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Research Article

HEAT TRANSFER AND PRESSURE LOSS 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, the rotational speed of axis with blades and the viscosity of test fluid. From these data, the heat transfer coefficient and the pipe friction coefficient of 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 correlations of Nusselt Number obtained previously 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 loss, double-tube, heat exchanger, rotating blade*

1. INTRODUCTION

In the manufacturing process of food, the heat exchangers of the different types are in use of heating and cooling required 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.

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The overall heat transfer coefficients and the pressure drop 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 heat exchanger of the different types. 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 an overflow tank 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 test fluid and cooling water are measured by sheathed CA thermocouples with a diameter of 1 mm respectively, which have been inserted in the center of the flow section. Especially the bulk mean temperatures of the test fluid have been measured after passing the static mixer. Also, the head of pressure loss between the inlet and the outlet of 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 test fluid. Now the test process fluid and the cooling water are mutually flowing in counter direction.

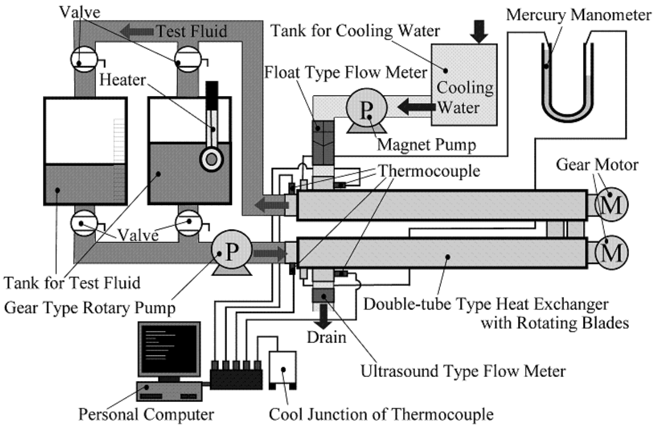

Fig. 1. Schematic diagram of an experimental apparatus.

Fig. 2. Photograph of an axis of rotation with blades.

Fig. 2 is a photograph of a rotational axis of outside diameter of 20 mm with blades of the length of 380 mm. A set of two blades are welded on the rotational axis which has the section without blades at the intervals of 380 mm in an axial direction, and a set of two blades is installed to being orthogonal mutually. Hence the rotation axis looks like having four blades of the right angle mutually as shown in a view from the axial direction of Fig. 3.

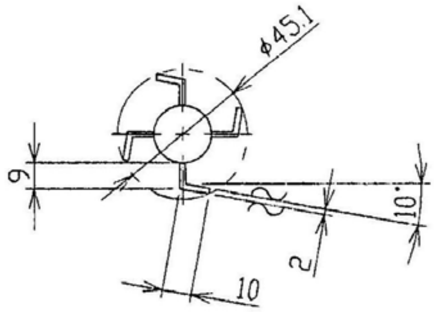

Fig. 3. Detail of an axis of rotation with blades.

The specifications of the test section in a double-tube heat exchanger are indicated in Table 1. The experiments were carried out under the conditions shown in Table 2.

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.

$$K = Q_c / (A_p \Delta T_m) \tag{1}$$

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

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

Table 1: Specifications of test section

Annular duct of test fluid	Inside diameter Outside diameter	20.0 mm 47.8 mm
Annular duct of cooling water	Inside diameter Outside diameter	50.8 mm 59.5 mm
Length of heat transfer	1.87 m $\times$ 2 stages	
Outside diameter of blade D	45.1 mm	

Table 2: Experimental conditions

Test fluid as process fluid	Corn syrup water solution
Mass fraction of an undiluted corn syrup	0.86 – 0.95
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

Based on the measured data, K is calculated from Eq. (1). The heat transfer coefficient 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.

$$Nu_c = \alpha_c de_c / \lambda_c = 0.023 \; \mathrm{Re}_c^{\,0.8} \; \mathrm{Pr}_c^{\,0.4} \tag{3}$$

From Eq. (2) using these above-mentioned values, the heat transfer coefficient 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 the heat transfer coefficient α_p, rotational Reynolds number Re_r and Prandtle number Pr are defined by the following equations, respectively.

$$\mathrm{Re}_p = Vde_p / \nu_p \tag{4}$$

$$Nu_p = \alpha_p de_p / \lambda_p \tag{5}$$

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

$$\mathrm{Pr} = \mu_p c_p / \lambda_p \tag{7}$$

The method of evaluating the physical properties of the test fluid is described in detail in our paper [2]. The values of the physical property used in this study are presented in Table 3.

Table 3: Physical properties of test fluid

Density ρ	1345 - 1395 kg/m^3
Viscosity μ_p	0.5 - 10 Pa$\cdot$s
Specific heat c_p	2476 - 2638 J/(kg$\cdot$K)
Thermal conductivity λ	0.434 - 0.461 W/(m$\cdot$K)
Prandtl number Pr	3000 - 50000

Well, it is though the scattering in this experimental data mainly depend on the measurement errors of the flow rate and the bulk mixed mean temperature on account of the low velocity and the high viscosity of the test fluid, and the estimated errors of physical properties of the test fluid. In addition, it is though the scattering of the experimental data is due to the instability of the phenomenon by being laminar flow in axial and rotational Reynolds number.

4. Experimental Results

4.1 Heat Transfer Coefficient

Fig. 4 shows Nusselt numbers Nu_p calculated by the above-mentioned method against axial Reynolds number Re_p. In this figure, Nu_p increase with an increase of Re_p in each rotational speed N rpm of blades. Especially, in the case of $N = 0$ without rotation 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 $N = 0$, 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 axial Reynolds number more than $\mathrm{Re}_p \approx 1$.

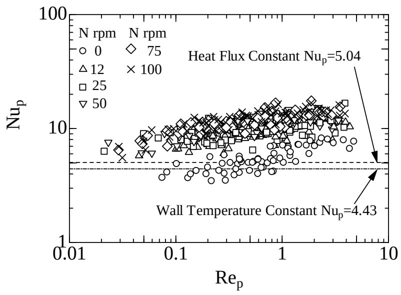

Fig. 4. Nusselt number vs. axial Reynolds number.

Fig. 5 shows the relation between Nusselt number $Nu_p / \mathrm{Pr}^{0.3}$ divided by 0.3 power of Prandtl number and axial Reynolds number Re_p in taking the rotational speed N rpm 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. Also, Nu_p in the case with rotation generally indicate about two times higher values than Nu_p in the case without rotation.

Fig. 6 shows the influence of the rotational speed N rpm on Nusselt number Nu_p in some cases of 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. 7 shows values of Nusselt number $Nu_p / \mathrm{Pr}^{0.3} / \mathrm{Re}_p^{0.3}$ divided respectively by 0.3 power of Prandtl number and axial 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.

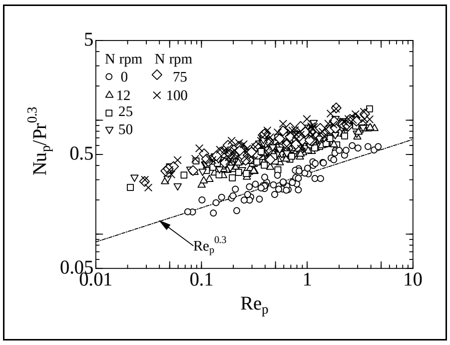

Fig. 5. Nusselt number divided by 0.3 power of Prandtl number vs. axial Reynolds number.

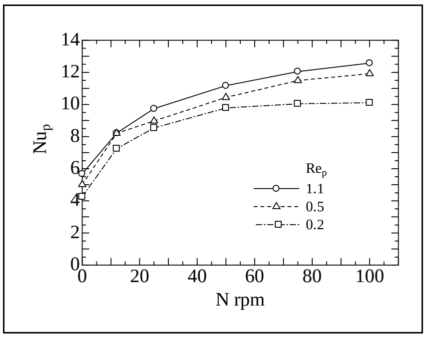

Fig. 6. Effect of rotational speed of blades on Nusselt number.

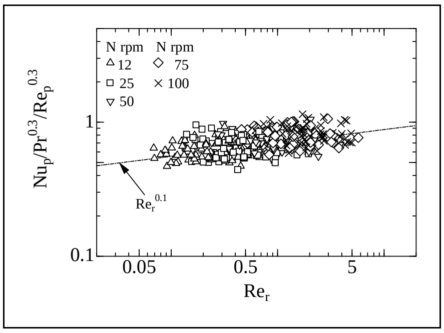

Fig. 7. Nusselt number divided respectively by 0.3 power of Prandtl number and axial Reynolds number vs. rotational Reynolds number.

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. 8 shows the relation between the pipe friction coefficient f and axial 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 of an increase in the surface area by having inserted the rotational axis with blades in inner tube.

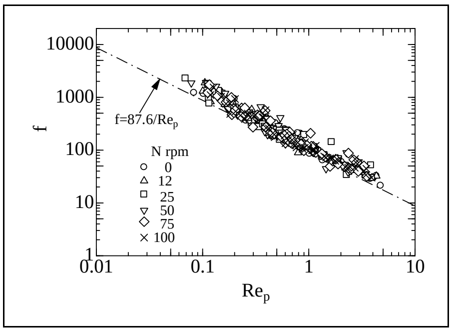

Fig. 8. Pipe friction coefficient vs. axial Reynolds number.

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 were compared with the following Eq. (9) of Sieder and Tate [3], which takes into account the viscosity gradient of fluids in the tubes.

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

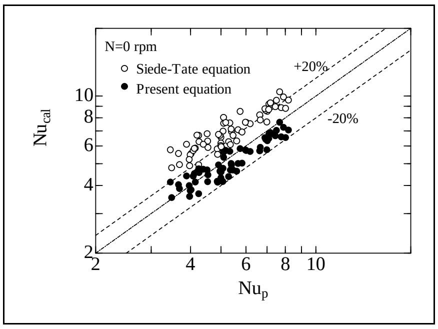

Fig. 9. Comparison of experimental values with calculated values of Nusselt number in case without rotation.

The comparison of the experimental values Nu_p of Nusselt number with calculated values Nu_{cal} of Nusselt number by Eq. (9) is shown by the white symbol of circle in Fig. 9. It is found that the calculated values from the equation of Sieder and 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) in consideration of the effect of viscosity gradient μ_w/μ_b in the case without rotation.

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

The comparison of the experimental values Nu_p with calculated values Nu_{cal} from Eq. (10) is shown by the black symbol of circle in Fig. 9. 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) in consideration of the effect of viscosity gradient as well as Eq. (10).

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

Fig. 10 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 %.

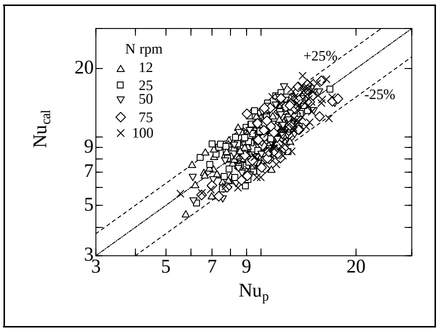

Fig. 10. Comparison of experimental values with calculated values of Nusselt number in case with rotation.

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 not depending on Re_r.

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

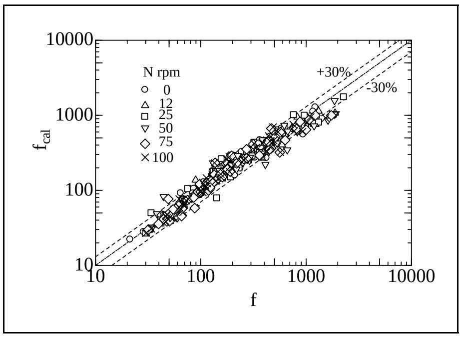

Fig. 11. Comparison of experimental values with calculated values of pipe friction coefficient in case with and without rotation.

Fig. 11 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 having installed the blades on the axis of rotation.

However, these correlations Eqs. (10) - (12) made about Nusselt number and pipe friction coefficient are effective within the range of this experimental conditions. By the way, the exponents of viscosity gradient in Eqs. (10) - (12) were given by -0.14 for Nusselt number and 0.5 for pipe friction coefficient in the case of cooling in a reference [1], respectively.

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

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 in the heat exchanger of the different types. 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 Eqs. (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. 12 shows the comparisons of Nusselt number $Nu_p/\mathrm{Pr}^{0.3}$ obtained from Eqs. (10, 11) and Eqs. (13) - (15) against Re_p among heat exchanger of three kinds of types. 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.

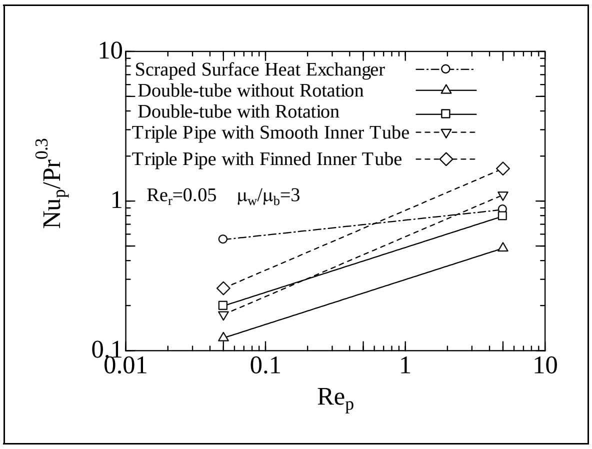

Fig. 12. Comparison of heat transfer performance in heat exchanger of different types.

7. Conclusions

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 and the pipe friction coefficient of the test fluid flowing in the inner annulus 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 the 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
cal	: calculation
h	: heat transfer
sus	: stainless steel of 304
c	: cooling water
f	: pressure loss
p	: test process fluid
w	: wall

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