A Study on Flow Characteristics and Heat Transfer in Countercurrent Water and Air Flows*

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Flow characteristics, and the heat and mass transfer in a countercurrent annular two-phase flow in a vertical circular tube, where hot water flows downward as a film and air at room temperature flows upward, have been investigated. It is shown by the experiment that the pressure loss, and the heat and mass transfer rate in the gas phase increase due to the roughness effect of the interfacial wave. It is further shown that when the temperature of water is high, condensation of vapor occurs in the gas phase, and the heat transfer rate is enhanced by the release of latent heat accompanied by mist formation. A theoretical model based on the low Reynolds number k - ϵ turbulence model is proposed, where an additional production term is considered to incorporate the wave effect. The calculated results of the friction factor and the Nu and Sh numbers are compared with the experimental ones.

Key Words: Multiphase Flow, Convective Heat Transfer, Phase Change, Mass Transfer, Turbulence, Countercurrent Flow, Liquid Film

1. Introduction

Heat and mass transfer between downward-flowing hot water film and upward-flowing air is important in many industrial applications, such as in a cooling tower and a heat recovery process from waste hot water. In such flows, interfacial waves between gas and liquid affect the flow characteristics near the interface and augment the heat and mass transfer coefficients as well as friction loss. Therefore, many papers have been published regarding these problems. Moalem-Maron et al.⁽¹⁾ have been investigating interfacial waves from the viewpoint of liquid film motion, but they have not been concerned with the transport phenomena in the gas phase. Akai et al.⁽²⁾ analyzed horizontal liquid metal two-phase flow by using a two-

equation turbulence model without considering the special generation of turbulent energy caused by the interfacial waves. The condensation heat transfer was calculated by Suzuki et al.⁽³⁾ in the high liquid film Reynolds number region where the effect of interfacial waves is important. However, when the gas phase consists of multicomponents, the transport process is rate-controlled in the gas phase and is affected by an interfacial wave. From these standpoints, we investigate the effect of an interfacial wave on the gas phase. The characteristics of liquid film, including the time mean and fluctuation component of the liquid film thickness, and the frequency and celerity of the wave are measured under various gas and liquid flow rate conditions. The pressure drop, and the heat and mass transfer in the gas phase are measured and correlated to the interfacial wave characteristics to clarify the effect of the wave. In the case of hot water and air two-phase flow, the phase change from vapor to water in the gas phase, namely mist formation, has an important effect on the transport phenomena. As the vapor diffuses from the hot water surface into cold air, the vapor concentration exceeds the saturated value and condensation occurs under certain

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conditions. Release of latent heat due to condensation enhances the heat transfer in the gas phase. In order to clarify this effect on the transport phenomena, heat and mass transfer coefficients are measured for various water temperatures.

Following the experimental study, a theoretical model based on the low Re number k - ε model is proposed. The turbulence generation effect by the interfacial wave is considered as an extra production term in the transport equation of k . The effect of mist formation was taken into account by assuming thermodynamic equilibrium between the concentration of vapor and temperature. A numerical calculation was carried out and compared with the experimental results.

Nomenclatures

- a : thermal diffusivity, m^2/s
 d : diameter, m
 D : diffusivity, m^2/s
 C_f : wall friction factor
 c_p : specific heat, $\text{J}/\text{kg}^\circ\text{C}$
 f : frequency, Hz
 G : mass flow, kg/s
 h_{fg} : latent heat of vaporization, J/kg
 k : turbulent kinetic energy, m^2/s^2
 Nu : Nusselt number
 P : pressure, Pa
 Pr : Prandtl number
 r : radial direction coordinate, m
 Re : Reynolds number
 Re_l : liquid film Reynolds number ($=4\Gamma/\mu_l$)
 Sc : Schmidt number
 Sh : Sherwood number
 T : temperature, $^\circ\text{C}$
 U : velocity, m/s
 y : distance from the gas-liquid interface, m
 z : distance upward measured from the bottom of the tube, m
 Γ : liquid film mass flow rate per unit width, kg/sm
 δ : liquid film thickness, m
 ε : dissipation of the turbulent kinetic energy, m^2/s^3
 λ : thermal conductivity, $\text{W}/\text{m}^\circ\text{C}$
 μ : dynamic viscosity, kg/sm
 ν : kinematic viscosity, m^2/s
 ρ : density, kg/m^3
 ω : mass fraction
Subscripts
 b : bulk
 c : value at the center line of the tube
 f : property at the film temperature

- i : interface
 in : inlet
 l : liquid
 m : mean value
 out : outlet
 t : turbulent
 v : vapor
 w : interfacial wave

2. Experimental Apparatus

A schematic diagram of the experimental apparatus is shown in Fig. 1. The test section was a Pyrex glass tube, whose inner diameter and length were 25.1 mm and 1.7 m, respectively. Hot water was supplied downward along the inner tube wall from a sintered metal located at the top of the test section. On the other hand, air flow, which was controlled to keep a constant temperature of 20°C , was supplied upward from the bottom of the tube. The temperature of the falling liquid film was measured at three points along the flow direction by thermocouples embedded in the inner wall. The exhaust air temperature was measured immediately downstream of the water-inlet section by thirteen thermocouples in the same cross section. The bulk temperature was calculated by the weighting average assuming a $1/7$ power law for the velocity profile. In order to measure the bulk concentration of water including vapor and mist components in the exhaust air, the air was heated in the exit chamber to fully vaporize the water content and concurrently mixed. A part of it was sampled and analyzed by a gas chromatograph. Water concentra-

Fig. 1 Experimental apparatus

tion in the inlet air was also measured by the gas chromatograph. By these measurements, heat and mass transfer rates from the hot water film to the air were calculated. The exhaust air was observed by a laser beam to judge the occurrence of mist formation in the gas phase.

In order to clarify the effect of the interfacial wave on the air flow, the pressure drop and the liquid film thickness were measured. The pressures were measured by a static tube inserted in the tube at $z=0.7\text{m}$, where the air flow is fully developed, and by a pressure tap at the top of the tube. The liquid film thickness was measured by conductance probes, as shown in Fig. 2, at three locations along the flow direction. At each location, three probes were settled in the perimeter to check the axisymmetry of the liquid film. Further, two probes were located closely along the flow direction, and the wave speed was calculated from the cross-correlation of these two signals.

3. Experimental Results

3.1 Behavior of liquid film

Variations of the liquid film thickness at $z=650\text{ mm}$ with time are shown in Fig. 3 for three kinds of liquid flow rate without air flow. It is seen that the higher liquid flow rate brings a higher wave height and a sharper shape. In order to compare these differences quantitatively, variations of the mean liquid film thickness, $\bar{\delta}$, and the root mean square of the fluctua-

Fig. 2 Conductance probe system

Fig. 3 Liquid film thickness fluctuation

tion, δ' , with liquid flow rate are plotted in Fig. 4. The solid line is the liquid film thickness predicted from Nusselt's solution for laminar falling film. The mean liquid film thickness without air flow, shown by the white circles, is slightly less than the theoretical value. The value of δ' reaches 30~40% of $\bar{\delta}$ independent of the Reynolds number. This figure also shows the effect of upward air flow, indicated by triangular symbols. The air flow rate in this study was lower than that of the onset of flooding, which was coincident with the Wallis correlation⁽⁴⁾. The air superficial velocity of this figure is nearly equal to the upper limit of stable operation, however, there is no appreciable change in the mean film thickness due to air velocity. The amplitude of the film thickness fluctuation increases slightly with air velocity. The wave frequency and celerity of the liquid film without air flow are listed in Table 1. The effect of air flow on these parameters was also measured, although the influence is very small. These results are used later in the theoretical calculation.

3.2 Pressure drop

The interfacial friction factor is shown in Fig. 5 for three kinds of liquid flow rates as the function of the gas-phase Reynolds number including the single-phase air flow. The friction factor, C_f' , and the Reynolds number, Re' , are based on the relative velocity between air and the liquid film, where the liquid film velocity was calculated from the liquid flow rate and the mean film thickness. Data for the air single-phase flow agree well with the Blasius' friction law. On the other hand, the interfacial friction increases

Table 1 Wave parameters

$\Gamma_l, \text{kg/sm}$	Re_l	$\bar{\delta}, \text{mm}$	δ', mm	f, Hz	$u_w, \text{m/s}$
0.182	590	0.38	0.14	6.5	0.93
0.393	1270	0.49	0.17	5.0	1.28
0.605	1960	0.57	0.24	4.5	1.46

Fig. 4 Variations of the mean liquid film thickness and the R. M. S. of its fluctuation with liquid flow rate

with an increase of the liquid flow rate, and the degree of increase is about 1.3 times the wall friction of the smooth pipe. The correlation of the friction factor for the rough pipe⁽⁵⁾ is also shown by the solid curves, in which δ' is used as the equivalent sand roughness, k_s . It is seen that the effect of the interfacial wave is explained by using the correlation of the rough pipe to some extent, however the measured values are higher than the correlation in the low Reynolds number region. One of the major differences between the interfacial wave and the sand roughness is the moving of the interfacial wave relative to the mean liquid-film velocity. A theoretical analysis considering the wave velocity is carried out and compared with the experimental results in which the friction factor can be well predicted even in the small Reynolds number region, as shown later.

3.3 Heat and mass transfer

The heat and mass transfer coefficients from the liquid film to the air flow were measured for three levels of water inlet temperature, 30°C, 50°C, 70°C. In Fig. 6, the mass fraction of water in the exhaust air is plotted against the exhaust air temperature. When the

Fig. 5 Friction factor

Fig. 6 Relations between water concentration and temperature of the exhaust air

water temperature is high, mist is observed in the exhaust air. In Fig. 6, the closed and open symbols denote that mist was and was not observed, respectively, and the solid line shows the saturated vapor concentration. From this figure, it is confirmed that mist formation occurs when the vapor concentration exceeds the saturation value, thus, it is not necessary to consider supercooling of the vapor in the analysis.

The mist formation in the gas phase also occurs in the case of the condensation of air and the vapor mixture on a cooled wall⁽⁶⁾. The mechanism of the mist formation is explained as follows. When the temperature and the vapor mass fraction of air at the centerline are 20°C and 0.00453 (point A in Fig. 6, being the typical inlet condition in this experiment), the liquid film surface temperature is 30°C (point B), and the Lewis number is unity, the relation of the temperature and the vapor concentration at each point in the cross section lies on the straight line A-B; the vapor concentration is smaller than the saturated value everywhere in the pipe. However, when the surface temperature is either 50°C or 70°C, the vapor concentration in the cross section varies along the lines A-C and A-D, respectively; thus, the vapor becomes supersaturated locally and mist formation occurs.

The heat and mass transfer coefficients in the gas phase are shown in Figs. 7 and 8, respectively. In order to estimate the forced convective part of the heat transfer, the effect of transport by blowing at the gas-liquid interface is eliminated in the calculation of the Nu and Sh numbers as follows:

$$Nu = \frac{d_i}{\lambda_f} \frac{(G_{out} c_p T_{b,out} - G_{in} c_p T_{b,in} - \Delta G \cdot c_{pv} T_{im})}{(\Delta T)_{im} \pi d_i L}, \quad (1)$$

$$Sh = (1 - \omega_{vi}) \frac{d_i}{(\rho D)_f} \frac{\Delta G}{(\Delta \omega_v)_{im} \pi d_i L}, \quad (2)$$

Fig. 7 Comparison of experimental and calculated results for the Nu number

where ΔG is the total amount of vaporization, i.e., $(G_{out} - G_{in})$, and T_{im} is the mean temperature of gas-liquid interface. $(\Delta T)_{lm}$ and $(\Delta \omega_v)_{lm}$ are the logarithmic mean differences of the bulk and the interface between the inlet and outlet. In Eq. (1), the third term in the parentheses on the right-hand side is the sensible heat transferred to the gas phase by vaporization. The temperature of the interface is estimated from the measured temperature of the liquid film by using the following correlation for the heat transfer of the liquid film:

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (3)$$

It might be a rather crude estimation, however, the estimated temperature difference in the liquid film is less than 6% of the total temperature difference and thus the accuracy of the estimation does not affect the results significantly.

As shown in Figs. 7 and 8, in the case of no mist formation ($T_{lin} = 30^\circ\text{C}$), the heat and mass transfer coefficients are larger than the conventional correlations for the turbulent flow in a smooth pipe (Eq. (3)). The difference between them is considered as an augmentation effect of the interfacial wave. In the low Reynolds number region, the increase of the heat and mass transfer coefficients by the wave effect is smaller than those in the high Reynolds number region, as opposed to the friction factor. Further, in the case of $T_{lin} = 50, 70^\circ\text{C}$, heat transfer is enhanced by the effect of vapor condensation which releases latent heat in the gas phase. However, the mass transfer rate is not influenced greatly by the mist formation because the effect of the mist formation on the concentration profile of vapor is small as shown later. In these figures, the theoretical results obtained in the next chapter are also plotted as lines.

4. Theoretical Analysis

There have been many theoretical models for

Fig. 8 Comparison of experimental and calculated results for the Sh number

turbulent transport in annular two-phase flow, although most of them are concerned with co-current flow. Of the existing models, Suzuki et al. (7) introduced additional viscosity induced by a wave which was given by wave frequency and wave height. However, their model underestimated the friction factor when it was applied to the present experiment. Akai et al. (2) employed a low Re number k - ϵ model with a finite value of k at the interface to account for the wave effect. Since the boundary condition was successful for the liquid phase but failed for the gas phase, they used the wall function for a rough wall as the boundary condition of the gas phase. In this study, the low Re number k - ϵ model is employed to investigate the effect of additional production of turbulent kinetic energy due to a wave in the vicinity of the interface. As the heat resistance in the liquid film is small compared with the one in the gas phase, a numerical calculation is performed only in the gas phase.

The following assumptions are made.

1. The velocity field is fully developed because the tube is long enough compared with its diameter.
2. The effect of vaporization on the gas flow field is neglected.
3. Physical properties are constant and independent of the composition.

4.1 Flow field

In order to calculate the velocity field of air, the low Re number k - ϵ model by Jones and Launder (8) is employed as a fundamental equation. The governing equations are as follows.

Momentum equation;

$$0 = -\frac{1}{\rho} \frac{dP}{dz} + \frac{1}{r} \frac{\partial}{\partial y} \left(r(\nu + \nu_t) \frac{\partial U}{\partial y} \right) \quad (4)$$

Turbulent kinetic energy equation;

$$\frac{\partial k}{\partial t} = \frac{1}{r} \frac{\partial}{\partial y} \left[r \left(\nu + \frac{\nu_t}{\sigma_k} \right) \frac{\partial k}{\partial y} \right] + \nu_t \left[\frac{\partial U}{\partial y} \right]^2 + P_e - \epsilon - 2\nu \left[\frac{\partial \sqrt{k}}{\partial y} \right]^2 \quad (5)$$

Dissipation equation;

$$\frac{\partial \epsilon}{\partial t} = \frac{1}{r} \frac{\partial}{\partial y} \left[r \left(\nu + \frac{\nu_t}{\sigma_\epsilon} \right) \frac{\partial \epsilon}{\partial y} \right] + C_1 \frac{\epsilon}{k} \nu_t \left[\frac{\partial U}{\partial y} \right]^2 - C_2 f_2 \frac{\epsilon^2}{k} + 2\nu \nu_t \left[\frac{\partial^2 U}{\partial y^2} \right]^2 \quad (6)$$

Definition of eddy viscosity;

$$\nu_t = C_\mu f_\mu \frac{k^2}{\epsilon} \quad (7)$$

The dependency of proportional constants on the Re number;

$$f_\mu = \exp \left[\frac{-3.4}{(1 + R_t/50)^2} \right] \quad (8)$$

$$f_2 = 1 - 0.3 \exp(-R_t^2) \quad (9)$$

where $R_t = k^2/\nu\epsilon$.

The model constants are as follows.

$$C_\mu=0.09, \quad C_1=1.44, \quad C_2=1.92, \\ \sigma_k=1.0, \quad \sigma_\epsilon=1.3 \quad (10)$$

The boundary conditions at the interface ($y=0$) and the center of the tube ($r=0$) are given as

$$\left. \begin{aligned} y=0: U &= -U_{lm}, \quad k=0, \quad \epsilon=0 \\ r=0: \partial U/\partial y &= \partial k/\partial y = \partial \epsilon/\partial y = 0 \end{aligned} \right\} \quad (11)$$

where U_{lm} is the mean velocity of the liquid film, i. e., $\Gamma_l/(\rho_l \delta)$, and is given by the experimental data. In Eqs. (5) and (6), unsteady terms are included for convergence of iterative solutions to these coupled equations. Equations (4), (5) and (6) are solved iteratively for a given dP/dz .

In Eq.(5), the extra production term, Pe , is added to consider the turbulence generation effect by the interfacial wave. It is assumed that turbulence is produced by a velocity difference between the interfacial wave and the gas phase behind the wave due to flow separation as shown in Fig.9. The turbulent kinetic energy production behind the wave is considered to be proportional to $(U_h + U_w)^2/2$ and f_w , where U_h and U_w are the gas-phase velocity at the wave crest ($y=h$), and wave speed, respectively, and f_w is the frequency of the wave. Then,

$$\left. \begin{aligned} 0 \leq y \leq h: Pe &= (U_h + U_w)^2/2 \cdot f_w \\ y > h: Pe &= 0. \end{aligned} \right\} \quad (12)$$

The wave height, h , is expressed as the function of the R. M. S. value of the liquid film thickness fluctuation, $h=1.73\delta'$. (13)

On the right-hand side of Eq.(13), the multiplier constant is determined by the calculated friction factor most fitting to the experimental results.

4.2 Heat and mass transfer

Variations of the temperature, and the concentration of vapor and mist along the flow direction are calculated by the following equations:

$$\rho c_p U \frac{\partial T}{\partial z} = \frac{1}{r} \frac{\partial}{\partial y} \left[r \rho c_p \left(a + \frac{\nu_t}{Pr_t} \right) \frac{\partial T}{\partial y} \right] + h_{fg} \dot{m}, \quad (14)$$

$$\rho U \frac{\partial \omega_v}{\partial z} = \frac{1}{r} \frac{\partial}{\partial y} \left[r \rho \left(D + \frac{\nu_t}{Sc_t} \right) \frac{\partial \omega_v}{\partial y} \right] - \dot{m}, \quad (15)$$

Fig. 9 Wave model

$$\rho U \frac{\partial \omega_l}{\partial z} = \frac{1}{r} \frac{\partial}{\partial y} \left[r \rho \frac{\nu_t}{Sc_t} \frac{\partial \omega_l}{\partial y} \right] + \dot{m}, \quad (16)$$

where $\dot{m}$ is the mist-formation rate per unit volume, and Pr_t and Sc_t are the turbulent Prandtl and Schmidt numbers. The boundary conditions are given as follows:

$$\left. \begin{aligned} y=0: T &= T_i, \quad \omega_v = \omega_{vs}, \quad \omega_l = 0 \\ r=0: \partial T/\partial y &= \partial \omega_v/\partial y = \partial \omega_l/\partial y = 0 \end{aligned} \right\} \quad (17)$$

In the above equations, Pr_t and Sc_t are taken to be equal to 0.8, which is determined by comparing the calculated results with the conventional correlation for fully developed pipe flow. As we did not measure the mist size, the turbulent Schmidt number of the mist diffusion is assumed to be equal to that of the vapor in Eq.(16), and any other mechanisms for controlling the motion of mist were neglected. The condensation rate, $\dot{m}$, is eliminated from Eqs.(14), (15) and (16), and the following equations are derived:

$$\begin{aligned} U \frac{\partial H}{\partial z} &= \frac{1}{r} \frac{\partial}{\partial y} \left[r \left(a + \frac{\nu_t}{Pr_t} \right) \frac{\partial H}{\partial y} \right] \\ &+ h_{fg} \frac{1}{r} \frac{\partial}{\partial y} \left[r (D - a) \frac{\partial \omega_v}{\partial y} \right], \end{aligned} \quad (18)$$

$$\begin{aligned} U \frac{\partial \omega}{\partial z} &= \frac{1}{r} \frac{\partial}{\partial y} \left[r \left(D + \frac{\nu_t}{Sc_t} \right) \frac{\partial \omega}{\partial y} \right] \\ &- \frac{1}{r} \frac{\partial}{\partial y} \left[r D \frac{\partial \omega_l}{\partial y} \right]. \end{aligned} \quad (19)$$

In Eqs.(18) and (19), the total enthalpy, H , and the total water concentration, ω , are defined as follows:

$$H = c_p T + h_{fg} \omega_v, \quad (20)$$

$$\omega = \omega_v + \omega_l. \quad (21)$$

Since thermodynamic equilibrium is assumed on the condensation in the gas phase, the vapor and mist concentration are given by using the saturated vapor concentration, ω_{vs} , which is given as a function of total enthalpy;

$$\omega_{vs} = \omega_{vs}(H), \quad (22)$$

$$\left. \begin{aligned} \omega \leq \omega_{vs}: \omega_v &= \omega, \quad \omega_l = 0 \\ \omega > \omega_{vs}: \omega_v &= \omega_{vs}, \quad \omega_l = \omega - \omega_{vs} \end{aligned} \right\} \quad (23)$$

The convection terms in Eqs.(18) and (19) are discretized by the upwind difference scheme, and these equations are integrated along the flow direction numerically.

4.3 Results of calculation

Examples of the calculated profiles of velocity, turbulent kinetic energy and its production rate are compared between the results with and without liquid flow. As shown in Fig.10 in which dP/dz is the same, the turbulent kinetic energy and its production rate are increased near the interface due to an extra production term, and the velocity profile shifts down due to this effect as shown in Fig. 10(a). The extra production of the turbulent energy exists only in the vicinity of the interface, and its production rate is only

7% of the normal production term. However, it increases the normal turbulent energy production near the interface as shown in Fig. 10(b).

The calculated friction factor is shown in Fig. 11 with the aforementioned experimental results. The calculated result for a smooth tube without a liquid film agrees well with Blasius' friction law, although it is slightly higher at the low Re number region. The calculated results for the countercurrent two-phase flow agree well with the experimental ones including the dependency on the Reynolds number.

Typical calculated profiles of the temperature and concentration at $z=0.5$ m are shown in Fig. 12(a) for the cases with and without mist formation. In the case of $T_{lin}=30^\circ\text{C}$ where mist formation does not occur, the temperature and vapor concentration profiles are almost the same. On the contrary, in the case of $T_{lin}=50^\circ\text{C}$ where mist formation does occur, the profiles differ significantly from each other because the vapor concentration is directly determined from the temperature profile by a saturation condition. It is seen that the vapor concentration is less affected by mist formation, which is the reason why the Sh number is not affected greatly by mist formation as shown in Fig. 8.

(a) Velocity profile

(b) Profiles of the turbulent kinetic energy and its production rate

Fig. 10 Calculated results for the flow field

The temperature profile at the top of the tube is shown in Fig. 12(b) with its measured values. As the temperature difference in the cross section is small at this location, the difference caused by mist formation is small, which is observed for the calculated and

Fig. 11 Comparison of calculated and experimental results for the friction factor

(a) $z=0.5$ m

(b) $z=1.73$ m

Fig. 12 Calculated results of temperature and concentration profiles

measured values. The calculated results of the Nu and Sh numbers are shown in Fig. 7 and Fig. 8, respectively. For $T_{lin}=30^{\circ}\text{C}$ without mist formation, the numerical results agree well with the experimental ones, as do the results for the friction factor. Therefore, it can be concluded that the modelling of the wave effect is appropriate. When mist formation occurs, i. e., $T_{lin}=50, 70^{\circ}\text{C}$, the calculated results can predict a marked increase of the Nu number, and an insensitiveness of the Sh number to the mist formation. Strictly speaking, however, the numerical results underestimate the Nu number. This is considered to be brought about by simplifications made in this analysis, for example, the neglect of the effect of vaporization on the flow, which causes some error in the high-water-temperature region. Such an effect must be considered for more accurate analysis.

5. Conclusions

Flow characteristics, and the heat and mass transfer in a countercurrent two-phase flow of hot water and air were investigated, and the following conclusions were obtained.

- (1) The increase of friction loss due to the interfacial wave can be roughly estimated by the correlation for the flow in the rough-wall tube.
- (2) The heat transfer coefficient increases remarkably due to the release of latent heat when condensation occurs in the gas phase. However, the Sh number is not affected by mist formation.
- (3) The turbulence model proposed in this study, which includes an extra production term, can predict

the friction loss and the heat and mass transfer coefficient including the wave effect, and also the effect of condensation in the gas phase.

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