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Enhancement of CHF in open thermosyphon with heated bottom chamber

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Abstract

An experiment has been carried out to elucidate the critical heat flux (CHF) of an open thermosyphon with a bottom heated chamber and to enhance the CHF using a concentric-tube connecting between the heated chamber and an upper cooling chamber. The CHF data are measured for the saturated liquid of R113 at a different pressure and different dimension of concentric tubes. The CHF data without the inner tube are in good agreement with the existing correlation and analytical result. Heat supplied to the bottom chamber is absorbed by evaporation of liquid and vapor generated thereby makes a counter-current flow of vapor and liquid in an ordinary thermosyphon without the concentric-tube. On the other hand, in the thermosyphon with the concentric-tube by which the counter-current flow can be controlled well, the CHF can be improved as much as several times of the CHF without the inner tube at an optimum condition. The CHF is increased with an increase in the diameter of inner tube up to its optimum diameter and then decreases continuously as the inner tube diameter approaches the outer tube diameter.

Keywords: Phase change; Critical heat flux; Thermosyphon; Improvement

1. Introduction

Heat transfer equipment using phase change process plays an indispensable role in many technological applications because of very high heat transfer coefficients. Miniaturization of equipment, such as electronic devices, needs more powerful cooling system by which heat transfer coefficient as well as the maximum heat removal can be improved. A two-phase thermosyphon is commonly used as a heat transport device since it has a large heat transport capacity. Therefore, the heat transfer coefficient and the critical heat flux (CHF) are extensively studied experimentally and theoretically for the two-phase thermosyphon.

Imura et al. [1], for example, measured the CHF in the two-phase thermosyphon and proposed the following correlation:

$$\frac{4L}{D}\frac{q_{\mathrm{co}}/(\rho_{\mathrm{g}}H_{\mathrm{lg}})}{\sqrt[4]{\sigma g(\rho_{\mathrm{l}}-\rho_{\mathrm{g}})/\rho_{\mathrm{g}}^{2}}}=0.64(\rho_{\mathrm{l}}-\rho_{\mathrm{g}})^{0.13} \tag{1}$$

On the other hand, Monde [2] analyzed maximum falling liquid film flow rate during counter-current flow of vapor and liquid in a tube and cleared a relationship between maximum flow rate and critical heat flux in the two-phase thermosyphon. The CHF value pre-

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Nomenclature

A_g	area of annulus $(= \pi/4(D_i^2 - d_o^2))$
A_h	area of heated surface
A_l	area of inner tube $(= \pi/4 d_i^2)$
D_i	inner diameter of the outer tube
d_i	inner diameter of the inner tube
d_o	outer diameter of the inner tube
H_{lg}	latent heat of vaporization
Ku	Kutateladze number $(= q_{co}/(\rho_g H_{lg})/\sqrt[4]{\sigma(\rho_l - \rho_g)g/\rho_g^2})$
L	length of annular passage
P	system pressure
Q	heat transfer rate
q	heat flux
q_{co}	CHF for saturated boiling
s	annular gap
ρ_l, ρ_g	densities of liquid and vapor, respectively
σ	surface tension

dicted by his analysis is found to agree well with Eq. (1).

It is expected recently, to improve the CHF in two-phase thermosyphon due to an increase in a heat dissipation rates from more compact devices. One way to improve the CHF is to insert an inner tube in the thermosyphon by which an influence of counter-current flow can be reduced [3,4]. Islam et al. [4] have measured the CHF for a concentric two-phase thermosyphon in which a tube is concentrically inserted into an uniformly heated tube and succeeded in significantly improving the CHF and deriving correlation by which the CHF can be predicted well.

The present objective is to increase removal of heat dissipated from a heating assembly closely attached to actual electronic devices. To accomplish the objective, a concentric tube is employed to avoid counter-current flow in a feeding tube connecting the heated chamber with upper liquid plenum. The CHF is measured for this configuration similar to a concentric two-phase thermosyphon and then its characteristic will be cleared.

2. Experiment

2.1. Test facility

Fig. 1 shows the whole experimental apparatus. The heated chamber is attached to a pressure vessel. A concentric tube is mounted at the center of the vessel to connect a lower heated chamber to an upper plenum. The bottom end of the inner tube has a large hole enough to supply liquid and is fixed at the bottom of the heated chamber. A level of the test liquid is always kept at 150 mm higher than the exit of inner tube. A special attachment is settled at the upper part of the inner tube by which vapor escaping through the annular passage is not sucked again into the inner tube, while liquid is smoothly supplied into the inner tube as shown in Fig. 1.

Fig. 2 shows the details of copper block used as a vertical heated cylinder in the heating capsule. A plate type of electrical heater is separately mounted into 36 grooves to heat the copper block uniformly. Two C–A thermocouples are radially inserted at a distance of 1 and 7 mm from the surface into the copper block to measure the heat flux and give the surface temperature. The heat flux and the surface temperature are measured at two different points of lower and upper parts of the vertical heated cylinder. The heat loss in the z-direction may be reduced by making both ends of heated block a labyrinth configuration as shown in Fig. 2, and in the r-direction to the back side by covering it with an insulated material. In a preliminary experiment, the vapor, which is generated on the heated surface in a pool boiling at heat fluxes covering the present experiment, is actually accumulated in a collector to measure the heat flux across the heated surface. When the heat flux determined thereby is compared with the electric input, it is found that the heat loss from the copper block is always equal to about 23% of the total electric input at any heat flux. In addition to this, it is found except for near critical heat flux that the heat flux thereby is in good agreement with the heat flux calculated from the temperature difference at two measuring points. The reason why a large difference sometimes appears near critical heat flux, is that the temperature monitored near the surface randomly fluctuates due to recurrence of wet and dry of the surface and the temperature difference between two points is not large enough to ignore the temperature fluctuation. Therefore, as the heat flux, the one calculated from the electric input by taking the heat loss into account is employed.

Finally, it should be necessary to say that except for near critical heat flux, there is a difference in only a few percent between the heat fluxes calculated from the electric input and the difference in the temperatures measured at two points.

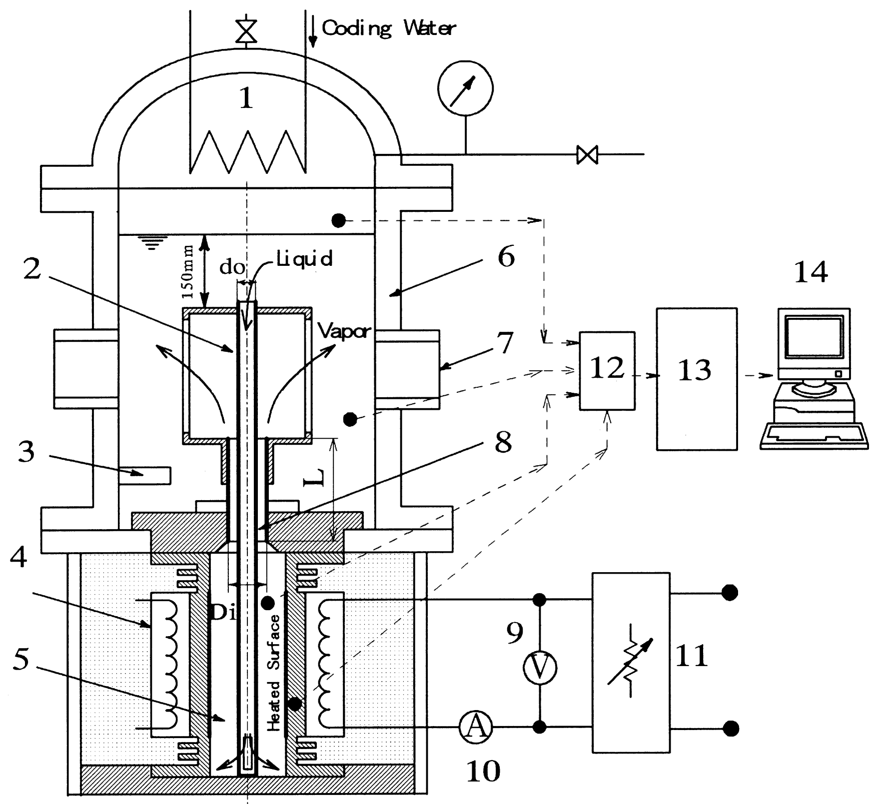

Fig. 1. The experimental apparatus. 1. Cooler, 2. inner tube, 3. tank heater, 4. slot heater, 5. boiling space, 6. pressure vessel, 7. window, 8. annular passage, 9. voltmeter, 10. ammeter, 11. variable transformer, 12. ice box, 13. digital thermometer, 14. CPU.

2.2. Procedure of measurement for CHF

Test liquid is first heated up to saturation temperature at a designated pressure. The electric input is increased stepwise that is less than 2% of each preceding heat flux during which the heat flux curve against the wall superheat (see Fig. 3) suddenly changes from steep to slow slopes. It usually takes about 15 min for the temperature in the block to reach a steady state, after heat flux is increased. The CHF is here determined at the heat flux at which the slope of this curve turns. At that point, the CHF is determined with an

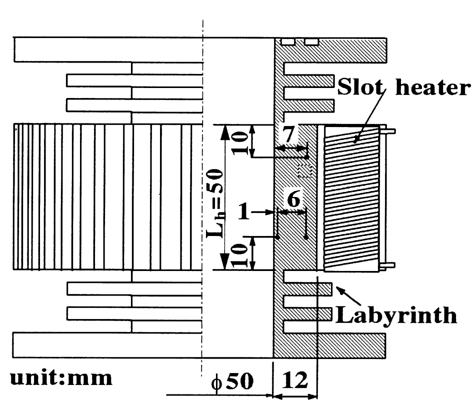

Fig. 2. The details of heated block in bottom chamber.

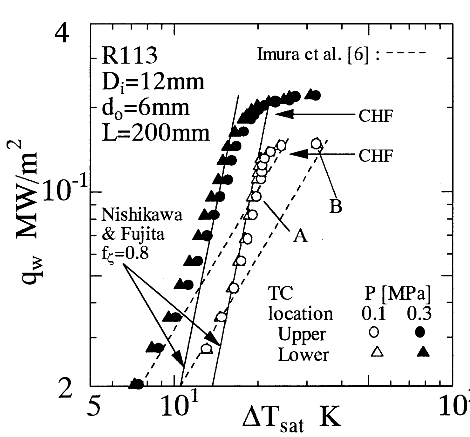

Fig. 3. Boiling curve.

uncertainty of 2%. The experimental conditions are listed in Table 1.

3. Results and discussions

3.1. Boiling curve

Fig. 3 shows boiling curves of R113 measured at upper and lower parts in the vertical heated cylinder, as an example, for $D_i = 12$ mm, $d_o = 6$ mm and $L = 200$ mm at two different pressures of 0.1 and 0.3 MPa. The solid line in Fig. 3, is the Nishikawa and Fujita [5] prediction for ordinary nucleate pool boiling and a broken line is the Imura et al. [6] prediction for two-phase thermosyphon. It may be necessary to say that the boiling curves obtained for other experimental conditions are similar to those in Fig. 3.

From Fig. 3, there is no difference in boiling curves between upper and lower parts so that the boiling on the vertical cylindrical surface appears uniformly. In addition, boiling heat transfer is enhanced at higher pressure, as generally accepted. Fig. 3 shows that the heated surface temperature gradually increases with a large increase in heat flux, while beyond a certain heat flux, for example, $q = 0.12$ MW/m^2 for $P = 0.1$ MPa, a slight increase in heat flux brings about a large rise in heated surface temperature. Comparing the temperature changes between two different points A and B, the temperature at point A almost does not change after a steady state is reached, while the temperature at point B, randomly and largely fluctuates with a temperature change. Therefore, as the temperature at point B, a mean value of the fluctuated temperature is given in Fig. 3. This fluctuation may allow us to estimate that at point B, amount of the liquid supplied through the inner tube becomes deficient in keeping ordinary nucleate boiling resulting in an unsteady state that the heated surface is alternately dried and wetted. On the basis of the characteristic of boiling curve, the occurrence of CHF in the present experiment can be defined as the point at which the curve turns from gradual to sharp increase in the temperature with an increase in heat flux as shown in Fig. 3.

Table 1
Experimental conditions

Test fluid	R113
P (MPa)	0.1, 0.2 and 0.3
ρ_l/ρ_g	8.76, 12.04, and 17.45
L (mm)	100, 200, and 300
$D_i(d_o)$, mm(mm)[a]	12(10, 8, 6, 4, 0), 9(6, 4, 0)
s (mm)	1–4

[a] $D_i(d_o)$ means a combination of concentric tube.

3.2. Characteristic of critical heat flux

Fig. 4 shows the critical heat flux measured for two different tubes of diameters $D_i = 12$ and 9 mm at a different pressure, plotted against the inner tube diameter. In Fig. 4, the Kutateladze [7] prediction (2) with a constant of $K = 0.16$ is given to compare the present CHF value, although this geometric configuration is quite different from ordinary pool boiling.

$$\frac{q_{co}/(\rho_g H_{lg})}{\sqrt[4]{\sigma g(\rho_l - \rho_g)/\rho_g^2}} = K(=0.16) \tag{2}$$

It may be interesting to compare the present CHF data with the Monde [2] analytical prediction and the

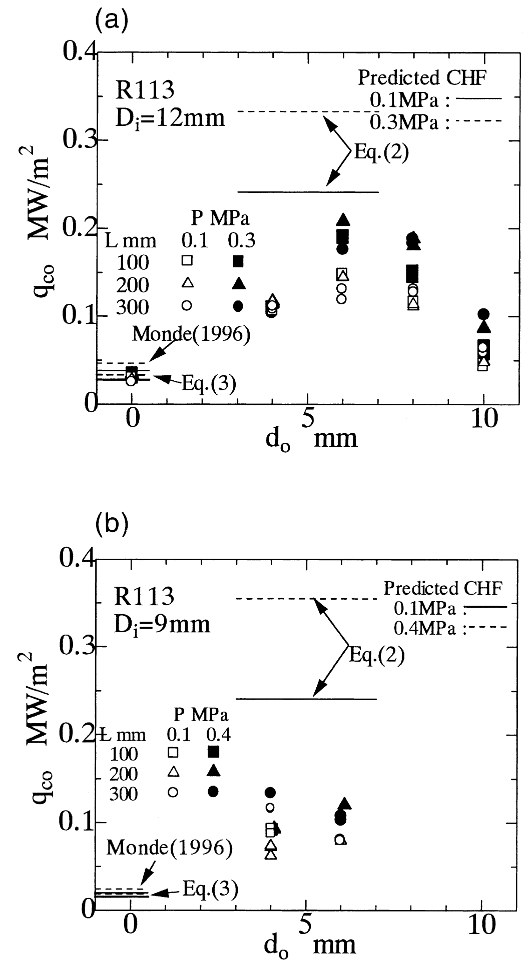

Fig. 4. Effect of inner tube diameter on CHF for (a) $D_i = 12$ mm and (b) $D_i = 19$ mm.

Imura et al. [1] prediction (1) for the two-phase thermosyphon, since this configuration is just like a two-phase thermosyphon when the inner tube is removed. Recalling that Eq. (1) was derived from the CHF for two-phase thermosyphon in which a tube is directly heated different from the present heating type, we may rearrange Eq. (1) for the present configuration by taking the maximum vapor flow through the tube into account, yielding:

$$\frac{A_h}{A_g}\frac{q_{co}/(\rho_g H_{lg})}{\sqrt[4]{\sigma g(\rho_l-\rho_g)/\rho_g^2}} = 0.64(\rho_l/\rho_g)^{0.13} \tag{3}$$

where A_h is area of the heated surface and A_g is cross-sectional area of tube.

It is seen from Fig. 4 that the CHF increases up to a certain diameter of the inner tube and then decreases monotonically as the inner tube diameter approaches the outer tube diameter. The maximum of the CHF for $D_i = 12$ mm appears near $d_o = 6$ mm. The optimum tube diameter at which the CHF becomes the maximum seems to be about half of the outer tube diameter when taking account of the result for the other case of $D_i = 9$ mm. The CHF is not almost influenced by the tube length in the range of 100–300 mm. The increasing trend of the CHF with a rise in pressure is similar to that predicted by the Kutateladze correlation (2) for pool boiling.

3.3. Critical heat flux without inner tube

Fig. 4 shows that the CHF data agree well with the values predicted by Eq. (3) and the Monde [2] analysis. It may be interesting to note that both Eq. (3) and the Monde analysis are not subject to how to supply heat into tube or heated chamber and focus on the flow aspect under the vapor–liquid counter-current flow at the exit of heated part. In addition, Monde [2] mentioned that this type of CHF can be governed by the condition of vapor–liquid counter-current flow near the exit of heated tube and may not be influenced by the length of adiabatic part.

The fact that the CHF data can be predicted well by Eq. (3) and the Monde analysis, would allow us for the CHF to take place when the falling liquid film flow rate reaches the maximum. The maximum flow rate results from the fact that amount of liquid supplied through the tube to the heated chamber is constrained by the counter flow of escaping vapor. In addition to this, the results that the CHF is hardly influenced by the adiabatic tube length and becomes much smaller than that predicted by Eq. (2), may also support that this type of CHF is concerned with the maximum liquid supply to the heated section through the tube connecting with the upper liquid plenum.

It should be noted finally that the above-mentioned fact allows us for the CHF to be hardly influenced by how to supply heat to the two-phase thermosyphon.

3.4. Enhancement of CHF due to inner tube

Fig. 4 shows that the CHF is markedly improved by inserting the inner tube and that there is an optimum diameter of the inner tube at which the improvement of CHF becomes the maximum. The maximum value of the CHF reaches about seven times as large as that without the inner tube. Nevertheless, the maximum value of CHF is still about 40% lower than that predicted by Eq. (2). Consequently, even in the case of the optimum condition, the amount of the liquid needed to maintain nucleate boiling becomes short due to a blockage of the escaping vapor.

In order to calculate heat transfer rate consumed by evaporation of the liquid which circulates through the concentric tube between the plenum and the heated

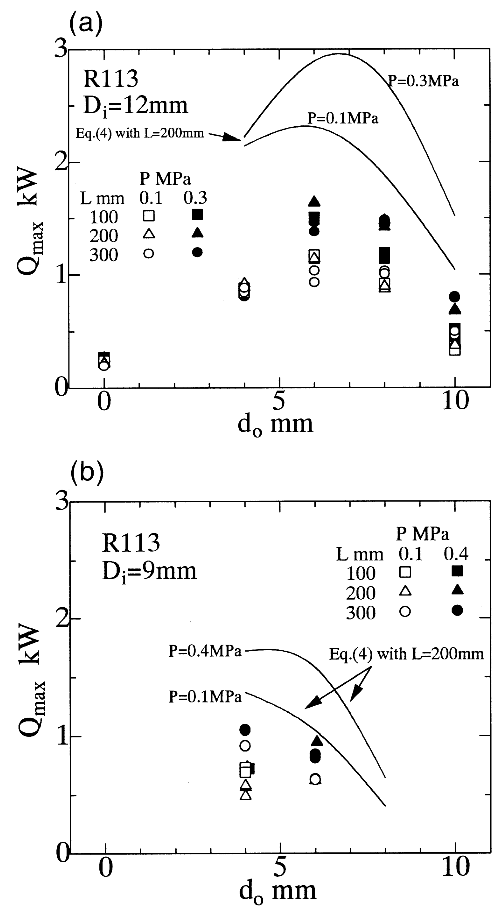

Fig. 5. Relationship between maximum heat transfer rate and inserted tube diameters: (a) $D_i = 12$ mm and (b) $D_i = 9$ mm.

space, we tentatively assume that liquid is not subject to any frictional force and the generated vapor flows out only from the annulus. As the result, the maximum heat transfer rate is expressed as:

$$Q = \rho_l A_l H_{lg} \sqrt{\frac{2(\rho_l - \rho_g) g L}{\rho_l \left\{ \frac{\rho_l}{\rho_g} \left(\frac{A_l}{A_g} \right)^2 + 1 \right\}}} \tag{4}$$

where A_l is area of the inner tube.

The maximum heat flux can be easily obtained by dividing the heat transfer rate given by Eq. (4) by the actual area of heated surface.

From engineering point of view in designing a cooling equipment such as electric devices, we would prefer to focus on the maximum heat transfer rate rather than the CHF.

Fig. 5 shows the maximum heat transfer rate rearranged from the CHF in Fig. 4 plotted against the inner tube diameter in which the value calculated from Eq. (4) is also shown for comparison. The CHF characteristic is qualitatively similar to that predicted by Eq. (4). If a frictional effect of the wall on vapor and liquid flows, would be correctly evaluated and be taken its effect into account, then it may be possible to predict the CHF more accurately.

Incidentally, there is a discrepancy in the effect of the tube length between the experimental result and Eq. (4). This may be attributed to no consideration of the frictional effect and to a trade off between an increase in frictional force on the wall and buoyant force with an increase in the tube length.

4. Similarity between two-phase concentric thermosyphon and present system

Islam et al. [4] measured the CHF in a concentric two-phase thermosyphon in which an outer tube is uniformly heated and unheated tube is inserted into the heated tube and reported that the CHF can be markedly improved by the inner tube and a correlation was proposed which can predict the optimum diameter of the inner tube at which the improvement of the CHF becomes maximum.

Comparing the present CHF with the CHF described by Islam et al., it is found that both CHF characteristics are very similar in the following points: (1) two CHFs without the inner tube are predicted well by Eq. (1) and the Monde prediction and (2) the optimum diameter of inner tube appears near half of the inner diameter of the outer tube. However, a direct comparison between both CHFs seems to be difficult, since it is not known how to evaluate heated equivalent diameter for the present system.

5. Conclusions

The experimental study has been carried out to improve the CHF in the thermosyphon with the heated bottom chamber by inserting the inner tube into the tube connecting between the heated chamber and the upper liquid plenum, and the following results are obtained:

1. The maximum heat transport rate without the inner tube is governed by the counter-current flow at the connecting tube for this type of the two-phase thermosyphon and corresponds to the amount of heat calculated from maximum liquid flow rate falling along the connecting tube.
2. The CHF can be improved about seven time by using the optimum concentric tube compared with that without the inner tube.

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