Effect of Non-condensable Gas on the Vapor Flow in Thermosyphons

By

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1. INTRODUCTION

Recently Thermo-fluid dynamics in the enclosure with condensation and evaporation has been paid a wide attention in the various engineering fields, such as chemical plants, heat exchanger, or heat pipes, but a basic mechanism of condensation and evaporation has not yet fully established.

In this paper a detailed flow field survey was conducted to get a better insight into condensation mechanism of vapor flow onto the cooled plate in the enclosure. We used a two-dimensional thermosyphon as an experimental apparatus and paid a special attention to the effect of non-condensable gas in the thermosyphon. Generally it is understood that the non-condensable gas in the thermosyphon in its steady state operation occupies upper region of the condensation section, and that there exists an interfacial layer separating vapor flow from non-condensable gas which will move vertically according to the heat input rate applied to the thermosyphon [1]. However the structural properties of this interfacial layer and the boundary layer on the wall of condensation section, such as location of layers, thickness, temperature or density variation across them, have not been thoroughly investigated.

We have continued further investigation by using a real time laser holographic interferometer to visualize a flow field in the thermosyphon, and measuring its temperature distribution to evaluate the total heat transfer coefficient in the condensation section.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

2.1 Thermosyphon model

Figure 1 illustrates schematically the thermosyphon model used in this experiment. The vessel is made of Pyrex glass cylinder with 55 mm in inner diameter and 360 mm in length. In order to visualize a flow field a pair of parallel glass plates are installed as the optical windows. In the condensation section two parallel rectangular cooling panels are set backed with *U*-shaped pipes to be cooled by running water. An electric coil heater and a working fluid at the bottom of the vessel constitute an evaporator section. To measure temperature distribution in the flow six copper-constantan thermocouples are installed as shown schematically in Fig. 1. A real time laser holographic interferometer [3], [3] was used to visualize the flow field.

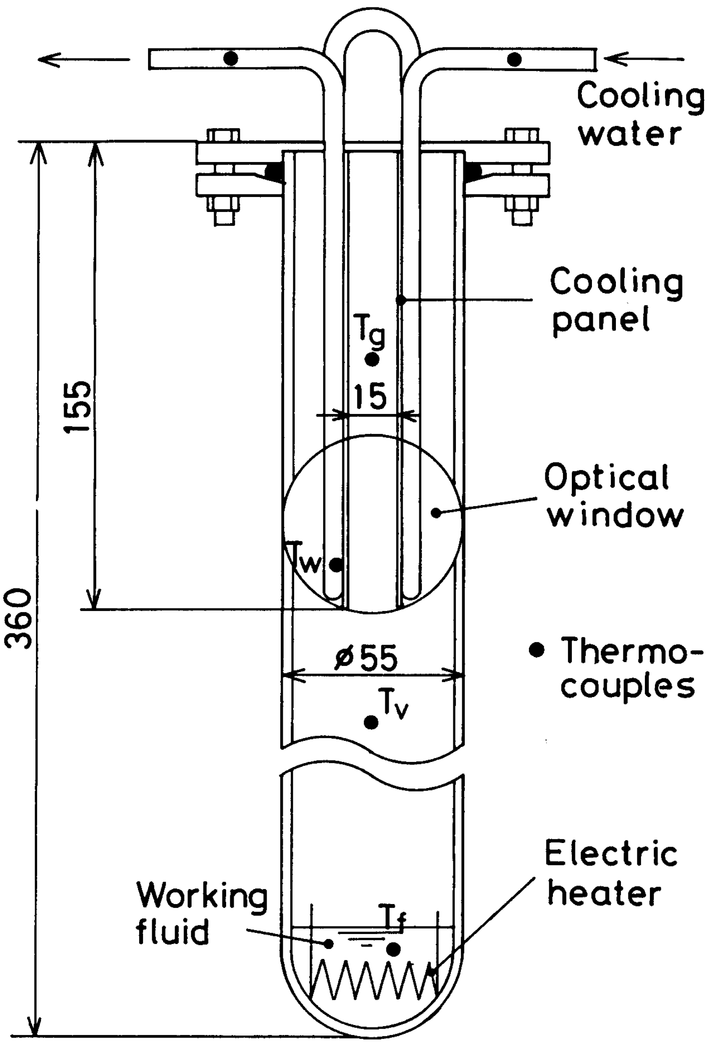


Fig. 1. Schematic view of the thermosyphon model.

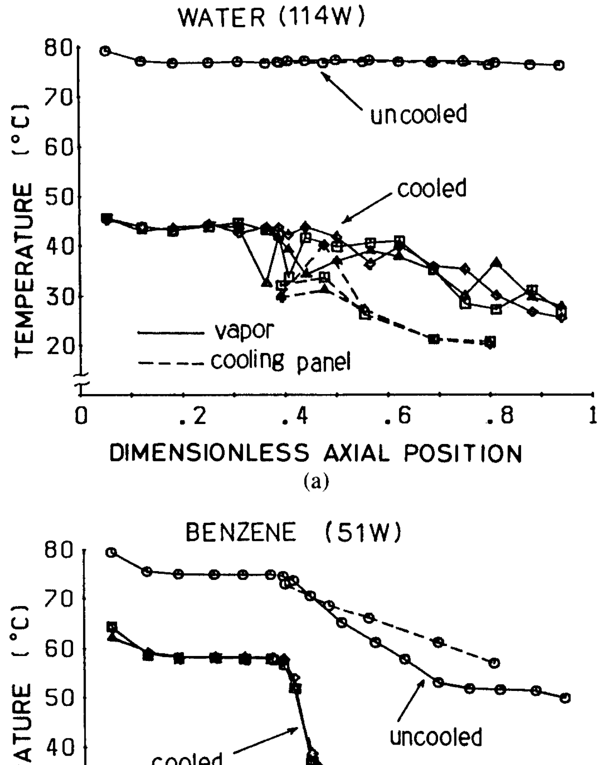
2.2 Measurements

Flow visualization and temperature measurement were carried out simultaneously for several conditions of heat input and partial pressure of contained non-condensable gas. Heat input was calculated from electric current and voltage applied to the electric heater. In the flow field visualization pure benzene was exclusively used as a working fluid, and ambient air, as a non-condensable gas, while in the measurement of temperature distribution, several other fluid were used, such as distilled water, acetone and Freon 11. The air was put into the thermosyphon using glass syringe after measuring its volume under a condition of 0.1 MPa and 20°C. Data of temperature and heat input were recorded by microcomputer-linked data acquisition system. All of the data and pictures were taken after the thermal steady state condition had been established in the thermosyphon. The heat input rates applied in our experiment are in the range of 15–370 W, and partial pressure of non-condensable gas are varied in the range of 0–15 KPa.

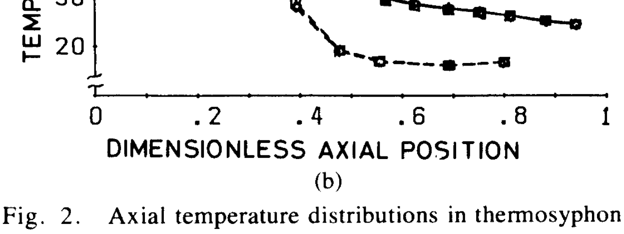
3. Results and Discussions

Figure 2 shows vertical temperature distributions along the centerline of the thermosyphon, taken by a series of thermocouple arrays specifically installed in the vessel. Figure 2(a) shows a distribution of distilled water as a working fluid, and Fig. 2(b) is that of benzene. The upper distributions in each figure indicate the temperature distributions when no cooling water is circulated, whereas the lower ones indicate those under cooling water circulation. In the distilled water a large temperature fluctuation can be seen even after a thermal steady state condition had been established in the vessel. A large temperature gradient at the lower end of cooling panels for benzene indicates the existence of an interfacial layer of vapor and non-condensable gas.

Figure 3 shows temperature changes in the vessel, where T_f, T_v, T_g and T_w represent temperatures of working fluid, vapor, non-condensable gas, and wall of a cooling panel, respectively, with heat input rate as a parameter. Somewhat large values of $T_{f/f\text{-}010}$ noticed when heat input is small is considered to indicate a transition of the pattern of evaporation, from surface boiling to nucleate boiling. Figure 3(a) shows the case of no non-condensable gas, and no large temperature differences are observed among the

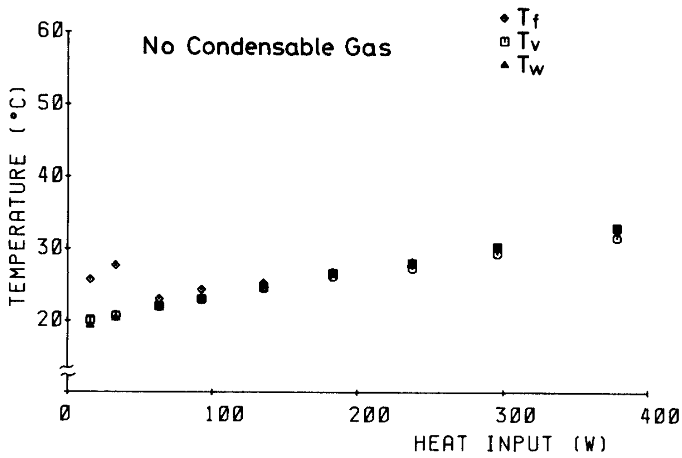


(a)

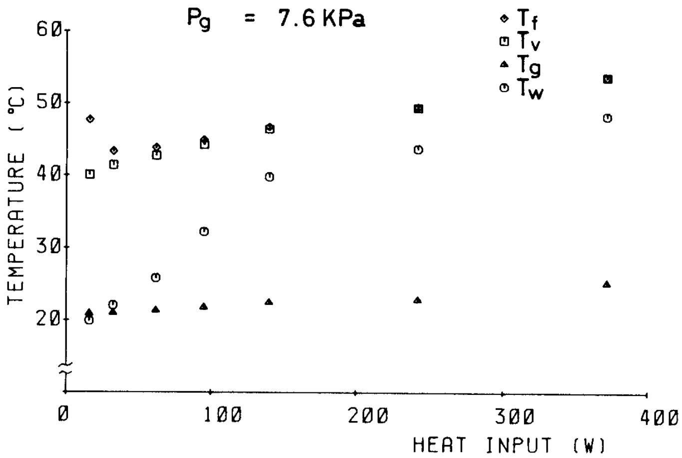


(b)

Fig. 2. Axial temperature distributions in thermosyphon.



(a)



(b)

Fig. 3. Temperature change in thermosyphon vs. heat input.
(a) without non-condensable gas.
(b) with non-condensable gas.

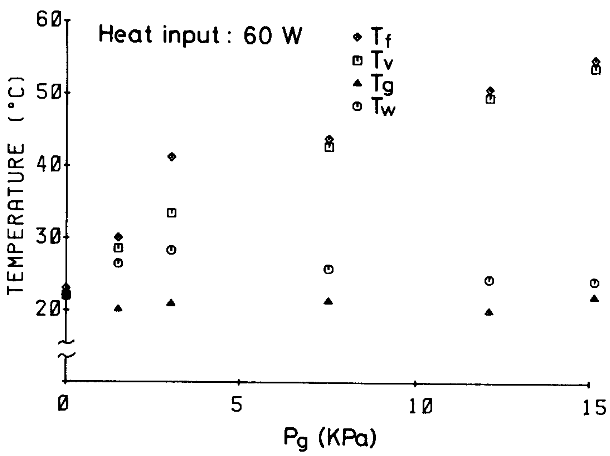


Fig. 4. Temperature distributions with partial pressure of non-condensable gas.

values of T_f, T_v, T_W. In the condition of containing 7.6 KPa of non-condensable gas larger temperature differences is observed between T_v and T_g, as shown in Fig. 3(b), and T_f and T_v seems to depend more on partial pressure of non-condensable gas rather than heat input rate. Figure 4 shows a temperature distribution under a condition of constant heat input with partial pressure of non-condensable gas as a parameter, where, T_f and T_v gradually rise with increasing partial pressure of non-condensable gas, whereas T_g remains constant. This tendency coincides with the data of vertical temperature distribution in Fig. 2, indicating that heat transfer from vapor to gas is very small and that non-condensable gas does not contribute much to heat transfer in the condensation section.

Two non-dimensional parameters of Reynolds number Re and Nusselt number Nu were used to arrange heat transfer rate and mean vapor flow velocity. Reynolds number is defined by (1),

$$Re=\frac{\rho_v U_v d}{\mu_v}, \tag{1}$$

where ρ_v and μ_v are density and viscosity of vapor of working fluid respectively, U_v, a mean velocity of vapor flow, and d, a distance between two cooling panels. A mean velocity, U_v, is evaluated from heat input H, latent heat L of working fluid, and cross sectional area of thermosyphon A_1, under the assumption that whole heat input is transferred to the condensation section in the form of latent heat, as expressed in the relation (2).

$$U_v=\frac{H}{A_1 L \rho_v}. \tag{2}$$

We could also evaluate U_v, from the temperature rise and mass flow rate of the cooling water before and after passing through the cooling panels. But the measurement error may be larger than those evaluated from heat input because of the relatively small values of temperature rises of about 0.1−1.0°C. A relative difference of heat transfer rates between two positions at the electric heater and at the cooling panel has been evaluated to be about 5−15% of the heat input rate.

Nusselt number at the condensation section is evaluated by (3),

$$Nu=\frac{hd}{k}=\frac{Hd}{A_2 k(T_v-T_w)}, \tag{3}$$

where k is thermal conductivity of a vapor, and A_2 is a total surface area of the cooling panels which is fixed constant regardless of an existence of non-condensable gas.

Figure 5 shows a relation of Nu and Re under several partial pressures of non-condensable gas and corresponding U_v in the range of 2−70 cm/s. Data of no non-condensable gas indicate relatively large Nu, whereas those with non-condensable gas indicate much lower values particularly when Re is small. That is, the effect of non-condensable gas on the heat transfer rate is large when vapor flow velocity is small.

Figure 6 shows Nu with respect to a ratio of partial pressure of non-condensable gas to total pressure of the vessel, P_g/P_t, where P_t stands for total pressure, and is esteemed as a saturated vapor pressure of the working fluid at T_v. Again Nu is observed to decrease with increasing non-condensable gas, and when heat input becomes large, value of Nu

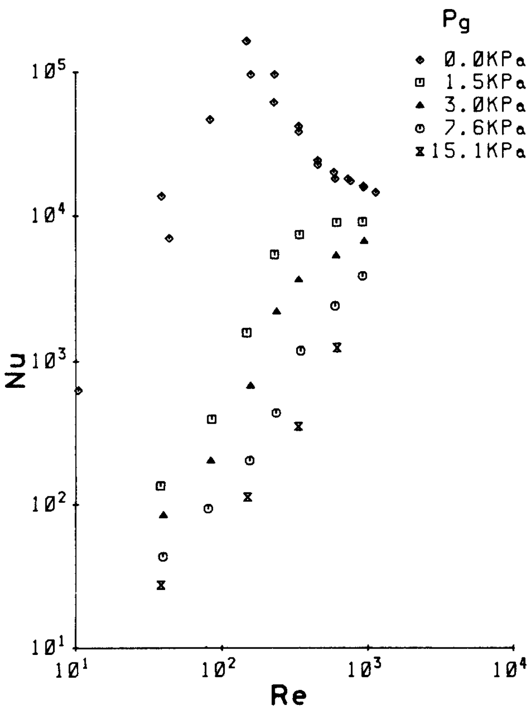


Fig. 5. Nusselt number vs. reynolds number.

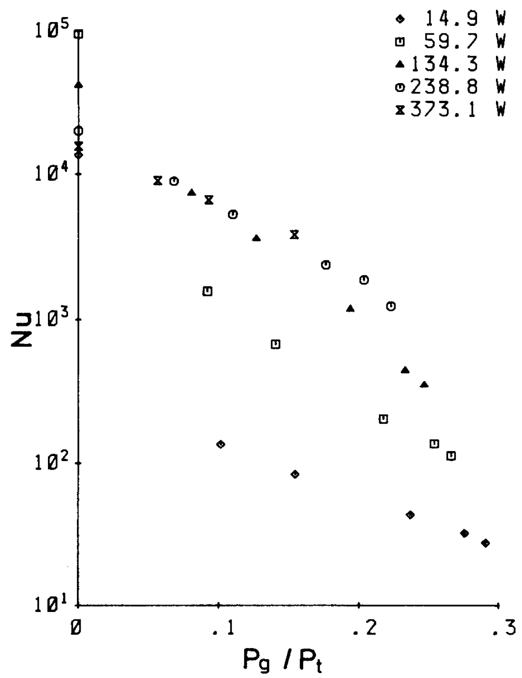
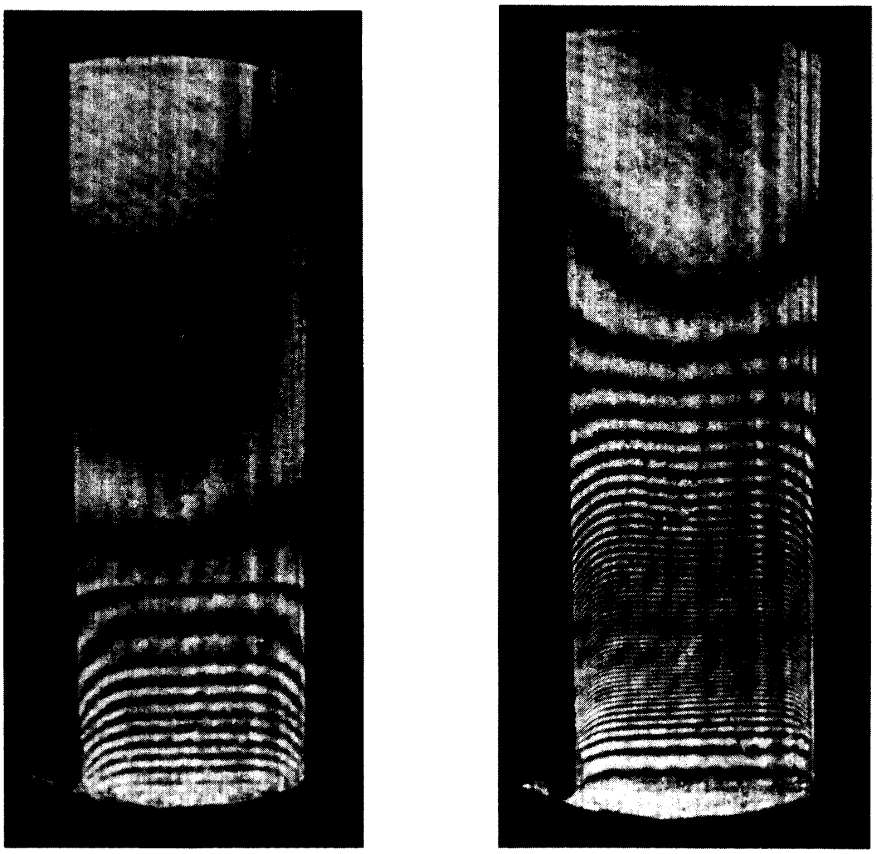


Fig. 6. Nusselt number vs. Pg/Pt.



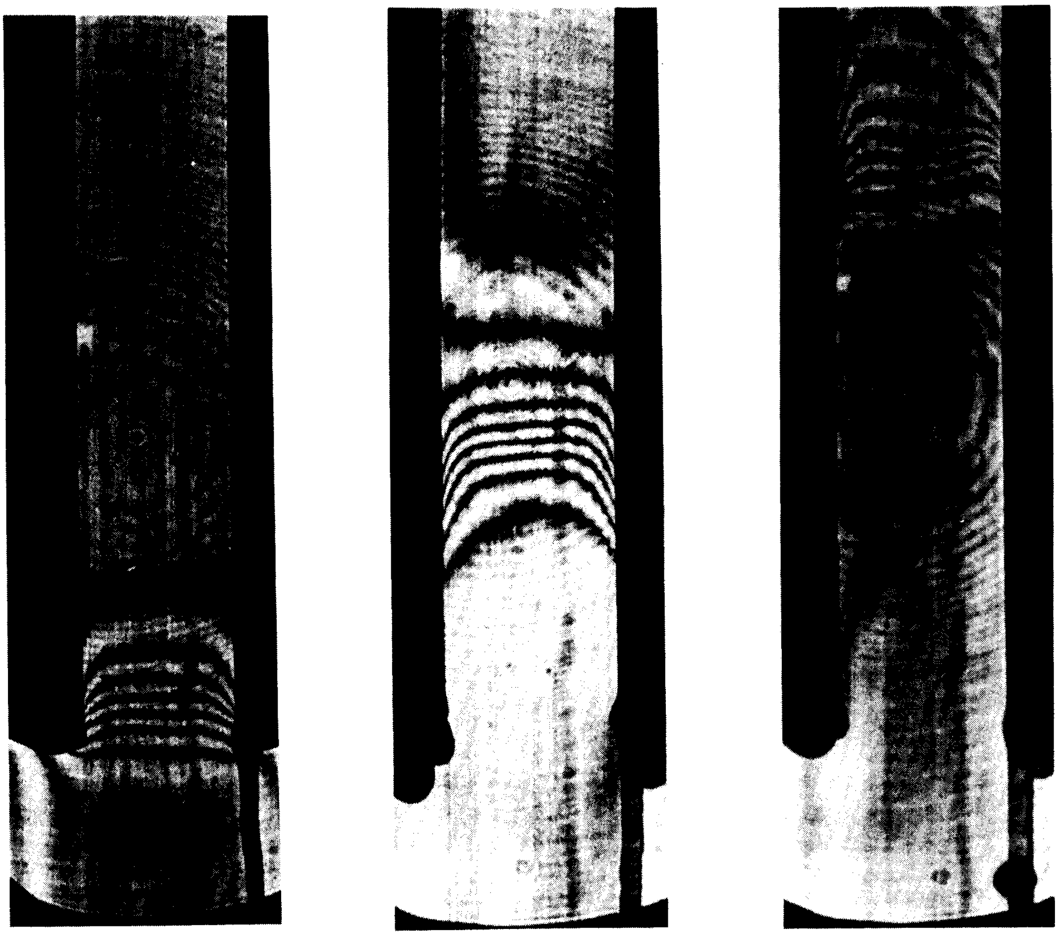
(a) ($H=16W$, $P_g=7.6KPa$) (b) ($H=60W$, $P_g=15.1KPa$)

Fig. 7. Density contour figure pattern of condensation section.

seems to depend more on P_g/P_t than on heat input rate. However further investigation should be due to get any conclusive remarks on these phenomena.

Figure 7(a) shows density contour fringe pattern, using benzene as a working fluid. Heat input rate and partial pressure of non-condensable gas are 16 W and 7.6 KPa, respectively. This picture views a lower end of about one-third of the entire condensation section. A relatively broad interfacial layer with large density gradient is seen at the lower end of cooling panels. Figure 7(b) shows a similar one with larger heat input and partial pressure of gas of 60 W and 15.1 KPa. It is observed in this case that an interfacial layer shifts upward and its thickness and density gradient across it are larger than those in Fig. 7(a). Further, a density boundary layer is clearly seen on the cooling panels and fringe line gradients along the surface of cooling panel are reversed in the vapor and in the gas region. As to these phenomena it is considered that differences in refractive index and densities of vapor and gas are closely involved. Quantitative interpretation of these fringe patterns is under study.

Figure 8 visualize movement of the interfacial layer with increasing heat input, and extension of condensation section which contributes heat transfer.



(a) (H=22W) (b) (H=60W) (c) (H=130W)

Fig. 8. Movement of interfacial layer according to heat inpt rate.

4. Conclusions

An interfacial layer between vapor and non-condensable gas was visualized, and its thickness was found to be about 2–3 cm in the case of present experimental conditions which increases with increasing heat input and partial pressure of non-condensable gas. A fairly thick boundary layer is observed on the wall of condensation section. A non-condensable gas contained in the condensation section affects adversely heat transfer rates of vapor flow in the thermosyphon particularly when flow velocity and /or heat input is small.

Further study on the relationship between refractive index and density or temperature of gas mixtures shall be continued to be able to evaluate density contour fringe pattern quantitatively.

References

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