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Theoretical and experimental research on heat transfer performance of the semi-open heat pipe

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Abstract: This paper focuses on the heat transfer performance of semi-open heat pipe which is a new type of heat pipe. After analyzing its condensation heat transfer mechanisms theoretically, several semi-open heat pipes in different length ratios and upper hole diameters are studied experimentally and compared with the same dimensions closed heat pipes. Experimental results show that the heat transfer performance of semi-open heat pipe becomes better by increasing heat transfer rate. At the first transitional point, the heat transfer performance of semi-open heat pipe approaches the level of the closed heat pipe. It is suitable to choose upper small hole about 1 mm in diameter and length ratio larger than 0.6 for the semi-open heat pipe.

Key words: Semi-open heat pipe, Heat transfer performance, Closed heat pipe, First transitional point
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INTRODUCTION

Heat pipe is regarded as the superconductive heat transfer component. It is researched and widely applied to many fields in the world. After finishing some experiments, Esen and Esen (2005) brought the closed heat pipe in solar water heater. An aeration-thermosyphon heat pipe for controlling the temperature in a bulk silo was also developed (Dussadee *et al*., 2007). At present, many researchers (Monde *et al*., 1996; Li *et al*., 2003) have done many experiments on several types of heat pipes. Birkholzer (2006) proposed a simple temperature-profile method that uses high resolution temperature data for deriving the differences in the gradients inside and outside heat-pipe regions. Liu *et al*. (2006) investigated the effects of the evaporator length, vapor temperature, and power throughput on the critical values of the upper and lower boundaries in the looped separate heat pipe. Simulation results show that the length of the evaporator makes almost no influence on the upper boundary, but great effect on the lower boundary. Ren *et al*.(2007) developed a 2D mathematical model of the loop heat pipes. In this model, they simulated the flow and evaporation of interface and explained the auto-driving mechanism of inverted meniscus type evaporators. Hussein (2007) investigated the wickless heat pipes flat plate solar collector theoretically and experimentally. Farsi *et al*.(2003) have also done some researches in the transient heat transfer performance of the closed heat pipe experimentally and theoretically. But they were aimed only at the performance of closed heat pipe or open heat pipe. A new type of heat pipe which is invented by us, the semi-open heat pipe, has the advantages of lower cost and more security in operating compared with the closed heat pipe. Although there were some researches (Tu and Hong, 1989; Mo and Li, 2006) on the characteristics and applications of the semi-open heat pipe, many other performances were not been found. Schmidt (1951) investigated open heat pipe with evaporation section only and showed that its heat transfer performance could approach the level of the closed heat pipe. But the analytical paper of such heat pipe with condensation sections is rarely seen until now. To our knowledge, no research has been re-

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ported on the heat transfer performance of semi-open heat pipe with both evaporation and condensation sections.

In this paper, a series of experiments are made for the heat transfer of semi-open heat pipe with both evaporation and condensation sections. Comparing with the theoretical analyses, experimental results are satisfactory and a series of important conclusions are obtained.

ANALYSIS

The semi-open heat pipe is a heat pipe with an upper small hole which connects to the outside water seal (as shown in Fig.1). Its working principle is also similar to that of the closed heat pipe at normal working conditions except that its inter-pressure depends on the outside pressure (Zhu *et al.*, 2001).

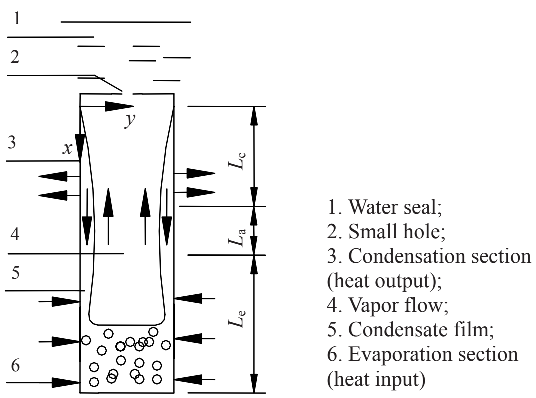


Fig.1 A semi-open heat pipe
L_c, L_a, L_e are lengths of condensation section, adiabatic section and evaporation section, respectively

The working state inside the condensation section of semi-open heat pipe is different from the closed heat pipe due to the existence of the first transitional point. The heat transfer process of the condensation section in the closed heat pipe is just film condensation. However, the process is variable with working conditions in the semi-open heat pipe.

The inner heat transfer area is increasing when the working state comes to near the first transitional point, where the whole surface of the condensation section is heat transfer area. After passing the first transitional point, direct contact bubble condensation enters the heat transfer process in addition. Nevertheless, film condensation is the major heat transfer means in the semi-open heat pipe as show in Fig.1. Thus direct contact bubble condensation is neglected here.

The effect of non-condensable gas is neglected since the non-condensable gas generated in operation can exit via upper hole. The curvature effect of the pipe wall also can be ignored as the film thickness is much smaller than the pipe radius. The governing equations are

$$\rho_l\left(u\frac{\partial u}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial x}+(\rho_l-\rho_v)g+\mu_l\frac{\partial^2 u}{\partial y^2}, \quad (1)$$

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0, \quad (2)$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=a\frac{\partial^2 T}{\partial y^2}, \quad (3)$$

where ρ is density, u is x-component velocity, v is y-component velocity, p is pressure, subscripts 'l' and 'v' represent liquid and vapor, μ_l is liquid viscosity, T is temperature, a is thermal diffusivity coefficient, g is acceleration of gravity.

Assume that the thermal properties of vapor and condensate are constant and the condensate flow is laminar with smooth surface. The fluid inertia and energy convection of the film can be neglected. The effects of film super-cooling and vapor shear stress are considered. Hence the governing equations become the momentum and energy equations:

$$0=\mu_l\frac{d^2u}{dy^2}-\frac{dp}{dx}+(\rho_l-\rho_v)g, \quad (4)$$

$$0=\frac{d^2T}{dy^2}. \quad (5)$$

The boundary conditions are

$$u=0,\ T=T_w,\ \text{at } y=0, \quad (6)$$

$$\mu_l\frac{du}{dy}=-\tau_v,\ T=T_v,\ \lambda_l\frac{dT}{dy}=\dot{\Gamma}h'_{fg},\ \text{at } y=\delta, \quad (7)$$

where τ_v is shear stress at the condensate film and vapor interface, λ_l is liquid thermal conductivity, $\dot{\Gamma}$ is condensation coefficient, h'_{fg} is latent heat of vaporization with super-cooling, δ is film thickness and T_w is wall temperature. $\dot{\Gamma}$, h'_{fg} and τ_v are given as

$$\dot{\Gamma}=\frac{\mathrm{d}\Gamma}{\mathrm{d}y}=\frac{\mathrm{d}}{\mathrm{d}x}\int_0^{\delta}\rho_{\mathrm{l}}u\mathrm{d}y, \quad (8)$$

$$h'_{\mathrm{fg}}=h_{\mathrm{fg}}+3C_{\mathrm{p}}\Delta T/8, \quad (9)$$

$$\tau_{\mathrm{v}}=\tau_{\mathrm{f}}+\tau_{\mathrm{l}}=(1+1400F)\rho_{\mathrm{v}}u_{\mathrm{v}}^2C_{\mathrm{f}}/2+m(u_{\mathrm{v}}-u_{\mathrm{l}}), \quad (10)$$

where C_f is friction coefficient, h_{fg} is latent heat of vaporization, C_p is specific heat, F is modification factor of Fanning friction coefficient and m is mass flow rate.

Introducing some parameters:

$$G=1+\frac{1}{(\rho_{\mathrm{l}}-\rho_{\mathrm{v}})}\frac{\mathrm{d}p}{\mathrm{d}x},\quad \beta=\frac{(\rho_{\mathrm{l}}-\rho_{\mathrm{v}})}{\rho_{\mathrm{l}}},$$
$$g^*=gG\beta,\qquad \Delta T=T_{\mathrm{s}}-T_{\mathrm{w}},$$

where G is parameter of pressure gradient, β is parameter of liquid vapor density, g^* is modified gravity, ΔT is temperature difference, T_s is the saturated temperature.

The velocity and temperature distributions of condensate layer can be obtained from Eqs.(4) and (5):

$$u=\rho_{\mathrm{l}}g^*(\delta y-y^2/2)/\mu_{\mathrm{l}}-\tau_{\mathrm{v}}y/\mu_{\mathrm{l}}, \quad (11)$$

$$T=\Delta Ty/\delta+T_{\mathrm{w}}. \quad (12)$$

Let

$$f=\delta/R,\ \psi=x/L_{\mathrm{c}},\ B=3/(2\rho_{\mathrm{l}}g^*R),$$
$$E=\lambda_{\mathrm{l}}\Delta T(\mu_{\mathrm{l}}h'_{\mathrm{fg}}),\ A=R^3g^*/(3v_{\mathrm{l}}^2E),$$
$$\xi=1-B\tau_{\mathrm{v}}/f=1-3\tau_{\mathrm{v}}/(2\rho_{\mathrm{l}}g^*\delta),$$

where f is dimensionless condensate film thickness, δ is condensate film thickness, ψ is the length ratio, R is radius, A and B are two intermediate parameters, ξ is vapor shear effect coefficient and E is condensation factor.

The meaning of the vapor shear effect coefficient ξ is that the condensate layer has no effect of vapor shear stress when ξ=1, and ξ<1 means that the vapor shear affects condensate flow when condensate layer flows are countercurrent.

The distribution of condensate layer thickness can be obtained:

$$\psi=\frac{AR}{12L_{\mathrm{c}}}f^4(1+8\xi),\quad f=\left[\frac{12L_{\mathrm{c}}\psi}{AR(1+8\xi)}\right]^{1/4}. \quad (13)$$

From energy conservation equations, we obtain

$$a_x\Delta T\mathrm{d}x=\lambda_{\mathrm{l}}\frac{\Delta T}{\delta}\mathrm{d}x=\dot{\Gamma}h'_{\mathrm{fg}}. \quad (14)$$

The local heat transfer coefficient of condensation is:

$$a_x=\{[g^*\rho_{\mathrm{l}}^2\lambda_{\mathrm{l}}^3h'_{\mathrm{fg}}(1+8\xi)]/(36\mu_{\mathrm{l}}\Delta Tx)\}^{1/4}. \quad (15)$$

The mean heat transfer coefficient in length of 0→x:

$$\overline{a}_x=f^3\xi A\lambda_{\mathrm{l}}/(L_{\mathrm{c}}\psi). \quad (16)$$

Let hydraulic diameter d_e=4Rf, thus the mean Nusselt number for condensation section is

$$\overline{Nu_x}=\overline{a}_xd_{\mathrm{e}}/\lambda_{\mathrm{l}}=4RAf^4\xi/(L_{\mathrm{c}}\psi). \quad (17)$$

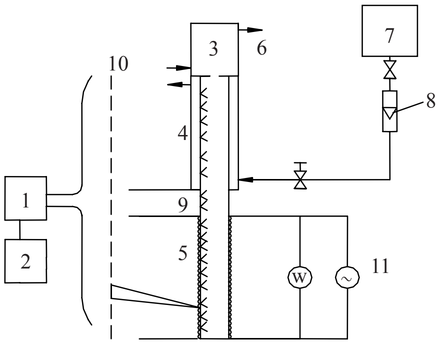
EXPERIMENTS

The test semi-open heat pipes were made of copper tubes and used water as working fluid. The dimensions of test semi-open heat pipes are shown in Table 1. A series of closed heat pipes in the same dimensions as the semi-open heat pipes but without the upper hole were tested too.

18~20 pairs of thermocouples were chosen for measuring the temperature of pipe wall and cooling water for each of test semi-open heat pipe and closed heat pipe. Cooling water is supplied by a high level tank and measured by a rotameter. An electrical furnace is used for heating. The schematic diagram of test apparatus for both semi-open and closed heat pipes is shown in Fig.2.

Table 1 The dimensions of test semi-open heat pipes

Tube	Outer diameter (mm)	Length ratio of condensation and evaporation sections L_r	Upper hole diameter Φ (mm)
A	25	0.74	0.7
C	25	0.54	0.7
B	25	0.67	0.7, 1.5 or 2.1
D	25	0.49	0.7, 1.5 or 2.1



1: digital voltmeter; 2: computer; 3: water seal; 4: condenser; 5: evaporator; 6: water jacket; 7: high level tank; 8: rotameter; 9: adiabatic section; 10: thermocouples; 11: constant-voltage power supply

Fig.2 Schematic diagram of experimental apparatus

The heat transfer performance of semi-open heat pipe is indicated by overall thermal resistance R_t and compared with the closed heat pipe. The experimental data for the test semi-open heat pipes A~D are shown in Fig.3. From experiments, it is clear that the heat transfer performance of semi-open heat pipe becomes better by increasing heat transfer rate in a certain scale. The optimal operating state is at the first transitional point and near it. The heat transfer performance of the closed heat pipe changes little with the variation of heat flux.

The upper small hole diameter has few effects on heat transfer performance due to capillary force, but the effect can be ignored when the diameter is small. So, it is better to choose upper small hole diameter as small as possible, i.e., 1 mm or less is suitable for the dimensions of our experimental samples.

The heat transfer performance of semi-open heat pipe in optimal working state could be better than that of the closed heat pipe if the length ratio L_r is large. From Fig.3, it is apparently to find that overall thermal resistances of tubes A (L_r=0.74) and B (L_r=0.67) decrease about 30%~50% compared with tubes C (L_r=0.54) and D (L_r=0.49). And tubes A and B also have smaller overall thermal resistances R_t than those of closed heat pipes with the same dimensions.

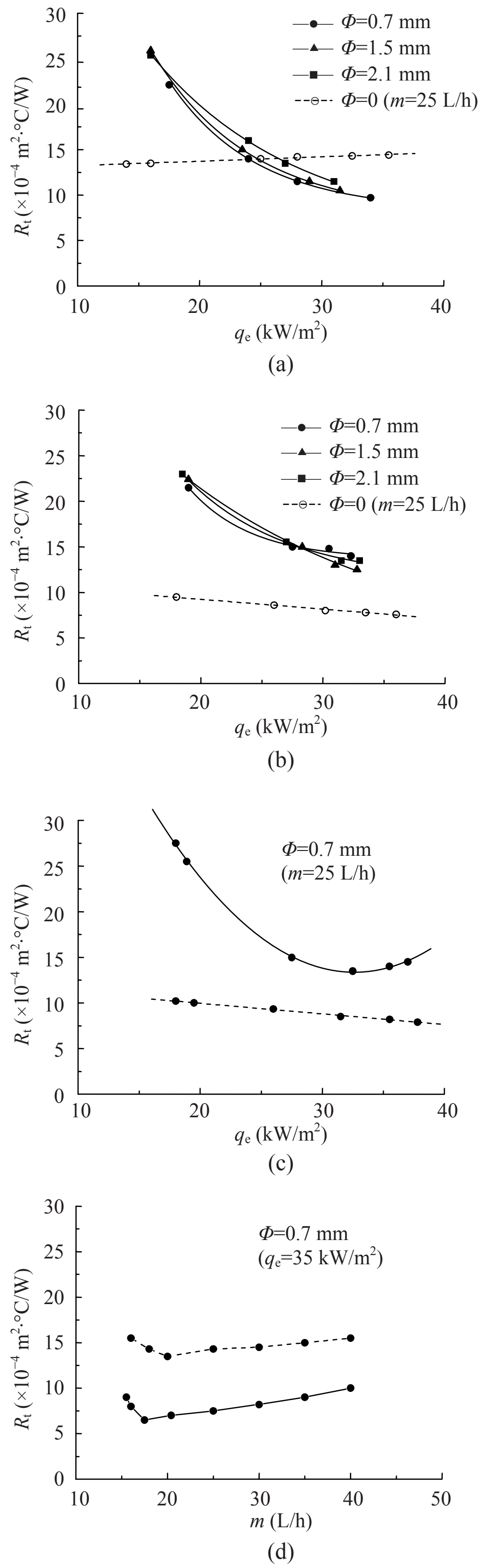


Fig.3 The heat transfer performances of the semi-open heat pipe. (a) Tube B; (b) Tube D; (c) Tube C; (d) Tube A

— Semi-open heat pipe - - - - Closed heat pipe

COMPARISON AND DISCUSSION

The results of theoretical analysis and experiments are compared in Figs.4 and 5. The value of theoretical results is smaller than that of Nusselt solution (ξ=1) and agrees with the experimental data better. The forty experimental data in Fig.5 are chosen

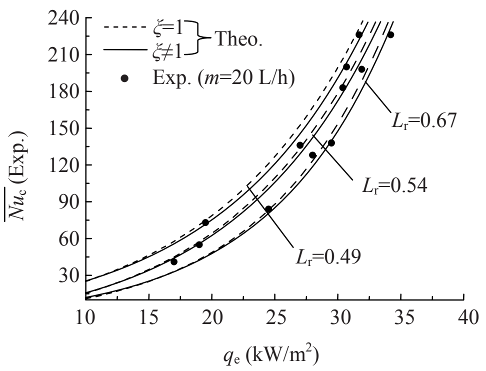


Fig.4 Comparison of theoretical Nusselt and experimental data

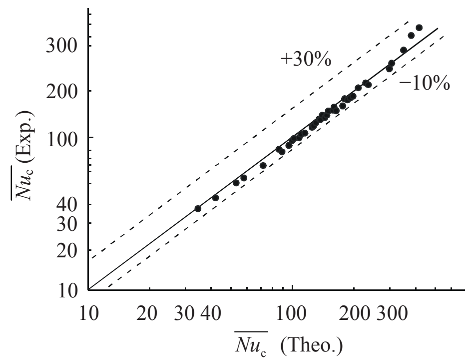


Fig.5 Comparison of theoretical solutions and experimental results

at random. When the mean Nusselt number for condensation section is less than or equal to 300 ($\overline{Nu_c} \leq 300$), the error is from 0 to –10% because of the existence of heat loss. When $\overline{Nu_c} > 300$, the error is from 0 to 30%. The value of experimental data is bigger than theoretical calculation value that always happens at small condensation factor ($E \leq 0.0053$). The bubble direct contact heat transfer may be the reason. It needs further study. In all, such results are satisfactory.

CONCLUSION

(1) Semi-open heat pipe is the kind of heat pipe which has a bright future. It has the advantages of low cost, reliable performance and security. It is widely applied in industry and operating successfully.

(2) It is important to consider the effect of vapor shear stress in analysis. Among two factors causing vapor shear stress, friction and condensation, the latter is the dominant factor.

(3) The heat transfer performance of semi-open heat pipe becomes better by increasing heat transfer rate. There is an optimal working state, the first transitional point, at which the heat transfer performance of semi-open heat pipe approaches the level of the closed heat pipe.

(4) The length ratio affects the heat transfer performance while the effect of upper small hole is small. It is suitable to choose upper small hole about 1 mm in diameter and length ratio larger than 0.6 for semi-open heat pipe.

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