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Phase change driving mechanism and modeling for heat pipe with porous wick

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According to heat pipe theory, capillary force is the only driving force for the circle of working fluid in heat pipe with porous wick. By developing a simulating circuit of liquid and vapor flow in heat pipe with porous wick, this paper presents a new driving mechanism which is from phase change of fluid. Furthermore, by analyzing transport process of working fluid between evaporation and condensation interfaces, a mathematical model is developed to describe this driving mechanism. Besides, calculating examples are given for heat pipe with water as working fluid to predict its driving force and flow resistance. By applying the model presented in the paper, thermal design and calculation for heat pipe with porous wick, especially for miniature heat pipe, can be made correctly, and phase change driving mechanism of working fluid can be explained, which thereby leads to a better understanding of heat transfer limitation of heat pipe with porous wick.

Heat pipe, porous wick; phase change; driving mechanism; modeling

Heat pipe is a kind of heat transfer device and its operating principle is based on evaporation and condensation of working fluid[1–7]. Heat pipe with porous wick is a kind of heat pipe which utilizes capillary pumping forces to ensure fluid circulation[8]. Heat pipe with porous wick has obvious advantages, for example, it is capable of passively transporting more heat over long distance with small temperature difference, and there are no moving parts for pumping working fluid. Therefore, heat pipe with porous wick is widely used in the field of cooling for equipment both in space and ground.

Figure 1 shows the structure of a typical heat pipe with porous wick. It is made up of evaporation section, adiabatic section and condensation section, and its inner wall is covered by porous wick. The working process of a heat pipe with porous wick is that liquid evaporates in the evaporation section, vapor comes to the condensation section via vapor line and condenses to liquid, then liquid flows back to the evaporation section driving by capillary force of the porous wick.

1 Operation mechanism of a heat pipe with porous wick

The driving mechanism of a heat pipe with porous wick can be conventionally expressed as the fact that capillary force is a driving force to make working fluid circulate by overcoming pressure drops of liquid flow and vapor flow as well as gravity. The operational condition of a heat pipe with porous wick is[9]

$$\Delta p_{\text{total}} = \Delta p_{\text{capillary}} > \sum_{i=n} \Delta p_i \,, \qquad (1)$$

where $\Delta p_{\text{capillary}}$ is capillary pressure head, Δp_i is subsection pressure drop of working medium circle ($i=1, 2, \cdots, n$) and n is the number of subsections.

The capillary force is calculated by the following Young-Laplace equation:

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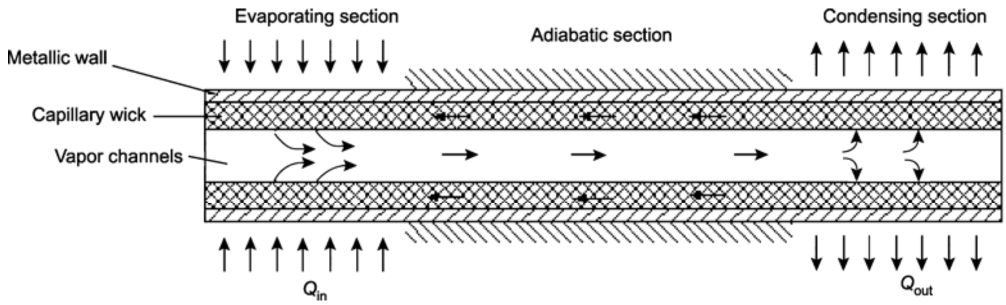

Figure 1 Schematic of a heat pipe with porous wick.

$$\Delta p_{\text{capillary}} = p_v - p_l = \frac{2\sigma\cos\theta}{r}, \tag{2}$$

where p_v and p_l are vapor and liquid pressures across evaporation interface of capillary wick respectively, σ is surface tension coefficient, θ is liquid contact angle, r is capillary radius.

The conventional expression for driving mechanism is, however, imperfect. Ishizuka[1)] established a prototype system realizing the circle of working fluid FC-12 by vapor energy from liquid phase change and gravity of condensation liquid without other driving force. This indicates that capillary force is not the only driving mechanism for the operation of heat pipe with porous wick or other similar thermal system. In other words, the system can circle without capillary force. We consider that driving mechanism of a heat pipe with porous wick is related to two types of interface actions. The first one is interface pressure head $\Delta p_{dr,l}$ which represents the capillary driving force. The capillary force on evaporation interface drives liquid circulation and overcomes reverse capillary force on condensation interface. The second one is interface pressure head $\Delta p_{dr,v}$ which represents the phase-change driving force. The driving mechanism can be explained as the fact that due to the action of heat source on evaporating interface and heat sink on condensation interface, the vapor circulates along the vapor line.

According to heat-electric analogy, a simulating loop of driving head and pressure drop in heat pipe with porous wick is established as shown in Figure 2. It indicates that driving function of a heat pipe with porous wick includes two parts: $\Delta p_{dr,l}$ drives liquid movement and $\Delta p_{dr,v}$ drives vapor movement. Ignoring pressure losses across evaporation and condensation interfaces, the total driving force equals the total flow resistances. As shown in the figure, Δp_v is vapor pressure loss via vapor line, Δp_l is liquid pressure loss via porous wick, $\Delta p_{eq,l}$ is reverse capillary force on the condensation interface, i.e. $\Delta p_{eq,l} = 2\sigma/r_c$, where r_c is curvature radius of liquid meniscus on condensation interface. So $\Delta p_{eq,l}$ is an equivalent resistance on condensation interface. The earthing point in Figure 2 is a reference potentid in the system. If the reference potentid is defined to be zero, the voltage in the vapor side is positive and the voltage in the liquid side is negative. The point with zero voltage is located at the condensation interface, where heat is transferred out of heat pipe.

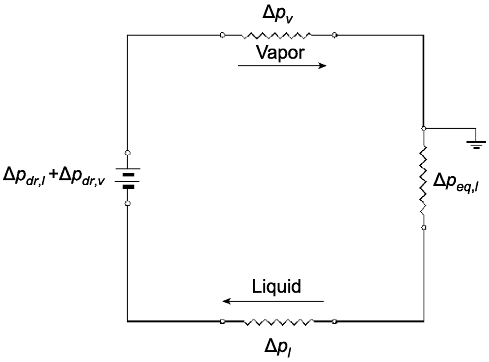

Figure 2 Heat-electric loop of capillary force and pressure drop in heat pipe with porous wick.

Based on the above analysis, the steady operation status for a heat pipe with porous wick shown in Figure 2 can be expressed as

$$\Delta p_{\text{total}} = p_v - p_l = \Delta p_{dr,l} + \Delta p_{dr,v} = \sum_{i=n} \Delta p_i , \tag{3}$$

with

$$\Delta p_{dr,l} = \Delta p_l + \Delta p_{eq,l} , \tag{4}$$

and

$$\Delta p_{dr,v} = \Delta p_v , \tag{5}$$

1) Ishizuka M, Yakagawa S, Koizumi K. Development of a self-cooling system utilizing waste heat from electronic equipment. Proceeding of the Sixth International Conference on Enhanced, Compact and Ultra-Compact Heat Exchangers: Science, Engineering and Technology, Potsdam, Germany, 2007

where p_v and p_l are vapor and liquid pressures respectively across evaporation interface, and driving heads are

$$\Delta p_{dr,l} = \frac{2\sigma}{r_e}, \tag{6}$$

and

$$\Delta p_{dr,v} = p_v - p_c . \tag{7}$$

In eqs. (6) and (7), r_e is curvature radius of liquid meniscus on evaporation interface, and p_c is saturation pressure of vapor above condensation interface.

The capillary force $2\sigma/r_c$ in eq. (6) is to balance pressure drop in porous wick as well as reverse capillary force on the condensation interface. When heat pipe works steadily, the capillary interface does not move, and there is no coupling relation for variables between equations of liquid and vapor across the interface. Therefore, capillary head $\Delta p_{dr,l}$ can not drive vapor flow. So, capillary force on evaporation interface only acts on the liquid within porous wick, and driving mechanism of vapor flow needs to be described as a new mathematical model.

2 Phase change driving model for heat pipe

According to the analysis for heat and mass transfer in the heat pipe shown in Figure 1, we can know that:

(1) The energy schlepped by vapor is far larger than that by liquid because of the latent heat added by phase change of liquid;

(2) Compared with liquid, relatively larger specific volume and vapor velocity results in more energy loss in the flow;

(3) The vapor produced by phase change of liquid raises the pressure in heat pipe, which helps to form the driving potential of vapor in heat pipe.

As vapor energy at evaporation and condensation interfaces can be expressed as $\dot{m}_e A_e(h_e + 1/2u_e^2 + gz_e)$ and $\dot{m}_c A_c(h_c + 1/2u_c^2 + gz_c)$ respectively, energy loss between the two interfaces is written as

$$\Delta E = \dot{m}_e A_e(h_e + 1/2u_e^2 + gz_e) - \dot{m}_c A_c(h_c + 1/2u_c^2 + gz_c). \tag{8}$$

The mass flux in steady state is given as

$$\dot{m} = \dot{m}_e = \dot{m}_c \frac{A_c}{A_e} = \frac{q}{h_{fg}}, \tag{9}$$

where $\dot{m}$ is phase change mass, A is interface area, h is enthalpy, u is vapor velocity, q is heat flux, h_{fg} is latent heat, g is gravity acceleration, z is relative height of interface. Subscript e and c represent evaporation and condensation interfaces respectively.

As energy loss of vapor can be expressed by exergy loss, eq. (8) is rewritten as

$$\Delta E_x = \dot{m}_e A_e(h_e - T_0 s_e + 1/2u_e^2 + gz_e) - \dot{m}_c A_c(h_c - T_0 s_c + 1/2u_c^2 + gz_c). \tag{10}$$

Mass equilibrium equation between the two interfaces is $\rho_{v,e} u_e A_e = \rho_{v,c} u_c A_c$, and vapor density changes in accord with temperature. If gravity is neglected, eq. (10) can be rewritten as

$$\Delta E_x = \dot{m} A_e \begin{bmatrix} (h_e - h_c) - T_0(s_e - s_c) \\ + \dfrac{\dot{m}^2}{2\rho_{v,e}^2}\left(1 - R_A^2\left(\dfrac{\rho_{v,e}}{\rho_{v,c}}\right)^2\right) \end{bmatrix}, \tag{11}$$

where ρ is density, $R_A = A_e / A_c$ is area ratio, s is entropy. Pressure head between the two interfaces will be used to overcome flow resistance of vapor based on the assumption that vapor flow between the two interfaces is adiabatic.

According to physical formula $\Delta E = \Delta p \cdot \rho u A$, the driving force of vapor can be expressed as

$$\Delta p_{dr,v} = p_v - p_c = \rho_{v,e}\left[(h_e - h_c) - T_0(s_e - s_c)\right] + \frac{\dot{m}^2}{2\rho_{v,e}}\left(1 - R_A^2\left(\frac{\rho_{v,e}}{\rho_{v,c}}\right)^2\right). \tag{12}$$

Combining eqs. (3), (6) and (12), we have

$$\Delta p_{\text{total}} = p_v - p_l = \frac{2\sigma}{r_e} + \rho_{v,e}\left[(h_e - h_c) - T_0(s_e - s_c)\right] + \frac{\dot{m}^2}{2\rho_{v,e}}\left(1 - R_A^2\left(\frac{\rho_{v,e}}{\rho_{v,c}}\right)^2\right). \tag{13}$$

If area ratio R_A = 1, eq. (13) is predigested as

$$\Delta p_{\text{total}} = p_v - p_l = \frac{2\sigma}{r_e} + \rho_{v,e}\left[(h_e - h_c) - T_0(s_e - s_c)\right] + \frac{\dot{m}^2}{2\rho_{v,e}}\left(1 - \left(\frac{\rho_{v,e}}{\rho_{v,c}}\right)^2\right). \tag{14}$$

As entropy difference and momentum variety can be neglected because of their smaller magnitudes, eq. (14) can be rewritten as

$$\Delta p_{\text{total}} = \frac{2\sigma}{r_e} + \rho_{v,e}(h_e - h_c). \tag{15}$$

Thus vapor driving head is

$$\Delta p_{dr,v} = p_v - p_c = \rho_{v,e}\Delta h. \quad (16)$$

Although eq. (14) fully shows the total driving force of heat pipe with capillary wick, which includeS capillary force and phase change force, eq. (15) provides enough precision in actual calculation.

If vapor density is constant: $\rho_{v,e} = \rho_{v,c} = \rho_v$, eq. (16) can be rewritten as

$$\Delta p_{dr,v} = \rho_v (h_e - h_c) = \rho_v \Delta h \,. \quad (17)$$

Enthalpy difference in Eq.(17) is given as

$$\Delta h = \int_c^e c_{p0} \mathrm{d}T - (h_{r,e} - h_{r,c}) \,, \quad (18)$$

where c_{p0} is specific heat, and can be expressed as function of temperature:

$$c_{po} = c_0 + c_1 T + c_2 T^2 + c_3 T^3 \,. \quad (19)$$

In eq. (18), surplus enthalpy h_r is given as

$$h_r = a_r + T s_r + (p_0 v_0 - p v) \,, \quad (20)$$

where pressure is given as

$$p = \frac{RT}{v-b} - \frac{a}{T^{0.5} v (v+b)} \,, \quad (21)$$

and surplus free energy a_r and surplus entropy s_r are given as

$$a_r = RT \ln \frac{v-b}{v} + \frac{a}{T^{0.5} b} \ln \frac{v+b}{v} + RT \ln \frac{v}{v_0} \,, \quad (22)$$

and

$$s_r = -R \ln \frac{v-b}{v} + \frac{a}{2bT^{1.5}} \ln \frac{v+b}{v} - R \ln \frac{v}{v_0} \,, \quad (23)$$

where constants a, b and c can be obtained from ref. [10].

From eqs. (18)—(23), we can calculate the enthalpy difference of practical vapor. Then, from eq. (17), we can find driving pressure head formed by phase change in the heat pipe with porous wick. Generally speaking, theoretical driving force of phase change will be a little larger than the practical value of vapor pressure drop because vapor flow in heat pipe is not adiabatic, i.e.

$$\Delta p_{dr,v} > \Delta p_v \,.$$

3 Heat transfer limitation of heat pipe with porous wick

As we know, there exists a capillary limitation for heat pipe with porous wick. Then a working heat flux limitation $Q_{\max}$ can be defined correspondingly, and factor of heat transfer capability for heat pipe with porous wick is expressed as[11]

$$Q_{\max} l_{\text{eff}} = \frac{\dfrac{2\sigma}{r_e} - \rho_l g (l \sin\phi + d\cos\phi)}{\dfrac{\mu_l}{K \rho_l h_{fg} A_w} + \dfrac{32\mu_v}{\rho_v h_{fg} A_v d_v^2}} \,, \quad (24)$$

where μ is viscosity coefficient, A_w is flow cross section area of porous wick, A_v is flow cross section area of vapor, d_v is vapor channel hydraulic diameter, K is permeability of porous wick, d is diameter of heat pipe, ϕ is angle between heat pipe and horizontal plane, l is length of heat pipe, l_{eff} is effective length of heat pipe: $l_{\text{eff}} = l_e/2 + l_a + l_c/2$. Subscripts l and v stand for liquid and vapor respectively, a means adiabatic.

According to driving pressure head eq. (12), factor of heat transfer capability for heat pipe with porous wick should be amended as follows:

$$Q_{\max} l_{\text{eff}} = \frac{\dfrac{2\sigma}{r_e} + \rho_{v,e}\left[(h_e - h_c) - T_0 (s_e - s_c)\right] + \dfrac{\dot{m}^2}{2\rho_{v,e}}\left(1 - R_A^2 \left(\dfrac{\rho_{v,e}}{\rho_{v,c}}\right)^2\right) - \rho_l g (l\sin\phi + d\cos\phi)}{\dfrac{\mu_l}{K\rho_l h_{fg} A_w} + \dfrac{32\mu_v}{\rho_v h_{fg} A_v d_v^2}} \,. \quad (25)$$

Eq. (25) may be a little bit complex. If ignoring variation of kinetic energy, density and entropy of vapor, we can have

$$Q_{\max} l_{\text{eff}} = \frac{\dfrac{2\sigma}{r_e} + \rho_v \Delta h - \rho_l g (l\sin\phi + d\cos\phi)}{\dfrac{\mu_l}{K\rho_l h_{fg} A_w} + \dfrac{32\mu_v}{\rho_v h_{fg} A_v d_v^2}} \,. \quad (26)$$

Eqs. (25) and (26) show that there exist both capillary driving and phase change driving mechanism in heat pipe with porous wick at the same time. Obviously, for a heat pipe with porous wick, the value of heat transfer capability factor calculated by eq. (26) will be higher than that calculated by eq. (24), because a new driving force $\rho_v \Delta h$ is introduced.

4 An example of capillary wick heat pipe

Based on formulas of pressure drop[1], a thermal calculation was carried out for capillary wick heat pipe shown in Figure 1, and calculating results are shown in Table 1. Working medium in the heat pipe is water, and its temperature is 80℃. Related parameters are chosen as: lengths of evaporation section and condensation section $l_e = l_c = 80$ mm, length of insulation section $l_a = 40$ mm, wall thickness $\delta = 1$ mm, heat pipe inner radius $r_w = 4.5$ mm, vapor channel inner radius $r_v = 1.5$ mm, porous wick wire mesh $N = 7.87\times10^3$ m^{-1}, wire diameter $d = 6.25\times10^{-5}$ m, wire porosity $\varepsilon = 1 - \pi SNd/4 = 0.6$, and constant S is set as 1.05, wire mesh permeability $K = d^2\varepsilon^3/122(1-\varepsilon)^2 = 4.09\times10^{-11}$ m^2, liquid contact angle of condensation interface $\theta = 80°$.

According to Darcy's Law, liquid flow resistance in capillary wick is

$$\Delta p_l = \frac{\mu_l \dot{m}_l l_{\text{eff}}}{\rho_l K A_w}, \quad (27)$$

and vapor flow resistance is

$$\Delta p_v = \frac{4\mu_v l Q}{\pi \rho_v r_v^4 h_{fg}}, \quad (28)$$

where total heat load of heat pipe is $Q = qA_e$.

From the data in Table 1, when heat flux $q = 3$ W/cm^2, maximum capillary pressure head provided by the capillary wick in evaporation section is $\Delta p_{\text{capillary}} = 1956$ Pa (not considering variation of surface tension coefficient with temperature). Real value of capillary pressure head provided by the capillary wick is $\Delta p_{dr,l} \geqslant \Delta p_l + \Delta p_{eq,l} = 1069$ Pa. Phase change driving pressure head is $\Delta p_{dv,l} \geqslant \Delta p_v = 87$ Pa. The results in Table 1 show that for parameters used in this paper, vapor flow resistance is about one-eighth of liquid flow resistance in the capillary wick heat pipe. In general, if more heat is transferred by heat pipe, length of heat pipe is relatively longer, and diameter of vapor channel is less, vapor flow resistance will be larger. Meanwhile, phase change driving force should also be enlarged. Therefore, estimation of phase change driving force in heat pipe with capillary wick has great significance in the design of heat pipe.

Table 1 Capillary force and pressure drop of heat pipe with capillary wick

q (W/cm^2)	$\Delta p_{\text{capillary}}$ (Pa)	$\Delta p_{eq,l}$ (Pa)	Δp_l (Pa)	Δp_v (Pa)	Δp_{total} (Pa)
1	1956	340	243	29	612
2	1956	340	486	58	884
3	1956	340	729	87	1156

5 Conclusions

A method of heat-electricity analogy is used to obtain a simulating circuit of driving force and resistance drop for heat pipe with porous wick. A new driving mechanism related to phase change is presented, and a corresponding mathematical model is developed. The results indicate that although capillary interface is a vapor-liquid interface, capillary force formed on evaporation interface is just for driving liquid flow in capillary wick, and balancing a reverse capillary force formed on condensing interface. Therefore, for heat pipe with porous wick, the total driving force for working medium circulation includes phase change driving force caused by heat effect except capillary driving force, which represents driving mechanism for vapor flow. Compared with conventional theory model, mathematical model presented in this paper reveals operating mechanism of heat pipe with porous wick objectively, which thereby leads to a more precise calculation for thermal process and engineering design.

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