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Study on Heat Pipe*

(2nd Report, Heat Transfer Mechanism)
in the Evaporator Zone of Heat Pipe

By Shinzo SHIBAYAMA** and Shinichi MOROOKA***

The primary object of this study is to gain a better understanding of the heat transfer mechanism in the evaporator of a heat pipe.

To evaluate the effects of working pressure, working fluid, pipe inclination and noncondensable gas on the heat transfer mechanism, sintered powders were used as the wick, and pure water and R113 as the working fluid. Experimental results for the heat transfer coefficient are compared with predicted values based on the mechanism proposed by authors. The results indicate the mechanism to be substantially correct.

1. Introduction

Recently studies on a heat pipe(1)~(9) have been extensively conducted in various fields. The objective of this study is to gain a better understanding of the heat transfer mechanism in the evaporator. Although a number of past studies(5)~(9) have contributed to the knowledge in this area, the understanding is far from complete.

Ferrel et al.(5)(6) studied vaporization from wicks of monel and glass beads with water as the working fluid at atmospheric pressure and concluded that the heat transfer mechanism was conduction. Based on this mechanism, they solved the three-dimensional conduction problem of a bed of packed spheres and found the overall heat transfer coefficient. However, the generality of their analytical results is poor due to a narrow range of their applications. On the other hand, study using sintered powder as the wick was conducted by Alexander et al.,(9) who used a simplified model of a heat pipe. However, no obvious result has been obtained due to a narrow range of experimental conditions and use of the model heat pipe. Therefore, the authors tried to conduct an experimental study over a wider range of conditions, using an actual heat pipe (not a model heat pipe) and to develop a heat transfer model to predict the evaporator heat transfer coefficient. To evaluate the effects of the working pressure, the working fluid, the pipe inclination and noncondensable gas on the heat transfer mechanism, sintered metal powders were used as the wick, and pure water and R113 as the working fluid. Experimental results for the heat transfer

coefficient are compared with predicted values based on the mechanism proposed by authors. The results indicate the mechanism to be substantially correct.

2. Equipment and Procedures

Drawing of the apparatus and specifications of the heat pipes are shown in Fig. 1 and Table 1, respectively.

The heat pipe container studied was a 304 stainless steel seamless tube with 28 mm I.D. and 32 mm O.D. It is generally known that the surface condition and contamination of the wick are important factors influencing the performance and life of a heat pipe. Therefore, during assembly of a heat pipe, care was taken

- A : Overflow tank
- B : Preheating chamber
- C : Pipe heater
- D : Mixing chamber
- E : Cold junction
- F : Switch
- G : Self-registering thermometer
- H : Digital voltmeter
- I : Wattmeter
- J : Flow control valve
- K : Transformer
- L : Heat pipe

Fig. 1 Drawing of the apparatus

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** Professor Emeritus, Waseda University.

*** Research Fellow, Nuclear Engineering Laboratory, Toshiba Corporation, Kawasaki, Japan.

Table I Heat pipe specification

	pipe length	vapor space dia.	material	sintered metal powders		
				particle size range μ	average particle size μ	porosity
HP1	483	23.6	bronze	246 ~ 495	340	0.42
HP2	480	22.4	bronze	495 ~ 701	412	0.364
HP3	491	19.1	bronze	351 ~ 495	433	0.323
HP4	490	23.6	bronze	250 ~ 420	335	0.360
HP5	480	21.9	copper		495	0.461

in minimizing the contamination in a heat pipe, and the wick surface was oxidized. Both ends of the heat pipe were sealed off with flanges and a leak test was done. When the heat pipe was vacuum tight, the working fluid sufficiently deprived of air was fed into it. A cylindrical heater block, made of 5 cm long aluminum wound with electrical resistance wire, was used as the heat source and located at one end of the pipe and the input power to the heater was controlled by a transformer. A heat-resistant agent having a good thermal conductivity filled up the space between the outside of the pipe and the inside of the heater block in order to make a good contact and the outside of the heater block is heat insulated with glass wool. Heat was picked up by the cooling water passing through a jacket fitted to the opposite end. Temperatures of the pipe outer wall, inlet and outlet of the cooling water, and the vapor were detected by 0.3 mm - dia. copper-constantan thermocouples, recorded by a self-registering thermometer and measured by a digital voltmeter after confirming attainment of steady state.

Experiment was conducted holding the vapor temperature constant by regulating the inlet temperature and flow rate of the cooling water. Heat input was progressively increased after collecting the data.

3. Arrangement of experiment results

Heat transfer rate, evaporator heat flux and superheat have been calculated: [Heat transfer rate]

Heat transfer rate was calculated from heat picked up by the cooling water at the condenser.

$$Q_e = G_{\ell} C_{p\ell} \Delta t_c \quad (1)$$

Fig. 2 Typical axial temperature distribution

G_{ℓ} : cooling water flow rate kg/h

$C_{p\ell}$: cooling water specific heat kcal/°C

Δt_c : difference between inlet and outlet temperatures °C

[Evaporator heat flux]

Average heat flux based on the area of the evaporator inner wall A_1 was calculated from the equation

$$q_1 = Q_e / A_1$$

[Superheat]

The temperature of the inner heated surface of the evaporator was calculated from the temperature of the outer heated surface, and superheat ΔT_{sat} was defined as the difference between the inner heated surface and the vapor temperature.

4. Results and Discussion of Results

4.1 The effect of some factors on heat transfer

The data ① in Fig. 2 shows a typical plot of the temperature profile along the axis of the heat pipe. Temperature in the vapor space is constant for most of the experiments. The data ② shows an example of a rapid drop in the vapor temperature in the condenser due to the existence of noncondensable gas.

Figures 3 and 4 shows typical result about the effect of noncondensable gas and the pipe inclination angle on the heat flux-superheat relation (heat transfer pattern) and the heat transfer coefficient. Both are considered to have no sensible effect on the heat transfer of the evaporator in the range of this experimental condition.

Fig. 3 Effect of noncondensable gas on heat transfer data

Figures 5 and 6 show the effect of the working fluid and the pressure on the heat transfer pattern. In these figures, symbols U, L and A show results based on the upper wall temperature (heat flows downward), the bottom wall temperature (heat flows upward) and the average wall temperature of the evaporator, respectively. Results for R113 indicate that the pressure has a small effect on the heat transfer pattern and the heat transfer coefficient vs. the heat flux-superheat relation is linear below a certain heat flux. On the other hand, results for pure water indicate that the heat flux-superheat relation is considered to be linear. However, the pressure is considered to have more sensible effect on the heat transfer phenomena as compared with the results for R113. In this case, the heat transfer coefficient decreases as the pressure decreases and there appears a clear difference in the heat transfer pattern between the upper and bottom walls in a low pressure range. Figure 7 shows a result for HP5 and pure water in order to investigate the difference in the heat transfer pattern between the upper and bottom walls, together with a change in the evaporator wall temperature. In the low heat flux range, the heat transfer coefficient for the upper and bottom walls increases with heat flux. The heat flux-superheat relation for the

Fig. 4 Effect of pipe inclination angle on heat transfer data

Fig. 5 Evaporator heat transfer data for HP 5

(a) HP1

(b) HP2

(c) HP5

Fig. 6 Evaporator heat transfer data

upper wall is considered to be linear in the range of $2 \sim 6 \times 10^4$ kcal/m²h, but beyond this upper limit the heat transfer coefficient decreases as heat flux increases, and the operating limit which is considered due to pressure loss occurs at point E (about 1.3×10^5 kcal/m²h). On one hand, the heat flux-superheat relation for the bottom wall remains linear up to a high heat flux region. The heat transfer coefficient for the bottom wall is larger than for the upper wall. Confirmation experiment was conducted by rotating the pipe by 180° and it was made clear that the difference in sintered powder local conditions was not a cause for this. Judging from the fact mentioned above, this trend in the low pressure range would be expected because of the difference in the liquid-vapor interface recession caused by the difference in heat flow direction. No definite explanation for this difference is given at the present moment.

4.2 Heat transfer mechanism

As shown in Figs. 5, 6 and 7(a), the heat transfer pattern in the evaporator of the heat pipe with the sintered metal powder wick may be represented by the model as shown in Fig. 8 in the range of this experimental condition, in spite of

(a) Evaporator heat transfer data for HP5

(b) Change in the evaporator wall temperature

Fig. 7

the differences among limits of each region according to the experimental conditions.

The state of the wick at the evaporator in each region is known from this diagram and the heat transfer mechanism is estimated. Because the heat flux-superheat relation is linear in the region II of Fig. 8, the heat transfer mechanism in this region is considered to basically consist of the thermal resistance of conduction through the wick-liquid matrix and the evaporation resistance at the liquid-vapor interface, and the regions I and II are considered to be transitional regions from or to the region III. The region I where the heat transfer coefficient increases with heat flux corresponds to the states of (a) ~ (d) in Fig. 8, where the liquid layer in the evaporator wick becomes thinner.

At the beginning the state is (a). When power is supplied, evaporation occurs at the surface of the liquid which fills the wick completely. When the evaporation rate increases with input, mass balance between the liquid supply and the evaporation rate no longer holds and the part where the capillary pore diameter is large becomes dry as in (b). With further increase in heat input, the liquid-vapor interface retreats gradually and the wick ends up in the state (d). The heat transfer in the region II where the heat transfer coefficient is constant corresponds to this state.

In the region III where the heat transfer coefficient decreases with an increase of heat flux, the difference among the local surface temperatures becomes remarkable as shown in Fig. 7(b). From this fact the state of the region III is considered to be the state (e) in which the heated surface is covered with vapor locally. Then, the area of the surface which is covered with vapor becomes larger with an increase of input. And as a result, the heat transfer coefficient decreases and reaches the operating limit

Fig. 8 Heat transfer pattern

which is caused by pressure loss. In the region III, since there are some dry gaps, the liquid flow area is made narrower and the maximum heat transfer rate caused by pressure loss may be lower than that in normal state. In Fig. 7(a), the maximum heat flux caused by pressure loss is 2.7×10^5 kcal/m²h by the equation which authors⁽¹⁰⁾ showed in a previous report. The experimental value is 1.3×10^5 kcal/m²h and is lower than the above-mentioned value just as has been expected.

5. Derivation of an equation for an overall heat transfer coefficient

5.1 Fundamental idea

In this chapter, by considering that there is no nucleate boiling but evaporation only from the liquid-vapor interface in the region II, an equation for the overall heat transfer coefficient applicable to this region is derived. The heat transfer mechanism is considered to be the thermal resistance of conduction through the wick-liquid matrix and the evaporation resistance at the liquid-vapor interface. Ferrell et al.⁽⁵⁾⁽⁶⁾ studied vaporization from the wicks of monel and glass particles with pure water as the working fluid at atmospheric pressure and concluded that the heat transfer mechanism was conduction through the saturated wick-liquid matrix to the liquid-vapor interface located approximately at the radius of particles above the heated surface. Based on this mechanism, they derived an equation for an overall heat transfer coefficient. There are some points in question about the range of applications and the assumption on the state of contact between the particles and the heated surface in their analysis. However, it well indicates the heat transfer coefficient for a packed bed to a certain degree. Furthermore, because of magnitude of capillary force and mechanical stability at the radius of particles above the heated surface, their assumption is considered to be appropriate. In this analysis, assumption is made that the final liquid-vapor interface is located at the radius of particles above the heated surface. Figure 9 shows the proposed model. $x = 0$ and δ show the location of the heated surface and the liquid-vapor interface, respectively.

Fig. 9 Proposed model

As shown in Fig. 9(b), a temperature difference at the liquid-vapor interface occurs due to the evaporation resistance. Now, ΔT_{eva} and ΔT_{con} express the temperature differences due to the evaporation resistance and between $x = 0$ and δ , respectively. Therefore, superheat ΔT_{sat} can be written as follows.

$$\begin{aligned}\Delta T_{sat} &= t_{we} - t_v \\ &= \Delta T_{con} + \Delta T_{eva}\end{aligned}\quad (3)$$

t_{we} : heated surface temperature

t_v : vapor temperature

The overall heat transfer coefficient h_e can be expressed in heat flux q_1 based on the area of the heated surface:

$$h_e = \frac{q_1}{\Delta T_{eva} + \Delta T_{con}} \quad (4)$$

The individual terms in eq. (4) are described in the remainder of this chapter.

5.1.1 Temperature difference due to evaporation ΔT_{eva} ⁽¹¹⁾⁽¹⁵⁾

As described above, thermodynamic equilibrium does not strictly hold on the liquid-vapor interface with evaporation. Katto⁽¹¹⁾, Silver⁽¹²⁾ and Peter⁽¹³⁾ gave an approximate equation for the relation between ΔT_{eva} and heat flux q_2 based on the area of the liquid-vapor interface:

$$\Delta T_{eva} = \left(1 - \frac{\rho_v T_v \sqrt{2\pi R g T_v}}{\rho_l J g \rho_v \alpha h_{fg}}\right)^2 q_2 \quad (5)$$

ρ_v : vapor density kg/m³

ρ_l : liquid density kg/m³

h_{fg} : latent vaporization heat kcal/kg

α : evaporation-condensation coefficient

R : gas constant m²/°K

Some reported values of α for pure water are given in Table 2. There is little difference among these values and therefore, $\alpha = 0.03$ reported by Katto is used here.

5.1.2 Temperature difference due to conduction ΔT_{con}

The layer between $x = 0$ and δ is a mixed bed of the sintered metal powders and the working liquid. λ_e shows the effective thermal conductivity of the mixed bed and if all the heat supplied from the heated surface is assumed to transfer in the bed only by conduction, ΔT_{con} is written as follows.

$$\Delta T_{con} = \frac{q_1 \delta}{\lambda_e} = \frac{q_1 D_p}{\lambda_e} \quad (6)$$

where D_p is the particle diameter.

Table 2 α for pure water

reference	α
11	0.03
12	0.038
15	0.04

In the condenser, a phase change occurs on the wick surface (16) and a convection by the condensate is negligible (16) (17). Therefore, λ_e is calculated from the following equation by taking into consideration the condensation resistance on condition that the heat transfer in this section is conduction only.

$$\lambda_e = \frac{Q_e}{2\pi} \frac{\log_e(D_i/D_v)}{\ell_c(t_{wi}-t_{wc})} \quad (7)$$

$$t_{wi} = t_v - \frac{1}{h_{cv}} \frac{Q_e}{\pi D_v \ell_c} \\ = t_v - \left(1 - \frac{\rho_v}{\rho_\ell} \frac{T_v \sqrt{2\pi R_g T_v}}{J_g \rho_v \alpha h_{fg}^2} \right) \frac{Q_e}{\pi D_v \ell_c} \quad (8)$$

D_i : pipe inner diameter m

D_v : vapor space diameter m

ℓ_c : condenser length m

h_{cv} : overall heat transfer coefficient at the liquid-vapor interface in the condenser

kcal/m²h°C

t_v : vapor temperature °C

t_{wi} : wick surface temperature °C

t_{wc} : mean temperature of the condenser inner wall °C

Table 3 shows a comparison between λ_e measured in the condenser and the values calculated from Yagi and Kunii's equation (18). Since the agreement is good, λ_e is calculated from Yagi and Kunii's equation

$$\frac{\lambda_e}{\lambda_g} = \epsilon + \frac{1 - \epsilon}{\phi + (2/3)(\lambda_g/\lambda_s)} \quad (9)$$

λ_g : thermal conductivity of the fluid kcal/mh°C

λ_s : thermal conductivity of solid kcal/mh°C

Fig. 10 Particle arrangement

Table 3 Comparison of experimental values for λ_e and Yagi-Kunii's equation

	vapor temp °C	number of measurement	average experimental value for λ_e kcal/m h °C	λ_e by Yagi-Kunii's equation
HP 1	45	30	4.09	3.86
	60	22	3.63	
	100	16	5.44	
HP 2	60	11	5.50	5.08
	45	21	5.50	
HP 5	30	20	5.00	5.21
	45	35	5.40	

ϵ : porosity

ϕ : (thickness of a slab of stationary fluid near the contact point between powders)/(powder diameter)

ϕ may be obtained theoretically, but the explanation is omitted for brevity's sake.

5.1.3 q_1 and q_2

A_1 and A_2 denote the area of the heated surface and the liquid-vapor interface (the evaporation area), respectively. All the heat supplied from the heated surface is assumed to be used up to evaporate liquid from the liquid-vapor interface ($q_2 A_2 = q_1 A_1$) and the ratio of A_1 to A_2 is defined as the evaporated fraction coefficient a . Therefore, the relation between q_1 and q_2 can be written as follows

$$q_2 = a q_1 \quad (10)$$

5.1.4 Evaporated fraction coefficient a

A rectangular parallelepiped the sides of which are of unit length and the height is equal to the particle diameter D_p is considered and N is defined as the number of particles in it, ϵ as porosity, and the liquid-vapor interface as a plane located at the radius of particles above the heated surface is considered. Therefore, the area of the liquid-vapor interface A_2 can be written as follows

$$A_2 = 1 - \frac{\pi D_p^2}{4} N = \frac{3\epsilon - 1}{2} \quad (11)$$

According to the definition of a , we have

$$a = \frac{A_1}{A_2} = \frac{2}{3\epsilon - 1} \quad (12)$$

Table 4 shows the values of a obtained from Fig. 10. Figure 10(a) shows the closest packing arrangement of particles with the same diameter in a two-dimensional field. Therefore, equation (12) is applicable when porosity is more than 0.4. If the porosity is less than 0.4, the packing can not be treated as two-dimensional arrangement. It is difficult to get the evaporation area with three dimensional consideration due to many unknown factors such as the shape of a pile of particles and so on. Therefore, when the porosity is less than 0.4, $a = 10.0$ which corresponds to the porosity of 0.4 is used.

Table 4 a calculated by Fig. 10

porosity	a
0.4	10.0
0.476	4.65

Substituting eqs. (5), (6) and (10) into eq. (4), we obtain

$$h_e = \frac{1}{\frac{D_p}{2\lambda_e} + (1 - \frac{\rho_v}{\rho_l}) \frac{T_v \sqrt{2\pi R_g T_v}}{J \rho_v g \alpha h_{fg}^2} a} \quad (13)$$

In the model of this study the effect of the pipe inclination angle on the heat transfer mechanism is not considered. As a matter of fact, it seems that the curvature of the liquid-vapor interface in the evaporator changes in accordance with the inclination angle and has an effect on the heat transfer mechanism. However, as shown in Fig. 4, the effect of the inclination angle is not confirmed qualitatively in the range of this experiment condition. Therefore, it is assumed that there is no effect of the pipe inclination angle on the heat transfer mechanism.

5.2 Comparison between experimental results and predicted values

Figure 6 shows comparisons between experimental results for pure water and predicted values. The solid line in the figures is predicted from eq. (13). Equation (13) makes it possible to predict the overall heat transfer coefficient for a sintered metal powder wick. The range of heat fluxes applied to this equation would vary according to the kind of the working fluid, the kind and the dimensions of sintered metal powders, the working pressure, etc., and in the range of this experimental condition the range of heat fluxes applied has been found to be approximately $0.5 \times 10^4 \sim 6 \times 10^4 \text{ kcal/m}^2\text{h}$ when the working fluid is pure water. The heat transfer mechanism in the region beyond and below the application range of heat fluxes - i.e. the regions I and III - was explained qualitatively in Section 4.2.

The relation between α and fluid property has not been fully studied yet (19) and measured value of α for R113 has not been reported yet as far as authors are aware of. Therefore, α for R113 is estimated here: The experimental heat transfer coefficient for HP5 at 0.37 ata is 2600 $\text{kcal/m}^2\text{h}^\circ\text{C}$. Using this value and eq. (13), α is calculated at 0.1. The solid lines shown in Fig. 5 are predicted from eq. (13) with $\alpha = 0.1$. It is found that the agreement between the predicted lines and the experimental results is excellent.

Table 5 shows the calculation results of the evaporation resistance, the conduction resistance and the ratio of the evaporation resistance to total resistance (the evaporation resistance + the conduction resistance) at each working pressure for

HP5 and R113. The reason for the small effect of the working pressure on the heat transfer coefficient with R113 seem to be that the ratio of the evaporation resistance to total resistance is small at each working pressure in the range of the experimental condition.

For the purpose of examining adequacy of this model, a comparison between this model and experimental results obtained by Ferrell et al. (5) is made. In Fig. 11, results for some kinds of particles at various angles of inclination are given and compared with the lines predicted from eq. (13). Same as our results, their results show that the pipe inclination angle has no sensible effect on the heat transfer pattern and the heat transfer coefficient, and the heat flux-superheat relation is linear below a certain heat flux. From

Fig. 11 Comparison of eq. (13) and experimental data for Ferrell et al.

Table 5 Calculated values of conduction resistance and evaporation resistance (HP5, R113)

pressure ata	conduction resistance R_{con} ($\text{kcal/m}^2\text{h}^\circ\text{C}$) ⁻¹	evaporation resistance R_{eva} ($\text{kcal/m}^2\text{h}^\circ\text{C}$) ⁻¹	$\frac{R_{eva}}{R_{con}+R_{eva}}$
0.37	3.07×10^{-4}	7.57×10^{-5}	0.91
0.55	3.15	5.75	0.15
0.79	3.23	4.52	0.12
1.033	3.27	3.72	0.10

Fig. 11, it is found the heat transfer pattern changes from the region II to III and furthermore to the operating limit as shown in Fig. 8. Transition heat flux to the region III decreases with an increase of the inclination angle. Therefore, the transition phenomenon is estimated to be related to the balance between capillary force and pressure loss, and to the heat flow direction. Equation (13) well indicates all the results for 40-50 mesh glass and monel particles and data for run 1 of 80-100 mesh monel particles. And the tendency indicated by this model is similar to that of data for runs 2 and 3 of 80-100 mesh monel particles, but the experimental values are less than the predicted ones. The comparison between the analysis by Ferrell et al. and these results shows the same matter. The two experimental results for the inclination angles of 0.3° and 30.2° (when the inclination angle is larger than or smaller than 15.1°) differ from the results for the inclination angle of 15.1° by the same value. Therefore, it is supposed that the cause for this difference is not the effects of the inclination angle but other factors (e.g. accumulation of impurities on the heated surface). However, equation (13) can indicate well the data obtained by Ferrell et al. qualitatively and quantitatively in a certain range of heat fluxes. Therefore, the heat transfer mechanism proposed here is considered to be substantially correct.

6. Conclusions

Experiments to investigate the heat transfer mechanism in the evaporator of a heat pipe were conducted and the following results were obtained:

- (1) Noncondensable gas and the pipe inclination angle are considered to have no sensible effect on the heat transfer pattern and the heat transfer coefficient in the range of this experimental condition.
- (2) In case of R113, the pressure has a small effect on the heat transfer pattern and the heat transfer coefficient vs. the heat flux-superheat relation is linear below a certain heat flux. On the other hand, in the case of pure water the heat flux-superheat relation is considered to be linear. However, the pressure is considered to have more sensible effect on the heat transfer phenomena as compared with the results for R113. The heat transfer coefficient decreases as the pressure decreases and there appears a clear difference in the heat transfer pattern between the upper and bottom walls in a low pressure range. This trend in the low pressure range would be expected because of the difference in the liquid-vapor interface recession caused by the difference in heat flow direction.
- (3) Heat transfer mechanism is considered to basically consist of the thermal resistance of conduction through the wick-liquid matrix and the evaporation resistance at the liquid-vapor inter-

face.

- (4) The model proposed by this study makes it possible to predict the overall heat transfer coefficient for a heat pipe with a sintered metal powder wick. The range of heat fluxes applied to this model would vary according to the kind of the working fluid, the kind and the dimensions of the sintered metal powder, the pressure etc., and in the range of this experimental condition the range of heat fluxes applied has been $0.5 \times 10^4 \sim 6 \times 10^4$ kcal/m²h when the working fluid is pure water.

This study has not been able to explain the effect of the liquid-vapor interface curvature and the heat transfer mechanism on the liquid-vapor interface recession process. Authors have a plan for further studies.

References

- (1) Dunn, P. D. and Reay, D. A., Heat Pipe, (1976), Pergamon Press.
- (2) Feldman, K. T., The Heat pipes, (1976-6), Univ. of New Mexico.
- (3) Ranken, W. A. and Kemme, J. F., Thermionic Specialist Conference (AIAA), (1965-10), p.325.
- (4) Parker, G. H. and Hanson, J. P., Intersociety Energy Conversion Engng. Conference, (1967-8), p.847, Advances in Energy Conversion Engng.
- (5) Ferrell, J. K. and Johnson, H. R., ASME Pub., No.70-HT/SPT-12 (1970).
- (6) Ferrell, J. K. and Alleavitch, J., Chem. Engng. Progr., Symp. Ser., Vol.66, No.102 (1970), p.83.
- (7) Allingham, W. D. and McEntire, J. A., J. Heat Transfer, Vol.33, No.1 (1961-2) p.71.
- (8) Kunz, H. R., et al., Vapor Chamber Fin Studies, NASA CR-812 (1967-6).
- (9) Alexander, E. G., et al., AIAA Selected Reprint Series, Vol.16, (1973-9), p.117.
- (10) Shibayama, S. and Morooka, S. et al., Trans. Japan Soc. Mech. Engrs. (in Japanese), Vol.44, No.380 (1978-4), p.1355.
- (11) Kato, Y., Dennetsu Gairon (in Japanese), (1964-12), p.285, Yokendo.
- (12) Silver, R. S., Engineering, Vol.161 (1946-5), p.505.
- (13) Sockol, P. M., NASA TM-X-2265 (1971-4), p.4.
- (14) Kennard, E. H., Kinetic Theory of Gasses, (1938), p.68, McGraw-Hill.
- (15) Sukhatne, S. P. and Rohsenow, W. H., J. Heat Transfer, Vol.88, No.1 (1966-2), p.19.
- (16) Sun, K. H. and Tien, C. L., AIAA J., Vol. 10, No.8 (1972-10), p.1051.
- (17) Sun, K. H. and Tien, C. L., Int. J. Heat & Mass Transf., Vol.18, (1975-5), p.363.
- (18) Yagi, S. and Kunii, D., Int. Developments in Heat Transfer, Part IV, (1961), p.742.
- (19) Mori, Y., et al., Trans. Japan Soc. Mech. Engrs. (in Japanese) Vol.33, No.332 (1974-4), p.1064.