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Start-up Characteristics of Swallow-tailed Axial-grooved Heat Pipe under the conditions of Multiple Heat Sources

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Abstract. A mathematical model was developed for predicting start-up characteristics of Swallow-tailed Axial-grooved Heat Pipe under the conditions of Multiple Heat Sources. The effects of heat capacitance of heat source, liquid-vapour interfacial evaporation-condensation heat transfer, shear stress at the interface was considered in current model. The interfacial evaporating mass flow rate is based on the kinetic analysis. Time variations of evaporating mass rate, wall temperature and liquid velocity are studied from the start-up to steady state. The calculated results show that wall temperature demonstrates step transition at the junction between the heat source and non-existent heat source on the evaporator. The liquid velocity changes drastically at the evaporator section, however, it has slight variation at the evaporator section without heat source. When the effect of heat source is ignored, the numerical temperature demonstrates a quicker response. With the consideration of capacitance of the heat source, the data obtained from the proposed model agree well with the experimental results.

1. Introduction

Swallow-tailed axial-grooved heat pipe as depicted in Fig. 1 is thermal device with extra high efficiency due to evaporation-condensation heat transfer, good temperature evenness and stable operation under microgravity environments, which has application in a wide variety of heat transfer related applications where stable heat dissipation is needed, such as aerospace heat control systems and chip cooling technique.

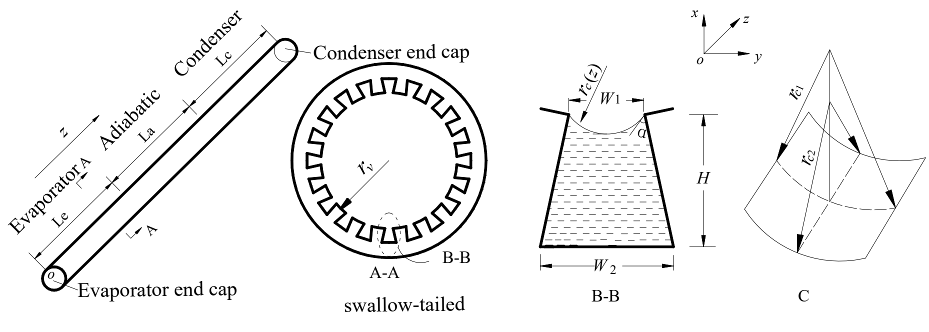

Fig. 1 Schematic diagram of Swallow-tailed Axial-grooved Heat Pipe

Most of the existing studies on start-up performance of axial-grooved heat pipes are centered on trigonal grooved heat pipes[1-5],and the proposed models mostly consider uniform heating in the evaporator section, non-uniform heating is not taken into consideration in the proposed model under multiple heat sources operating conditions, however, this working condition is ubiquitous when the heat pipe perform the thermal control, therefore, the intrinsic model face great challenge in the practical use. And moreover, most of the work dealing with the heat transfer characteristics of heat pipe under the conditions of multiple heat sources centered on the steady characteristics [6-11], the researches on the dynamic response subjected to heating with multiple heat sources are scarce, and consideration in the effect of heat source on the transient response is far less. Therefore, a one-dimensional thermal and hydrodynamic unsteady model is developed for swallow-tailed axial-grooved heat pipe under the conditions of multiple heat sources, the effect of heat source is included in the model. Nonlinear control equations are given a numerical solution. The time variations of wall temperature, capillary radius of curvature, liquid velocity are achieved from the initial start-up to steady state. The time constants in the start-up process are defined and given in different heat flux. The unsteady heat transfer characteristics of heat pipe is verified by the experiment.

2. Theoretical model

Based on kinetic theory, mass flux rate at the interface is expressed as:

$$m_i = \left(\frac{2a}{2-a}\right)\left(\frac{1}{\left(2\pi R_u / M\right)^{1/2}}\right)\left(\frac{P_v}{\left(T_v\right)^{1/2}} - \frac{P_l}{\left(T_l\right)^{1/2}}\right) \tag{1}$$

In which, a is the accommodation coefficient, In the case of phase equilibrium, a is equal to 2/3. Therefore, a is assumed to be 2/3 in this paper. R_u is the universal gas constant and the subscript v denotes the vapour.

The vapour pressure P_v can be achieved from the ideal gas equation of state:

$$P_v = \rho_v R_u T_v / M \tag{2}$$

The vapour density ρ_v can be calculated by the mass of vapour divided by the volume of vapour core, The vapour is supposed to be fully saturated, therefore, the vapour temperature is determined by the corresponding vapor density, $T_v = T_v(\rho_v)$.

The liquid pressure can be calculated by the Clausius-Clapeyron equation

$$\frac{R_u}{h_{fg}} \ln\left(\frac{P_l}{P_0}\right) = \frac{1}{T_0} - \frac{1}{T_l} \tag{3}$$

where P_0 and T_0 are reference values.

The evaporator section, adiabatic section and condenser section of axial-grooved heat pipe have different inner heat transfer process and external boundary conditions. Therefore, dynamic energy differential equation must be developed in the three sections respectively, and it can be expressed as

$$\rho_w C_{pw} A_{cw} \frac{\partial T_w}{\partial t} = \begin{cases} A_{cw} K_w \dfrac{\partial^2 T_w}{\partial z^2} + \dfrac{Q(z)}{L(z)} - m_i h_{fg} R_{wv} - h_{wl}(T_w - T_l) w_{wl}, 0 \le z \le L_e \\ A_{cw} K_w \dfrac{\partial^2 T_w}{\partial z^2} - h_{wl}(T_w - T_l) w_{wl}, L_e \le z \le L_e + L_a \\ A_{cw} K_w \dfrac{\partial^2 T_w}{\partial z^2} - h_{wl}(T_w - T_l) w_{wl} - m_i h_{fg} R_{wv} - h_\infty (T_w - T_\infty) \pi d_{out}, L_e + L_a \le z \le L_e + L_a + L_c \end{cases} \tag{4}$$

In which, $Q(z) = \begin{cases} Q_{hp}, & \text{(The position z has heat source)} \\ 0, & \text{(The position z hasn't heat source)} \end{cases}$, Q_{hp} is the heat input absorbed by the heat pipe the power of heat source, $L(z)$ is the length of heat source, R_{wv} is fin width, h_∞ is the convective heat transfer coefficient of outside surface at the condenser section, T_∞ is the temperature of cooling water.

For the heat pipes with axially microgrooves, the motive power of circulation comes from the capillary pressure generated by axial groove. As depicted in Fig. 1(B-B), the capillary pressure P_c can be achieved based on the Laplace-Young equation:

$$P_c = P_v - P_l = \sigma(\frac{1}{r_{c1}} + \frac{1}{r_{c2}}) \tag{5}$$

Compared to r_{c1}, r_{c2} is very big, $1/r_{c2}$ is very small and can be ignored. r_{c1} can be considered to be the capillary radius $r_c(z)$ .The differential form of the Laplace-Young equation can be given below:

$$\frac{dP_v}{dz} - \frac{dP_l}{dz} = -\frac{\sigma}{r_c^2(z)}\frac{dr_c(z)}{dz} \tag{6}$$

Working fluid flows in the microgrooves, the transient equation of energy conservation can be written as

$$\rho_l C_l A_l dz \frac{\partial T_l}{\partial t} = \rho_l C_l V_l A_l dz \frac{\partial T_l}{\partial z} + h_{wl}(T_w - T_l) w_{wl} dz - m_i h_{fg} R_{lv} dz \tag{7}$$

Where C_l is the specific heat of the liquid, h_{wl} is the convective heat transfer coefficient of the working fluid in the microgrooves, which is determined by the Nusselt number Nu, $h_{wl}=Nuk_l/D_h$, and k_l is the thermal conductivity of liquid.

Based on the transient momentum balance of liquid flow, the differential momentum equation can be given as

$$\frac{\partial(\rho_l A_l V_l)}{\partial t} = \rho_l A_l V_l \frac{\partial V_l}{\partial z} + A_l \frac{\partial P_l}{\partial z} - w_{wl}\tau_{wl} - R_{lv}\tau_{il} \tag{8}$$

Where V_l is the liquid velocity, τ_{wl} is wall shear stress, τ_{il} denotes liquid-vapor interfacial force.

The differential mass conservation equation can be written as follows:

$$\frac{\partial(\rho_l A_l)}{\partial t} = \frac{\partial(\rho_l A_l V_l)}{\partial z} - m_i R_{lv} \tag{9}$$

In which, ρ_l is the liquid density, τ_{wl} is the wall tangential stress, τ_{il} is the liquid-vapor interfacial shear stress, h_{fg} is the working liquid latent heat of vaporization, D_h is the liquid flow hydraulic diameter, $D_h=4A_l/w_{wl}$,A_l is the working substance radial section surface. R_{lv} is the length of liquid-vapor interface, $R_{lv}=(\pi-2\alpha)r_c$， α is the contact angle of liquid working substance.

3. Experimental verification

The proposed model which predicted the thermal performance of a swallow-tailed axial-grooved heat pipe was validated by an experimental investigation [12]. The transient thermal responses in the evaporator section of heat pipe are shown in Fig.2. As can be seen in the figure, when accounting for the influence of heat source, the numerical result demonstrates a quicker response. With consideration of the heat capacitance, the numerical result have a slower thermal response, but it still exists minor difference between numerical and experimental result. This is because the heat capacitance of the insulating layer is not included in the model. During the start-up process, the heat sources also absorb part of the input power. Thus, the energy absorbed by the heat pipe is decreased so that the thermal response of heat pipe becomes slower.

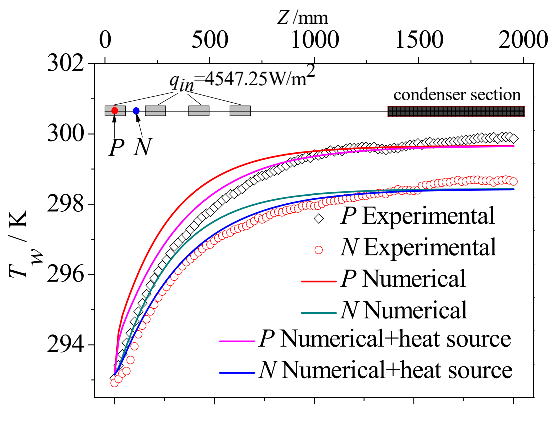

Fig.2 start-up characteristics comparison of the experimental and numerical results

4. Results and discussion

The evaporating mass rate of whole evaporator section for different heat inputs as a function of time is presented in Fig.3 when the heat pipe start subjected to single heat source. As can be seen from the figure, at the initial stage of start-up operation, the mass rate rises sharply, as times go on, it rises more slowly. This phenomenon shows that the heat input is mainly applied to improve the thermal capacity of heat pipe wall and working liquid at the initial process, however, the portion is decreased as the process continues. While the whole input heat is used to make the liquid evaporate into vapour, it is indicated that heat pipe reaches the steady state.

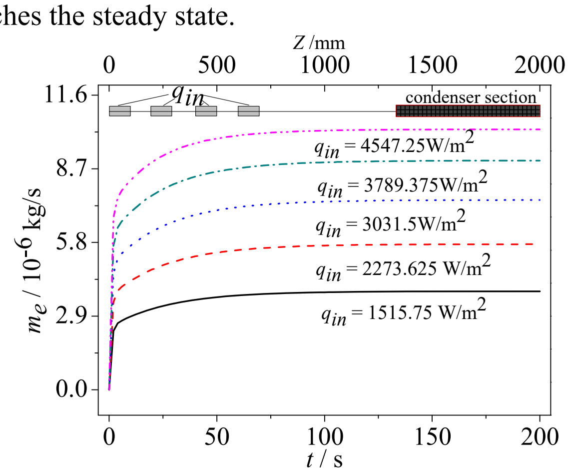

Fig. 3 Time variation of evaporating mass rate under different heat flux

Unsteady change of axial wall temperature of heat pipe during the start-up procedure is shown in the Fig.4. When the heat pipe operate subjected to heating with multiple heat sources, the temperature demonstrates step transition at the junction between the heat source and non-existent heat source on the evaporator because of heat conduction of wall and wick. As is shown in the figure, temperature distribution from the start-up to steady state shows similar trend, temperature changes with time and becomes slower gradually. It is shown that at the initial time of the start-up, phase change heat transfer have less effect compared to heat conduction, as time goes on, phase change heat change plays more and more important role. When the temperature of heat pipe comes into a constant value, heat transfer depends on liquid-vapour phase change. It can be seen from the Figure 3(a), when the influence of heat source is not included, the outside surface of evaporator is imposed on constant heat fluxes. It takes 80s for the heat pipe to attain the steady state. As is shown in the Figure 3(b), when the effect of heat source is considered, It takes 110s for the heat pipe to attain the steady state. Due to the existence of heat source, part of the input heat is absorbed by the heat source to raise its temperature. The heat flux imposed on heat pipe changes with time, it is no longer a constant value.

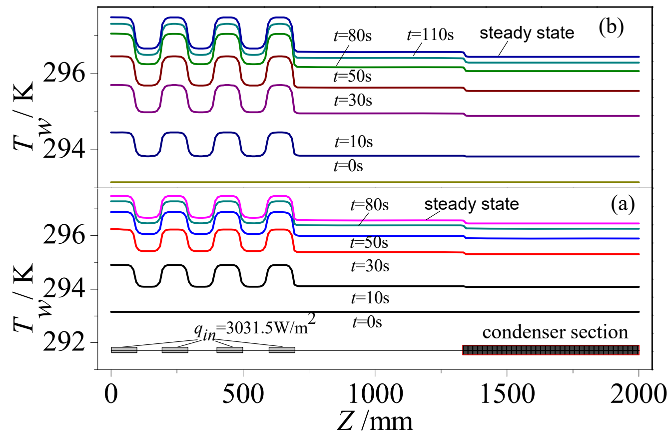

Fig. 4 The axial distribution of wall temperature during the start-up process: (a) without the effect of heat source (b) with the effect of heat source

The transient profile for liquid velocity is illustrated in Fig.5. As is shown in the figure, at the initial time working substance in the microgrooves keeps stationary and liquid velocity equals zero, working fluid flows from the condenser end to starting point of evaporator section. When the heat pipe lie on the horizontal position, once heat flux imposed on the heat pipe comes into the liquid-vapour interface, it is indicated that evaporation starts and liquid starts to flow towards evaporator section. The driving force of working fluid comes from the variation of capillary radius. The liquid velocity increase gradually, once the evaporation mass flux equals liquid replenishment of condenser section. It takes about 110 seconds for the liquid velocity to attain steady value. As is shown in the figure, the liquid velocity changes drastically at the evaporator section where subjected to the heat source, however, it has slight variation at the evaporator section without heat source. It decreases moderately in the adiabatic section. It drastically drops in the condenser section because saturated vapour condenses to make the meniscus radius raises sharply.

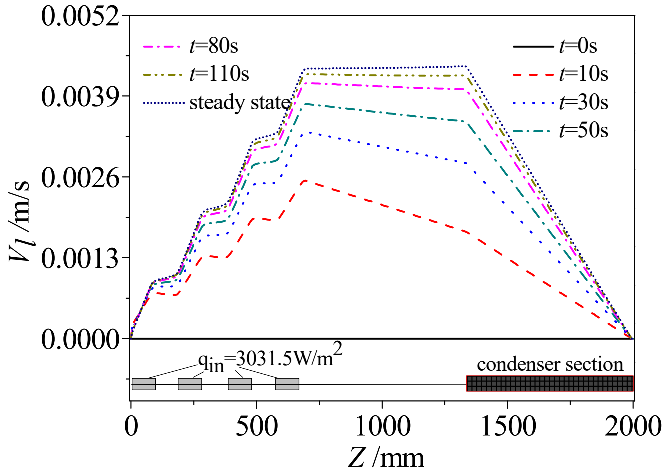

Fig.5 The axial distribution of liquid velocity during the start-up procedure

To predict the start-up characteristics of heat pipe, the time from initial start-up to steady state is an important yardstick. Fig.6 presents the effect of heat input on the time constant during the start-up. As is illustrated in the figure, the heat flux demonstrates minor effects on the time constants of heat pipe. An increase in heat flux slightly decreases the start-up time constants for wall temperature of heat pipe. As indicated in Fig.6 (a), when the influence of heat source is not included, the time constant during the start-up operation is around 26.7 s. As is shown in Fig. 6(b), when the effect of heat source is considered, the time constant during the start-up operation is approximately 31.9 s, this is due to the existence of the heat sources, they also absorb part of the input power. Thus, the energy entering the heat pipe varies with time, it results in the increase of time constant of start-up operation.

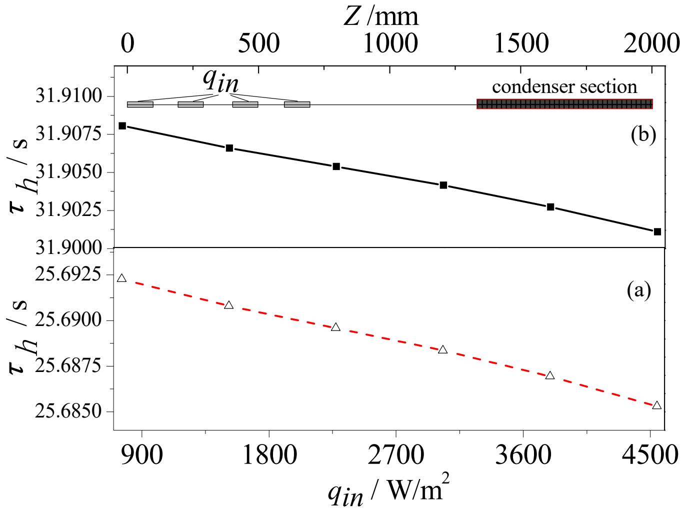

Fig.6 Effect of heat input on the time constant during the start-up: (a) without the influence of heat source (b) with the influence heat source

5. Conclusions

(1) At the initial stage of start-up operation, the mass rate rises sharply, as times go on, it rises more slowly. This phenomenon shows that the heat input is mainly applied to improve the thermal capacity of heat pipe wall and working liquid at the initial process, however, the portion is decreased as the process continues. While the whole input heat is used to make the liquid evaporate into vapour, it is indicated that heat pipe reaches the steady state.

(2) The temperature demonstrates step transition at the junction between the heat source and non-existent heat source on the evaporator because of heat conduction of wall and wick. When the temperature of heat pipe comes into a constant value, heat transfer depends on liquid-vapour phase change. Once the influence of heat source is not included, the outside surface of evaporator is imposed on constant heat fluxes. It takes 80s for the heat pipe to attain the steady state. when the effect of heat source is considered, It takes 110s for the heat pipe to attain the steady state. The heat flux imposed on heat pipe changes with time, it is no longer a constant value.

(3) When accounting for the influence of heat source, the numerical result demonstrates a quicker response. With consideration of the heat capacitance, the numerical result have a slower thermal response, but it still exists minor difference between numerical and experimental result. This is because the heat capacitance of the insulating layer is not included in the model. During the start-up process, the heat sources also absorb part of the input power. Thus, the energy absorbed by the heat pipe is decreased so that the thermal response of heat pipe becomes slower.

Acknowledgements

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