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Unsteady heat transfer performance of heat pipe with axially swallow-tailed microgrooves

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Abstract. A mathematical model is developed for predicting the transient heat transfer and fluid flow of heat pipe with axially swallow-tailed microgrooves. The effects of liquid convective heat transfer in the microgrooves, liquid-vapor interfacial phase-change heat transfer and liquid-vapor interfacial shear stress are accounted for in the present model. The coupled non-linear control equations are solved numerically. Mass flow rate at the interface is obtained from the application of kinetic theory. Time variation of wall temperature is studied from the initial startup to steady state. The numerical results are verified by experiments. Time constants for startup and shutdown operation are defined to determine how fast a heat pipe responds to an applied input heat flux, which slightly decreases with increasing heat load.

1. Introduction

Heat pipes with axially swallow-tailed microgrooves as shown in Fig. 1 are highly efficient thermal systems due to phase change heat transfer of the internal working fluid. It has demonstrated excellent thermal performance, high degree of temperature uniformity and reliable operation under microgravity conditions, which are being used in the area of efficient and reliable heat removal, such as spacecraft thermal control systems and microelectronic cooling systems.

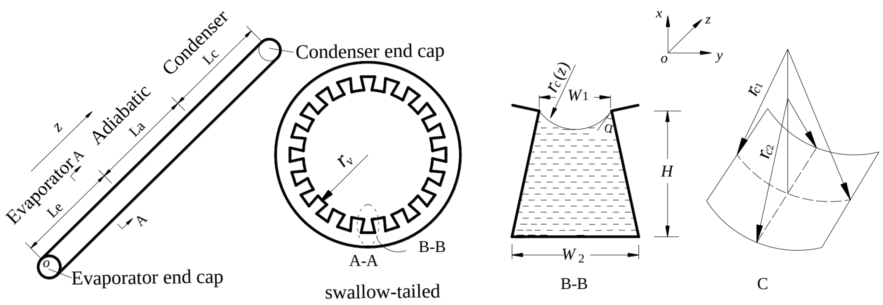

Fig. 1 Schematic diagram of heat pipe with axial swallow-tailed microgrooves

In many thermal control applications, heat pipes typically do not operate at steady state conditions. Grarimella [1] pointed out that many heat pipes operate in transient conditions to manage heat dissipation problems and heat fluctuations of electronic components. Colwell and Chang [2] made a series of tests on an 80-cm long Freon(R11) stainless-steel heat pipe under normal and supercritical startup. Wang [3,4]performed experimental investigation on a flat plate copper heat pipe during startup and shutdown cycles, experimental results were compared with the analytical model of Zhu [5] and were found to be in very good agreement. Most of the available researches on transient characteristics of heat pipes with axially microgrooves are focused on the triangular microgrooves [6-9], while there have been very few reports on heat pipes with axially swallow-tailed microgrooves. Unsteady investigation made on the heat pipe with the axially swallow-tailed microgrooves not only understand the startup / shutdown and load variation characteristics of heat pipe, also contribute to preliminary transient analyses of loop heat pipes and capillary pumped loops. Therefore, in the current study, a one-dimensional fluid flow and heat/mass transfer unsteady model is developed for heat pipe with axially swallow-tailed microgrooves and coupled nonlinear governing equations are solved numerically.

2. Theoretical model

In the evaporator section, the heat imposed on the heat pipe wall together with the difference of the heat entering and leaving the control element by conduction is partly used to raise the heat pipe wall temperature while the remaining heat is absorbed by the working liquid in the grooves through convection. Therefore, the unsteady energy balance equation of the heat pipe wall in the evaporator section can be given by

$$\rho_w C_w A_w dz \frac{\partial T_w}{\partial t} = A_w dz K_w \frac{\partial^2 T_w}{\partial z^2} + \frac{Q_{in} dz}{L_e} - h_{wl}(T_w - T_l) w_{wl} dz \tag{1}$$

In the adiabatic section, variation of wall temperature as a function of time not only comes from the condensation of vapour in the fin and swallow-tailed microgrooves, but also comes from the working liquid heat convection in the swallow-tailed microgrooves and the difference of the heat entering and leaving the control element by conduction, the unsteady state energy balance of the heat pipe wall can be expressed as:

$$\rho_w C_w A_w dz \frac{\partial T_w}{\partial t} = A_w dz K_w \frac{\partial^2 T_w}{\partial z^2} - m_i h_{fg} R_{wv} dz - h_{wl}(T_w - T_l) w_{wl} dz \tag{2}$$

In the condenser section, the heat taken up by the cooling liquid and the difference of the heat entering and leaving the control element by conduction not only comes from the condensation of vapour in the fin and swallow-tailed microgrooves, but also comes from the working liquid heat convection in the swallow-tailed microgrooves , so the unsteady state energy balance of the heat pipe wall is written as

$$\rho_w C_w A_w dz \frac{\partial T_w}{\partial t} = A_w dz K_w \frac{\partial^2 T_w}{\partial z^2} - h_{wl}(T_w - T_l) w_{wl} dz - m_i h_{fg} R_{wv} dz - h_\infty (T_w - T_\infty) \pi d_{out} dz \tag{3}$$

The energy entering the control volume includes two parts, one is the sensible heat of working liquid flowing into the element, the other is the heat taken up by the working liquid from the heat pipe wall. The energy leaving the control volume also include two portion, one is the sensible heat of working liquid flowing out of the element, the other is the heat leaving the element by evaporation. Therefore, the unsteady energy conservation equation of the liquid in the microgrooves in differential form is expressed as

$$\rho_l C_l A_l dz \frac{\partial T_l}{\partial t} = \rho_l C_l V_l A_l dz \frac{\partial T_l}{\partial z} + h_{wl}(T_w - T_l) w_{wl} dz - m_i h_{fg} R_{lv} dz \tag{4}$$

The rate of accumulation of the convective momentum is equal to the momentum change due to liquid flow in the microgrooves, pressure force acting on the element, the wall shear force and liquid-vapour interfacial force. So the differential form of unsteady state momentum balance of liquid can be expressed as

$$\frac{\partial(\rho_l A_l V_l)}{\partial t} = \rho_l A_l V_l \frac{\partial V_l}{\partial z} + A_l \frac{\partial P_l}{\partial z} - w_{wl} \tau_{wl} - R_{lv} \tau_{il} \tag{5}$$

The variation of mass as a function of time is equal to the difference of change of liquid mass due to convection and the mass evaporated from the control volume. So the differential form of mass balance at unsteady state can be expressed as

$$\frac{\partial(\rho_l A_l)}{\partial t} = \frac{\partial(\rho_l A_l V_l)}{\partial z} - \frac{m_i h_{fg} R_{lv}}{h_{fg}} \tag{6}$$

3. Experimental validation

The present mathematical model is verified by an experiment on the heat transfer performance of a swallow-tailed heat pipe with the geometry shown in Table 1. The schematic of the experimental setup is shown in Fig.2. Heat removal at the condenser was provided using a clamped cold water jacket, a Thin-film heater was mounted on the evaporator region to simulate the heat source. Adequate contact between the water jacket and the condenser region of the heat pipe, as well as the thin-film heater and evaporator region, was ensured using pads of thermal interface material. The thin-film heater was insulated on top using a thick piece of acrylic. Nine thermocouples (type-T), which had uncertainty of ±0.1°C，are attached on the outside wall of the heat pipe. All the thermocouples are calibrated against any variation in the room temperature. Before each test, heat pipe was allowed to reach a steady-state temperature condition by turning on the flow through the cooling jacket. This provided the initial condition for the experiments as well as for the model. The heater was then turned on, and temperature measurements continued until a new steady state was reached. Temperature measurements were recorded during the transient heating period as well as at steady state. The axial temperatures and the water inlet and outlet temperatures are recorded at an interval of 1 second using the data acquisition Instrument. Steady state temperatures reported at each power level were averaged over a period of one minute. Working temperature is regarded as the mean value of these measured data of the adiabatic segment. Given a heat load, Q_{in}, the working temperature of the heat pipe can be controlled by regulating the temperature and velocity of the cooling water supplied by the constant temperature water bath.

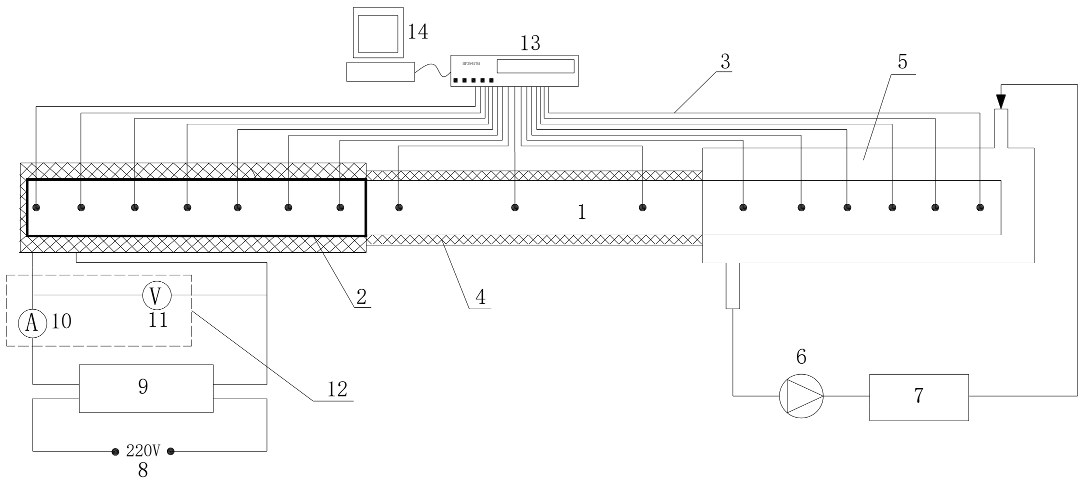

Fig.2 Schematic diagram of experimental setup

1- Heat pipe, 2-Thin-film electrical heater, 3-Thermocouple, 4-Thermal insulation, 5-Tank,6-Pump, 7-Constant temperature water bath, 8-power meter, 9-Voltage regulator, 10-Amperemeter, 11-Voltmeter, 12-Power supply, 13-Data acquisition instrument, 14-Computer

4. Results and discussion

To make the comparison meaningful, the transient behaviour of an aluminium-ammonia axially grooved heat pipe was studied using the experimental working condition. Heat pipe started at ambient temperature, T=293.15, constant heat load of 200W was applied and condenser section was cooled by convection. Both the experimental and numerical computational temporal temperature response for the evaporator section of heat pipe are presented in Fig.3. In view of the uncertainty in the internal dimensions and wick properties of the heat pipe tested, the agreement between the numerical prediction and experiment is reasonable.

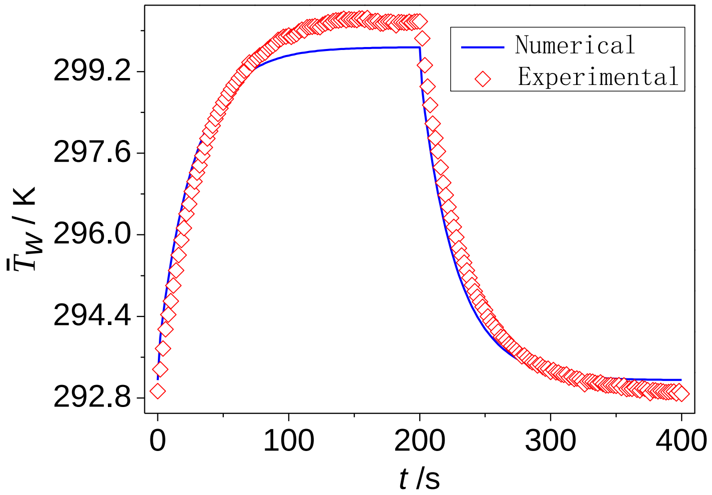

Fig.3 Comparison of the experimental and numerical wall temperature of the evaporator section

Model predictions for transient variation of axial wall temperature distribution are shown in Fig.4. It can be seen from the figure, the temperature distribution at each section (i.e. the evaporator, adiabatic and condenser sections) during the startup process is almost uniform, and the major temperature difference occurs at the junctions among neighbouring sections. The uniformity of the temperature distribution at the evaporator section is vital for the heat pipe to be used to thermal control of electronic product. It can be seen from the figure that temperature increase gets more and more slowly with time. This phenomenon also indicated that, during the initial startup, heat transported by phase change play less important role in the whole heat transfer, and become more and more important as the process continues. Once the heat pipe reaches the steady state, the input heat is completely transported through evaporation and condensation. In this context, heat transport device consisting of grooved heat pipe can reduce weight of electronic cooling system, which is of significance to aero- and space flight. In addition, the wall temperature of heat pipe maintain at a relatively low level for the high heat flux imposed.

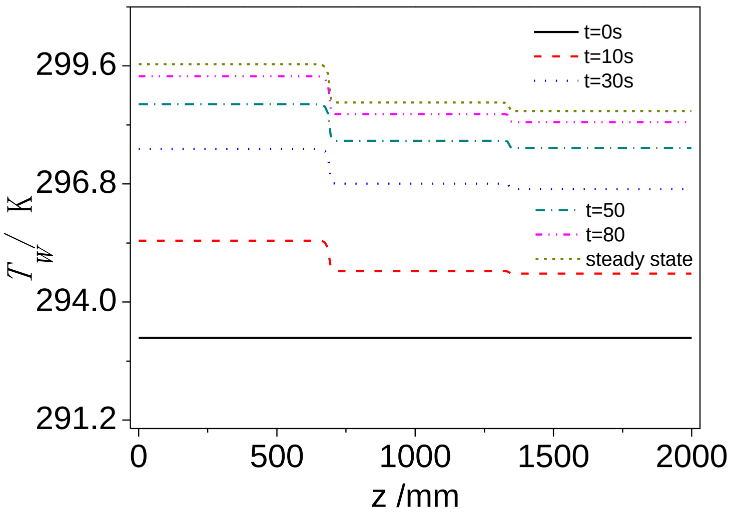

Figure 4 Transient profiles of axial temperature

The time required to reach steady state is an important parameter for the startup/shutdown of a heat pipe. The concept of the time constant, τ, as proposed by El-Genk and Huang [10] is utilized in the analysis to comprehensively simulate startup/shutdown operation.

Figure 5 presents time constants for both the startup and shutdown operation for different heat inputs. As can be seen from the figure, heat flux minimally affects the time constants of heat pipe. Increasing the heat flux slightly lowers the time constants. The results also show that time constant for the shutdown operation is a little more than that for the startup operation.

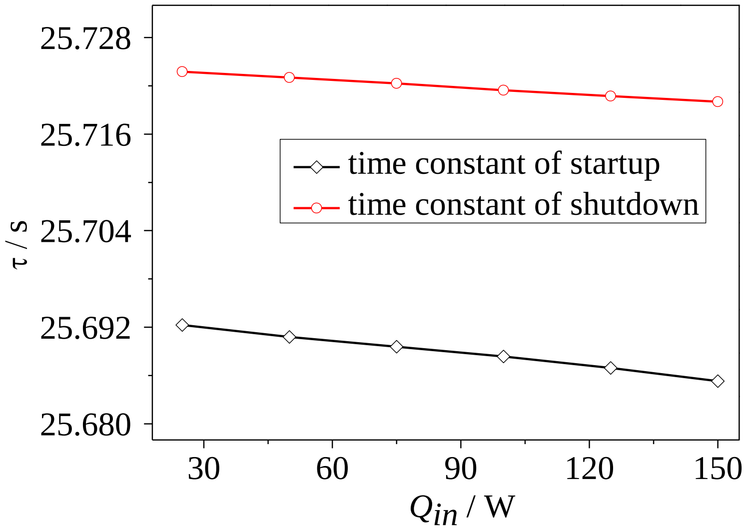

Fig. 5 Startup/shutdown time constant profiles as a function of heat inputs

Fig.6 shows the temporal variation for the effective thermal conductivity. As can be seen from the figure, the effective thermal conductivity increases gradually until the heat pipe achieve the steady state during the startup operation, and retains a constant value during the shutdown operation. In addition, there is a sharp increase in the value of the effective thermal conductivity when the heat pipe operate from the steady state to shutdown operation.

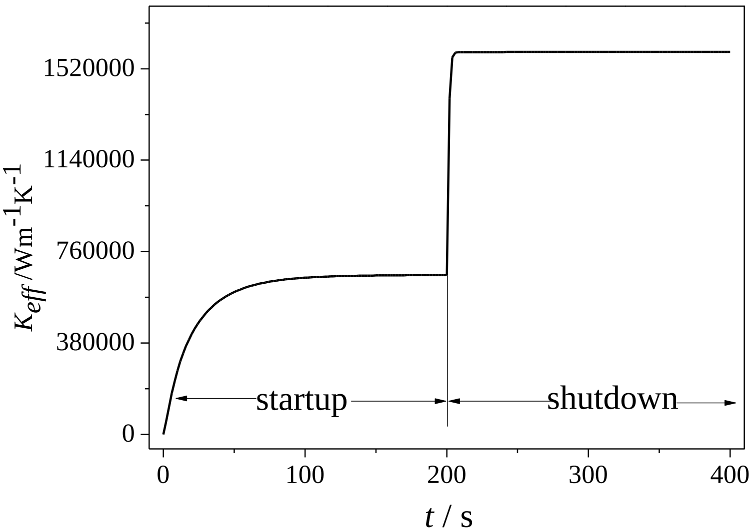

Fig. 6 Transient variation of the effective thermal conductivity

5. Conclusions

A one-dimensional analysis model for a heat pipe with axial swallow-tailed microgrooves is developed and analyzed numerically to predict the transient heat transfer and fluid flow characteristics. The transient thermal performance of heat pipe is investigated by the experiment. The results obtained from the proposed model are in close agreement with the experimental data in terms of startup and shutdown temperature response. The conclusions can be summarized as: Temperature distribution at each section (i.e. the evaporator, adiabatic and condenser sections) during the startup process is almost uniform, and the major temperature difference occurs at the junctions among neighbouring sections. Heat flux minimally affects the time constants of the heat pipe. Increasing the heat flux slightly lowers the time constants for wall temperature response of heat pipe. The results also show that the time constant for the shutdown operation is a little more than that for the startup operation. Effective thermal conductivity increases gradually until heat pipe achieve the steady state during the startup operation, and retains a constant value during the shutdown operation.

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