Heat pipe: a simple one-dimensional model and an experimental study

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In this work the authors proposed to use 1-D model for heat and mass transfer calculations inside a porous heat pipe. Several limits are taken into account: sound, boiling, capillary and dry out of evaporator. To calculate the heat flow, we analyzed processes in four zones: the evaporator, the condenser, the porous body in which fluid flows, and the cavity in which steam flows. We assume that in the evaporator all the incoming liquid evaporates, and in the condenser, all the vapor condenses. The liquid and the vapor are in a state of saturation and, therefore, the pressure in the condenser and the evaporator is equal to the saturation pressure. The good agreement between theoretical and experimental results is observed.

1. Introduction

A heat pipe is a passive device that can dissipate a large amount of heat due to evaporation and condensation of the working fluid. The working fluid is evaporating in the heater part. Due to the pressure difference created, the steam is supplied to the condenser, where it releases heat and changes its phase to liquid. The liquid is returned to the evaporator zone using the wick capillary force. Since the latent heat of evaporation is usually large, heat pipes can efficiently transfer a significant amount of heat over a long distance with a relatively small temperature drop. The choice of working fluid depends on the working temperature of the heat pipes. Because of the wide range of operating temperatures, heat pipes can be used in many areas, such as the accumulation of thermal energy, the cooling of electronic and electrical equipment [1, 2], heat exchangers and heat pumps [3, 4].

The phenomenon of liquid penetration into a porous medium under the action of capillary forces is widely used, including microfluidic devices [5, 6], fuel cells [7, 8], inkjet printing [9] and much more. Such fluid movement in a porous medium is due to negative capillary pressure. To create a model, a porous medium can be simplified to a hollow tube with an effective capillary radius. The first description of the dynamics of capillary movement in the pipe was made by Lukas [10] and Washburn [11] at the beginning of the last century. They suggested that the distance of penetration of a liquid into a porous medium $L$ and the time of penetration $t$ satisfy the diffusion relation $L = \sqrt{Dt}$, where $D$ is the diffusion coefficient, depending on the size of the pipe and the properties of the liquid. It was also established that the diffusion correlation between the distance and the time of fluid penetration in porous media corresponds to the Lukas-Washburn law for both unidirectional [12, 13] and radial penetration [14, 15, 16, 17].

In the present work, one-dimensional analysis of the steady state of a heat pipe was performed. To calculate the maximum heat flux, it is necessary to consider the main constraints that impede its growth. First of all, these are capillary restrictions, sound effects, liquid entrainment from the surface
of the wick and boiling up of the coolant. These restrictions, in turn, depend on the various properties of the coolant, the structure of the wick and the geometry of the heat pipe.

2. Theory description

A detailed theoretical and experimental interpretation of the sound limit is presented in the works of Levy [18], Kemme [19] and Deveral [20]. In a heat pipe with a constant diameter of the vapor channel, the flow is accelerated and slowed down due to the supply of steam in the evaporator and drainage in the condenser. The sound limit is reached when the steam velocity at the evaporator outlet becomes sound. The description of the sound limit can be obtained from the one-dimensional theory of steam flow, in which it is assumed that the properties of steam obey the ideal gas law, inertial effects are predominant, friction effects can be neglected. Then the sound limit of the heat pipe can be expressed as follows [18]:

$$Q_s,\text{max} = A_v \rho_v \lambda \left[ \frac{\gamma_v R_i T_v}{2 (\gamma_v + 1)} \right]^{1/2}$$  \hspace{1cm} (1)

Here $A_v$ is the cross-sectional area of the steam channel, $\rho_v$ is the vapor density, $\lambda$ is the latent heat of vaporization, $R_i$ is the vapor constant, $T_i$ is the vapor temperature.

Since in the heat pipe the liquid and vapor move in opposite directions, a shearing force arises at the interface. If the vapor velocity is high enough, then the limit can be reached when the liquid will break away from the surface of the wick and be carried away by the flow of steam. The occurrence of this phenomenon leads to instant drying of the wick in the evaporation zone. The phenomenon of liquid entrainment was detected by the sound emitted by liquid droplets hitting the end of the condenser, and the evaporator overheating sharply [21]. The maximum heat flux, taking into account the restriction on the entrainment of liquid is expressed as follows [22]:

$$Q_e,\text{max} = A_v \lambda \left( \frac{\sigma \rho_v}{2r_h} \right)$$  \hspace{1cm} (2)

Here $\sigma$ is the coefficient of surface tension, $r_h$ is the hydraulic radius of the pore surface of the wick.

The main mechanism of heat transfer in the evaporator and the heat pipe condenser is thermal conductivity with evaporation and condensation. The passage of heat through a wick saturated with liquid is accompanied by the appearance of a radial temperature gradient in the liquid. With an increase in heat flow in the wick, vapour bubbles may begin to form. Their appearance is undesirable because they lead to the occurrence of overheated areas and obstructing the circulation of fluid. The heat flow restriction associated with vaporization in a heat pipe is called boiling restriction. There is a fundamental difference between this restriction and the rest, since here the restriction is imposed on the radial heat flux, and in the rest on the axial one. The expression for limiting boiling power is:

$$Q_b,\text{max} = \frac{2\pi L_e k_e T_v}{\lambda \rho_v \ln \left( r_i / r_v \right)} \frac{2\sigma}{r_n}$$  \hspace{1cm} (3)

Here $L_e$ is the length of the evaporation zone, $k_e$ is the effective thermal conductivity of the wick, $r_i$ is the inner radius of the tube envelope, $r_v$ is the radius of the vapor channel, $r_n$ is the critical radius of the nucleus.

In the course of steady-state operation of the heat pipe, the coolant flows as a vapor phase from the evaporation zone to the condensation zone and returns to the evaporator in the form of a liquid phase. Since steam moves from the evaporator to the condenser, there is a pressure gradient along the vapor
channel in the steam flow. There is also a pressure gradient in the fluid, which drives it back from the condenser to the evaporator. For the required pressure balance, it is necessary that the pressure from the liquid at the liquid-vapor interface over the entire length of the pipe is different from the pressure from the vapor, except for the point where this difference is minimal and equal to zero. This pressure difference between the phases is called capillary pressure and is created by the surface tension and the porous structure of the wick. There is a maximum capillary pressure that can be achieved with the selected coolant-wick pair. If the heat pipe is operating continuously without drying the wick, the working capillary pressure should not exceed its maximum value. This restriction, which is superimposed on the transferred thermal power, is known as a capillary restriction:

\[
Q_{c,\text{max}} = \frac{(QL)_{c,\text{max}}}{0.5L_c + L_{\text{at}} + 0.5L_e}
\]  

(4)

Here \((QL)_{c,\text{max}} = \frac{2\sigma / r_c}{F_l + F_v}\) is the capillary restriction of the transmitted power factor, \(F_l = \frac{\mu_l}{KA_w \rho_l \lambda}\) is the liquid friction coefficient, \(F_v = \frac{(f_v \Re_v) \mu_v}{2A_v r_h^2 \rho_v \lambda}\) is the vapor friction coefficient, \(r_c\) is the capillary radius, \(L_c\) is the condenser length, \(L_{\text{at}}\) is the transport zone length, \((f_v \Re_v)\) is the hydraulic resistance coefficient, \(\mu_v\) is the vapor viscosity, \(\mu_l\) is the liquid viscosity, \(K\) is the wick permeability, \(A_w\) is the area cross section wick.

2.1. Simple 1d model
To calculate the heat flow, we will analyze the processes in each zone of the heat pipe. We distinguish four such zones: the evaporator, the condenser, the porous body in which fluid flows, and the cavity in which steam flows. To estimate the speed of steam, we use the well-known dependency:

\[
\Delta p = \lambda \frac{L}{D_h} \frac{\rho_v \mu_v^2}{2}
\]  

(5)

In the case of laminar motion, the drag coefficient:

\[
\lambda = \frac{64}{\Re_v} = 64 \frac{\mu_v}{\rho_v \mu_v D_h}
\]  

(6)

For the turbulent case:

\[
\lambda = \frac{0.316}{\Re_v^{0.25}}
\]  

(7)

Substituting (6) or (7) into (5), we find the stationary velocity of the vapor.

To estimate the fluid velocity in a porous body, we turn to the theory of Lukas-Washburn:

\[
\frac{dl}{dt} = \frac{2\sigma \cos \theta / r - (P_1 - P_2)}{8\mu} \left( r^2 + 4\varepsilon r \right)
\]  

(8)

\[
\frac{l^2}{t} = \frac{2\sigma \cos \theta / r - (P_1 - P_2)}{8\mu} \left( r^2 + 4\varepsilon r \right) = C
\]  

(9)
\[ l = \sqrt{C \cdot t} \]  
(10)

\[ \frac{dl}{dt} = u_t = \frac{\sqrt{C}}{2\sqrt{t}} \]  
(11)

\[ t_f = \frac{L^2}{C} \]  
(12)

\[ u_t = \frac{C}{2L} \]  
(13)

We assume that in the evaporator all the incoming liquid evaporates, and in the condenser, all the vapor condenses. In this case, the liquid and vapor are in a state of saturation and, therefore, the pressure in the condenser and the evaporator is equal to the saturation pressure.

When calculating the heat flow, it is necessary to keep track of it so that it is not larger than the maximum possible. Thus, a mathematical model is presented for calculating the maximum heat flux in a heat pipe depending on the properties of the coolant and the characteristics of the porous medium.

3. Experimental results
A heat pipe 22 cm long was used to study heat transfer, where the measurements of the condenser, evaporator, and transport zones are 2.5 cm; 2.2 cm and 17.3 cm, respectively. K-type thermocouples are installed in the evaporator and condenser zones, which fix the temperature. Sintered copper particles with a characteristic size of 50 μm were used as a porous medium. Figure 1 shows a schematic representation of a heat pipe and an experimental setup. The evaporator power was set by the power source to which the heater is connected. The condenser zone was cooled with a thermostat.

![Schematic representation of the heat pipe and experimental setup.](image)

Fig. 1. Schematic representation of the heat pipe and experimental setup.

Figure 2 shows a comparison of the calculated heat flux and experimental data at the same values of the evaporator and condenser temperatures. It can be seen from the figure that as the heat flux increases, the temperature difference between the evaporator and the condenser also increases. In this case, it is possible to distinguish two zones with different types of dependence. In the first zone, with a small heat flux (up to 15 W), the derivative of the graph is larger, and the heat pipe operates without any restrictions. In this area, one can see a fairly good agreement between experimental and calculated data.
Fig. 2. Experimental and calculated dependences of the heat flux on the temperature difference between the condenser and the evaporator.

In the second part of the graph, with an increased heat flux (more than 15 W), the derivative of the graph decreases, which indicates the operation in the limiting mode. Based on the calculation data, it can be said that in this area the velocity of the steam, which reaches the sound velocity, significantly increases. However, the results of the calculation give overestimated values of the heat flux under the same conditions, which may indicate that some other restrictions that are not taken into account in the numerical model are starting to impact.

4. Conclusion

In this paper the 1-D model is developed for mass- and heat transfer inside the heat pipe with porous media. A comparison of theoretical and experimental results is carried out. The satisfactory agreement between theoretical and experimental results are demonstrated for heat flux versus temperature difference between evaporator and condenser.

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