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Measurement of Condensation Curve for Dropwise Condensation Heat Transfer

滴状凝縮熱伝達における凝縮曲線の測定

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1. Introduction

The object of this study is to observe the possible change in the mode of dropwise condensation which may occur when one increases the degree of surface subcooling.

In boiling it is well known that the mode of heat transfer shifts from convection to nucleate boiling, then to transition boiling, and finally to film boiling, along with the increase of surface superheat. The heat flux also varies with the degree of superheat. The curve that represents such a change of heat flux with the superheat is called boiling curve and has first been measured by Nukiyama[1].

In dropwise condensation, a similar curve, which may be called "condensation curve", must be obtained. In the case of dropwise condensation, however, it is very difficult to realize large surface subcooling by the usual means of cooling owing to the very large heat transfer coefficient. Thus, no measurement was made until recently.

The first measurement on steam has been done by Takeyama and Shimizu[2]. For organic vapors (aniline, ethanediol and nitrobenzene) a paper by Wilmshurst and Rose[3] has been published. Takeyama and Shimizu[2] have devised a means of cooling and a heat transfer block of the shape of a truncated cone, and have observed the change of the mode of condensation. Their results are shown in Figs. 10 and 11 as a thin solid line. In Fig. 10 it is seen that the heat flux increases with the increase of surface subcooling ΔT, quite similar to the boiling curve. Further increase of ΔT brings about a state in which the entire surface gets blanketed by a film of condensate. Such phenomenon occurs when the rate of drop departure is exceeded by the rate of condensation. But this is not to be called film condensation, in the strict sense of the words. The reason has been stated elsewhere[4]. When the surface temperature becomes lower than the freezing point by further cooling, a thin layer of ice is formed on the surface and the vapor condenses onto it as a film. This mode of condensation, named glacial condensation by Takeyama and Shimizu, is actually a film condensation on the ice layer. The authors would like to propose a term "on-ice condensation" for this process.

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At any rate the first condensation curve for steam has been obtained by Takeyama and Shimizu. But as Westwater[5] and Ochiai, Tanasawa and Utaka[6] have pointed out, there remains some doubt on the accuracy of their measurement. In the present study, a care has been taken of the method of measurement to attain sufficient accuracy. The condensation curves have been measured taking as a parameter the departing drop diameter which is the most important factor affecting the coefficient of heat transfer by dropwise condensation.

2. Apparatus

The heat transfer block designed to obtain sufficient

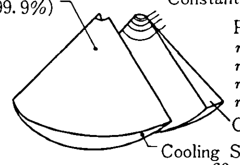

Fig. 1 Heat Transfer Block and Location of Thermocouples

accuracy of measurement and attain large surface subcooling is shown in Fig. 1. It is a copper block of the shape of a right cone with 90° vertical angle truncated by two concentric spherical surfaces; the inner surface being condensing surface and the outer surface being cooling surface. This block is of the split-type like the previous cylindrical one[7]. Constantan wires of 0.08 mm dia. for thermoelectric thermometry are fixed in concentric grooves cut on one of the split surfaces. The heat flux and the temperature of the condensing surface are obtained by extrapolation of four temperatures inside the block assuming one-dimensional heat conduction inside a sphere, i.e. the change in temperature is inversely proportional to r, where r is the radial distance from the virtual apex of the cone.

Before every measurement the condensing surface was polished with fine alumina powder and then oleic acid was applied as the promoter.

The heat transfer block was fitted to a insulator block made of bakelite and was settled in the condensing chamber as shown in Fig. 2.

The steam was forced to flow closely along the spherical

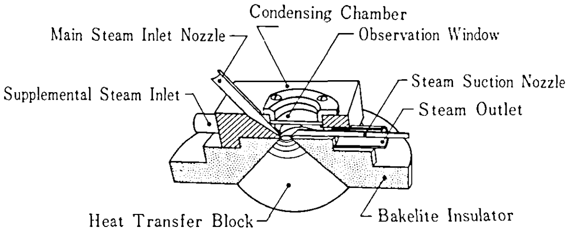

Fig. 2　Condensing Chamber

surface for two reasons. The first was to remove non-condensable gas. The second was to make change widely the departing drop size and keep the maximum drop size small enough to make the effect of the radius of curvature of the condensing surface negligible. A certain device was needed to make the steam flow along the concave surface having large curvature. For this purpose, the steam was introduced into the condensing chamber from an inlet nozzle with a flat section, tangentially to the upper stream edge of the condensing surface. In addition, in order to prevent possible decrease of steam velocity toward downstream, the steam was sucked by another nozzle. Furthermore, excessive steam was supplied to the chamber from another supplemental nozzle so that the chamber was always filled with the sufficient amount of steam.

To determine appropriate ratio between inlet and outlet steam velocity, a hot-wire anemometer (Hayakawa, HC 24) was used. Probes with thin platinum wires with diameters 5 μm and 20 μm were manufactured, and the mean velocities at the inlet and outlet and the local velocity at a distance 0.4 mm from the center of the surface were measured. Determination of the relationship between these velocities will be mentioned later.

Water, alcohol-dry ice mixture, and liquefied nitrogen (LN_2) were used as coolants. Cooling chamber used for water or alcohol-dry ice mixture is shown in Fig. 3.

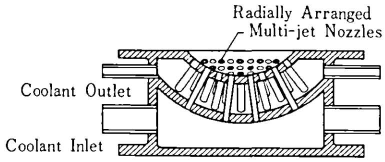

Fig. 3　Cooling Chamber (for Water and Alcohol)

Most measurements were done in steady state. Besides these, quasi-steady measurement was made in which the temperature of the condensing surface was varied very slowly. LN_2 was used as the coolant. When LN_2 was used in the steady measurement, silicone resin (Shin-etsu Chemicals Co., X31-060G) containing a fillers was uniformly applied on the cooling surface and the heat flux was adjusted by varying the thickness of the layer. In quasi-steady measurement a spherical shell made of thermally resistant material was attached to the cooling surface which decelerated the change in the temperature of the condensing block. The thermal e.m.f.'s from the thermocouples were recorded with a penrecorder.

Drop diameters were observed by a 16-mm cinecamera through microscope, filming rates being 500–2500 pictures per second.

3. Results

3. 1 Ratio of steam velocities

The outlet-to-inlet steam velocity ratios were determined so that the mean inlet velocity coincided with the velocity at the center of the surface. The relationship between the outlet-to-inlet steam velocity ratio and the inlet velocity is shown in Fig. 4. Since this relationship seems independent of the heat flux, a relationship shown by the solid line in the figure was used to determine the steam velocities.

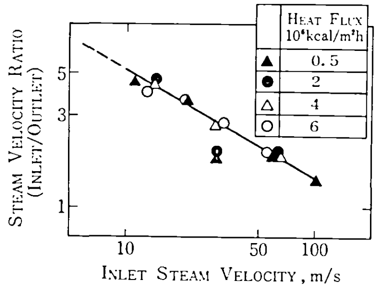

Fig. 4　Steam Velocity Ratio vs. Inlet Steam Velocity

3. 2 Effect of steam velocity

Heat transfer coefficient, departing drop diameter and mean maximum drop diameter are plotted against the steam velocity in Figs. 5, 6 and 7, respectively. Heat flux is taken as the parameter. Here the mean maximum drop diameter is the mean diameter of the drops at the central part of the surface just before absorbed by a departing drop.

According to Fig. 5, the heat transfer coefficient decreases with the increase of the heat flux. On the other hand, the drop diameters shown in Figs. 6 and 7 are independent of the heat flux. From these results the de-

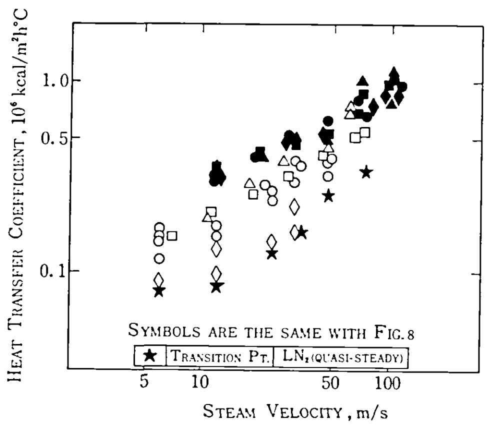

Fig. 5　Heat Transfer Coefficient vs. Steam Velocity

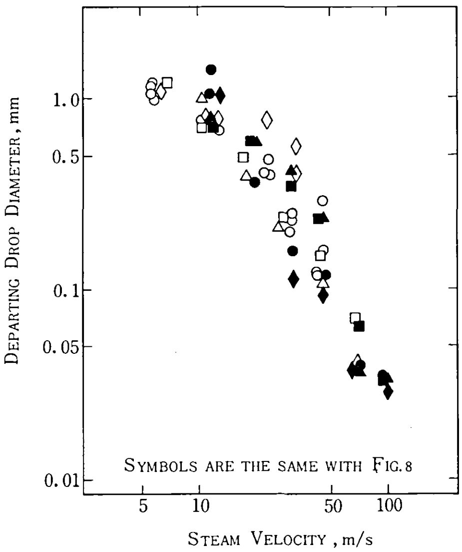

Fig. 6 Departing Drop Diameter vs. Steam Velocity

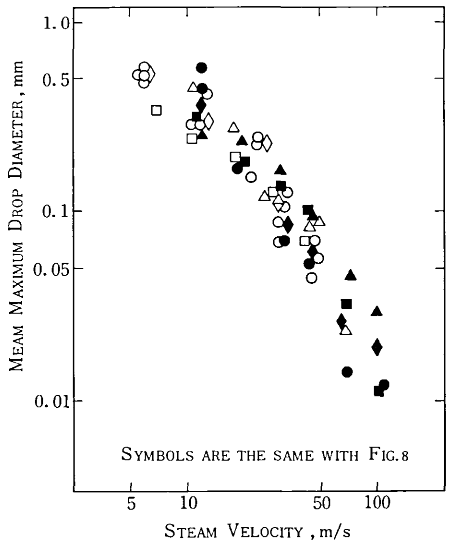

Fig. 7 Mean Maximum Drop Diameter vs. Steam Velocity

pendences of the heat transfer coefficient upon the departure diameter and the maximum diameter are obtained as shown in Figs. 8 and 9. From both figures it is found that the heat transfer coefficient depends uniquely upon the drop diameter when the heat flux is between 0.5×10^6 and 4×10^6 kcal/m^{2}h (shown by black marks in the figures). Following expressions are obtained for these data by least squares' calculation:

$$h=3.25\times10^5 D_f^{-0.304} \qquad (1)$$

$$h=2.52\times10^5 \overline{D}_{\max}^{-0.290} \qquad (2)$$

where h is the heat transfer coefficient in kcal/m^{2}h°C, D_f is the departing drop diameter in mm and $\overline{D}_{\max}$ is the mean maximum drop diameter in mm.

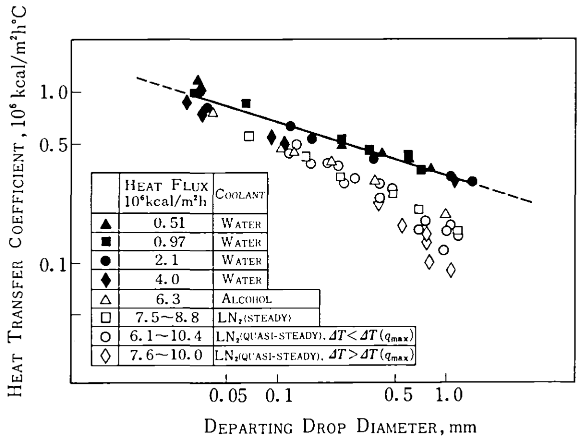

Fig. 8 Heat Transfer Coefficient vs. Departing Drop Diameter

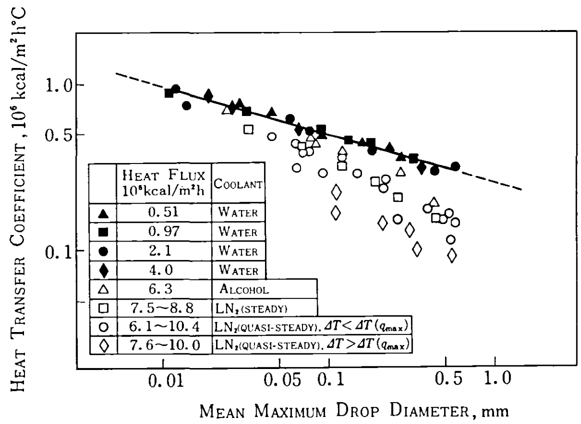

Fig. 9 Heat Transfer Coefficient vs. Mean Maximum Drop Diameter

Equation (1) is very close to the result of previous measurement by the authors[7)] in which the heat flux was 0.5×10^6 kcal/m^{2}h. At larger heat fluxes the heat transfer coefficient becomes smaller than is expressed by Eqs. (1) and (2). The tendency is more remarkable where the drop sizes are larger.

3.3 Effect of surface subcooling

The heat flux and the heat transfer coefficient are plotted against the degree of surface subcooling as shown in Fig. 10 and 11, respectively. The mean maximum drop diameter is taken as the parameter. For all five curves, the plotted points were obtained from the data of steady measurements up to the heat flux of about 8×10^6 kcal/m^{2}h: Expressions were obtained from the data shown in Figs. 6, 7, 8 and 9 using the least squares' calculation, and the points corresponding to appropriate subcoolings were determined. The quasi-steady measurement was done for the heat fluxes exceeding 6×10^6 kcal/m^{2}h. The result agreeded well with that of steady measurement where the values of heat fluxes overlapped. Although the mean maximum drop diameter was taken as the parameter in Figs. 10 and 11, the similar results must have been obtained, if the departing drop diameter or the steam velocity had been chosen as the parameter. The values corresponding to the maximum diameters are listed in the figures.

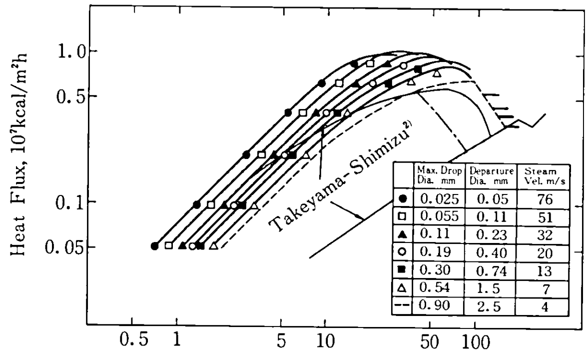

Fig. 10 Condensation Curves

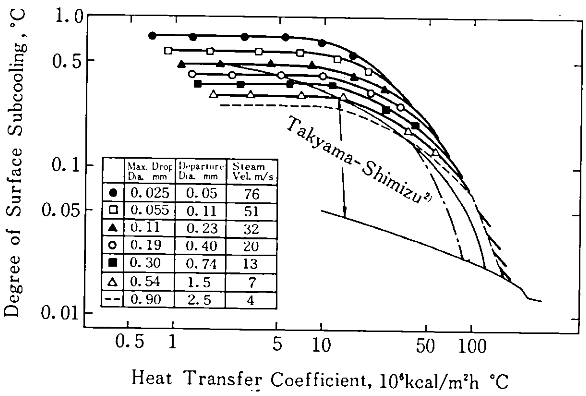

Fig. 11 Heat Transfer Coefficient vs. Surface Subcooling

The effect of non-condensable gas should be taken care of, especially at large heat fluxes. However, the concentration of air in the steam supplied from the boiler was found to be less than a few tens of p.p.m. and it was confirmed by another experiment, in which a known amount of nitrogen gas was added to the steam, that the effect of non-condensable gas was minimized by the steam flow.

The every condensation curve may be divided into three regions: In the first region the heat transfer coefficient remains constant and is expressed by Eq. (2). In the second region the heat transfer coefficient decreases, while the heat flux increases until it reaches the maximum. In the third region both the heat flux and the heat transfer coefficient decrease and finally dropwise condensation shifts to pseudo-film or on-ice condensation. This result coincides qualitatively with that of Takeyama and Shimizu, but quantitatively it does not. The broken lines in Figs. 10 and 11 were obtained by the extrapolation of the present results so that the departing drop diameter agreeded with that of Takeyama and Shimizu (i.e., D_f=2.5 mm). As seen in Fig. 11 the result of Takeyama and Shimizu shows the decrease in the heat transfer coefficient even at smaller ΔT and the magnitude there is relatively higher. It is also found that dropwise mode still continues even when the condensing surface temperature reaches below the freezing point. It is very likely that these facts are due to errors in measurement caused mainly by the shape of the block and the way of controlling the surface temperature. The effect of non-condensable gas may also be possible.

4 Conclusion

The condensation curves of steam at atmospheric pressure were measured taking the maximum drop diameter as the parameter. It was found that dropwise condensation continued up to considerably large subcooling, then shifted unsteadily to pseudo-film and on-ice condensation. The maximum heat flux as high as 10^7 kcal/m²h was observed.

Drop size distribution and rate of drop growth will be obtained from cinefilms and the mechanism of the shift of mode of condensation will be made clear.

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