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Experimental Study on Liquid Film Flow State and Heat Transfer Characteristics of a Falling-Film-Type Heat Exchanger

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Abstract- The falling-film-type heat exchanger used in this study is expected to achieve high efficiency in heat exchange by utilizing the falling film. With this heat exchanger, the heating medium flows inside the heat exchanger tube and the falling liquid film flows on the outer surface of the heat exchanger tube to exchange heat. Our previous studies revealed the basic heat transfer characteristics of the falling-film-type heat exchanger; developed design equations applicable to a limited scope; and demonstrated the possibility of improving performance by inserting a variety of shafts with screw blades into the heat exchanger tube. The present study focuses on the state of the liquid film flowing on the outer surface of the falling-film-type heat exchanger's tube, and examines the effect on heat transfer characteristics depending on: the diameter of the tube; and whether the tube is coated with Titanium dioxide (TiO_2) or not. The results showed that improving the liquid film flow state on the outer surface of the heat exchanger tube enhances the heat exchange performance.

Keywords – Falling-film-type heat exchanger, Heat exchanger tube, Liquid film, Liquid film flow state, Heat transfer characteristics

I. INTRODUCTION

Energy and resource conservation measures have become an urgent need in recent years due to escalating environmental destruction, the depletion of energy resources, soaring oil prices, and other issues. Therefore, the effective use of natural energy and resources, such as waste heat from factories, is considered to be a very important energy-saving measure, and the development of high-performance and high-efficiency heat exchangers for effective utilization of various types of energy is strongly desired. For this reason, this study focuses on falling-film-type heat exchangers.

Falling-film-type heat exchangers are a type of surface heat exchanger [1-3], in which a low-temperature liquid (or high-temperature liquid) flows inside a heat exchanger tube and a high-temperature liquid (or low-temperature liquid) flows on the outer surface of the tube as a liquid film for the purpose of heat exchanging. Highly efficient heat exchange performance is expected for falling-film-type heat exchangers by using falling liquid film as well as the heat of vaporization released from the surface of the liquid film to the external air when a high-temperature liquid flows down the tube. Since one of the liquids flows outside of the heat exchanger tube, maintenance of the tube is easy, and thus, falling-film-type heat exchangers are expected to be utilized for heat exchange of relatively corrosive liquids, as well as for heating/cooling technologies related to food products. However, there do not seem to be many studies reporting on the heat transfer characteristics of falling-film-type heat exchangers. The current design of falling-film-type heat exchangers generally applies the design method for the surface heat exchangers that consist of closed flow channels, with parts changed as necessary [4], and few experimental studies have been conducted on the heat transfer characteristics of falling-film-type heat exchangers. In addition, conventional studies do not take into account the effect of heat of vaporization released from the surface of the liquid film when high-temperature liquid flows down the tube, and thus the validity of the design method is yet to be fully evaluated. Therefore, the authors in the previous study examined the basic heat transfer characteristics of the falling-film-type heat exchanger and developed design equations that can be applied to a limited scope [5]. The authors also examined how the state of liquid flow inside the tube affects the heat transfer characteristics of the falling-film-type heat

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exchanger, and clarified the possibility of improving performance by inserting a shaft with a screw blade (hereafter referred to simply as "screw") inside the tube [6].

On the other hand, the present study focuses on the flow state of the liquid film on the outer surface of the falling-film-type heat exchanger tube and examines the effect of the outer surface's wettability on the heat exchange performance. More specifically, it examines how the different tube diameters as well as the presence or absence of Titanium dioxide (TiO_2) coating on the outer surface of the tube affect the heat transfer characteristics of the falling-film-type heat exchanger.

II. EXPERIMENTS

2.1 Experimental apparatus and methodology

Fig. 1 shows the setup of the experimental apparatus. The experimental apparatus consists of a falling-film-type heat exchanger, a falling system of heat-exchanged fluid (Distributor), and other measuring instruments. Two types of stainless steel round tubes shown in Fig. 2 and 3 were used for the falling-film-type heat exchanger tubes. Fig. 2 shows details of the large-diameter tube: a round stainless steel tube (wall thickness: t = 1.2 mm; length: l = 488 mm) with an external diameter d_o = 48.6 mm, which is the optimum diameter achieved in our previous study [6] with the stainless steel screw (diameter of screw shaft D = 38 mm, pitch of screw blade P = 100 mm, height of screw blade H = 3.5 mm) inserted inside the tube. The screw was made by: cutting out a screw blade (open annulus) for a pitch from a one-millimeter-thick stainless steel plate; pressing it for reshaping into the necessary form;

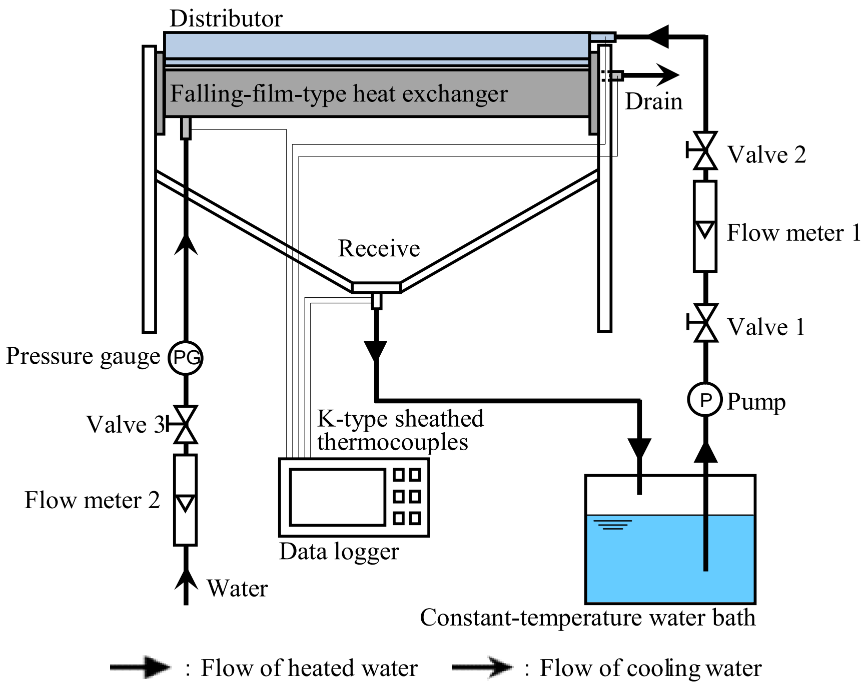

Figure 1. The experimental apparatus

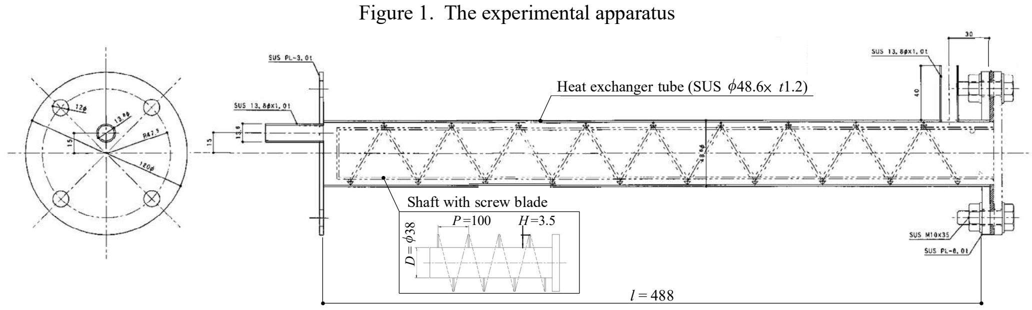

Figure 2. The large-diameter tube

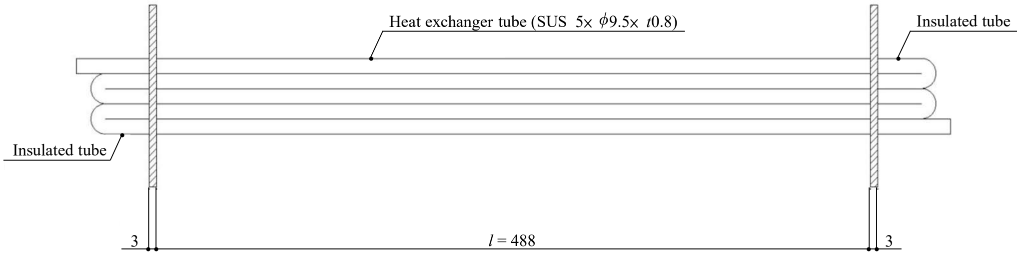

Figure 3. The small-diameter tube

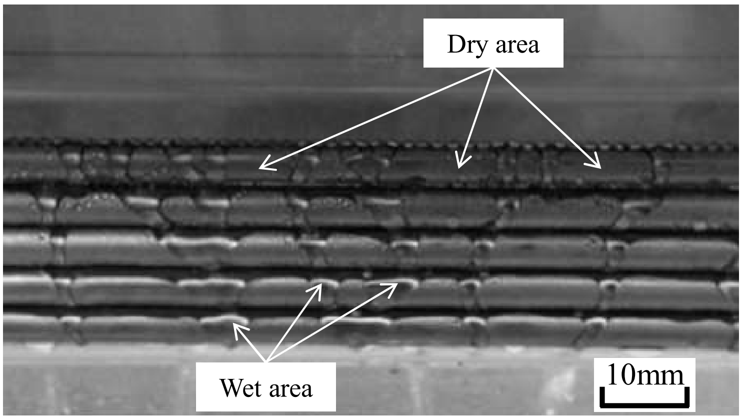

Figure 4. Condition of falling film in the case of small-diameter tube (TiO_2-uncoated tube)

preparing the required number of screw blades; and welding them to the screw shaft (hollow inside, closed at both ends). Fig. 3 shows details of the small-diameter tube: a round stainless steel tube (wall thickness: t = 0.8 mm, length: l = 488 mm) with an outer diameter d_o = 9.5 mm. Five tubes were placed in contact with each other and connected by insulated tubes in order to keep the outer surface area of the heat exchanger tube close to that of the large-diameter tube. The experiment was conducted on large- and small-diameter tubes with and without TiO_2 coating (Sagan Coat, Japan Photocatalyst Center) on the outer surface of the tubes. In the case of the large- and small-diameter tubes coated with TiO_2, the liquid film was formed uniformly over the entire outer surface of the tubes. On the other hand, in the case of the small-diameter tube without TiO_2 coating, the liquid film flowed down while splitting, resulting in dry patches, as shown in Fig. 4.

Table 1. Experimental conditions

Type of tube	Large-diameter d_o=48.6mm	Small-diameter d_o=9.5mm
Falling film	Heated water (50°C)	
Heat carrier	Cooling water (about 21°C)	
Flow rate of falling film G_h [L/min]	2.5, 5, 10	
Flow rate of heat carrier G_c [L/min]	5, 10, 15	0.85, 1.7, 2.5

As shown in Fig. 1, heated water (heat-exchanged fluid) that was maintained at a predefined temperature in the constant-temperature water bath was sent to the Distributor and flowed down as a liquid film on the outer surface of the falling-film-type heat exchanger tube. The heated water liquid film flow rate was controlled by Valve 1 and Flow meter 1. The heated water was cooled by the falling-film-type heat exchanger, collected into the liquid receiving unit (Receive), and returned to the constant-temperature water bath for circulation. Cooling water (heat carrier) was supplied from the tap, and flowed inside the heat exchanger tube, with the flow rate controlled by Flow meter 2 and

Valve 3. The cooling water that passed through the heat exchanger tube was drained. A digital thermo-hygrometer was used to measure room temperature and humidity. Temperatures were measured using K-type sheathed thermocouples at six locations: one at the inlet and outlet of the cooling water; one at the inlet of the heated water; two at the outlet of the heated water; and one above the apparatus for the room temperature. Temperatures were recorded and analyzed with a data logger (measurement accuracy: ±(0.05% of reading +1.0°C)).

2.2 *Experimental conditions*

Table 1 shows the experimental conditions. The heated water temperature was set at a constant 50°C, and to examine the effect of the liquid film flow state, the liquid film flow rate of heated water G_h was changed from 2.5 to 5, and then 10 L/min. The flow rate of cooling water G_c was changed from 5 to 10, and then 15 L/min for the large-diameter tube; and from 0.85 to 1.7, and then 2.5 L/min for the small-diameter tube, considering the Reynolds number (*Re*) inside the heat exchanger tube. That is, the Reynolds numbers inside the large-diameter tube (hydraulic mean diameter was calculated by assuming a rectangular flow path with a pitch of screw blade and height screw blade as its sides) and small-diameter tube (diameter of the round tube) were about Re = 1600 to 5000 and Re = 2600 to 8000, respectively, which enabled us to compare the heat exchange performance by the Reynolds numbers inside the heat exchanger tube. The inlet port temperature of the cooling water was about 21°C (19.0-23.9°C). The temperatures of each section reached a steady state two to three minutes after starting the experiment in any condition. Therefore, the temperature measurement duration in each condition was determined to be 10 minutes (measurement interval: one second), and the arithmetic mean value of the temperature data for the latter five minutes was used for the analysis of the experimental data in each experimental condition.

III. RESULTS OF EXPERIMENTS AND DISCUSSIONS

3.1 *Exchanged heat*

Fig. 5 shows the effect of the heat exchanger tube's external diameter on the exchanged heat Q in the case of the large- and small-diameter tubes without TiO_2 coatings. The exchanged heat Q was calculated using Eq. (1):

$$Q = (\dot{m}c)_h (T_{h1} - T_{h2}) \tag{1}$$

where $\dot{m}$ is the mass flow rate, c is the constant pressure specific heat of fluid, T is the temperature, subscript h is the heated water, subscript 1 is the inlet port, and subscript 2 is the outlet port. Density and constant pressure specific heat of cooling water were calculated by linear interpolation using a water properties table, where each property value was indicated by every 10 K of temperature [7]. The correlation coefficient was 0.98 between the exchanged heat Q on the heated water side calculated by Eq. (1) and the exchanged heat calculated by the temperature difference, etc. on the cooling water side.

In Fig. 5, the ranges of the Reynolds number Re between the large- and small-diameter tubes were different, but the comparison of exchanged heat Q was possible in the range Re = 2000 to 5000, where the Re in each tube overlapped.

Fig. 5 shows that the exchanged heat Q was larger in the case of the large-diameter tube than the small-diameter tube in the range Re = 2000 to 5000. This is considered because in the case of the small-diameter tube, the high Raynolds number was obtained at the relatively low flow rate and the heat capacity flow ratio was lower than that of the large-diameter tube; and because of the turbulent flow generated by the screw insertion, heat transfer on the cooling water side was promoted better in the case of the large-diameter tube than the small-diameter tube for the same Raynolds number.

Fig. 6 and 7 show the effect of the TiO_2 coating on the exchanged heat Q for large- and small-diameter tubes, respectively. The solid line in the figures represents the TiO_2-uncoated tube, and the dashed line represents the TiO_2-coated tube.

As shown in Fig. 6 and 7, the values of exchanged heat Q are larger for both tubes coated with TiO_2 than those without TiO_2 coating. This is considered because the application of TiO_2 improved the wettability of the outer surface of the heat exchanger tube, and thus the state of the liquid film reached the ideal state. In particular, the exchanged heat Q increased remarkably in the range where the liquid film flow rate of heated water G_h was small.

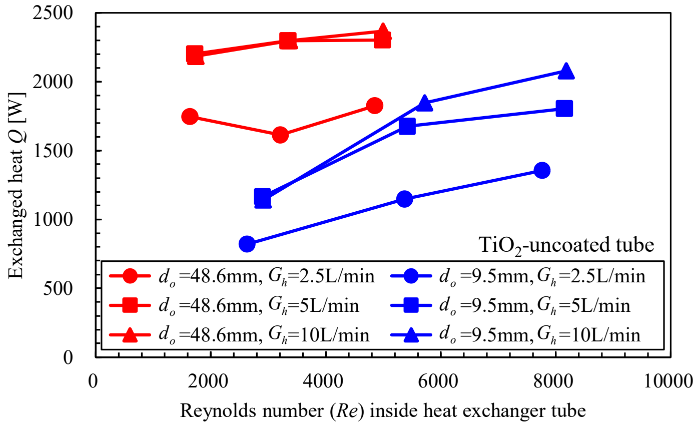

Figure 5. The effect of the external diameter of the heat exchanger tube on the exchanged heat Q (TiO_2-uncoated tube)

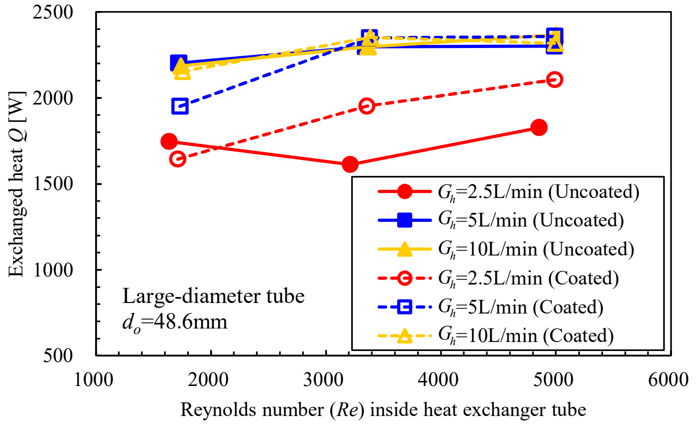

Figure 6. The effect of the coating TiO_2 on the exchanged heat Q (d_o = 48.6mm)

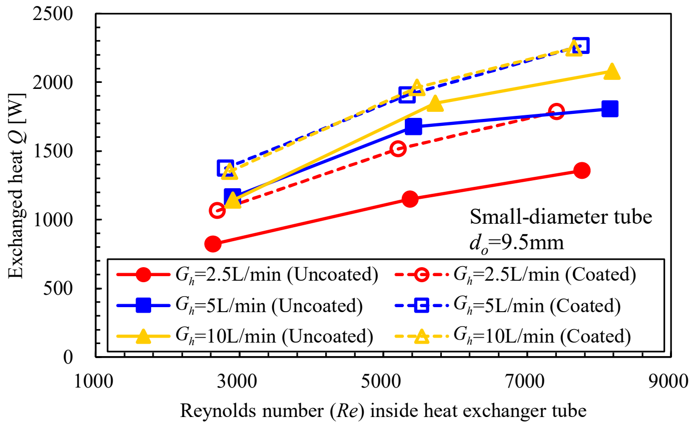

Figure 7. The effect of the coating TiO_2 on the exchanged heat Q (d_o = 9.5mm)

This result shows that the wettability improvement by the application of TiO_2 is more effective when the liquid film flow rate of heated water is smaller because when the flow rate is small, it is difficult for a uniform film to form over the entire tube.

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3.2 Overal heat transfer coefficient

Fig. 8 and 9 show the effect of the TiO_2 coating on the overall heat transfer coefficient K for large- and small-diameter tubes, respectively. The solid line in the figure represents the TiO_2-uncoated tube, and the dashed line represents the TiO_2-coated tube. The overall heat transfer coefficient K was calculated using Eq. (2):

$$K = \frac{Q}{A\Delta T} \tag{2}$$

where A represents the heating surface area, and ΔT represents the logarithmic mean temperature difference, which is calculated using Equation (3):

$$\Delta T = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln\left(\frac{T_{h1} - T_{c2}}{T_{h2} - T_{c1}}\right)} \tag{3}$$

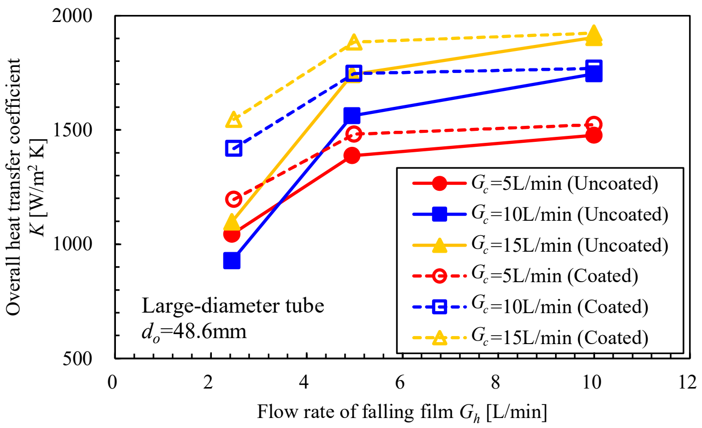

Figure 8. The effect of the coating TiO_2 on the overall heat transfer coefficient K (d_o = 48.6mm)

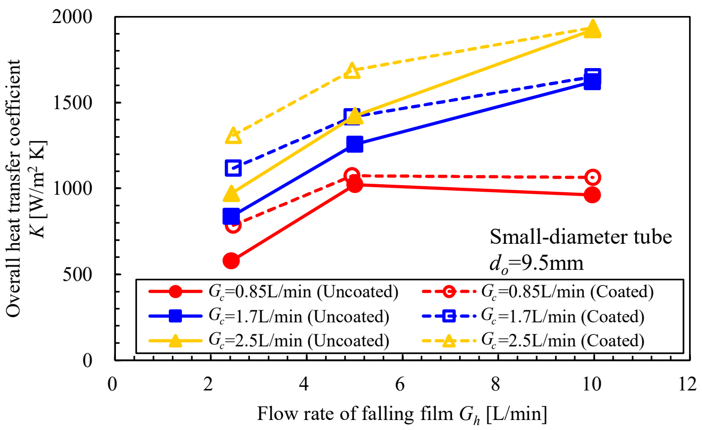

Figure 9. The effect of the coating TiO_2 on the overall heat transfer coefficient K (d_o = 9.5mm)

where subscript c represents cooling water.

As shown in Fig. 8 and 9, the application of TiO_2 improved the overall heat transfer coefficient K for both large- and small-diameter tubes. This is considered because the wettability improvement on the heat exchanger tube's outer surface reduced the liquid film thickness δ and increased the flow velocity, resulting in enhanced heat transfer on the liquid film side. In addition, in the range where the liquid film flow rate of heated water G_h was reasonably high, the increase of the overall heat transfer coefficient K that was achieved by the application of TiO_2 was small. This is considered because when the liquid film flow rate of the heated water was relatively high, sufficiently thick liquid films were formed even on the tube without TiO_2 coating, and the film flow velocity increased to a certain extent, suggesting that the effect of the wettability on the outer surface of the tube is small when the heated water liquid film flow rate is reasonably high. Moreover, comparing Fig. 8 and 9, the overall heat transfer coefficient K in the case of the large-diameter tube is mostly larger than that of the small-diameter tube.

3.3 Heat transfer coefficient on the liquid film side

This section discusses the heat transfer coefficient as a parameter governing the heat transfer mechanism on the liquid film side. As described in the previous section, the overall heat transfer coefficient K was increased by improving the wettability on the outer surface of the heat exchanger tube. This is considered because the liquid film thickness δ became thinner and the flow velocity became faster, resulting in enhanced heat transfer on the liquid film side.

Next, the value of the heat transfer coefficient on the liquid film side h_h was calculated using Eq. (4), which is a modified version of the equation associated with heat transfer passing through a circular tube:

$$h_h = \frac{1}{\frac{1}{K} - \frac{d_o}{2\lambda_t}\ln\frac{d_o}{d_i} - \frac{d_o}{d_i h_c}} \tag{4}$$

where h_c is the heat transfer coefficient on the cooling water side and λ_t is the thermal conductivity of the heat exchanger tube.

In the case of the small-diameter tube, the heat transfer coefficient h_c was calculated by using an equation that is commonly used to obtain the heat transfer coefficient for flow in a tube, and the heat transfer coefficient on the liquid film side h_h was obtained by substituting the value of h_c into Eq. (4). On the other hand, in the case of the large-diameter tube, since the screw was inserted inside the heat exchanger tube, applying the above equation could result in a large error, and the value of h_h needed to be obtained by a method different from that for the small-diameter tube. The following sections show calculation methods and values of the h_h for large- and small-diameter tubes.

3.3.1 In the case of the large-diameter tube

A graph was plotted with the experimental value for the inverse $1/K$ of the overall heat transfer coefficient K on the vertical axis; and the experimental value for the inverse $1/Re$ of the Reynolds number inside the heat exchanger tube on the horizontal axis, and the intercept, $(1/K)_{1/Re=0}$, of the linear approximation equation was calculated for each experimental condition. $1/Re = 0$ is relevant to the infinity of the Reynolds number inside the heat exchanger tube, which means the infinity of the flow rate of cooling water G_c. The heat transfer coefficient on the cooling water side h_c in this case is infinity, making the value of the $1/h_c$ in Eq. (4) zero. Therefore, the experimental value of the heat transfer coefficient on the cooling water side h_h for the large-diameter tube was obtained using Eq. (5):

$$h_h = \frac{1}{\left(\frac{1}{K}\right)_{1/Re=0} - \frac{d_o}{2\lambda_t}\ln\frac{d_o}{d_i}} \tag{5}$$

3.3.2 In the case of the small-diameter tube

The heat coefficient on the liquid film side h_h for the small-diameter tube was calculated using Eq. (4). However, the heat transfer coefficient K in Eq. (4) was calculated using experimental values, and the heat transfer coefficient on the cooling water side h_c was calculated using Eq (6).

$$h_c = \frac{Nu\lambda}{d_i} \tag{6}$$

where Nu is the Nusselt number and λ is the thermal conductivity of the fluid. The Nu was obtained using Eq. (7) developed by Hausen for laminar flow in a tube [8], and Eq. (8) developed by Dittus-Boelter for turbulent flow [9]:

$$Nu = 3.66 + \frac{0.0668}{\left\{(l/d_i)/(RePr)\right\} + 0.04\cdot\left\{(l/d_i)/(RePr)\right\}^{1/3}} \tag{7}$$

$$Nu = 0.023Re^{0.8}Pr^{0.4} \tag{8}$$

where Pr is the Prandtl number. The applicable range of Eq. (7) was $Re \leq 2300$, $10^{-4} \leq (l/di)/(RePr) \leq 10^4$, and that of Equation (8) was $0.7 \leq Pr \leq 160$, $2500 \leq Re \leq 1.24 \times 10^5$, and $l/di > 60$.

As a reference, the authors' previous study had already obtained Eq. (9), the estimating equation of the heat transfer coefficient on the cooling water side in the turbulent flow region of the large-diameter tube with a screw inserted [6].

$$Nu = 0.06Re^{0.8}Pr^{1/3} \tag{9}$$

Comparing this Eq. (9) with Eq. (8) used for the small-diameter tube, it is evident that the insertion of the screw into the large-diameter tube significantly improved the heat transfer on the cooling water side, which is considered to be the main reason why the exchanged heat Q and the heat transfer coefficient K for the large-diameter tube were greater than those for the small-diameter tube described in Section 3.1. and 3.2.

3.3.3 Calculation results

Fig. 10 shows the relationship between the heat transfer coefficient on the falling film side h_h and the flow rate of the falling film G_h. The solid line and dashed line in the figures represent TiO_2-uncoated and -coated tubes, respectively.

According to Fig. 10, the values of h_h show a tendency to increase as the values of G_h increase under all conditions, suggesting that the increased liquid film flow rate of the heated water G_h accelerates the velocity of the liquid film flow, which in turn improves the heat transfer. In addition, increased values of the heat transfer coefficient are observed in the case of TiO_2-coated heat exchanger tubes. As described in Section 3.2, this is considered to be due to the improved wettability on the outer surface of the heat exchanger tube, which led to the formation of a thinner and wider liquid film and made the flow velocity faster. In addition, a comparison of the experimental values between the large- and small-diameter tubes shows that the heat transfer coefficient on the liquid film side h_h for the small-diameter tube was mostly larger than that for the large-diameter tube. This is considered because the larger the curvature ratio of the heat exchanger tube, the faster the liquid film flows down on the tube.

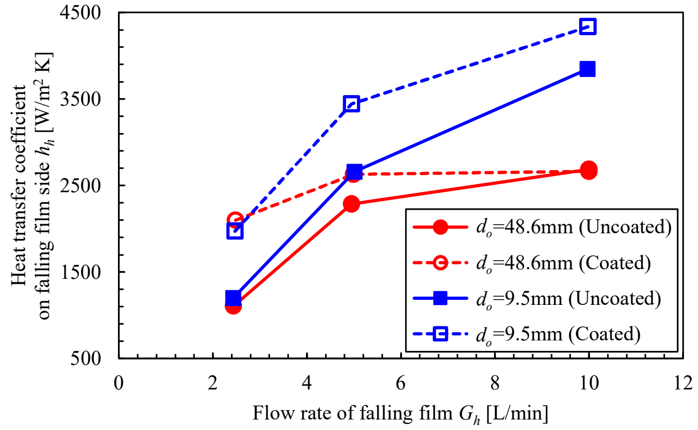

Figure 10. The relationship between the heat transfer coefficient on the falling film side h_h and the liquid film flow rate of heated water G_h

IV. Theoretical examinations and comparison with experimental values

4.1 Theoretical examination of heat transfer coefficient on the liquid film side

In the authors' previous study, the ideal heat transfer model of a liquid film shown in Fig. 11 was considered [6], which allowed approximating the assumptions of Nusselt's theory of liquid film [10] and [11], and then the theoretical Eq. (10) was developed to calculate the local heat transfer coefficient on the liquid film side h_x at an arbitrary position x on the outer surface of the round heat exchanger tube used in this study.

$$h_x = \frac{3\lambda \sin^{1/3}\varphi}{2\left\{\frac{240\mu\lambda d_o}{19\rho^2 cg}\int_0^{\varphi}\sin^{1/3}\varphi d\varphi + \left(\frac{3\mu\dot{m}}{2\rho^2 gl}\right)^{4/3}\right\}^{1/4}} \qquad (10)$$

where φ is the slope of the cooling surface, μ is the viscosity coefficient of the fluid, ρ is the density of the fluid, and g is the acceleration of gravity.

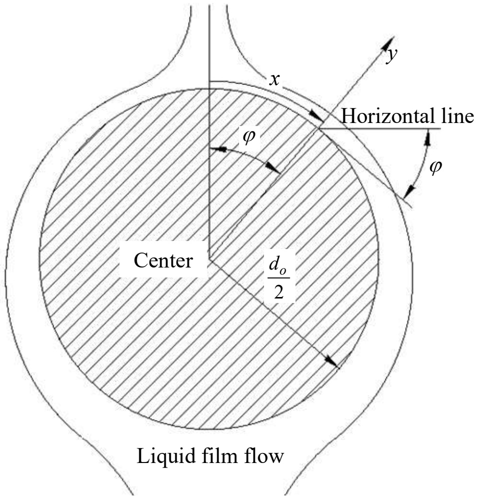

Figure 11. Cross-section of falling film

The local heat transfer coefficient on the liquid film side h_x was numerically calculated using Eq. (10). The average heat transfer coefficient h_h was obtained by calculating the local heat transfer coefficient on the liquid film side h_x for the entire surface of the heat exchanger tube and then averaging it.

4.2 Comparison between experimental and theoretical values

This section describes a comparison between the experimental values presented in Section 3.3.3 and the theoretical value of the heat transfer coefficient on the liquid film side h_h obtained from Eq. (10) (hereafter, "theoretical h_h value"). Fig. 12 and 13 show a comparison between experimental and theoretical values of the heat transfer coefficient on the liquid film side h_h for large- and small-diameter tubes, respectively. For reference, the figures also show the theoretical value of the heat transfer coefficient on the liquid film side h_h (hereafter, "Hoffman's h_h value") obtained by Hoffman's formula (11) [12].

$$h_h = 0.698\left(\frac{\left(\dot{m}/l\right)^{0.38}}{d_o^{\ 0.535}}\right)\left\{\frac{c^{0.535}\rho^{0.31}\lambda^{0.46}}{\left(\mu/g\right)^{0.155}}\right\} \tag{11}$$

As shown in Fig. 12 and 13, comparing the values for the large- and small-diameter tubes, the qualitative tendency is consistent with the heat transfer coefficient on the liquid film side h_h being larger in the case of the

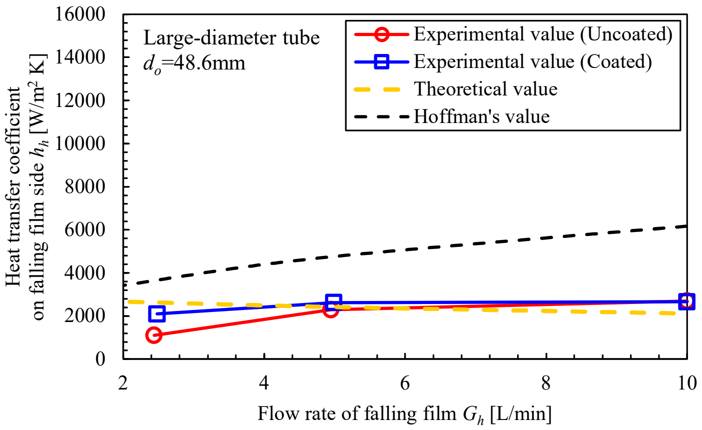

Figure 12. The comparison between theoretical and experimental values of the heat transfer coefficient on the falling film side h_h (d_o=48.6mm)

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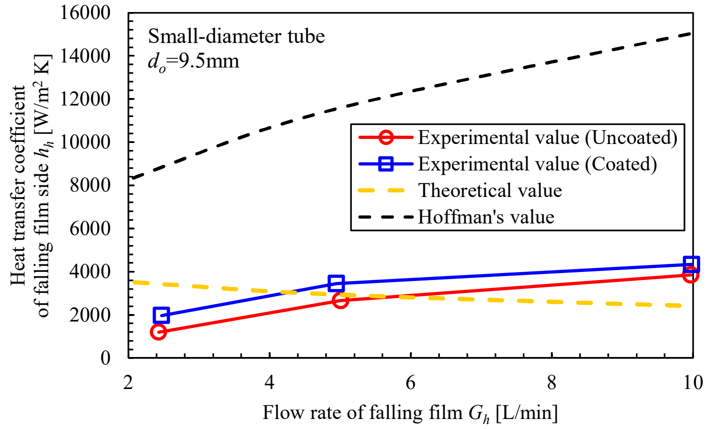

Figure 13. The comparison between theoretical and experimental values of the heat transfer coefficient on the falling film side h_h (d_o=9.5mm)

small-diameter tube than in the case of the large-diameter tube for all experimental, theoretical, and Hoffman's h_h values. In addition, for both large- and small-diameter tubes, the theoretical h_h values are closer to the experimental values than the Hoffman's h_h values. However, in the range where the liquid film flow rate of the heated water G_h is small, the theoretical h_h values show larger values than the experimental values.

One of the reasons for this could be that the liquid film was difficult to form uniformly in the range where the liquid film flow rate of heated water G_h was small due to the wettability of the heat exchanger tube in the experiment. Moreover, the theoretical h_h values decreased as the liquid film flow rate of the heated water G_h increased, which was a different tendency from Hoffman's h_h values and the experimental values. One of the reasons for this tendency is considered to be that the assumption of the liquid film flow being laminar when obtaining the theoretical h_h value was contrary to the actual status. In a laminar flow, the heat transfer amount due to convection is considered small. Therefore, when obtaining the theoretical h_h value, it is considered that a decrease of the heat transfer amount in the liquid film due to the liquid film thickness δ becoming larger has more effect than acceleration of heat exchange due to the increase of the flow velocity of the liquid film, as the liquid film flow rate of the heated water G_h increase. Hence, we consider the film Reynolds number Re_h [13] given by Eq. (12).

$$Re_h = \frac{4\dot{m}}{\mu l} \tag{12}$$

Here, it is generally known that the liquid film can maintain laminar flow at $Re_h \leq 1400$ [14]. In this experimental scope, the film Reynolds number Re_h calculated by Eq. (12) was Re_h = 520-2400, suggesting that, under a certain condition where the liquid film flow rate of the heated water G_h is large, the flow condition of the liquid film shifts from laminar to turbulent, as the surface of the liquid film becomes rippled. Therefore, the effect of heat transfer enhancement associated with the shift from laminar to turbulent flow is included in the experimental values, and in addition, it is known that wave-like turbulence occurs even in laminar flow regions in the actual phenomenon [15]; however, these effects were not considered when obtaining the theoretical h_h values.

Given the above reasons, further studies are needed in the future, such as studies into wettability improvement of the heat exchanger tube and a review of assumptions for obtaining the theoretical h_h value.

V. Conclusion

This study focused on the liquid film flow state on the outer surface of the falling-film-type heat exchanger's tube and examined the effects on heat transfer characteristics depending on the tube diameter and the presence or absence of TiO_2 coating applied on the outer surface of the tube. As a result, in the scope of this study, the following conclusion was obtained.

(1) For both large- and small-diameter tubes, both the experimental and theoretical values of the heat transfer coefficient on the liquid film side increased by applying TiO_2 coating to the outer surface of the heat exchanger tube to improve wettability, leading to increases in the exchanged heat and heat transfer coefficient.

(2) For the same Reynolds number inside the heat exchanger tube, the exchanged heat and heat transfer coefficient were larger in the case of the large-diameter tube than in the case of the small-diameter tube.

(3) Both the experimental and theoretical values of the heat transfer coefficient on the liquid film side were larger for the small-diameter tube than for the large-diameter tube.

(4) In our previous study, the theoretical formula to calculate the heat transfer coefficient on the liquid film side for the falling-film-type heat exchanger was obtained by applying the assumption of Nusselt's theory of liquid film.

In this study, for both large- and small-diameter tubes, the theoretical values calculated from the theoretical equation were close to the experimental values.

This study showed the possibility of improving the performance of the falling-film-type heat exchanger by enhancing the flow state of the liquid film on the outer surface of the heat exchanger tube. Further study will be needed to expand the experimental scope as well as to improve the theoretical formulas.

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NOMENCLATURE

A	Heating surface area [m^2]
c	Constant pressure specific heat of fluid [J/kg K]
d_i	Internal diameter of a heat exchanger tube [m]
d_o	External diameter of a heat exchanger tube [m]
D	Diameter of screw shaft [m]
g	Acceleration of gravity [m/s^2]
G_c	Flow rate of cooling water [L/min]
G_h	Liquid film flow rate of heated water [L/min]
H	Height of screw blade [m]
h_c	Average heat transfer coefficient on the cooling water side [W/m^2 K]
h_h	Average heat transfer coefficient on the liquid film side [W/m^2 K]
h_x	Local heat transfer coefficient on the liquid film side [W/m^2 K]
K	Overall heat transfer coefficient [W/m^2 K]
l	Length of a heat exchanger tube [m]
$\dot{m}$	Mass flow rate [kg/s]
Nu	Nusselt number
P	Pitch of screw blade [m]
Pr	Prandtl number
Q	Exchanged heat [W]
Re	Reynolds number inside a heat exchanger tube
Re_h	Film Reynolds number
T	Temperature [K]
t	Thickness of screw blade [m]
Greek symbols	
δ	Thickness of a liquid film [m]
ΔT	Logarithmic mean temperature difference [K]
λ	Thermal conductivity of fluid [W/m K]
λ_t	Thermal conductivity of a heat exchanger tube [W/m K]
μ	Viscosity coefficient of fluid [Pa s]
ρ	Density of fluid [kg/m^3]
φ	Slope of a cooling surface [°]
Subscripts	
c	Cooling water
h	Heated water
1	Inlet port
2	Outlet port

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