Heat-Transfer Characteristics of the R113 Annular Two-Phase Closed Thermosyphon*

(Heat Transfer in the Condenser)

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Visual observation of flow patterns in the condenser and heat transfer measurements were conducted for heat transfer rate ranges of 18-800 W using a vertical annular device with various quantities of R113 as a working fluid. As a result of Visual observations, it was shown that ripples (interfacial waves) were generated on the condensate film surface when the condensate film Reynolds number exceeded approximately 20, and the condensation heat transfer was promoted. A simple theoretical analysis was presented in which the effects of interfacial waves and vapor drag were both considered. This analysis agreed very well with experimental results when the working fluid quantity was small enough so that the two-phase mixture generated by boiling the working fluid didn't reach the condenser. The effects of interfacial waves and vapor drag on condensation heat transfer were also investigated theoretically.

Key Words: Thermal Engineering, Thermosyphon, Heat Pipe, Film Condensation, Wavy Flow, Vapor Drag

1. Introduction

Very complex phenomena, such as evaporation, boiling and condensation, occur in the two-phase closed thermosyphon. Therefore, the majority of the existing studies have dealt with the over-all heat transfer and thermal performance of thermosyphons.

A few theoretical analysis have been conducted on the condenser performance, and the effect of counter-flowing vapor on condensation heat transfer was investigated. Spengel theoretically analyzed the laminar vapor-laminar condensate film flow (1), (2). In his analysis, Nusselt's theory of film condensation was applied for the condensate film, and a numerical analysis was made of the simplified two-dimensional Navier-Stokes equation for the vapor flow. A simple analysis based on Nusselt's theory considering the

interfacial shear stress was made by Takuma et al. (3) and Chen et al. (4). They showed that the vapor flow increased the condensate film thickness and decreased the local and average condensation heat-transfer coefficients. However, it was observed that the experimental results was higher than the theoretical one. One of the reasons for this discrepancy was considered as the presence of interfacial waves observed on the condensate film surface.

In recent years, the effect of interfacial waves on condensation has been theoretically investigated. Sofrata (5) and Uehara et al. (6) assumed the shape of an interfacial wave was a sine wave, and obtained a semi-empirical equation which they compared to the existing experimental equation of wavy film condensation. The effect of fluid motion caused by interfacial waves was considered by Suzuki et al. (7) and condensation heat-transfer coefficients were obtained for a wide range of condensate film Reynolds numbers.

In this paper, flow patterns of condensate film in the condenser are observed visually and condensation heat transfer characteristics are also investigated experimentally using a vertical annular two-phase

* Received 20 th July, 1987. Paper No. 85-1028 A

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closed thermosyphon. Based on the visual observations, a simple analysis considering both the effects of interfacial waves and vapor drag is made for the condensation heat transfer in the annular thermosyphon. The theoretical analysis is then compared with the experimental results.

Nomenclature

A_r : ratio of friction drag to vapor drag
 C_P : specific heat at constant pressure J/(kg·K)
 D_H : hydraulic diameter m
 D_i : inner diameter of the annulus m
 D_o : outer diameter of the annulus m
 f : friction factor
 g : gravitational acceleration m/s²
 h_{fg} : latent heat J/kg
 k : thermal conductivity W/(m·K)
 L : length m
 N_R : ratio of the condensation number to that predicted by Nusselt's theory of film condensation
 N^*_{tu} : condensation number
 Pr : Prandtl number
 Q : heat-transfer rate W
 q : heat flux W/m²
 Re : Reynolds number
 So : dimensionless number
 T : temperature K or °C
 ΔT : temperature difference between saturated vapor and cooled wall in the condenser K
 u : condensate film velocity in x-direction m/s
 V_m : average vapor velocity m/s
 V^* : working fluid fill rate ; volumetric ratio of working fluid to evaporator
 We : Weber number
 x, y : coordinates m
 α : heat-transfer coefficient W/(m²·K)
 β : dimensionless amplitude of interfacial wave
 Γ : mass flow rate of condensate film per unit periphery kg/(m·s)
 γ : liquid-vapor density ratio of working fluid
 Δ : characteristic length m
 δ : average condensate film thickness m
 ζ : ratio of sensible heat to latent heat
 η : dimensionless variable
 θ : dimensionless temperature
 λ : wavelength m
 μ : viscosity Pa·s
 ν : kinematic viscosity m²/s
 ξ : dimensionless number
 ρ : density kg/m³
 σ : surface tension N/m
 τ : shear stress N/m²

subscripts

c : condenser

i : liquid-vapor interface

L : liquid

m : average

N : value calculated by Nusselt's theory of film condensation

v : saturated vapor

x : local value

superscript

$*$: dimensionless number

2. Experimental Program

2.1 Experimental apparatus and procedures

The experimental apparatus used for flow visualization and the heat-transfer experiment is schematically shown in Fig. 1. This annular device was a concentrically-spaced pipe which consisted of a transparent Pyrex glass envelope with an outer diameter of 60 mm, wall thickness of 4.8 mm and total length of 750 mm, and a brass pipe with an outer diameter of 30 mm, wall thickness of 3 mm and total length of 750 mm. The heat-transfer surface was formed by this hollow brass rod with a built-in calorimeter at the top (condenser) and a built-in cartridge heater with a copper spacer at the bottom (evaporator). The effective working length of this thermosyphon was 700 mm.

Fig. 1 Schematic diagram of the experimental apparatus

mm, and the evaporator, adiabatic and condenser lengths were 240, 60, 400 mm, respectively. Fifteen copper-constantan (Cu-Co) thermocouples were soldered on the heat-transfer surface from the inside at 20 and 80 mm intervals in the evaporator and condenser, respectively, and the leads were guided along the calorimeter and spacer to the outside environment. A stainless-steel-sheathed Cu-Co thermocouple with an outer diameter of 1.2 mm was inserted through the top end-cap and used to measure the vapor temperature in the annular space. In order to minimize the heat leakage through the glass envelope during the operation of the thermosyphon, the temperature around the experimental device was kept almost equal to that of the vapor in the system. Prior to assembling the device, all parts were thoroughly cleaned with acetone and distilled water. Before operation, the system was vacuum-dried and checked for leaks.

A series of experimental data was taken with an almost constant operating temperature (vapor temperature 22–45 °C).

Power was supplied to the heater of the evaporator, and the operating temperature was adjusted by controlling the coolant flow rate and the inlet coolant temperature to the calorimeter. When a steady-state condition had been reached, all thermocouple readings were recorded on a data logger for one second every three minutes; the time-averaged temperatures were calculated, and the coolant flow rates were measured. Visual observation of flow patterns of the condensate film was also conducted simultaneously and still

photographs were taken by a single-lens reflex camera.

2.2 Observations of flow patterns in the condenser

Flow patterns in the condenser are not affected by the working fluid quantity, except that the quantity is large enough so that the two-phase mixture generated by boiling the working fluid reaches the condenser.

Photographs of flow patterns of the condensate film in the condenser are shown in Fig. 2 for the case of $V^+ = 0.3$. In Fig. 2-(a), the film Reynolds number $Re_L (= \frac{4\Gamma}{\mu_L})$ at the bottom of the condenser is equal to 7.6, and the condensate film surface is smooth. This experiment showed that weak ripples were observed on the condensate film surface when Re_L exceeded approximately 20. Ripples are clearly seen in Fig. 2-(b) ($Re_L = 30$) and small ripples (secondary waves) behind the large waves are recognized in Fig. 2-(c) ($Re_L = 112$). It is observed in Fig. 2-(d) ($Re_L = 313$) that the waves begin to lose their shape. Even in such a case, the condensate film does not seem to change into turbulent flow.

2.3 Heat transfer in the condenser

Figure 3 shows the condensation heat transfer which represents the relationship between the average heat flux in the condenser (q_m) and the difference between saturated vapor and average condenser wall temperatures ($T_v - T_c$). The solid line in this Figure indicates the analytical solution by Nusselt's theory of film condensation in which the effects of interfacial waves and vapor drag are neglected. Good agreements

Fig. 2 Flow patterns of condensate film in the condenser

are obtained in the case that temperature differences or heat fluxes are relatively small ($q_m \leq 4 \times 10^3 \text{ W/m}^2$). However, the experimental results become larger than the analytical solutions with increasing temperature differences. The reason for this is considered to be the presence of interfacial waves observed on the condensate film surface, which will be analyzed theoretically in the next section. The experimental data in the case of $V^+ = 1$ are considerably larger than those in the other cases. This may be due to the fact that the two-phase mixture generated by boiling of the working fluid in the evaporator reaches the central part of the condenser.

3. Theoretical Program

A simple theoretical analysis with the effects of vapor drag and interfacial waves which were observed experimentally is presented and these effects are investigated in this chapter. The theoretical analysis is compared with the experimental results.

3.1 Analytical model and basic governing equations

Since the condensate film thickness is extremely thin compared with the pipe radius, a two-dimensional orthogonal coordinate system can be applied. Figure 4 shows the analytical model and the coordinate system used in this study.

For the condensate film flow, neglecting both the inertia term of the momentum equation and the convective term of the energy equation, the simplified basic governing equations are obtained as follows.

$$\mu_L \frac{\partial^2 u}{\partial y^2} + g(\rho_L - \rho_v) = 0 \quad (1)$$

$$k_L \frac{\partial^2 T}{\partial y^2} = 0 \quad (2)$$

Boundary conditions are :

$$y = 0 ; u = 0 \quad (3)$$

Fig. 3 Relation between temperature difference and average heat flux in the condenser

$$y = 0 ; T = T_c \quad (4)$$

$$y = \delta ; \mu_L \left(\frac{\partial u}{\partial y} \right) = -\tau_i - \frac{d\Gamma}{dx} (V_m + u_i) \quad (5)$$

$$y = \delta_x ; T = T_v \quad (6)$$

Equation (5) represents the vapor drag which was introduced by Meyhew et al. (8). τ_i and $\frac{d\Gamma}{dx} (V_m + u_i)$ on the right-hand side of Equation (5) are the expressions of drag forces due to the friction and momentum variation at the vapor condensation, respectively. Hereafter, the total drag forces of the friction and momentum is defined as the vapor drag.

In order to simplify the analysis, the shape of interfacial waves is assumed to be given by the following equation applying the analytical methods used by Sofrata (5) and Uehara et al. (6) in which the wavy film condensation in the stagnant saturated vapor was investigated.

$$\delta_x = \delta \left(1 + \beta \sin \frac{2\pi x}{\lambda} \right) \quad (7)$$

In the above equation, δ is the local value of average condensate film thickness within a wavelength, and β is the dimensionless amplitude.

Integrating Eq. (2) under the boundary conditions of Eqs. (4) and (6), the temperature distribution in the condensate film is obtained as follows.

$$\frac{T - T_c}{T_v - T_c} = \frac{y}{\delta_x} \quad (8)$$

The local heat flux dq_x passing through the infinitesimal wave surface illustrated in Fig. 4 becomes Eq. (9) using Eq. (8).

$$dq_x = k_L \left(\frac{\partial T}{\partial y} \right) dx = k_L \frac{\Delta T}{\delta_x} dx \quad (9)$$

where $\Delta T = T_v - T_c$

Assuming that δ and β are constant within an

Fig. 4 Analytical model

arbitrary wave-length, one obtains the following equation as the average heat flux q_λ per unit wavelength using Eq. (9).

$$q_\lambda = \frac{1}{\lambda} \int_0^\lambda \frac{dq_x}{dx} dx = \frac{k_L \Delta T}{\delta(1-\beta^2)^{0.5}} \quad (10)$$

The local heat flux dq_δ passing through the average condensate film surface is given by Eq. (11). (5), (6).

$$dq_\delta = q_\lambda dx \quad (11)$$

The thermal balance of the condensate film yields

$$\frac{d\Gamma}{dx} = \frac{k_L \Delta T}{h_{fg} \delta(1-\beta^2)^{0.5}} \quad (12)$$

Integrating Eq. (1) under the boundary conditions of Eqs. (3) and (5) and using Eq. (12), the temperature distribution in the condensate film becomes

$$u = \frac{g}{\mu_L} (\rho_L - \rho_v) \left(y\delta - \frac{y^2}{2} \right) - \left[\frac{k_L \Delta T (V_m + u_i)}{h_{fg} \delta \mu_L (1-\beta^2)^{0.5}} + \frac{\tau_i}{\mu_L} \right] y \quad (13)$$

The mass flow rate Γ of the condensate film per unit periphery is

$$\Gamma = \int_0^\delta \rho_L u dy = \frac{g}{\mu_L} \rho_L (\rho_L - \rho_v) \frac{\delta^3}{3} - \frac{\rho_L \tau_i}{2\mu_L} \delta^2 - \frac{\rho_L k_L \Delta T (V_m + u_i) \delta}{2\mu_L h_{fg} (1-\beta^2)^{0.5}} \quad (14)$$

The following dimensionless numbers are introduced in order to nondimensionalize the above equations (4).

$$\gamma = \frac{\rho_v}{\rho_L}, S = \frac{x}{L_c}, \Delta = \left[\frac{\nu_L^2}{(1-\gamma)g} \right]^{1/3} \quad (15)$$

$$\zeta = \frac{C_p \Delta T}{h_{fg}}, \xi = \frac{\zeta}{Pr} = \frac{k_L \Delta T}{\mu_L h_{fg}}$$

$$\delta_N = (4\xi \Delta^3 x)^{1/4}, \eta = \frac{y}{\delta_N}, \delta^* = \frac{\delta}{\delta_N}$$

$$u^* = u / \left(\frac{4\nu_L^2 \xi x}{\Delta^3} \right)^{1/2}, V_m^* = V_m / \left(\frac{4\nu_L^2 \xi x}{\Delta^3} \right)^{1/2}$$

$$\Gamma^* = \int_0^{\delta^*} u^* d\eta = \Gamma / \left(\frac{64\mu_L^4 \xi^3 x^3}{\Delta^3} \right)^{1/4} \quad (15)$$

$$\tau_i^* = \tau_i / (4\rho_L g^4 \xi x \Delta^3)^{1/4}, \theta = \frac{T - T_c}{T_v - T_c}$$

Using the above equations, Eqs. (12), (13) and (14) are transformed into the following:

$$u^* = \left(\delta^* \eta - \frac{\eta^2}{2} \right) - \left[\frac{\xi \eta}{\delta^* (1-\beta^2)^{0.5}} (V_m^* + u_i^*) + \frac{\tau_i^*}{1-\gamma} \eta \right] \quad (16)$$

$$\Gamma^* = \frac{\delta^{*3}}{3} - \frac{\tau_i^* \delta^{*2}}{2(1-\gamma)} - \frac{\xi (V_m^* + u_i^*) \delta^*}{\delta^* (1-\beta^2)^{0.5}} \quad (17)$$

$$3\Gamma^* + 4S \frac{d\Gamma^*}{dS} = \frac{1}{\delta^* (1-\beta^2)^{0.5}} \quad (18)$$

Substitution of $\eta = \delta^*$ into Eq. (16) yields the condensate film velocity u_i^* at the liquid-vapor interface.

$$u_i^* = \frac{\frac{\delta^{*2}}{2} - \frac{\tau_i^* \delta^*}{1-\gamma} - \frac{\xi V_m^*}{(1-\beta^2)^{0.5}}}{1 + \frac{\xi}{(1-\beta^2)^{0.5}}} \quad (19)$$

Since the vapor flow path is closed by the end cap at the top of the condenser, vapor velocity and interfacial shear stress become zero there. $\delta^* = 1$ is considered because of the negligible momentum variation at the vapor condensation. The above consideration gives the following initial conditions.

$$S \rightarrow 0; \delta^* = 1, \tau_i^* = 0, V_m^* = 0, 1 - \beta^2 = 1$$

$$\Gamma^* = \frac{1}{3} - \frac{\xi}{4(1+\xi)}, u_i^* = \frac{1}{2(1-\xi)} \quad (20)$$

The vapor velocity, effect of interfacial waves and interfacial shear stress must be known in order to perform the calculation. As the mass conservation law can be applied at arbitrary positions in the thermosyphon, one obtains $V_m = 4\Gamma/(\rho_v D_H)$. Thus,

$$V_m^* = \frac{4\delta_N \Gamma^*}{\gamma D_H} \quad (21)$$

$$\text{where } D_H = (D_o^2 - D_i^2)/D_i \quad (22)$$

The effect of interfacial waves is given by the semi-empirical equation obtained by Uehara et al. (6).

In the case of $Re_L \leq 0.156 So$, no waves are observed.

$$1 - \beta^2 = 1 \quad (23)$$

In the case of $Re_L > 0.156 So$, waves are generated.

$$1 - \beta^2 = 0.47 We^{-0.138} \quad (24)$$

where,

$$Re_L = \frac{4\Gamma}{\mu_L} = 4\Gamma^* \left(\frac{\delta_N}{\Delta} \right)^3 \quad (25)$$

$$So = \left(\frac{3\sigma^3}{\rho_L^3 g \nu_L^4} \right)^{1/5} \quad (26)$$

$$We = \left(\frac{Re_L}{4So} \right)^{5/3} \quad (27)$$

The interfacial shear stress τ_i^* due to friction is given in the following manner using the friction factor for the annular flow path (9).

In the case of laminar vapor flow ($Re_v \leq 2000$):

$$f = \frac{16}{Re_v} \frac{[1 - D_i/D_o]^2}{1 + (D_o/D_i)^2 - \frac{1 - (D_i/D_o)^2}{1n(D_o/D_i)}} \quad (28)$$

In the case of transient vapor flow regime ($2000 < Re_v < 4000$):

$$f = \frac{Re_v^{-5.39 \times 10^{-2}}}{55.3} \quad (29)$$

In the case of turbulent vapor flow ($Re_v \geq 4000$):

$$f = 0.0919 Re_v^{-0.25} \quad (30)$$

Equation (29) was obtained by interpolating Eqs. (28) and (30). Using the above friction factors, τ_i , τ_i^* and Re_v become:

$$\tau_i = f \frac{\rho_v (V_m + u_i)^2}{2} \quad (31)$$

$$\tau_i^* = \frac{f}{2} \gamma (1-\gamma) \left(\frac{\delta_N}{\Delta} \right)^3 (V_m^* + u_i^*)^2 \quad (32)$$

$$Re_v = \gamma \left(\frac{\mu_L}{\mu_v} \right) \left(\frac{D_o - D_i}{\delta_N} \right) \left(\frac{\delta_N}{\Delta} \right)^3 (V_m^* + u_i^*) \quad (33)$$

The local condensation heat-transfer coefficient

α_x and local condensation number Nu^* are :

$$\alpha_x = \frac{k_L}{\delta(1-\beta^2)^{0.5}}, \quad Nu^* = \frac{\alpha_x(\nu_L^2/g)^{1/3}}{k_L} \quad (34)$$

Neglecting the effects of waves ($\beta = 0$) and vapor drag, the expression of condensate film thickness becomes $\delta = (4\xi\Delta^3x)^{1/4} = \delta_N$ which is identical to Nusselt's solution in stagnant vapor.

3.2 Numerical scheme

Equation (18) is integrated first by the Runge-Kutta-Gill method under the boundary conditions of Eq. (20). Equation (17) is solved for δ^* by the Newton-Raphson method, and local condensate film thickness is then obtained. Many values are calculated next using Equations (19) through (34). These Procedures are repeated until the bottom of the condenser. The present theoretical analysis can also be applied to the conventional two-phase closed thermosyphon having a circular cross section with slight modifications such as the use of a friction factor for a circular pipe.

3.3 Results of theoretical analysis

Numerical calculations were made for the same annular thermosyphon model as the experimental device.

Figure 5 shows the analytical result when the working fluid is R 113; operating (vapor) temperature and temperature difference in the condenser are $T_v = 320\text{K}$ and $\Delta T = 15\text{K}$, respectively. In this figure, the ratio N_R of the local condensation number of the present analysis to that predicted by Nusselt's theory increased suddenly, the position of the increase corresponds to that where the transition of condensate film from laminar to wavy flow occurs and the effect of interfacial waves appears. Condensate film thickness δ^* , interfacial velocity of the condensate film U_i^* and mass flow rate of the condensate per unit periphery Γ increase with the presence of wave. These trends are

Fig. 5 Analytical result (I)

the same as those of N_R . When the waves are generated, condensation is prompted and heat transfer rates increase. Consequently, the flow rates of vapor and condensate increase, and condensate film thickens.

A_T in Fig. 5 is the ratio of friction drag to vapor drag, and is defined by Eq. (35).

$$A_T = 1 / \left[1 + \frac{\xi(1-\gamma)(V_m^* + u_i^*)}{\delta^* \tau_i^* (1-\beta^2)^{0.5}} \right] \quad (35)$$

The momentum drag by vapor condensation occupies almost all of the vapor drag at the top of the thermosyphon, and the effect of friction drag τ_i^* appears in the range of $-1 \leq \log(S) \leq 0$ which is equivalent to 90% of the whole condenser.

Figure 6 is the analytical result for water as a working fluid under the same operating conditions as in Fig. 5. The trends in Fig. 6 are different from those in Fig. 5. The influence of vapor drag is much larger than the effect of interfacial waves. Especially, the influence of vapor drag appears noticeably on the interfacial velocity of the condensate film u_i^* which is retarded by the vapor drag. In the case of water, as the liquid-vapor density ratio of the working fluid is very small, the velocity of the rising vapor flow is much faster than that of the falling condensate flow.

The effects of temperature difference ΔT and operating temperature T_v on N_R are illustrated in Figs. 7 and 8 respectively. Heat transfer rates change with ΔT under the same working fluid and operating temperature. The position at which interfacial waves are generated moves upwards in the condenser with increasing ΔT and T_v . Even if T_v changes, there are no evident variations in heat-transfer rates but, in thermophysical properties of the working fluid and condensate film and vapor Reynolds numbers,

Figure 9 shows the effect of the kinds of working fluids on N_R for the case of constant T_v and ΔT . The

Fig. 6 Analytical result (II)

ΔT . The effect of interfacial waves is larger than that of vapor drag for R113, ethanol and R11, but the opposite tendency is seen for water. This may be due to the small liquid-vapor density ratio ($\gamma = 7.234 \times 10^{-5}$) of water at this operating temperature ($T_v = 320\text{K}$). The analytical results for water at $T_v = 400\text{K}$ ($\gamma = 1.460 \times 10^{-3}$) are shown in Fig. 10. The similar trends for freon and ethanol were obtained in Fig. 10 for water.

4. Comparison of the Theoretical Analysis with Experimental Results

The results of the theoretical analysis stated in the preceding section are related to the local values, while the experimental results shown in the experimental (§ 2) are related to the average values of the whole condenser. Accordingly, it is necessary to calculate the relations between heat transfer rates and temperature differences in the condenser using the present analysis. Figure 11 is the comparison of the theoretical analysis with experimental results in the

Fig. 7 Effect of ΔT on N_R

Fig. 8 Effect of T_v on N_R

Fig. 9 Effect of the kind of working fluid on N_R

Fig. 10 Analytical result (III)

Fig. 11 Comparison of theoretical analysis with experimental results

form of the average condensation number versus the condensate film Reynolds number estimated at the bottom of the condenser. In this figure, the solid line is the Nusselt's solution in which neither vapor drag nor interfacial waves are considered, the dot-dash line is the case with vapor drag only and the broken line is the present analysis with effects of both vapor drag and interfacial waves. The solid line overlaps the dot-dash line and it is indicated that the effect of vapor drag is negligible for R 113 with a large density ratio of vapor to liquid. It is also shown that the present analysis agrees very well with the experimental data except $V^+ = 1$. The disagreement between theory and experiment may be considered as the two-phase mixture generated in the evaporator which reaches the central part of the condenser. Relatively large deviations are seen for small condensate film Reynolds numbers. This is caused by the deterioration of the accuracy of temperature difference measurements with lower heat transfer rates and smaller temperature differences in the condenser and the coolant inlet and outlet.

5. Conclusions

In this paper, visual observations of flow patterns of the condensate film and condensation heat transfer characteristics were investigated using a vertical annular two-phase closed thermosyphon. Based on the visual observations, a simple theoretical analysis for condensation heat transfer with the effects of interfacial waves and vapor drag was presented. The following conclusions can be drawn from this work.

(1) When the condensate film Reynolds number exceeds approximately 20, interfacial waves (ripples) are observed on the condensate film surface and the condensation heat-transfer coefficient becomes higher than that predicted by Nusselt's theory of film condensation in the stagnant saturated vapor.

(2) Condensation heat transfer is not affected by the working fluid quantity, except that the quantity is large enough so that the two-phase mixture generated

by boiling the working fluid reaches the condenser.

(3) The present theoretical analysis agrees very well with the experimental results for R 113 as a working fluid.

(4) The liquid-vapor density ratio of the working fluid plays an important role in considering the effects of interfacial waves and vapor drag.

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