Heat transfer characteristics of a radial heat pipe

By

Fumito KAMINAGA*, Yoshizo OKAMOTO*, Terumitsu YOTSUKURA**
Haruhiko ITO***, Takasi SAITO***, Hiroo AMEZAWA***

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Summary: Heat transfer characteristics of a newly designed heat pipe which transports thermal energy in a radial direction of the heat pipe are examined experimentally using Freon R-113 working fluid. The heat pipe produces a much higher overall heat transmission than the current heat pipe due to a direct contact condensation instead of a filmwise condensation. An unheated vapor space above a heated section and a higher charge rate over 70% are required to use it in a wide range of heat flux. Normal screen wick presents an insufficient capillary force to pump up the working fluid to the heated section against the gravity force.

NOMENCLATURE

g	: Acceleration of gravity	Re^*	: Film Reynolds number $=4qL/\mu_l h_{fg}$
h	: Heat transfer coefficient	s	: Dimensionless number $=(3\sigma^3/\rho_l^3 g\nu_l^4)^{1/5}$
h_{fg}	: Latent heat of vaporization	T	: temperature
k	: Overall heat	X	: defined by Eq. (1)
L	: Axial length	λ	: Thermal conductivity
Nu	: Nusselt number	μ	: Dynamic viscosity
Nu^*	: $=h(\nu_l^2/g)^{1/3}/\lambda$	ν	: Kinematic viscosity
p	: Pressure	ρ	: Density
Pr	: Rrandtl number	σ	: Surface tension
Q	: Linear heat rate	Subscript	
q	: Heat flux	l	: Liquid
r	: Radius	v	: Vapor
Re	: Reynolds number		

1. INTRODUCTION

The heat pipe is defined as a thermal device to transport thermal energy from one part of the tube structure to another part by means of phase changes, vaporization of a liquid and condensation of a vapor. Therefore, energy is transferred by a flow of a saturated vapor in a longitudinal direction, that is, an axial direction of the pipe as shown in Fig. 1−(a). This type of heat pipe is referred to an axial heat pipe hereafter. On the other hand, when a coolant as a heat sink can be obtained close to a heat source, heat can be transferred in a transversal direction, that is, a radial direction in case of a cylindrical geometry as shown it Fig. 1−(b). This type of heat pipe is referred to a radial heat pipe. A concept of the radial heat pipe was originally proposed to be used as a steam generator of the Fast Breeder Nuclear Reactor[1]. But experimental verification of the concept was not attempted at all. The radial heat pipe might be expected to produce an efficient

* Faculty of Engineering, Ibaraki University
** Graduate Student, Ibaraki University
*** Japan Atomic Energy Research Institute, Oarai

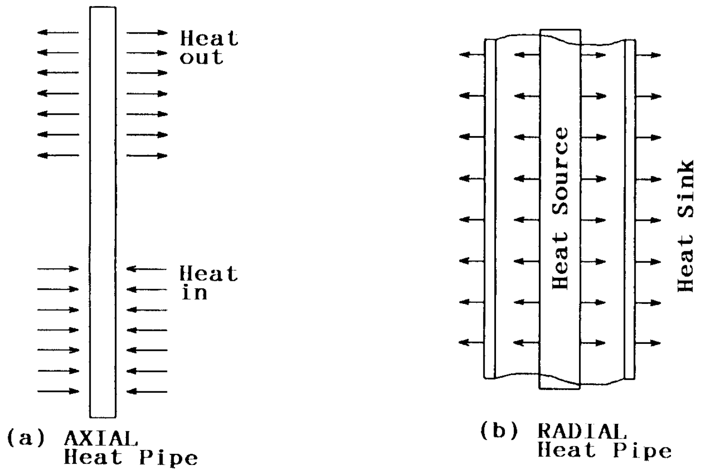


Fig. 1. Conception of radial heat pipe.

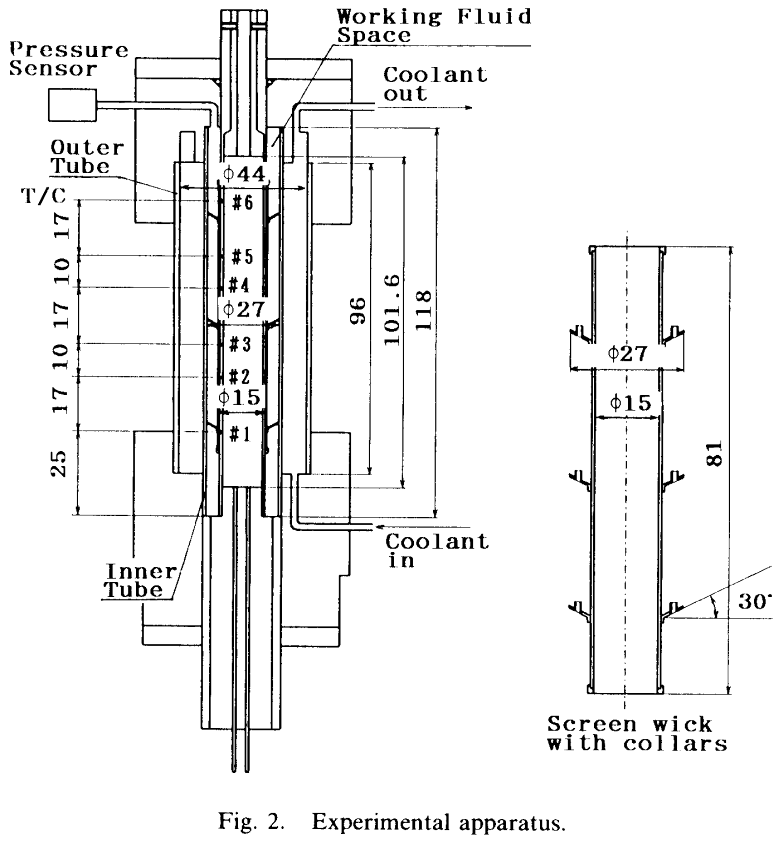


Fig. 2. Experimental apparatus.

transport of thermal energy, since thermal energy is removed by a subcooled boiling-from the heat source and is released by a direct contact condensation to the heat sink instead of a filmwise condensation in the axial heat pipe. But negative effects are expected: The heat pipe scarcely maintains a sufficient working fluid in the heat source part to make nearly isothermal operation and might present a large pressure fluctuation due to the subcooled boiiing and the direct contact condensation.

The purpose of this study is to obtain a basic understanding of heat transfer characteristics of the radial heat pipe.

2. Experimental Apparatus and Procedure

Figure 2 shows a test piece of the radial heat pipe. A heater as the heat source is placed at the center. An inner and an outer tubes are equipped concentrically with the heater to form a double annular spaces. In the inner annular space Freon R–113 is charged as a working fluid. In the outer space, water coolant circulates upward with a constant flow rate to condense vapor of the working fluid tnrough the inner tube. Two kinds of materials, a pyrex glass tube of 27 mm I. D. and 0.8 mm thickness and a copper tube of 27 mm I. D. and 1.5 mm thickness are used as the inner tube.

The outer tube is a pyrex tube. The pyrex glass inner tube makes it possible to visualize a flow behavior of the working fluid.

The experimental parameters are charge rate of the working fluid and heat flux. Table 1 shows experimental ranges of the parameters. Inner structures, a 50 mesh screen wick, a screen wick with collars shown in Fig. 2, and brass balls filling the inner space are tested to examine how to maintain the working fluid on the heater wall. Type 6 tests use a shorter heater by a quarter to enlarge an unheated vapor space above the actual heated part. The experiments are continued until the heater wall temperature or a working fluid pressure exceeds limitations for a reliable test, 200°C or 3 bar.

3. Experimental Results and Discussions

Figure 3 indicates the effect of the collar attached on the screen wick shown in Fig. 2. The abscissa is a length from the bottom of the heater. The screen wick and the collar both are not effective to provide the liquid as indicated by a large temperature rise at the upper part of the heater. The collar presents a lower tempergture due to a conduction below the liquid level and a higher temperature due to a larger system pressure at the upper part of the heater.

Figure 4 compares a screen wick effect on temperature distributions. The 50 mesh screen wick is indicated to have a negligible effect on the distributions, since the mesh size can not produce a capillary force enough to pump up the fluid against the gravity force. For the radial heat pipe the gravity force works negatively for provision of the working fluid to the heating part, on the other hand, for the axial heat pipe it also work negatively in the heating part, but it works positively for condensed fluid to fall back to the part.

Figure 5 shows the effect of the charge rate for the Type 4 tests: a copper inner tube

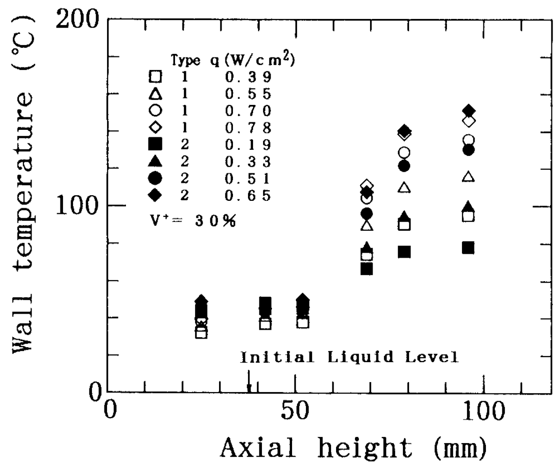


Fig. 3. Heater wall temperature distributions (Type 1 and 2).

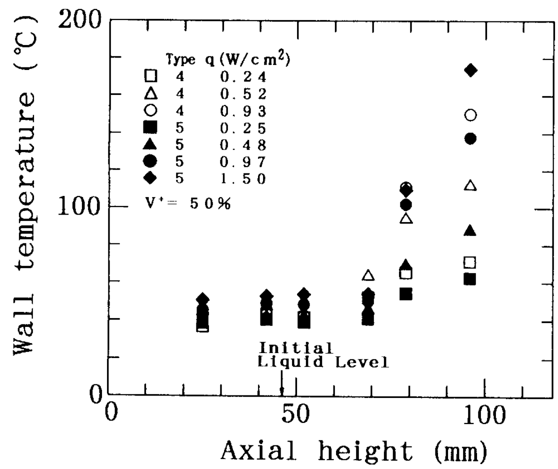


Fig. 4. Heater wall temperature distributions (Type 4 and 5).

test piece without the screen wick. The heat pipe fails to work at the smaller charge rate as shown in (b). Since at low heat flux, the liquid level hardly rise over the initial liquid level, temperatures are higher at the locations above the level as similarly shown in (a) and (b). For the high charge rate an axial heat conduction keeps the temperature not to increase so much. But for the low charge rate, uncovered area is too large to transfer the heat by the conduction, therefore, dryout occurs. As shown in (a), when the heat flux is large, nucleated bubbles push the liquid level up to the almost whole length of the heater and the temperature has a nearly uniform distribution.

Figure 6 shows the results using the shorter heater. For the shorter heater, the maximum heat flux is much higher than that for the longer heater shown in Fig. 5–(a), since a larger unheated vapor space results in a lower system pressure. This unheated space is a key parameter to determine an operating range of the heat pipe.

Figure 7 shows bolling heat transfer coefficients calculated by heat flux and temperature difference between a mean of measured wall temperatures and saturation temperature corresponding to the measured pressure. The abscissa X_k is derived from Kutateladze's correlation[2] and is defined as follows.

$$X_k = Pr^{0.35} \cdot \left[\frac{q}{\rho_v \cdot h_{fg} \cdot \nu_l}\sqrt{\frac{\sigma}{g(\rho_l-\rho_v)}}\right]^{0.7}\left[\frac{P}{\sigma}\sqrt{\frac{\sigma}{g(\rho_l-\rho_v)}}\right]^{0.7} \cdot \lambda_l\sqrt{\frac{g(\rho_l-\rho_v)}{\sigma}} \qquad (1)$$

Kutateladze's correlation is

$$h_e = 7.0\times10^{-4} X_R. \qquad (2)$$

The measured results is correlated by the following correlation except for experimental results of Type 3 and low heat transfer results under 0.02 W/cm^2.

$$h_e = 1.2\times10^{-3} X_R. \qquad (3)$$

This correlation has the same gradient but a higher value than Eq. (2). As for the Type 3 results, the measured pressure might be different from the actual value near the heater surface because a pressure drop through the packed small balls is large. As for the low heat transfer results, visual observation indicates that boiling locally occurs only at a upper part of the heater. The higher value of the measured results might be due to a vertical orientation and surface roughness of the heater. The same gradient indicates that dependence of the heat transfer coefficient on physical properties could be evaluated by the Kutateladze's correlation.

Figure 8 presents measured and predicted overall coefficient of heat transmission. The measured ones are defined by

$$K = Q/(T_w - T_c). \qquad (4)$$

where, T_w is a mean of measured heater wall temperatures and T_c is a mean of inlet and outlet coolant temperatures. The predicted ones are calculated by

$$K = \frac{2\pi}{\frac{1}{h_w r_1} + \frac{1}{h_i r_2} + \frac{1}{\lambda}\ln\frac{r_3}{r_2} + \frac{1}{h_c r_3}} \qquad (5)$$

where, r_1 is outer radius of the heater, r_2 inner radius of the inner tube, and r_3 an outer radius of it.

Boiling heat transfer coefficient, h_w, is predicted by Eq. (2). Condensation heat tranfer coefficient is calculated by a maximum value of the following three equations:

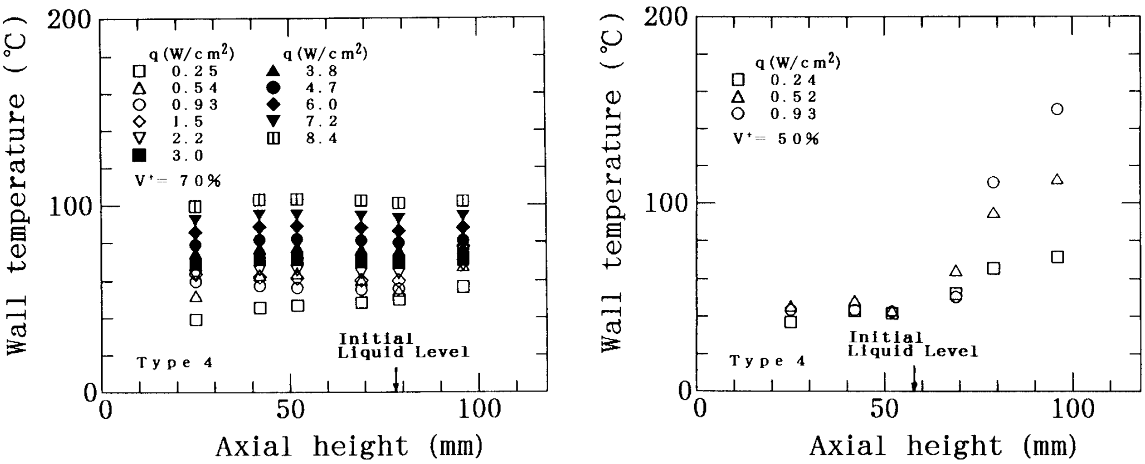
$$Nu^* = 1.47 Re^{*-1/3} \text{ (Nusselt Eq., For laminar flow)}^{(2)} \qquad (6)$$

$$Nu^* = 1.82 s^{-0.115} Re^{*-0.218} \text{ (For wavy flow)}^{(3)} \qquad (7)$$

$$Nu^* = 0.035 Pr^{2/5} Re^{*1/6} \text{ (For turbulent flow)}^{(4)} \qquad (8)$$

Forced convective heat transfer coefficient, h_c, is evaluated by Dittus-Boelter's equation[2]:

$$Nu = 0.02 Re^{0.8} Pr^{0.4} \qquad (9)$$



(a) Results at a charge rate of 70%. (b) Results at a charge rate of 50%.

Fig. 5. Heater wall temperature distributions.

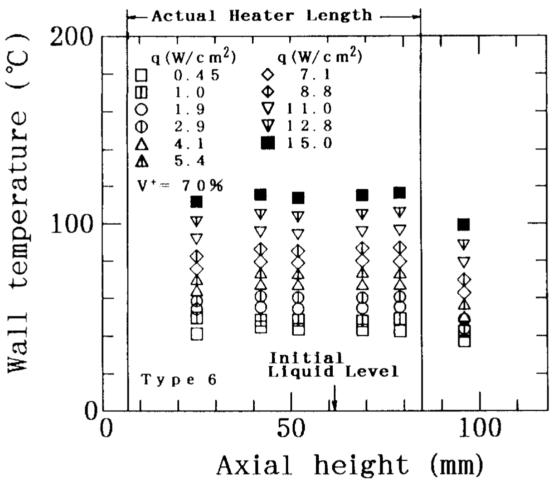


Fig. 6. Heater wall temperature distributions.

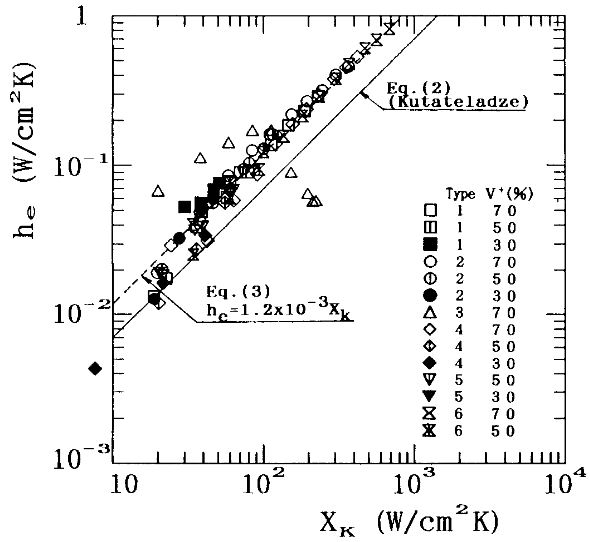


Fig. 7. Comparison of measured data with Kutateladze's correlation as for boiling heat transfer coefficient.

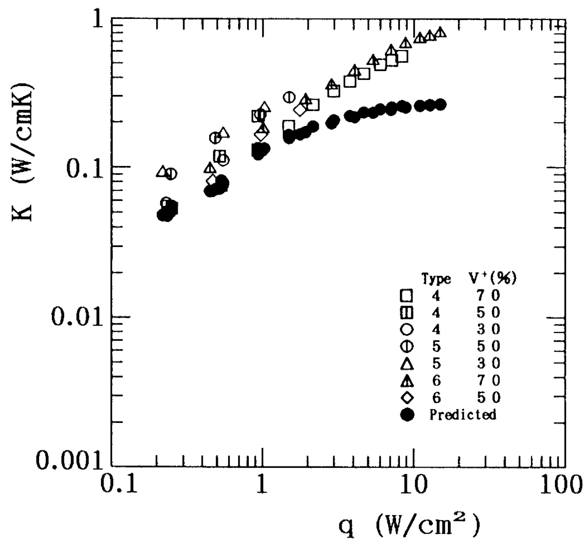


Fig. 8. Comparison of measured with predicted results as for overall coefficient of heat trasmission.

Measured results of the radial heat pipe indicates much higher overall coefficient of heat transmission than predicted results, shown by black circle symbols, of the normal axial heat pipe operated by the nucleate booling and the filmwise condensation. This higher value is due primarily to a higher condensation heat transfer.

4. Conclusions

Experimental studies of heat transfer characteristics of the radial heat pipe lead to the following conclusions.

1. The radial heat pipe indicates much higher overall coefficient of heat transmission than the current axial heat pipe operated by the nucleate boiling and the filmwise condensation due primarily to a higher condensation heat transfer. This higher efficiency can afford a smaller heat transfer area to design a compact heat exchanger.
2. Particular treatment besides a screen wick is required to provide a sufficient working fluid to the whole length of the heater. A charge rate over 70% provides sufficient working fluid to the whole length of the heater.
3. The larger unheated vapor space above the heated section results in a lower system pressure to use the heat pipe in a wider heat flux range.

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