limitation of an oscillating heat pipe

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ARTICLE INFO

Article history:
Received 24 April 2015
Received in revised form 8 November 2015
Accepted 15 November 2015

Keyword:
Oscillating heat pipe

ABSTRACT

The oscillating heat pipe (OHP) has been widely studied as a promising heat transfer device, which can efficiently transport heat by thermally exciting oscillating motions. In the current investigation, the operating limitation of an OHP is theoretically studied to determine its maximum heat transport capability. When the heat transfer rate added to the evaporator section reaches a critical point, the vapor generation rate increases resulting in a significant increase in momentum, which can penetrate the liquid plug causing the slug flow to become annular flow. When this phenomenon takes place, the spring–mass system consisting of a train of liquid plugs and vapor bubbles disappears, and the OHP reaches the maximum heat transport capability. Based on this mechanism, a new mathematical model to predict the OHP operating limitation is developed. The results indicate that the operation limitation of an OHP depends on its working fluid, latent heat, operating temperature, turn number, filling ratio, heat pipe dimensions, and liquid plug length.

1. Introduction

Oscillating heat pipes (OHPs) have been investigated extensively over the last two decades [1–10]. For an OHP, heat is transported from the evaporator to the condenser by thermally-excited oscillating motions, which is very different from conventional heat pipes [11]. Similar to the conventional heat pipe, a number of heat transport limitations exist in an OHP [12–18]. Many researchers [11–16] believe that the critical heat flux limits the maximum heat transport capability or the operating limitation in an OHP. Kew and Cornwell [17] investigated the dryout limit of fluid flow in microchannels and they found that existing correlations of flow boiling heat transfer could not predict the critical heat flux occurring in microchannels with characteristic dimension of an order of the nucleating bubble release diameter. In other words, the evaporation heat transfer in microchannels is significantly affected by the channel size and surface tension. Kew and Cornwell [17] used the Confinement number, Co, i.e.,

$$Co = \left[\frac{\sigma}{g(\rho_l - \rho_v)D_h^2}\right]^{1/2} \tag{1}$$

to consider the effects of the channel size and surface tension on the evaporation heat transfer in microchannels. For an OHP to operate, the surface tension and the meniscus radius of the liquid–vapor interface in microchannels must be utilized to form a train of liquid plugs and vapor bubbles, where the Bond number or Eötvös number, i.e.,

$$Bo = \frac{g(\rho_l - \rho_v)D_h^2}{\sigma} = Eo^2 \tag{2}$$

is used to determine the hydraulic diameter of the microchannels to be employed in an OHP. Substituting Eq. (1) into Eq. (2) yields

$$Co = Bo^{-0.5} = 1/Eo \tag{3}$$

It indirectly shows that the critical heat flux occurring in microchannels might be one of the operating limitations in an OHP.

Khandekar et al. [12] conducted experimental investigations with different working fluids (water, ethanol and R-123), and different orientations (vertical and horizontal). Results showed that the heat flux level and filling ratio significantly affect the thermal performance of an OHP and experimentally found that there exists a limitation of heat flux, wherefore partial dry-out of the evaporator was detected below a certain range of filling ratio. Meena et al. [13] conducted an investigation of the effects of evaporator section length (changing from 5 to 15 cm) on the operation limitation of closed loop oscillating heat pipes with check valves (CLOHP/CV). The results showed that the critical heat transfer flux decreased when the evaporator lengths increased. Hudakorn et al. [14]

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2015.11.039
0017-9310/

Nomenclature

D	diameter, m
F	force, N
h_{lv}	the latent heat, J/kg
L	the length of section of OHP, m
m	mass, kg
M	liquid number
N	turn number
p	pressure, N/m^2
q	heat load, W
r	radius, m
t	time, s
u	velocity, m/s
x	axial position, m

Greek

α	contact angle, rad
β	inclination angle, rad
ϕ	filling ratio
μ	viscosity, N/s-m^2
ρ	density, kg/m^3
σ	surface tension, N/m
τ	shear force, N

Subscripts

a	advancing
c	critical
e	evaporator
h	hydraulic
i	inner
l	liquid
r	receding
v	vapor

investigated the effect of inclination angle on the operation limit of a closed loop OHP. They found that the critical heat flux decreases as the evaporator length increases for all inclination angles from quantitative results of tests with a controlled vapor temperature of 60 ± 5 °C, and developed a correlation of

$$\frac{q_{c,\beta}}{q_{c,90}} = \left(1.164\sin\beta + 0.53\cos\beta - 0.484\left(\frac{L_e}{D_i}\right)\right)^{0.1} \quad (4)$$

to predict the critical heat flux with the effect of the inclination angles ranging from 10° to 90°. Yang et al. [15] conducted an experimental study on the operational limitations of two closed loop oscillating heat pipes by employing R123 as the working fluid with filling ratios of 30%, 50% and 70%, respectively. Their results showed that the dry-out heat fluxes in the vertical bottom heating mode were about 12,420 KW/m^2 (1 mm ID) and 4300 KW/m^2 (2 mm ID) for axial heat transport, and about 320 KW/m^2 (1 mm ID) and 240 KW/m^2 (2 mm ID) for radial heat input, respectively. However, Mikielewicz et al. [16] conducted an experimental investigation and theoretical study of dryout in an annular flow with R123 as the work fluid in small diameter channels with similar dimensions (1.15 mm ID, 2.3 mm ID), and showed that the critical heat flux varied from 23 kW/m^2 to 107 kW/m^2 with 1.15 mm ID, and 65 kW/m^2 to 150 kW/m^2 with 2.3 mm ID due to different mass flow rates. The big difference between these results indicates that another type of limitation might exist due to the disappearance of the mass–spring system in OHPs.

A train of liquid plugs and vapor bubbles is the basis for generating oscillating motion which should directly affect the operating limitation that exists in an OHP. In the current investigation, an operation limitation based on the mass–spring system is studied. In other words, when the heat transfer rate is high resulting in high flow velocity in the channel, the slug flow (consisting of vapor bubbles and liquid plugs) transitions to annular flow. As a result, the mass–spring system no longer exists. When an OHP reaches this critical point, the oscillating motions disappear. Based on this phenomenon, an operation limitation is predicted in the current investigation.

2. Theoretical modeling

The spring–mass system consisting of a train of liquid plugs and vapor bubbles, as shown in Fig. 1a, plays a key role in an OHP. When heat is added to the evaporating section, liquid becomes vapor producing vapor volume expansion, which increases the vapor pressure. At the same time, when the heat is removed from the condensing section, vapor condenses into liquid producing vapor volume contraction, which reduces the vapor pressure. The pressure difference between the evaporator and condenser with a vapor spring constant between them generates the oscillation motion in the system. When the heat transfer rate increases, the oscillating motion increases and enhances the heat transport capability in an OHP [18,19]. As the heat transfer rate increases, the velocity of liquid plugs and vapor bubbles increases. When the vapor velocity is higher than a critical value, the vapor penetrates the liquid plug and produces an annular flow as shown in Fig. 1b. When this takes place, the mass–spring system consisting of a train of liquid plugs and vapor bubbles disappears. The oscillating motion does not exist anymore, and the OHP reaches the maximum heat transport capability.

Consider a liquid plug moving into the evaporator section as a control volume (Fig. 1a). In order for vapor phase to penetrate the liquid plug, the momentum produced by the vapor phase will be used to overcome the total forces acting on the liquid plug, i.e.,

$$\int_0^{r_0} 2\pi\rho_v u_v^2 r dr = (-p_1 + p_2)\pi r_0^2 + 2\pi r_0\sigma(\cos\alpha_r + \cos\alpha_a) + 2\pi r_0\left(\int_0^{L_l}\tau_w dx\right) \quad (5)$$

where $\int_0^{r_0} 2\pi\rho_v u_v^2 r dr$ is the momentum produced by the vapor phase due to the vapor volume expansion, $(-p_1 + p_2)\pi r_0^2$ is due to the pressure difference acting on the liquid plug by vapor bubbles, $2\pi r_0\sigma(\cos\alpha_r + \cos\alpha_a)$ is due to the surface tension forces on both menisci, and $2\pi r_0\left(\int_0^{L_l}\tau_w dx\right)$ is due to the shear stress between the wall and the fluid of the liquid slug, and L_l is the length of the liquid slug. The shear stress on the wall can be expressed as

$$\tau_w = \mu\left(-\frac{du_l}{dr}\right)_w \quad (6)$$

When the liquid plug moves to the evaporating section, the heat addition to the evaporating section increases vapor volume and vapor pressure. Because the pressure wave speed in the liquid phase is different from that in the vapor phase, an exciting force is generated, which helps to start up the oscillating motion [19].