Effects of NCG Charging Mass on the Operational Characteristics of Variable Conductance Heat Pipe

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Abstract: - Numerical analysis and experimental study are performed to investigate the effect of heat load and operating temperature on the thermal performance of several variable conductance heat pipe heat pipe (VCHP) with screen meshed wick. The heat pipe is designed in 200 screen meshes, 500 ㎜ length and 12.7 ㎜ outer diameter tube of copper, water (4.8 g) is used as working fluid and nitrogen as non-condensable gas. Heat pipe used in this study has evaporator, condenser and adiabatic section, respectively. Analysis values and experimental data of wall temperature distribution along axial length are presented for heat transport capacity, condenser cooling water temperature change, degrees of an inclination angle and operating temperature. These analysis and experiment give the follow findings: For the same charging mass of working fluid, the operating temperature of heat pipe becomes to be high with the increasing of charging mass of NCG. When the heat flux at the evaporator section increases, the vapor pressure in the pipe rises and consequently compresses the NCG to the condenser end part and increases the active length of the condenser. It is found out that the operating temperature can be effectively controlled and the analytical results are in good agreement with those of experiment.

Key-Words: - Mesh wick, working fluid charging mass, non-condensable gas (NCG), variable conductance heat pipe (VCHP)

1 Introduction

Of the various methods of transporting heat, the heat pipe is one of the most efficient device known today. The introduction of the heat pipe was first conceived by Gaugler(1944) of the General Motors Corporation in the U.S. Patent[1]. Heat applied to the evaporator section by an external source is conducted through the pipe wall and wick structure, where it vaporizes the working fluid. The resulting vapor pressure drives the vapor through the adiabatic section to the condenser, where the vapor condenses, releasing its latent heat of vaporization to the provided heat sink. The capillary pressure created by the menisci in the wick pumps the condensed fluid back to the evaporator section. Therefore, the heat pipe can continuously transport the latent heat of vaporization from the evaporator to the condenser section. In a conventional heat pipe, the operating temperature is determined by the heat source and heat sink conditions. Because of its nearly constant conductance, the conventional heat pipe is incapable of temperature control. The device designed for temperature control is called the variable conductance heat pipe (VCHP).

When the heat load at the source increases, the vapor pressure in the active portion of the pipe rises and consequently compresses the gas and increases the active length of the condenser. Increasing the active length of the condenser has the effect of reducing the interface thermal resistance at the condenser. Hence, the net effect of increasing the heat load is to reduce the interface thermal resistance at the condenser, which in turn counterbalances the increase in vapor temperature and heat transfer rate. As a result, the pipe wall temperature can be maintained at a certain temperature range in spite of the variation in heat load.

A survey of literatures reveals that Marcus[2] developed the model that analyzes the heat transport behavior of VCHP, and Chi[3] presented the modified model of VCHP in a steady-state operation. Bobco[4~5] has analyzed the thermal performance of VCHP from one-dimensional diffusion model, and the maximum heat transport capacity of VCHP. Kobayashi et al[6] and Sauciuc et al[7] studied analytically the thermal performance of thermosyphon with non condensable gas reservoir. Peterson[8] studied the phenomena of convection and diffusion of vapor and inert gas in a reservoir.

As a result, the thermal performance of VCHP based on diffusion model has been compared[9] with that of the conventional heat pipe, and the axial variation of pressure difference between vapor and liquid obtained. Additionally, it is found for the distribution of concentration of inert gas and temperature variation in the condenser region. The maximum heat transfer capacity of variable conductance heat pipe is provided by varying the operation temperature.

2 Modeling of VCHP

The maximum heat transport capability of a VCHP is investigated numerically to predict the fluid-thermal behavior of variable conductance heat pipes. Fig.1 depicts the schematic shape and cross sectional diagram of variable conductance heat pipe considered in the present work. For the steady-state operation of a variable conductance heat pipe, the conservation principle must hold over the cross section of the variable conductance heat pipe at any axial location for the continuity, momentum, and energy for the liquid and vapor. The Laplace-Young equation is additionally applied for the radius of curvature at the liquid-vapor interface. The mass conservation over the cross section of variable conductance heat pipe is given by

$$\rho_v A_v \overline{w}_v = \rho_l A_l \overline{w}_l \qquad (1)$$

where $\overline{w}_v$ and $\overline{w}_l$ are the mean vapor and liquid velocities in the axial direction, and $\dot{m}_l$ the mass flow rate of liquid.

The conservation of axial momentum equation for an incompressible vapor using the one-dimensional boundary-layer approximation is given by

$$\frac{d}{dx}(P_v + \rho_v g z \sin\phi + \rho_v \beta_v \overline{w}_v^2) = -f_v \frac{2\rho_v \overline{w}_v^2}{D_{h,v}} \qquad (2)$$

where β_v is the momentum flux coefficient, and f_v the friction coefficient. These coefficients for two-dimensional, laminar flow of channel can be

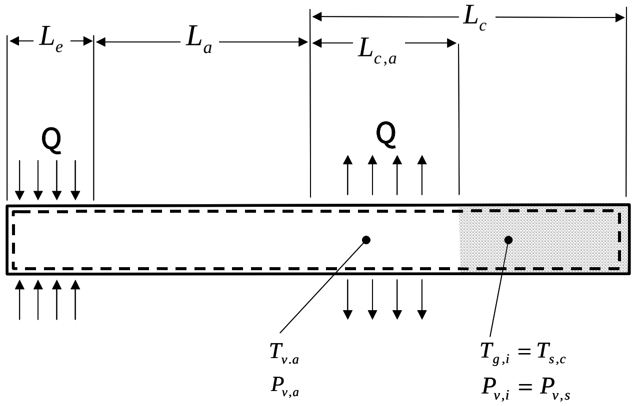

Fig.1. Schematic diagram of VCHP

taken from $(f\,\mathrm{Re})_v = 16$.

The conservation of energy equation for the liquid and vapor flows in a variable conductance heat pipe can be expressed in such a way that the mass flow rates of liquid and vapor varied along the axial direction by evaporation and condensation in the working fluid. The energy equation is as follows:

$$\frac{d}{dx}(\rho_l \overline{w}_l A_l) = \frac{1}{h_{fg}} \frac{dQ(x)}{dx} \qquad (3)$$

where $Q(z)$ denotes the local axial heat load function and h_{fg} the latent heat of the working fluid.

The interfacial radius of the meniscus curvature is related to the pressure difference between the vapor and liquid, and can be taken in differential pattern from the Laplace-Young equation as follows:

$$\frac{dP_l}{dx} = \frac{dP_v}{dx} - \frac{d}{dx}\left(\frac{\sigma}{r}\right) \qquad (4)$$

where σ denotes the surface tension coefficient for the liquid. For the above governing equations, the boundary condition for the liquid and vapor flows is

$$\overline{w}_l = \overline{w}_v = 0 \quad (x = 0 \text{ and } L_t) \qquad (5)$$

where L_t is the total length of VCHP. The boundary condition for the pressure of the liquid and vapor at the evaporator end cap is

$$P_v = P_{v0},\ P_l = P_{v0} - \frac{\sigma}{r_0} \quad (x = 0) \qquad (6)$$

In the VCHP, the heat and mass balance equation in the condenser port as follows:

$$\frac{d}{d}\left[k_p\pi(r_o^2-r_i^2)\frac{dT_p}{dx}\right]+\frac{2\pi r_i k_e}{r_i-r_v}(T_{wv}-T_p)-2\pi r_o h_{f,c}(T_p-T_s)=0$$

$$\frac{dm_v}{dx}=-\frac{2\pi r_i k_e}{\lambda(r_i-r_v)}(T_{wv}-T_p) \qquad (7)$$

where T_{wv} is the wick temperature and depends on the mass fraction of working fluid χ_g and the partial pressure of working fluid P_v.

The relation between the mass flow rate and mass fraction of working fluid can be given by

$$\dot{m}_v = A\rho D\frac{d}{dx}(\ln\chi_g) \qquad (9)$$

The total heat transfer rate conveyed by the VCHP can be obtained in the following equation:

$$Q=\int_0^{L_c}2\pi r_o h_{f,c}(T_p-T_s)dx \qquad (10)$$

The specification of variable conductance heat pipe mostly studied in the present work is listed in Table 1.

Table 1. Specification of variable conductance heat pipe used in this study

Description	Value
Container material	Copper
Pipe diameter(OD)	12.7 mm
Pipe diameter(ID)	11.1 mm
Total length	500 mm
Length of Condenser	100 mm
Length of Condenser	50 mm
Length of Condenser	350 mm
Working fluid	distilled water
Screen mesh number	200, 1 layer

3 Experiment

3.1 Experimental Apparatus

In this experiment, Apparatus consists mainly of three kinds of VCHP, data acquisition system and cooling system with constant temperature water bath. Heat pipes are vacuumed to 10^{-4} torr and then charged with the working fluid and NCG. Water as a working fluid is low-cost, non poisonous, high latent heat material and has good compatibility with copper. The charging quantity of working fluid is 4.8 g and calculated on condition that test pipe is thermosyphon. The non-condensable gas(NCG) is nitrogen gas and charged into the heat pipe with 1.0×10^{-6} kg (HP1), 3.4×10^{-6} kg (HP2) and 5.0×10^{-6} kg (HP3), respectively.

The evaporator section is heated by Nichrome heating coil surrounding this area and then insulated in 3 layers, which consists of Mica paper, Cerak wool and Urethane to minimize the heat loss to atmosphere. The condenser section is cooled by water to uniform the surface temperature through all the condenser section.

The wall temperatures of heat pipe are measured at several points from data acquisition system (2645A NetDAQ). All thermocouples are welded by graphite spark and then attached to the contact point of heat pipe wall. The thermal contact resistance is reduced by using the thermal grease at contact point. These thermocouples are allocated at every 25 mm interval and distributed in evaporator 3, adiabatic 1 and condenser 13.

The constant temperature water bath is used for condenser section cooling. This bath has thermostat to maintain the temperature of water with it. The bath temperature is controlled in the range of 276 K - 313 K. This coolant is circulated by pump which is installed in bath. Coolant water flow rate was measured using flow meter.

3.2 Experimental Methods

To test the performance of heat pipe, the amount of heat input and surface temperature should be measured at various conditions. The operating temperature of heat pipe is regarded as mean adiabatic temperature.

Because this study is focused on finding out the influence of NCG charging mass on the thermal characteristics of VCHP, we conducted the experiment in various conditions which are NCG charging pipe, heat input changing, cooling water temperature changing and inclination angle of heat pipe changing.

Heat input is gradually increased from 10 W and temperature is monitored. When the heat pipe reaches at the working limitation, the temperature of evaporator is rapidly increased, this point is dry-out. At that time we finished experiment to avoid deterioration of the heat pipe.

4 Results and Discussion

4.1 Wall Temperature Variation with Heat Load

Three kinds of heat pipe are installed at horizontal status and coolant temperature is 293 K. At this condition, the heat is imposed at step by step from 10 W to dry-out heat load.

Fig.2 shows the temperature distribution of NCG 1.0×10^{-6} kg heat pipe (HP1). When the heat load of 18.4 W is supplied to evaporator, surface mean temperature of evaporator section is 311.7 K and adiabatic temperature 310.4 K. When the heat load of 36.1 W is supplied to evaporator, surface mean temperature of evaporator section is 315.2 K and adiabatic temperature 311.5 K. Temperature deviation of evaporator is more than that of adiabatic section. This means the heat transfer is good. Heat load of 144.2 W caused heat pipe to dry-out.

Fig.3 shows the temperature distribution of NCG 3.4×10^{-6} kg heat pipe (HP2). Operating temperature for heat input of 18.4 W is made with 319 K and dry-out occurred at 135 W.

Fig.4 shows the temperature distribution of NCG 5.0×10^{-6} kg heat pipe (HP3). When the heat load of 18.1 W is supplied to evaporator, surface mean temperature of evaporator section is 335.4 K and adiabatic temperature 333 K. As the amount of heat input increases, the active length of condenser is increased. Compared with heat pipe of NCG 1.0×10^{-6} kg, the operating temperature increases to 22.6 K. As the NCG and heat input increase, the operating temperature is also increased.

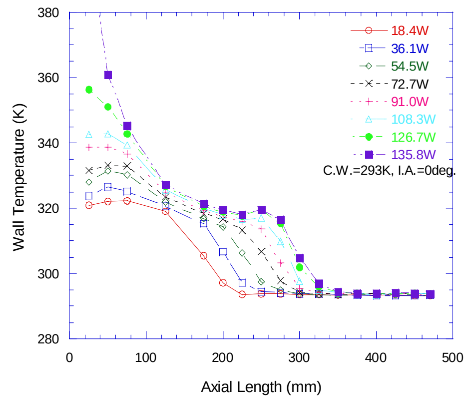

Fig.2. Wall temperature distribution with heat transport capacity (HP1)

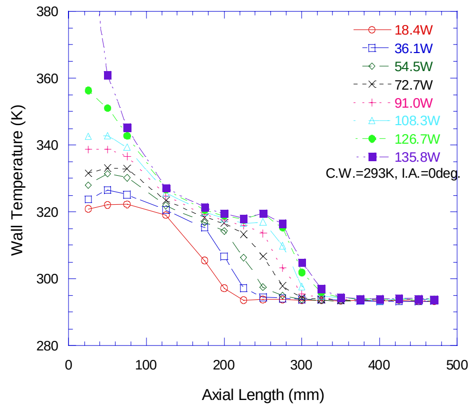

Fig.3. Wall temperature distribution with heat transport capacity (HP2)

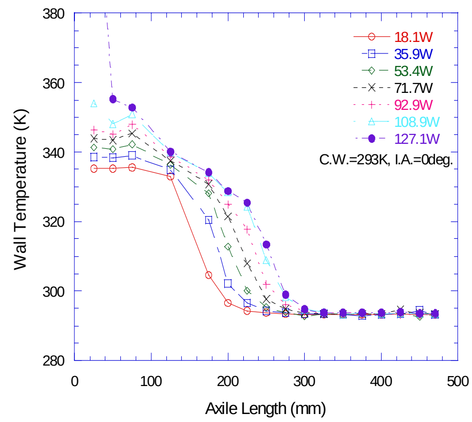

Fig.4. Wall temperature distribution with heat transport capacity (HP3)

4.2 Behavior of NCG Moving with Heat Load

Fig.5 shows measured temperature of HP1 which is installed at vertical status and coolant temperature 293 K. From this temperature distribution, we can estimate NCG behavior. At a low heat input of 9.3 W and 18.3 W, there is no NCG movement to condenser end side. At heat load of 36 W, the NCG begins to move toward condenser end side. As the amount of heat input increases, the active length of condenser apparently increases.

These phenomena show that as the heat load at the source increases, the vapor pressure in the active portion of the pipe rises and consequently compresses the gas and increases the active length of the condenser. The active length of the condenser has the effect of the interface thermal resistance at the condenser. As a result, the pipe wall temperature can be maintained at a certain temperature range in spite of the variation in heat load.

Additionally, at heat load 289.6 W, NCG is pushed to condenser end, the inactive portion of condenser section nearly disappeared and most of portion act as active portion to help heat rejection.

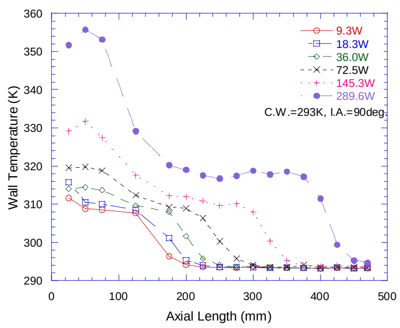

Fig.5. Axial distribution of wall temperature with HP1.

4.3 Wall Temperature Variation with Various Quantity of NCG

In this experiment, three kinds of heat pipe are used in vertical status and coolant temperature is maintained at 293 K. At this condition, the heat load of 110 W and 320 W is imposed on evaporator section and the surface wall temperature monitored along the axial length.

Fig.6 shows the wall temperature at heat input of 110 W. At NCG charging mass of 1.0×10^{-6} kg, surface mean temperature at evaporator section is 323.4 K and adiabatic temperature 314.5 K. At NCG 3.4×10^{-6} kg, surface mean temperature at evaporator section is 332.4 K and adiabatic temperature 324.3 K. At NCG 5.0×10^{-6} kg surface mean temperature at evaporator section is 347.7 K and adiabatic temperature 337.6 K. From above data, it is found out that NCG charging mass increases with the working temperature increasing.

These results from inactive condenser zone increased with NCG charging mass. As the charging mass increases, the operating temperature increases.

Fig.7 shows the wall temperature at heat input of 182 W, 256.2 W, 331 W. Comparison of the analyzed and experimental results for wall temperature with input power for HP2 shows that both are relatively coincided well.

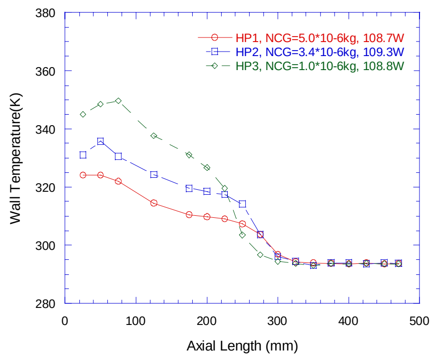

Fig.6. Wall temperature distribution with heat transport capacity at 110 W.

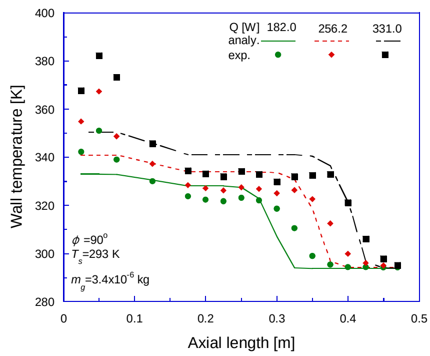

Fig.7. Comparison of the analyzed and experimental results for wall temperature with input power for HP2.

4.4 Comparison of the analytical and experimental results

The heat transport rate for each charging mass and operating temperature which is conducted in experiment and numerical analysis is presented in Fig.8. The CCHP has best heat transfer efficiency and as the amount of NCG charging mass increases, the heat transfer efficiency decreases. This resulted from , In case of NCG charging in heat pipe, the active length of condenser is shorten and then the heat emitting area decreased. In comparison of numerical and experimental results, the CCHP is mostly coincided among the above heat pipe. we assume that this result from the difference between actual NCG charging mass and theoretical NCG charging mass in manufacturing of VCHP, because of the difficulty of small amount charging of working fluid.

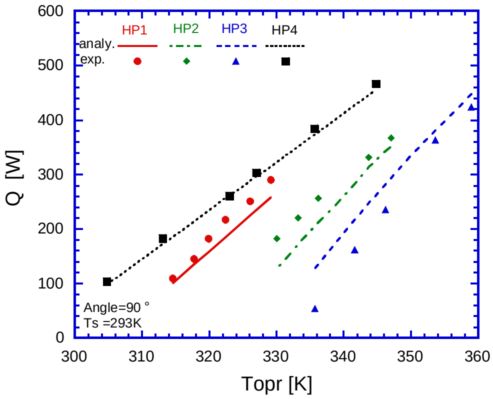

Fig.8. Comparison of the analyzed and experimental results for heat transfer rate with operating temperature in various heat pipes.

5 Conclusion

This work has been conducted to investigate experimentally the thermal performance of variable conductance heat pipe (VCHP) with screen mesh wick for thermal control of system. Water is used as a working fluid in the copper tube and nitrogen as a non-condensable gas. The results from this work can be summarized as follows:

(1) The operating temperature and wall temperature of heat pipe increases with the charging mass of NCG.
(2) It is found out that the wall temperature can be effectively controlled by using the VCHP which changes the active length of condenser with heat load.
(3) The operating temperature of heat pipe is lower in vertical position than that in horizontal position.
(4) The results from experiment is in good agreement with those of analysis

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