Transactions of the Korean Nuclear Society Autumn Meeting
Pyeongchang, Korea, October 30-31, 2014

Evaluating Program about Performance of Circular Sodium Heat Pipe

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1. Introduction

The heat pipe is a device of very high thermal conductance. Nowadays heat pipes are used for spacecraft thermal control, cooling of electronic gadgets and in ordinary commercial thermal devices. The superior heat transfer capability, structural simplicity, relatively inexpensive, insensitivity to the gravitational field, silence and reliability are some of its outstanding features.

We study about heat transfer equation of heat pipe and program predicting performance which is considering geometrical shape of heat pipe by the related heat transfer equation of heat pipe. The operating temperature is 450℃ ~ 950℃, working fluid is sodium, material for container is stainless steel, and type of wick is sintered metal.

2. Theoretical Background

2.1 Temperature dependence of surface tension

Surface tension decreases with increasing temperature, it is therefore important to take temperature effects into account when using the results of measurements at typical ambient temperatures. Bohdansky and Schins [1] have derived an equation that applies to the alkali metals. While Fink and Leibowitz [2] recommended equation (1)

$$\sigma_l = \sigma_0 (1 - \frac{T}{T_c})^n \qquad (1)$$

For sodium, where σ_0=240.5mN/m, n=1.126, T_c=2503.7K.

2.2 Gravitational head

The pressure difference, ΔP_g, due to the hydrostatic head of liquid may be positive, negative, or zero, depending on the relative positions of the condenser and evaporator. The pressure difference may be determined.

$$\Delta P_g = \rho_l g L \sin\Phi \qquad (2)$$

2.3 Capillary Pressure

The pressure drop across a curved liquid interface is

$$\Delta P = \frac{2\sigma_l}{R} \qquad (3)$$

We can see that $R\cos\theta = r$. Hence, the capillary head at the evaporator $\Delta P_e'$ is

$$\Delta P_e' = 2\sigma_l \frac{\cos\theta_e}{r_e} \qquad (4)$$

Similarly, for the condenser

$$\Delta P_c' = 2\sigma_l \frac{\cos\theta_c}{r_c} \qquad (5)$$

And the capillary driving pressure, ΔP_c, is given by $\Delta P_e' - \Delta P_c'$.

2.4 Homogeneous wicks

If ε is the fractional void of the wick, that is the fraction of the cross section available for the fluid flow, then the total flow cross-sectional area A_f is given by

$$A_f = A_w \varepsilon = \pi(r_w^2 - r_v^2)\varepsilon \qquad (6)$$

If r_c is the effective pore radius, then the Hagen-Poiseuille equation may be written

$$\dot{m} = \frac{\pi(r_w^2 - r_v^2)\varepsilon r_c^2 \rho_l}{8\mu_l} \frac{\Delta P_l}{L_{eff}} \qquad (7)$$

Relating the heat and mass flows, $\dot{Q} = \dot{m}\Lambda$ rearranging

$$\Delta P_l = \frac{8\mu_l \dot{Q} L_{eff}}{\pi(r_w^2 - r_v^2)\varepsilon r_c^2 \rho_l \Lambda} \qquad (8)$$

3. Analysis

Upper limits to the heat transport capability of a heat pipe may be set by one or more factors. These limits were illustrated in Fig. 1.

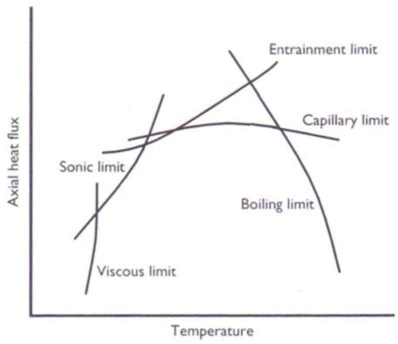

Fig. 1 Limitations to heat transport in a heat pipe.

3.1 Viscous Limit

At low temperature, viscous forces are dominant in the vapor flow down the pipe. Busse has shown that the axial heat flux increases as the pressure in the condenser is reduced, the maximum heat flux occurring when the pressure is reduced to zero. Busse carried out a two-dimensional analysis, finding that the radial velocity component had a significant effect, he derived the following equation.

$$\dot{q} = \frac{r_{\mathrm{v}} \wedge \rho_{\mathrm{v}} P_{\mathrm{v}}}{16 \mu_{\mathrm{v}} L_{\mathrm{eff}}} \tag{9}$$

This equation agrees well with the published data [3]. Because the vapor pressure difference naturally increases as the heat transported by the heat pipe rises, the constraint on this pressure difference thus necessitates that Q/A is limited. The limit is generally only of importance in some units during start-up. In other word, it doesn't have to be considered in performance calculation program.

3.2 Sonic Limit

At a somewhat higher temperature choking at the evaporator exit may limit the total power handling capability of the pipe because the situation, not increasing mass flow rate due to compressibility after Ma = 1. The sonic limit is given by

$$\mathrm{Q} = \rho_{\mathrm{v}} A c \wedge \tag{10}$$

3.3 Capillary Pressure

In a heat pipe, the vapor flows from the evaporator to the condenser and the liquid is returned by the wick structure. At the interface between the wick surface and the vapor, the latter will exert a shear force on the liquid in the wick. The magnitude of the shear force will depend on the vapor properties and velocity and its action will be to entrain droplets of liquid and transport them to the condenser end. This tendency to entrain is resisted by the surface tension in the liquid.

The Weber number, We, which is representative of the ratio between inertial vapor forces and liquid surface tension force provides a convenient measure of probability of entrainment. The Weber number is defined [4]

$$\mathrm{We} = \frac{\rho_{\mathrm{v}} v^2 z}{2\pi\sigma_l} \tag{11}$$

It may be assumed that entrainment may occur when We is of the order 1. The limiting vapor velocity, $\mathrm{v_c}$, is thus given by

$$v_{\mathrm{c}} = \sqrt{\frac{2\pi\sigma_l}{\rho_{\mathrm{v}} z}} \tag{12}$$

And, relating the axial heat flux to the vapor velocity using

$$\dot{q} = \rho_{\mathrm{v}} \wedge v \tag{13}$$

The entrainment limited axial flux is given by

$$\dot{q} = \sqrt{\frac{2\pi\rho_{\mathrm{v}} \wedge^2 \sigma_l}{z}} \tag{14}$$

3.4 Capillary Limit

For the heat pipe to operate, equation (15) must be satisfied

$$\Delta P_{\mathrm{c,max}} = \Delta P_l + \Delta P_{\mathrm{v}} + \Delta P_g \tag{15}$$

If we don't neglect pressure drop due to vapor flow, we can calculate mass flow by capillary limit

$$\dot{m}_{max} = \frac{[\frac{2}{r_{\mathrm{e}}} - \frac{\rho_l g L}{\sigma_l} \sin\Phi]}{(\frac{\mu_l L_{\mathrm{eff}}}{\rho_l K A_w} + \frac{8\mu_{\mathrm{v}} L_{\mathrm{eff}}}{\rho_v \pi r_{\mathrm{v}}^4})} \tag{16}$$

While $K = \frac{\varepsilon r_c^2}{b}$ and b=8 by the Hagen-Poiseuille Equation in case of circular heat pipe.

3. Conclusions

As a result of evaluating program about performance of circular sodium heat pipe based on MATLAB code, express correlation between radius and LHR, correlation between heat transfer length and LHR, correlation between wick and LHR, correlation between operating temperature and LHR.

Generally radius values of heat pipe are proportional to LHR because of increase of mass flow which is main factor of heat flow.

Heat transfer length values of heat pipe are inversely proportional to LHR and slightly inversely proportional to heat rate.

Pore size is proportional to LHR. Although increase of pore size decrease capillary pressure, decrease more pressure drop in liquid phase. As a result, mass flow and heat rate are increase. But we have to do additional consideration about pore size and voidage in the aspect of safety and production technique.

In the graph of correlation between operating temperature and LHR, output has maximum point in 1150K. Over that point, operating temperature is inversely proportional to linear heat rate that means unstable.

REFERENCES

[1] J. Bohdansky, H.E.J. Schins, The temperature dependence of liquid metals, J. Chem. Phys.49, 1968.
[2] J.K. Fink, L. Leibowitz, Thermodynamic and transport properties of sodium liquid and vapor, ANL/RE-95/2, Argonne National Laboratory, January 1995.

[3] C.A. Busse, Theory of the ultimate heat transfer limit of cylindrical heat pipes, International Journal of Heat and Mass Transfer, Volume 16, Issue 1, January 1973, Pages 169-186.
[4] B.H. Kim, G.P. Peterson, Analysis of the critical Weber number at the onset of liquid entrainment in capillary-driven heat pipes, International Journal of Heat and Mass Transfer, Volume 38, Issue 8, May 1995, Pages 1427-1442.