Heat Transfer and Pressure Drop in Double-tube Type Heat Exchanger with Rotating Blades

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Abstract

In order to enhance the heat transfer to heat and cool the process fluids of high viscosity or high heat sensitivity such as food, the axis with blades to agitate and mix the process fluid is rotated by force in the inner tube of double-tube type heat exchanger. In the experiment, the corn syrup water solution as a test process fluid is used, and the data of overall heat transfer coefficient and pressure drop were obtained by varying the flow rate and the viscosity of test fluid and the rotational speed of axis with blades. From these data, the heat transfer coefficient and the pipe friction coefficient for the test fluid flowing in the inner tube with rotating axis were calculated. The effects of axial Reynolds number, rotational Reynolds number, Prandtle number and viscosity gradient on Nusselt number and pipe friction coefficient were examined, and the characteristics of heat transfer performance and pressure loss were clarified. As a result, the correlations of Nusselt number and pipe friction coefficient were made in the case with rotation and without rotation respectively. The value calculated from these correlations can reproduce the experimental values by the accuracy within ±25% and ±30% respectively. In addition, the present correlation of Nusselt number was compared with the previously obtained correlations of Nusselt Number in the scraped surface heat exchanger and the triple pipe heat exchanger with smooth inner tube or finned inner tube respectively, and the range of application of the double-tube type heat exchanger was clarified in the heat transfer performance as not the contact surface heat exchanger like the scraped surface heat exchanger but the non-contact surface heat exchanger.

***Keywords**:* Heat Transfer, Pressure Drop, Double-tube, Heat Exchanger, Rotating Blade

1. Introduction

In the manufacturing process of food, the different types heat exchangers are used for heating and cooling needed in the process of sterilization. The double-tube type heat exchanger is one of the typical heat exchanger. However, it is necessary to attempt the heat transfer enhancement to heat and cool the process fluids of high viscosity or high heat sensitivity such as food. So we have tried to perform the experiments for double-tube type heat exchanger which has the axis with blades rotated by force to agitate and mix the process fluid in the inner tube. The overall heat transfer coefficients and the pressure loss were measured by varying the flow rate, the rotational speed of blades and the viscosity of the test fluid. It is purpose of this research to clarify in detail the effects of the rotation of blades on the heat transfer performance and the pressure loss. Based on the these experimental results, the correlations of Nusselt number and pipe friction coefficient were made, and the present correlation of Nusselt number was compared with the correlations [2,4] of Nusselt Number obtained previously in the several types heat exchangers. By the way, the heat exchanger with the direct contact surface of blades like the scraped surface heat exchanger has a higher heat transfer performance. However, the contact type heat exchanger has a serious problem in the safety of food because of the contamination by the metal powder etc. mixed in the process fluid. It is though that present research is useful to develop the non-contact surface heat exchanger in future.

2. Experimental apparatus and procedure

Fig. 1 shows a summary of the experimental apparatus. The test section is composed of double-tube heat exchanger of two stages, and the each rotational axis with blades is rotated by a gear motor respectively. The test fluid flows in the inner tube having the axis of rotating blades, and exchanges heat to the cooling water flowing in outer tube of the double-tube type heat exchanger. There are two tanks to store the test process fluid. The test fluid heated in one tank is circulated by a rotary pump of gear type and after passing in inner tube of the double-tube type heat exchanger, the flow rate is measured by another tank. The cooling water flowing in the outer tube of the double-tube type heat exchanger is supplied from the overflow tanks by a pump, and the flow rate of cooling water is measured by an ultrasonic flow-meter or a float type flow-meter. The temperatures in the inlet and the outlet of the test fluid and cooling water are measured respectively by sheathed CA thermocouples with a diameter of 1 mm. Also, the head of pressure loss between the inlet and the outlet of the test fluid is measured by a manometer of mercury. These measurements were carried out by changing the flow rate, the rotational speed of blades and the viscosity of the test fluid. Now the test process fluid and the cooling water are flowing mutually in counter direction.

Fig. 2 is a photograph of a rotational axis of outside diameter of 20 mm with blades of the

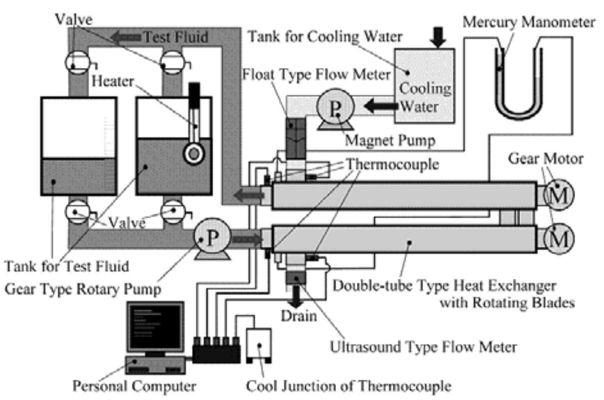

Fig. 1 Schematic diagram of an experimental apparatus

Fig.2 Photograph of an axis of rotation with blades

length of 380 mm. Two blades are welded alternately in the section with blades and in the section without blades at the intervals of 380 mm in an axial direction, and a set of two blades is orthogonally welded each other. Hence the rotation axis looks like having four blades of right angles mutually in a view from the axial direction.

The specifications of the test section in a double-tube heat exchanger are as follows.

Annular duct of test fluid : Inside diameter of 20.0 mm and outside diameter of 47.8 mm

Annular duct of cooling water : Inside diameter of 50.8 mm and outside diameter of 59.5 mm and heat transfer length of 1.87 m $\times$ 2 stages

Outside diameter of blade D : 43.2 mm

The experiments were carried out under the following conditions.

Test fluid as process fluid : corn syrup water solution

Viscosity of test fluid μ_p : 0.5 – 10 Pa$\cdot$s

Volume flow rate of test fluid : 25 – 643 ℓ/h

Temperature of test fluid : 14 – 50 $^\circ C$

Rotational speed N : 0, 12, 25, 75, 100 rpm

Prandtle number $\Pr$: 3000 – 50000

3. Definition of Heat Transfer Coefficient and Dimensionless Number

The overall heat transfer coefficient K in a double-tube heat exchanger is defined by the following equation.

$$K = Q_c / (A_p \Delta T_m) \qquad (1)$$

When the annular duct is approximated as the flat plate by disregarding the influence of curvature of circular tube, the overall heat transfer coefficient K can be expressed by the following equation.

$$1/K = 1/\alpha_p + \delta/\lambda_{sus} + A_p/(\alpha_c A_c) \qquad (2)$$

Based on the experimental data, K is calculated from Eq. (1). The heat transfer coefficients α_c of the cooling water is calculated from the following Eq. (3) of Dittus-Boelter [1] which can correlate the heat transfer coefficient very well in a fully developed turbulent region of circular tube.

$$Nu_c = \alpha_c de_c / \lambda_c = 0.023 \,\mathrm{Re}_c^{\,0.8} \,\Pr_c^{\,0.4} \qquad (3)$$

From Eq. (2) using these above-mentioned values, the heat transfer coefficients α_p of the test fluid flowing in the inner annular duct of the double-tube type heat exchanger can be evaluated.

Axial Reynolds number Re_p of the test fluid, Nusselt number Nu_p of these heat transfer coefficients, rotational Reynolds number Re_r and Prandtle number $\Pr$ are defined by the following equations, respectively.

$$\mathrm{Re}_p = V de_p / \nu_p \qquad (4)$$

$$Nu_p = \alpha_p de_p / \lambda_p \qquad (5)$$

$$\mathrm{Re}_r = D^2 N / (60\,\nu_p) \qquad (6)$$

$$\Pr = \mu_p c_p / \lambda_p \qquad (7)$$

The method of evaluating the physical properties of the test fluid is described in detail in our paper [2].

4. Experimental Results

4.1 Heat Transfer Coefficient

Fig. 3 shows Nusselt numbers Nu_p calculated by the above-mentioned method against Reynolds number Re_p. In this figure, Nu_p increase with an increase of Re_p in each rotational speed of blades. Especially, in the case of $N = 0$ rpm for the region of smaller Re_p less than 1, Nu_p almost agree with the theoretical values [1] for the heat transfer of laminar flow in an annular duct, that is, Nu_p =5.04 for heat flux constant and Nu_p =4.43 for wall temperature constant. However in the case of $N = 0$ rpm without rotation, Nu_p entirely increase with an increase of Re_p, and Nu_p are higher than the theoretical values because of the effect of flow in the region of Reynolds number more than $\mathrm{Re}_p \approx 1$. Also, Nu_p in the case with rotation generally indicate higher values than Nu_p in the case without rotation.

Fig. 4 shows the relation between Nusselt number $Nu_p / \mathrm{Pr}^{0.3}$ divided by 0.3 power of Prandtl and Reynolds number Re_p in taking the rotational speed N as a parameter. It is seen that the values of $Nu_p / \mathrm{Pr}^{0.3}$ increase in proportion to 0.3 power of Re_p in each rotational speed.

Fig. 5 shows the influence of the rotational speed N on Nusselt number Nu_p in the cases of some Reynolds numbers Re_p being constant as an example. It is confirmed that the values of Nu_p rise for an increase of N = 0 – 100 rpm, but the inclination of this increase has gradually become smaller with an increase of N rpm.

Fig. 6 shows values of Nusselt number $Nu_p / \mathrm{Pr}^{0.3} / \mathrm{Re}_p^{0.3}$ divided respectively by 0.3

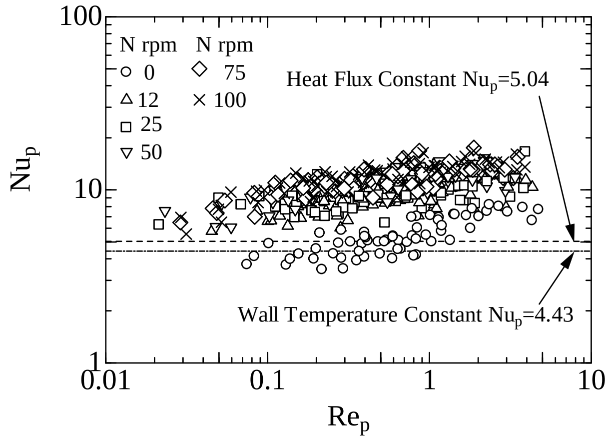

Fig. 3 Nusselt number vs. Reynolds number

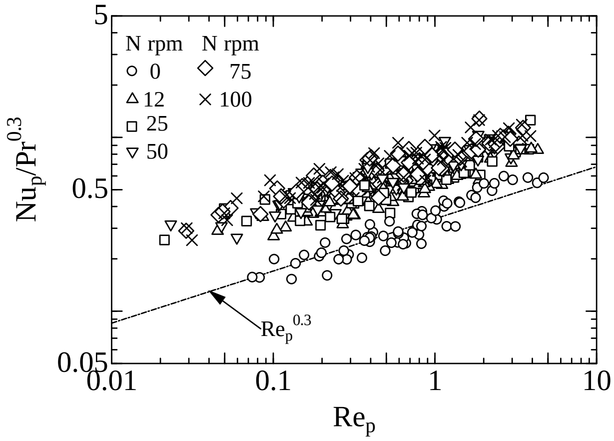

Fig. 4 Nusselt number divided by 0.3 power of Prandtl number vs. Reynolds number

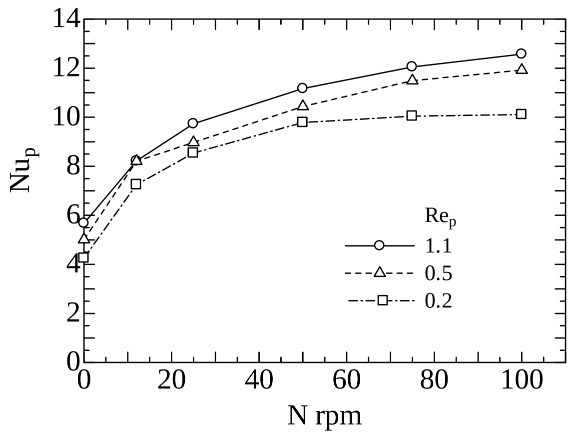

Fig.5 Effect of rotational speed of blades on Nusselt number

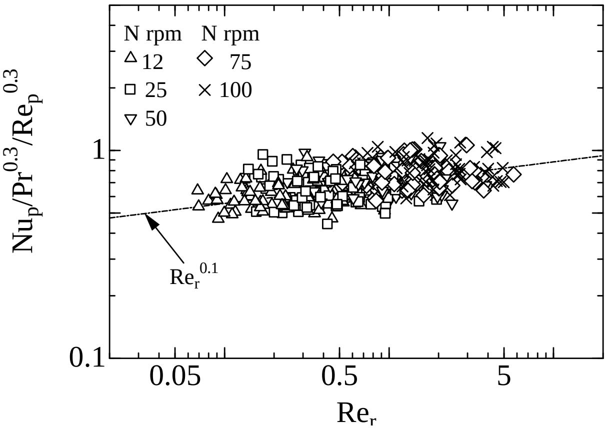

Fig. 6 Nusselt number divided respectively by 0.3 power of Prandtl number and Reynolds number vs. rotational Reynolds number

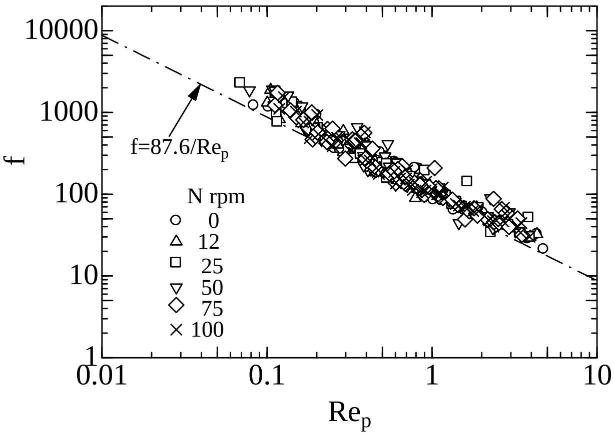

Fig. 7 Pipe friction coefficient vs. Reynolds number

power of Prandtl number and Reynolds number against rotational Reynolds number Re_r . The effect of rotational Reynolds number Re_r on $Nu_p / \mathrm{Pr}^{0.3} / \mathrm{Re}_p^{\ 0.3}$ is small, and it is seen that the values of $Nu_p / \mathrm{Pr}^{0.3} / \mathrm{Re}_p^{\ 0.3}$ increase in proportion to 0.1 power of Re_r in each rotational speed.

4.2 Pipe Friction Coefficient

The pipe friction coefficient f of the test fluid flowing in inner annular duct of the double-tube type heat exchanger was calculated from the following Eq. (8) by means of the measured data for the pressure loss ΔP .

$$f = 2\,\Delta P\, de_p \big/ (\rho_p\, L_f\, V^2) \tag{8}$$

Fig. 7 shows the relation between the pipe friction coefficient f and Reynolds number Re_p . In this figure, a theoretical equation [1] of pipe friction coefficient in the case of laminar flow in a smooth annular duct is indicated by an alternate long and short dash line. The pipe friction coefficients obtained in this experiment decrease in a relationship of inverse proportion with an increase of Re_p , and are generally higher in comparison with the values of theoretical equation in each rotational speed. This enlargement is thought to be an influence by installing the rotational axis with blades in inner tube of the double-tube type heat exchanger.

5. Correlation of Experimental Results

Based on the above characteristics of heat transfer and pressure loss, the correlations of Nusselt number and pipe friction coefficient were examined as follows respectively.

The experimental values of Nu_p in the case without rotation of blades are compared with the following Eq. (9) of Sieder and Tate [3], which takes into account the viscosity gradient of fluids in the tubes.

$$Nu_p = 1.86\,\mathrm{Pr}_p^{\ 1/3}\,\mathrm{Re}_p^{\ 1/3} \left(de/L_h \right) \left(\mu_w / \mu_b \right)^{-0.14} \tag{9}$$

The comparison of the experimental values Nu_p of Nusselt number with calculated values Nu_{cal} of Nusselt number by Eq. (9) are shown by the white symbol of circle in Fig. 8. It is found that the calculated values from the equation of Sieder and

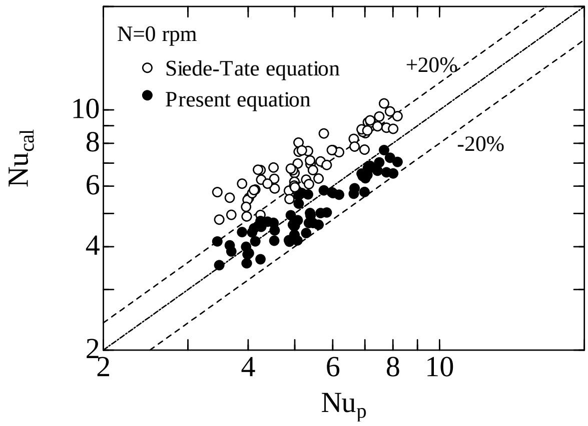

Fig. 8 Comparison of experimental values with calculated values of Nusselt number in case without rotation

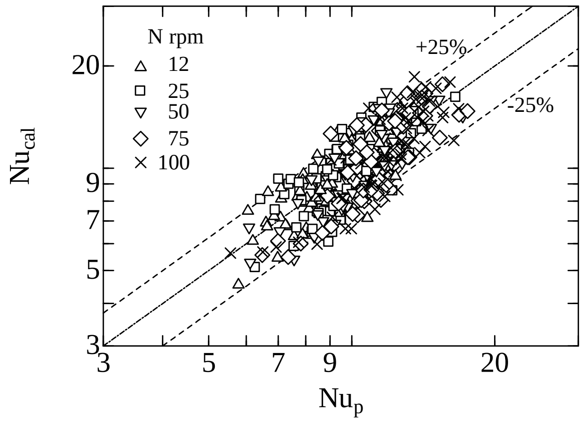

Fig. 9 Comparison of experimental values with calculated values of Nusselt number in case with rotation

Tate appear remarkably higher than the experimental values. Hence, an equation to correlate these experimental values of Nusselt number was made by the following Eq. (10) considering the effect of viscosity gradient μ_w/μ_b in the case without rotation.

$$Nu_p = 0.38\,\mathrm{Pr}_p^{\;0.3}\,\mathrm{Re}_p^{\;0.3}(\mu_w/\mu_b)^{-0.22} \qquad (10)$$

The comparison of the experimental values Nu_p with calculated values Nu_{cal} from Eq. (10) are shown by the black symbol of circle in Fig. 8. Both are well corresponding within an accuracy of ± 20 %. On the other hand, the experimental values of Nu_p of Nusselt number in the case with rotation of blades can be correlated by the following Eq. (11) considering the effect of viscosity gradient as well as Eq. (10).

$$Nu_p = 0.84\,\mathrm{Pr}_p^{\;0.3}\,\mathrm{Re}_p^{\;0.3}\,\mathrm{Re}_r^{\;0.1}(\mu_w/\mu_b)^{-0.22} \qquad (11)$$

Fig. 9 shows the comparison of the experimental values Nu_p of Nusselt number with calculated values Nu_{cal} from Eq. (11) in every rotational speed. It is confirmed that both are generally correlated within an accuracy of ±25 %.

Next, it is examined to correlate the experimental values of pipe friction coefficient. The following Eq. (12) was obtained by modifying a theoretical equation [1] of the pipe friction coefficient in the case of laminar flow in a smooth annular duct in consideration of the effect of viscosity gradient and being not depend on Re_r.

$$f = 1.14(87.6/\mathrm{Re}_p)(\mu_w/\mu_b)^{0.22} \qquad (12)$$

Fig. 10 shows the comparison between the experimental values f of pipe friction coefficient and calculated values f_{cal} from Eq. (12) in the case with rotation and without rotation. Both are generally corresponding within an accuracy of ±30 %. The numerical value of 1.14 in Eq. (12) is almost equal to the rate of an increase in the surface area by installing the blades on the axis of rotation.

By the way, the exponents of viscosity gradient in Eqs. (10) - (12) were given by -0.14 for Nusselt number in the case of cooling and

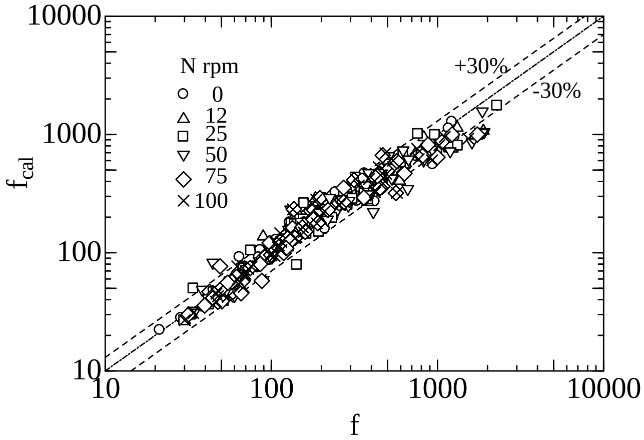

Fig. 10 Comparison of experimental values with calculated values of pipe friction coefficient in case with and without rotation

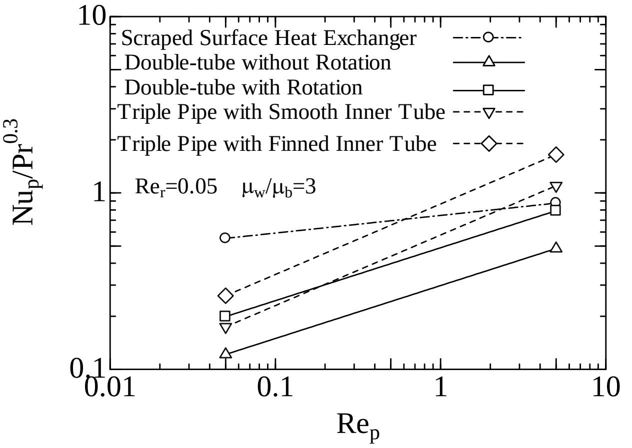

Fig. 11 Comparison of heat transfer performance in different types heat exchangers

0.5 for pipe friction coefficient in a reference [1], respectively.

6. Comparison of Heat Transfer Performance in Different Types Heat exchangers

In order to clarify the range of application of this double-tube type heat exchanger with rotating blades, an investigation was performed by comparing with the heat transfer performance of different types heat exchangers. They are scraped surface heat exchanger and triple pipe heat exchanger of which the heat transfer performance has been already clarified, and their correlations [2, 4] of heat transfer are given by the following Eq. (13) – (15) respectively.

For scraped surface heat exchanger

$$Nu_p = 4.5\,\mathrm{Pr}_p^{\ 0.3}\,\mathrm{Re}_p^{\ 0.1}\,\mathrm{Re}_r^{\ 0.6} \tag{13}$$

For triple pipe with smooth inner tube

$$Nu_p = \mathrm{Pr}_p^{\ 0.3}\,\mathrm{Re}_p^{\ 0.4}(\mu_w/\mu_b)^{-0.5} \tag{14}$$

For triple pipe with finned inner tube

$$Nu_p = 1.5\,\mathrm{Pr}_p^{\ 0.3}\,\mathrm{Re}_p^{\ 0.4}(\mu_w/\mu_b)^{-0.5} \tag{15}$$

Fig. 11 shows the comparisons of Nuselt number $Nu_p/\mathrm{Pr}^{0.3}$ obtained from Eqs. (10, 11) and Eqs. (13) – (15) against Re_p among three types of heat exchangers. Though this figure shows the particular case of lower Re_r of 0.05 as an example, it is seen that the heat transfer performance of the double-tube heat exchanger with rotation tends to approach the heat transfer performance of scraped surface heat exchanger as Re_p become higher. However, the heat transfer performance of scraped surface heat exchanger increases remarkably and the difference between both becomes larger with an increase of Re_r.

7. Conclusion

The overall heat transfer coefficient and the pressure loss were measured for cooling corn syrup water solutions as a test process fluid flowing in a double-tube heat exchanger with rotating blades. Based on these experimental data, the heat transfer coefficient of the test fluid in the inner annulus and the pipe friction coefficient were evaluated respectively.

As a result, the following conclusions were obtained.

(1) Nusselt numbers Nu_p increase in promotion to Prandtl number $\mathrm{Pr}^{0.3}$, axial Reynolds number $\mathrm{Re}_p{}^{0.3}$ and rotational Reynolds number $\mathrm{Re}_r{}^{0.1}$.

(2) Nusselt number can be increased about two times by rotating the blades.

(3) The pipe friction coefficients f are higher than a theoretical equation of laminar flow in the extent of increase in the surface area of blades.

(4) The correlations of Nusselt number and pipe friction coefficient were made in consideration of the effect of viscosity gradient within an accuracy of ±25 % and ±30 % respectively.

(5) The heat exchanger of this type shows the same heat transfer performance as the scraped surface heat exchanger in region of lower Re_r and higher Re_p.

8. Nomenclature

A : area of heat transfer surface, [m^2]
c_p : specific heat, [J/(kg·K)]
D : outside diameter of blade, [m]
de : equivalent diameter of annular duct, [m]
f : pipe friction coefficient, [-]
K : overall heat transfer coefficient, [W/(m^2·K)]
L : measurement length, [m]
N : rotational speed, [rpm]
Nu : Nusselt number, [-]
Pr : Prandtl number, [-]
Q : heat transfer rate, [W]
Re : axial Reynolds number, [-]
Re_r : rotational Reynolds number, [-]
V : mean velocity of test fluid, [m/s]
α : heat transfer coefficient, [W/(m^2·K)]
ΔP : pressure loss, [Pa]
ΔT_m :logarithmic mean temperature difference,[K]
δ : thickness of wall, [m]
λ : thermal conductivity, [W/(m·K)]
μ : viscosity, [Pa·s]
ν : kinematic viscosity, [m^2/s]
ρ : density, [kg/m^3]

Subscripts

b	: bulk	c	: cooling water
cal	: calculation	f	: pressure loss
h	:heat transfer	p	: test process fluid
sus	: stainless steel of 304	w	: wall

9. References

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