

**DEVELOPMENT OF GENERAL CORRELATION FOR HEAT TRANSFER IN SINGLE-PHASE
TURBULENT FLOW INSIDE INTERNALLY HELICALLY-GROOVED TUBES**

Norihiro INOUE and Masao GOTO

Department of Electronics and Mechanical Engineering, Tokyo University of Marine Science and
Technology, 2-1-6 Etschujima, Koto-ku, Tokyo 135-8533, JAPAN

*Corresponding author: Fax: +81-3-5245-7479 Email: inoue@kaiyodai.ac.jp

ABSTRACT

An experimental study of heat transfer in single-phase turbulent flow inside an internally helically-grooved tube was carried out using thirteen different kinds of grooved tubes and a smooth tube with 6.35 mm outer diameter and 1000 mm working length. The obtained data indicated that heat transfer coefficients were from 110 to 200% higher than that of the smooth tube. The heat transfer coefficients increased as both the ratio of the inner surface area of the grooved tube to that of the smooth tube and the helix angle increased, and decreased as the number of grooves increased. The correlations to predict the heat transfer coefficients were developed based on the measured data. The experimental data was successfully predicted within $\pm 20\%$, and data reported in the literature was correlated within $\pm 30\%$. The effect of groove shape upon the relationship between the pressure drop and heat transfer enhancement of the grooved tubes was clarified using the correlation of the friction factor obtained in a previous paper and the correlation of the heat transfer coefficient obtained.

INTRODUCTION

Helically-grooved tubes are capable of larger heat transfer in both single-phase and two-phase heat transfer without causing a large increase in pressure drop. In refrigeration engineering, many researchers have reported the characteristics of helically-grooved tubes, specifically condensation and

evaporation heat transfer and pressure drop. However, the performance in single-phase heat transfer, which occurs in superheated and subcooled regions in a heat exchanger, has not yet been fully elucidated, because the effect of the single-phase region on the performance of the heat exchanger is generally considered to be small. As a result of the development of system operation control techniques for air-conditioners, however, the importance of this region has recently been recognized. To improve the performance of heat exchangers in refrigeration systems, it is important to accurately predict the heat-transfer coefficients and pressure drop in the single-phase region. In this paper, we describe experimental results of the heat transfer in single-phase turbulent flow inside 13 internally helically-grooved tubes and a smooth tube with outer diameters of 6.35 mm. Also, using our experimental values, we propose a new general correlation for heat transfer coefficients inside internally helically-grooved tubes.

EXPERIMENTAL APPARATUS

In these experiments, we used water as a test fluid. The cooling water was set to a constant inlet temperature and flow rate and was supplied to a test section after passing through an inlet mixing chamber. The cooling water was discharged out of loop through an outlet mixing chamber. The test section was a section of internally helically-grooved copper-tube with an outer diameter (o.d.) of 6.35

mm and a total length of 1850 mm. A working length of 1000 mm, which was heated by an electric heater (Nichrome wire coils), was covered with gypsum after an approach length of 680 mm from the inlet. The pressure drop of a 1500-mm length could be measured by a differential pressure transducer by setting pressure taps at points 250 mm and 1750 mm from the inlet.

DATA REDUCTION

The Nusselt numbers Nu_i and Nu_{real} , which are expressed based on the maximum inside diameter of the tube and the real surface area of the internally helically-grooved tube, are given by:

$$Nu_i = \alpha_i d_i / \lambda \quad (1)$$

$$Nu_{real} = \alpha_{real} d_i / \lambda \quad (2)$$

where, the thermophysical properties of water were obtained from the literature (Fujii et al. 1977) by using the arithmetic mean temperature of the inlet/outlet test section as a representative length.

RESULTS AND DISCUSSION

Dimensionless Heat Transfer Coefficient

Fig. 1 shows the experimentally obtained Nusselt number, defined by Eq. (1), versus the Reynolds number. The various symbols denote the measured data for each tube. The thin line, the thick line, the broken line, and the dotted line represent the correlations for the heat transfer of single-phase turbulent flow of Colburn (1986), Gnielinski (1976), Petukov (1970), and Dittus-Boelter (1930),

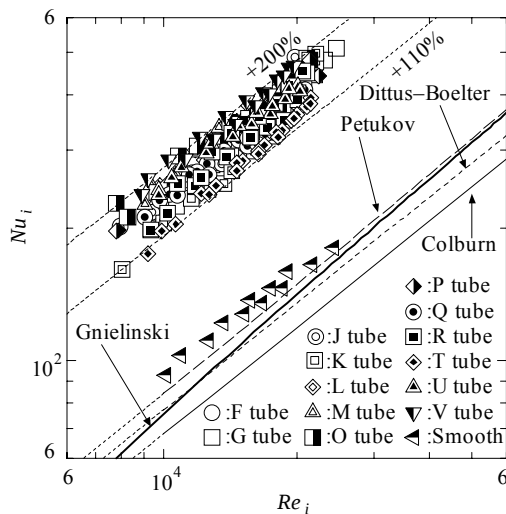


Figure 1. Nusselt number versus Reynolds number for all measured data

respectively. From the figure, it is found that, for the smooth and grooved tubes, the experimental Nusselt number versus Reynolds number shows a similar tendency to the values predicted by the Colburn correlation. Also, it is found that the experimental values for the smooth tube (symbol ◀) agree with the values predicted by the Petukov correlation and are about 30% higher than those predicted by the Colburn correlation. On the other hand, it can be seen that the experimental values for the grooved tubes are about 110% to 200% higher than those for the smooth tube and differ from the predicted values for each tube depending on geometry of the helical grooves. Namely, the experimental values for tube V (symbol ▼), which has the largest helix angle of 35 degree, were the largest among all helically-grooved tubes. In addition, the experimental values for tube T (symbol ◆), which has the smallest helix angle of 10 degree, were the smallest of all tubes. Fig. 2 shows the experimentally obtained Nusselt number, defined by Eq. (2), versus the Reynolds number. The solid line represents the Colburn correlation. The experimental values for the grooved tube were about 20% to 100% higher than the values predicted by the Colburn correlation and differ for each tube. These results show that the increase in heat transfer of the helically-grooved tubes is larger than the increase expected based on the magnifying ratio of the inner surface area. Furthermore, the enhancement ratios differ from tube to tube depending on the geometry of the helical grooves.

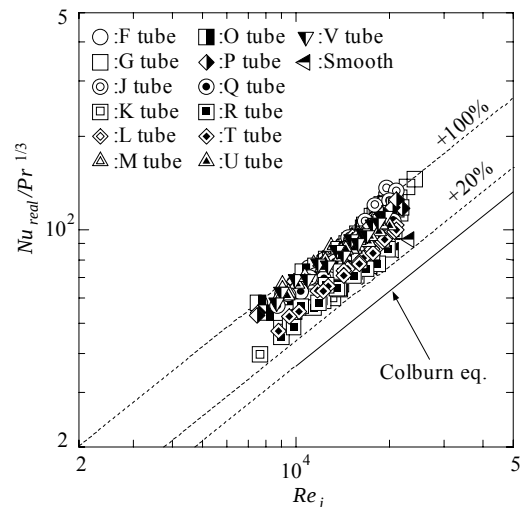


Figure 2. Nusselt number defined by real heat transfer area versus Reynolds number

Correlations of Heat Transfer Coefficients in Single-Phase Turbulent Flow

We attempted to apply the correlation for all internally helically-grooved tubes used in the experiment to predict the heat transfer coefficients in single-phase turbulent flow. We modified a coefficient of the Colburn correlation based on the experimental values for the smooth tube to define a smooth tube equation:

$$Nu_i = 0.031 Re_i^{0.8} Pr^{1/3} \quad (3)$$

In consideration of the similarity between the heat transfer and pressure drop inside tubes described by the above equation, the heat transfer coefficients were correlated by the same parameter used to correlate friction factor in the previous paper (Inoue et al., 2006). As the contact area of heat transfer between the fluid and the wall surface increases, the heat transfer rate of the internally grooved tubes increases. Therefore, we considered the surface area ratio η as one parameter, and we tried to correlate the experimental values using the following equation:

$$Nu_i = 0.031 Re_i^{0.8} Pr^{1/3} \eta^{0.8} \quad (4)$$

Here, the exponent of η was obtained by a trial-and-error method on the basis of the experimental values. The results are shown in Fig. 3(a). From the figure, it is found that all experimental values of the internally grooved tubes approach those of the smooth tube, and the data spread of several tubes is slightly smaller than the case of Fig. 1. Also, compared to the case where the other groove geometry parameters remain the same,

except for fin height (tubes J, O, and P), the experimental value for these tubes agree well with each other. Thus, the fin does not act as surface roughness affecting the heat transfer enhancement. We estimated the contribution of the fin to the enhancement as the heat transfer area was increased. Consequently, the surface area magnifying ratio can encompass the effect of the fin height, the same as in the case of pressure drop.

The fluid inside the internally grooved tubes formed a gyratory secondary flow along helix angle θ of the fins. We considered the effect of this on the heat transfer enhancement. To do so, we introduced a term $\cos \theta$ into Eq. (4), yielding the following equation:

$$Nu_i = 0.031 Re_i^{0.8} Pr^{1/3} \eta^{0.8} (\cos \theta)^{-1.3} \quad (5)$$

The results in Fig. 3(a) were then correlated with

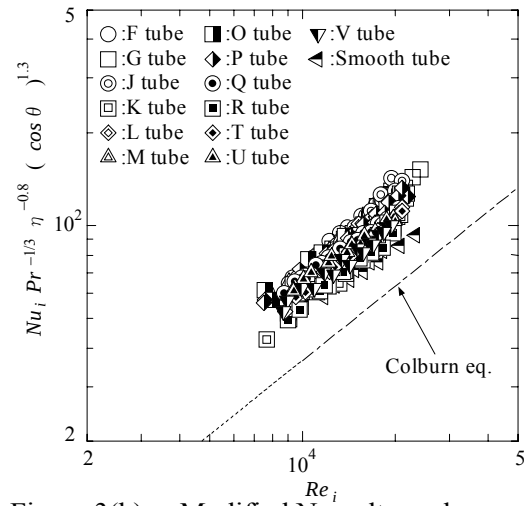


Figure 3(b). Modified Nusselt number

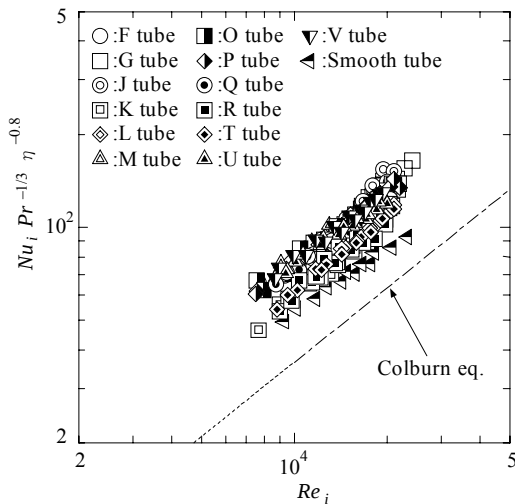


Figure 3(a). Modified Nusselt number

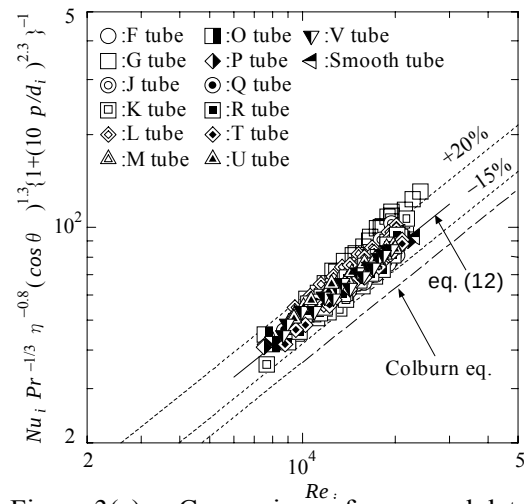


Figure 3(c). Comparison of measured data with present correlation

Eq. (5). Also, the exponent of the $\cos \theta$ term was obtained by a trial-and-error method on the basis of the experimental values. The results are shown in Fig. 3(b). From this figure, it is found that all experimental values of the internally grooved tubes approach those of the smooth tube, and the difference from tube to tube is smaller than that in the case of Fig. 3(a). Although the experimental values were modified by η and $\cos \theta$, the results still show dependence on the number of fins; specifically, the value decreased as the number of fins increased. The reason for the poor gyratory secondary flow inside tube was because of the fluid near the wall surface does not flow along the helix angle of the fin, but flows while "skidding" (sliding) on the fin. The fin pitch is defined by the following equation as a parameter for this effect:

$$p = (d_i \pi / n) \cdot \cos \theta \quad (6)$$

Using the ratio between the fin pitch p and maximum tube inside diameter d_i as a parameter, the results in Fig. 3(b) were further correlated using:

$$Nu_i = 0.031 Re_i^{0.8} Pr^{1/3} \eta^{0.8} (\cos \theta)^{-1.3} \times \left\{ 1 + (10 p/d_i)^{2.3} \right\} \quad (7)$$

The ranges used were $(0.036 \leq p/d_i \leq 0.07)$ and $(10^4 \leq Re_i \leq 10^5)$. The results are shown in Fig. 3(c). The coefficient and exponent of p/d_i were obtained by a trial-and-error method on the basis of the experimental values. From the figure, it is found

that all experimental values for the helically-grooved tubes can be correlated within $\pm 20\%$ using Eq. (7), which is obtained by adding the surface area magnifying ratio η , the helix angle θ , and the fin pitch p as parameters to Eq. (3) for smooth tubes.

Verified Practical Effectiveness of the Proposed Equation

The experimental values of Jensen et al. (1999), Webb et al. (2000), Koyama et al. (1992), and Inoue et al. (1995) were correlated to verify practical effectiveness of the proposed equation, using Eq. (7) itself. The results are shown in Fig. 4(a). From the figure, we found that the previous experimental values can be correlated within -30% to $+20\%$ by Eq (7), independent of the tube diameter and groove geometry. Furthermore, we tried similarly to correlate the experimental values of Jensen et al. for a large p/d_i value. Then, we thought that the effect of the number of fins could be ignored as a factor in the heat transfer enhancement, because of the large p/d_i value. We performed correlation using only the surface area magnifying ratio η and the helix angle θ with the following equation:

$$Nu_i = 0.027 Re_i^{0.8} Pr^{1/3} \eta (\cos \theta)^{-3} \quad (8)$$

The ranges used were $(0.09 \leq p/d_i \leq 0.34)$ and $(10^4 \leq Re_i \leq 10^5)$. The results are shown in Fig. 4(b). From this figure, it is found that the experimental values agree with the value of Eq. (8), within $\pm 10\%$.

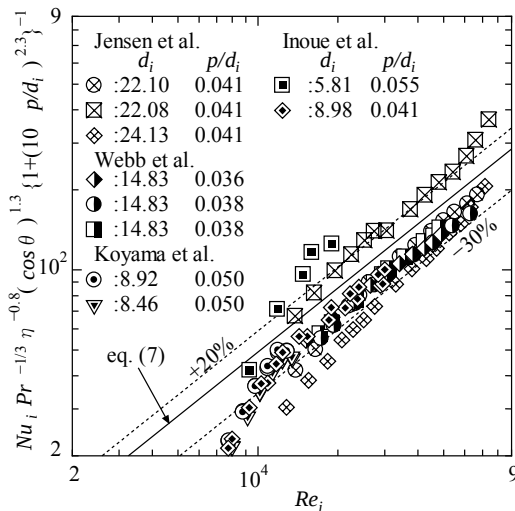


Figure 4(a). Comparison of Nusselt number of other researchers' data with the present correlation

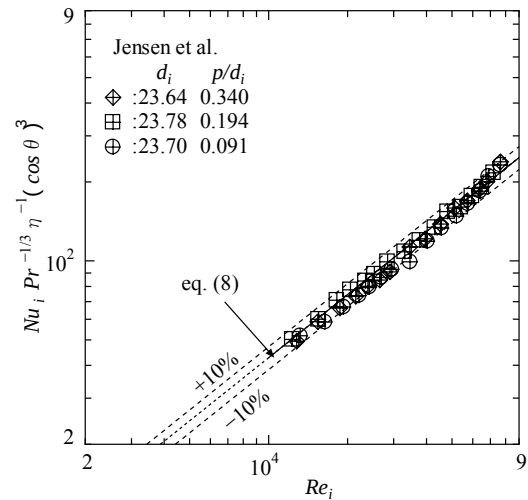


Figure 4(b). Comparison of the data in literature with the present correlation

Correlation Between Heat Transfer Coefficient and Friction Factor

Colburn produced the following relationship, called the Colburn analogy, between the heat transfer coefficient and friction factor by using the dimensionless correlation for the turbulent heat transfer coefficient and Eq. (11) below for the Fanning friction factor f_{fa} in association with an isothermal circular tube:

$$St Pr^{2/3} = f_{fa} / 2 \quad (9)$$

The left-hand part of Eq. (9) is called the Colburn j-factor for heat transfer, where St is the Stanton number defined by Eq. (10) below, and f_{fa} is the Fanning friction factor calculated from Eq. (11) below:

$$St = Nu / (Pr Re) \quad (10)$$

$$f_{fa} = 0.046 Re_i^{-0.2} \quad (11)$$

In order to clarify the relationship between the turbulent heat transfer coefficient and friction factor in a single-phase region of a helically-grooved tube, the relationship between $StPr$ and friction factor f_i was obtained, as in the Colburn analogy, using the correlation of the heat transfer coefficient obtained from the present experiment and the previously obtained correlation of the friction factor. The result is given as:

$$St Pr^{2/3} / f_i = 0.031 \left\{ 1 + (10 p/d_i)^{2.3} \right\} / \left[0.046 \left\{ 1 + (10.8 p/d_i)^{3.3} \right\} \right] \quad (12)$$

(0.036 ≤ p/d_i ≤ 0.07)

where f_i is calculated from:

$$f_i = 0.046 Re^{-0.2} \eta^{0.8} (\cos \theta)^{-1.3} \times \left[1 + (10.8 p/d_i)^{3.3} \right] \quad (13)$$

The equation shown below is obtained for grooved tubes having p/d_i values larger than 0.09:

$$St Pr^{2/3} / f_i = 0.59 \quad (0.09 < p/d_i < 0.34) \quad (14)$$

where f_i is calculated from:

$$f_i = 0.046 Re_i^{-0.2} \eta (\cos \theta)^{-3} \quad (15)$$

The relationship between Eq. (12) and Eq. (14) and the experimental values is shown in Fig. 5. This figure illustrates the experimental values in the form of graphs, where the vertical axis corresponds

to $StPr^{2/3}/f_i$ representing the ratio between the heat transfer coefficient and the friction factor, and the horizontal axis corresponds to p/d_i . The solid line and the broken line in the figure correspond to Eq. (12) and Eq. (14), respectively. It is apparent from the figure that the maximum experimental value, which is larger than the value of approximately 0.67 for a smooth tube ($p/d_i = 0$), is seen in the p/d_i range from 0.03 to 0.06. On the other hand, the experimental value is approximately 0.59, which is smaller than the value for a smooth tube, in the p/d_i range larger than 0.09. From these results, it can be concluded that a grooved tube with a p/d_i value from 0.03 to 0.06 has superior heat transfer performance in consideration of the pressure drop, because the rate of increase of heat transfer coefficient is larger than the rate of increase of pressure drop for the grooved tube in comparison with the smooth tube. On the other hand, for a grooved tube with a p/d_i value larger than 0.09, the rate of increase of pressure drop is estimated to be larger than the rate of heat transfer enhancement. These experimental values, though exhibiting some variations, agree substantially with the values from Eq. (12) and Eq. (14). Hence, the heat transfer performance and the pressure drop characteristics of various types of helically-grooved tubes with different groove shapes can be estimated by using these equations, and furthermore, the performance of a helically-grooved tube can be evaluated based on those characteristics.

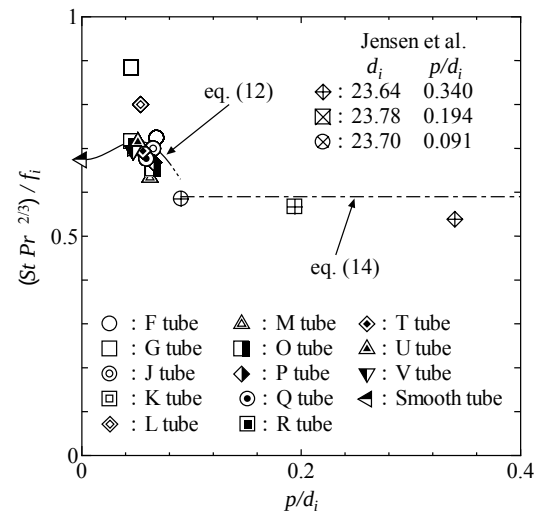


Figure 5. Relation of Stanton number with fin pitch divided by maximum inner diameter

CONCLUSIONS

A single-phase turbulent heat transfer experiment was conducted using 13 types of internally helically-grooved tubes with systematically different groove shapes to investigate how the groove shape affects the heat transfer coefficient and to find the correlation between heat transfer characteristics and pressure drop characteristics. As a result of this experiment, we found following:

(1) The heat transfer coefficient exhibited values approximately 110% to 200% larger than the experimental value for a smooth tube. Like the friction factor, the heat transfer coefficient increases according to increases in the helix angle and the area magnifying ratio of the grooved tube relative to the surface area of a smooth tube with the same inner diameter. In contrast, the heat transfer coefficient decreases as the number of grooves increases.

(2) The obtained empirical equation of turbulent heat transfer for a smooth tube yielded the correlation in Eq. (7) for the heat transfer coefficient of a helically-grooved tube, with the area magnifying ratio, the lead angle, and the groove pitch as parameters.

(3) Eq. (7) was verified based on experimental values reported to date. Consequently, it was found that these experimental values can be correlated within -30% to +20% with Eq. (7). Furthermore, Eq. (8), which can be applied to tubes with large groove pitches, was obtained from Eq. (7).

(4) The effect of groove shape upon the relationship between the pressure drop and heat transfer enhancement of the grooved tubes was clarified using the correlation of the friction factor obtained in a previous paper and the correlation of the heat transfer coefficient obtained here.

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