Thermal characteristics of annular flow with asymmetrically roughened surfaces

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Heat transfer characteristics were studied for confined fully developed turbulent flow in horizontal annuli with asymmetrically roughened surfaces. The present study used a helical angle ($\alpha=65°$), two pitch-to-height ratios (1.44 and 2.88), three aspect ratios ($D_o/D_i=2.68$, 3.48, and 5.1), and the corresponding ratios of roughness height-to-hydraulic diameters (e/D_H) were 0.192, 0.13, 0.08, and 0.095, 0.065, 0.040, respectively. The experiments were conducted in the range of $3000 \leqslant Re_H \leqslant 30{,}000$, with water as the cooling medium. Axial and transverse temperature distributions were measured along the downstream distance and at the downstream end of the test section. Data were extracted to correlate the Nusselt number with the surface roughness geometry and the fluid flow rate. Moreover, the baseline data from the smooth annuli with the same aspect ratios of 2.68 and 3.48 are also presented for comparison. The results are in good agreement with those of previous investigators.

Keywords: thermal analysis; annular flow; asymmetrically roughened surfaces

Introduction

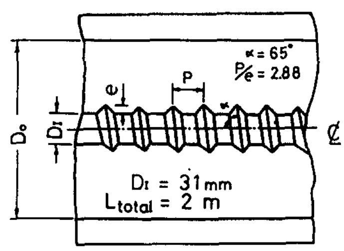
Heat transfer from rough surfaces has received a great deal of attention[1–5] due to the many practical applications of increasing the heat duty and due to the difficulty in obtaining analytical solutions for the temperature and velocity fields. Most published studies have been conducted on transverse-rib roughness with various rib shapes, heights, and spacings.[2–5] Because the increase in heat rate performed is accompanied by a friction factor increase, the preferred roughness geometry will yield a given heat transfer augmentation with minimum friction factor increase. Several investigators[6,7] have suggested that a helical, rather than a transverse rib, will yield a given St/St_s with a high efficiency index, $\eta=(St/St_s)/(f/f_s)$, and use an optimum helical angle of 30°–50° to obtain a higher efficiency index. In spite of this, the previous studies[2–5] focused more on large pitch-to-diameter ratios (for instance, $p/e \geqslant 8$), except Ref. 1, which used $2.0 \leqslant p/e \leqslant 4.7$ with triangular fin-type surface roughness.

It seems that data on small values of p/e are rather limited. Besides, the length of the test section in the previous studies was long enough to guarantee that the flow was fully developed. However, in many heat exchanger applications, the length of the heat exchangers is shorter and more compact than that used by previous investigators, owing to cost of material and space savings.

In the light of the foregoing discussion, a 2-m test section of helical-type roughened annuli was constructed and set up. An early result reported by Hsieh *et al.*[8] was on the aspect ratio effect of roughened double-pipe heat exchangers for a pitch-to-diameter ratio $p/e=1.44$ with aspect ratios $D_o/D_i=2.68$ and 5.1. In order to gather further information regarding the effect of pitch to diameter on heat transfer performance, one more case of $p/e=2.88$ was investigated—a 2-m test section of helical-type roughened annuli with a helical angle $\alpha=65°$, two pitch-to-diameter ratios of $p/e=1.44$ and 2.88, three aspect ratios of $D_o/D_i=2.68$, 3.48, and 5.1, and the corresponding ratios of roughness height-to-hydraulic diameters of 0.192, 0.13, 0.08, and 0.095, 0.065, 0.040, respectively, was conducted in the range of $3000 \leqslant Re_H \leqslant 30{,}000$. Moreover, the baseline data of smooth annuli of $D_o/D_i=2.68$, 3.48 were also taken for a comparison. Axial and transverse temperature distributions were measured along the downstream distance and at the downstream end of the test section. Data were obtained to find the relationship of the heat transfer coefficient to the surface roughness geometry and the fluid flow rate.

Experimental rig and instrumentations

A schematic of the water annuli, which has a 31-mm inner diameter with outer diameters of 83.08, 107.88, and 158.10 mm, is shown in Figure 1. The present study concentrates on the outer surface of the inner tube where it is roughened. Water as a cooling medium was forced to flow by a pump at room temperature through a flowmeter and over the test section of the roughened 2-m surface to a cooling tower, resulting in a flow temperature decrease to room temperature, and then was fed



D_o/D_i \ e/D_H	$P/e=1.44$	$P/e=2.88$
5.10	0.080	0.040
3.48	0.130	0.065
2.68	0.192	0.096

Figure 1 Typical illustration of present roughness element configuration

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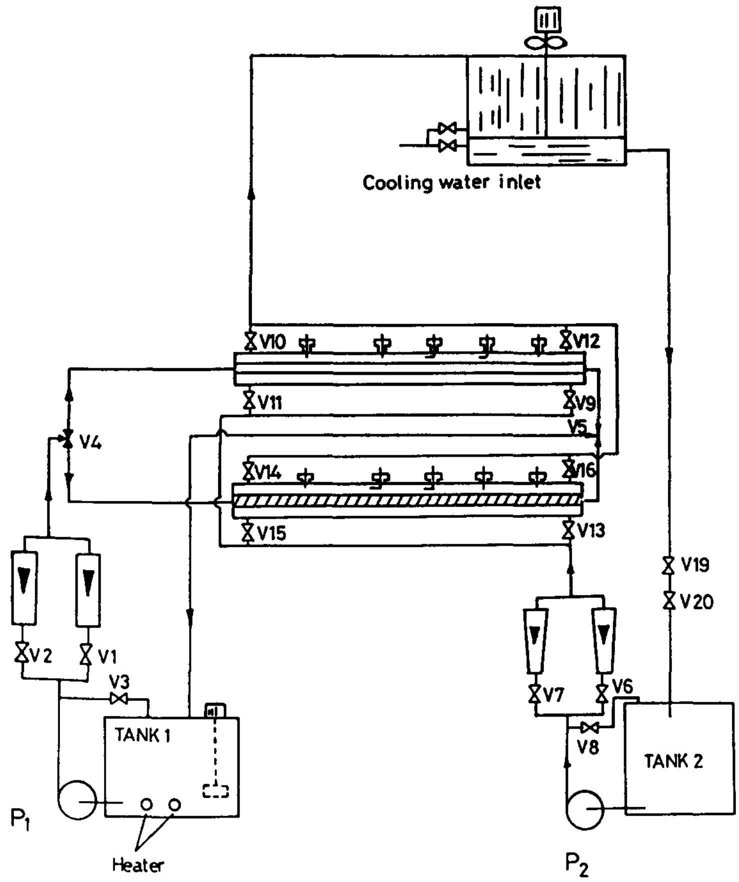

Figure 2 Schematics of the experimental rig

back by a pump. In the inner tube, 60% glycerin and 40% water were heated by two 30-kW heaters as a working fluid. In order to avoid transverse conduction heat loss, all outer pipes were wrapped with 25-mm-thickness insulation material of thermal conductivity 0.038 ($W\,m^{-1}\,K^{-1}$). A detailed description of the experimental apparatus was given in Ref. 8. A schematic diagram of the test apparatus is shown in Figure 2.

Fifteen Chromel-Alumel (AWG No. 28) thermocouples were used to measure the transverse temperature distribution of the inner pipe at the downstream end, and 12 thermocouples were positioned in the coolant to measure the temperature distribution along the downstream distance. The heating medium was first heated in a tank by a heater controlled by a PID temperature controller and then pumped into the pipeline. The thermocouples in the coolant were connected to a HP-3497A data acquisition system. By means of HP-85B PC software, the measured voltage was converted into temperature automatically, and the accuracy was ± 0.7 K.

Experiments were performed with different water flow rates in the range of Reynolds numbers $3000 \leqslant Re_H \leqslant 30{,}000$ for a convective heating in 2 m roughened annuli with varied pitch-to-height ratios (p/e) and relative surface heights (e/D_H) and the same physical dimensions of smooth test sections. The local temperature was measured along the downstream distance of the test section at a definite axial position, and data were taken to define the Nusselt number.

Notation

A	Surface area (m^2)
C	Correlation constant
C_P	Specific heat (J/kg K)
D_H	Hydraulic diameter (m)
D_o	Outer diameter of tube (m)
D_i	Inner diameter of tube (m)
f	Friction factor
f_s	Friction factor for smooth tube
h	Heat transfer coefficient (W/m^2 K)
k	Thermal conductivity (W/m K)
m	Correlation exponent
Nu	Nusselt number, hD_H/k
p	Roughness pitch (m)
Pr	Prandtl number, $\mu C_P/k$
Q	Total heat input (W)
q_w	Heat flux at wall (W/m^2)
q_w^+	Dimensionless heat flux at wall, $q_w D_H/kT_w$
Re_H	Reynolds number, vD_H/ν
St	Stanton number, Nu/(Pr Re)
St_s	Stanton number for smooth tube
T_i	Inlet temperature (K)
T_o	Outlet temperature (K)
T_b	Bulk temperature (K)
T_b^+	Dimensionless bulk temperature $(T_b - T_i)/(T_o - T_i)$
T_w^+	Dimensionless wall temperature $(T_w - T_i)/(T_o - T_i)$
v	Velocity (m/s)
α	Helical angle
η	Rough tube efficient index $(St/St_s)/(f/f_s)$
μ	Dynamic viscosity (kg/m s)
ν	Kinematic viscosity (m^2/s)

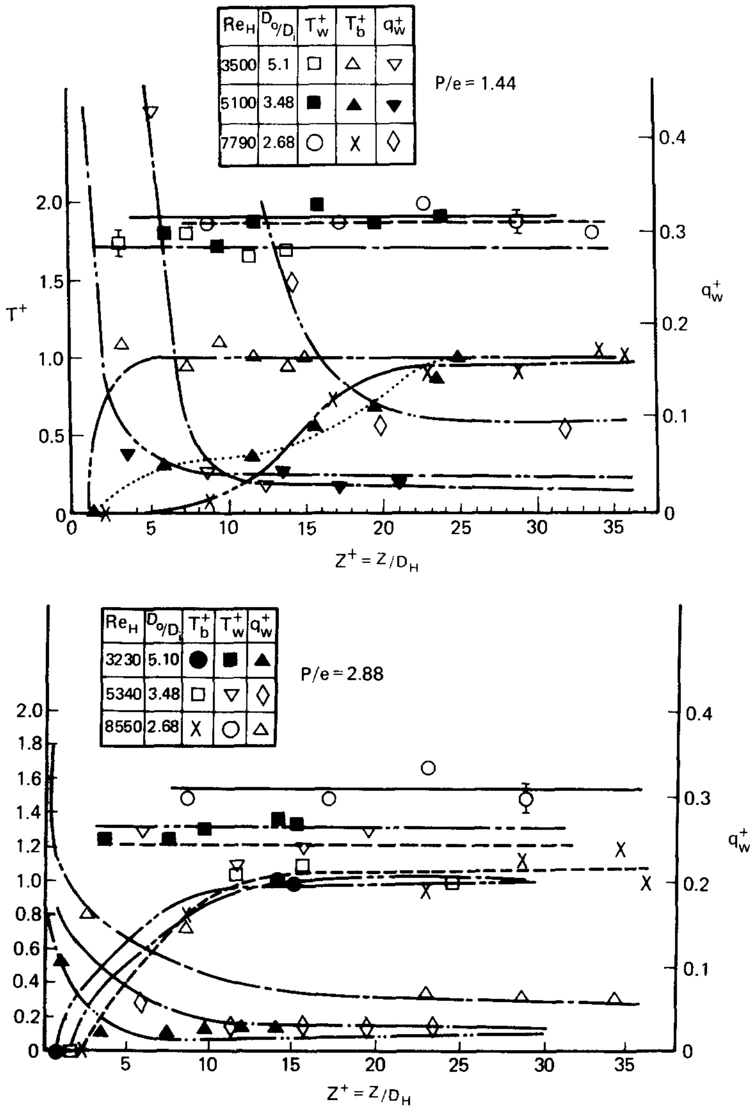

Figure 3 T_b^+, T_w^+, and q_w^+ versus $Z^+ = Z/D_H$

Results and discussion

The experimental data were taken in three helical-type double-pipe heat exchangers with a helical angle $\alpha = 65°$, two pitch-to-height ratios of 1.44 and 2.88, and three aspect ratios of 2.68, 3.48, and 5.1, and the corresponding ratios of roughness height-to-hydraulic diameter (e/D_H) of 0.192, 0.13, 0.08, and 0.095, 0.065, 0.040, respectively, for Reynolds number from 3000 to 30,000. The maximum value of the uncertainty of Reynolds number estimated for all mass flow rates is about $\pm 8\%$.

In the test section, the temperature and heat flux distributions through the wall along the downstream distance can be measured and calculated, of which one is shown in Figure 3 for different p/e ratios. It is found that the wall heat flux distribution q_w^+ decreased along downstream distance while the dimensionless bulk temperature of the flow increased, and each reached an approximately constant value in three operating conditions at Reynolds numbers 3230, 5340, and 8550 for three e/D_H of 0.04, 0.065, and 0.095, respectively.

Also shown in Figure 3 is the same plot for $p/e = 1.44$ at Reynolds numbers of 3500, 5110, and 7790 for e/D_H of 0.08, 0.13, and 0.192, respectively.[8] It appears that a highly disturbed turbulent flow due to the higher Reynolds number, higher surface roughness (e/D_H), and lower p/e would result in a shorter thermal entrance length based on these temperature distributions in Figure 3. The temperature distributions of the upper part of Figure 3 at the early region look strange for $D_o/D_i = 3.48$ and 5.1. They are perhaps due to the uncertainties of the temperature readings. Nevertheless, they both reach an asymptotic value as e/D_H increases, and the present Nusselt number was taken at the downstream end distance for data reduction. For $p/e = 1.44$ and 2.88, the results indicate that the thermal flow behavior can be considered as a constant wall temperature because the wall temperature difference at the upstream/downstream ends was found to be less than $\pm 8\%$ relative to the average fluid temperature. Actually, from Figure 3, it was found that the following relationship holds for certain downstream distances:

$$\frac{\partial}{\partial x}\left(\frac{T_w - T}{T_w - T_b}\right) = 0$$

which indicates that the present turbulent flow even in 2-m pipe would be thermally fully developed eventually in the test section for the three aspect ratios investigated.

The heat loss was estimated by measuring the temperature

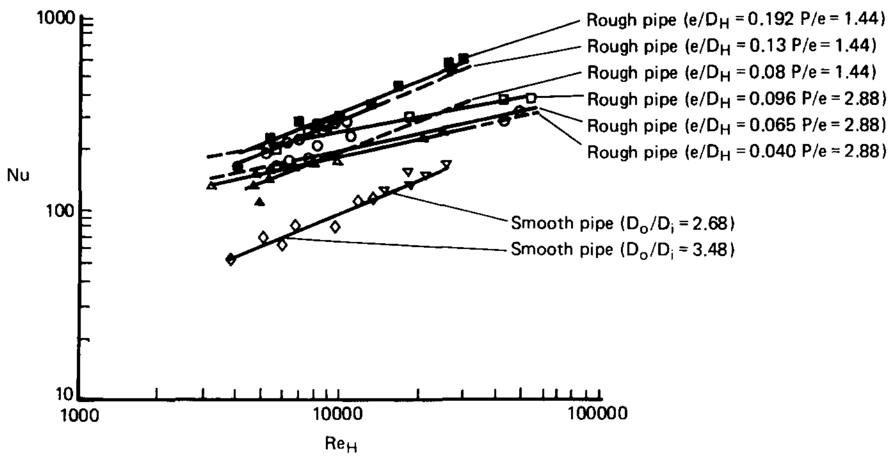

Figure 4 Dependency of $Nu = f(Re, e/D_H)$

Table 1 The values of C's and m's for Equation $Nu = C(Re)^m$

e/D_H	0.192		0.13		0.08		0.095		0.065		0.040	
p/e	C	m	C	m	C	m	C	m	C	m	C	m
1.44	1.965	0.552	1.293	0.586	1.076	0.568	—	—	—	—	—	—
2.88	—	—	—	—	—	—	1.814	0.545	1.08	0.572	1.056	0.574

— means data not available.

differences inside and outside the shell tube through the insulation material. The maximum heat loss was found to be 0.4% of the total heat input. With this value and the related uncertainties in the experimental measurements, the final uncertainty in the reduced variable Nusselt number was about $\pm 10\%$. The experimental data indicate that the heat transfer performance in this surface roughness configuration increases as the flow rate increases.

Through data regression, Figure 4 presents the result of fitting equations of the form $Nu = C(Re)^m$ to the data for the six roughened cases investigated. The values of C's and m's are tabulated in Table 1. The average of the six values of m's is $m = 0.57$. At the 95% confidence level, $m = 0.57 \pm 0.03$. The result also brackets the Reynolds number dependency of smooth baseline data, also shown in Figure 4, which measured $m = 0.58 \pm 0.13$, and so the Reynolds number dependency for roughened pipe and that for smooth pipe is approximately the same, which shows that the Reynolds number is independent of the Nusselt number for these two surface conditions. This finding agrees quite well with the previous result reported by Hsieh and Christensen.[1]

There seems to be no significant change in Nusselt number due to variations of the aspect ratio for smooth pipes in Figure 4. This may be partly caused by the insignificant change of curvature and partly because water is the working fluid and because of the flow pattern of forced convection in nature. The increase in the heat transfer coefficient of roughened pipe, relative to the coefficient of smooth pipe, was about 230–350% for the same aspect ratio.

The significance of the variation of the pitch-to-height ratio is due to a noticeable slope difference of the present results, which give a lower p/e ($= 1.44$) with a better heat transfer performance compared with $p/e = 2.88$. This is because the flow was much disturbed for $p/e = 1.44$. It seems that the heat transfer performance is strongly affected by both Reynolds number and the ratio of roughness height-to-hydraulic diameter. For the former, the result is quite obviously due to the present forced convection. For the latter, Figure 4 shows that the present results plotted as a set of lines have almost the same slope for different e/D_H ratios when they plotted against Re_H. It is therefore concluded that the thermal behavior can be written as

$$Nu = 5.90(Re_H)^{0.53}\left(\frac{e}{D_H}\right)^{0.54} \quad \text{for } \frac{p}{e} = 1.44$$

$$Nu = 5.23(Re_H)^{0.51}\left(\frac{e}{D_H}\right)^{0.33} \quad \text{for } \frac{p}{e} = 2.88$$

respectively, with an average deviation of $\pm 6\%$ and $\pm 9\%$ from the original data. The results are illustrated in Figure 5. From these two correlations, it is found that the relative surface roughness height for the high value of p/e has less effect on the Nusselt number than that of the low value of p/e.

The present result, finally, is compared with those obtained from Buller[9] and Wirtz.[10] Figure 6 illustrates such a comparison. The discrepancy is mainly due to the different surface roughness geometries. Nevertheless, the slopes of the plots shown are approximately the same, which indicates that the present result is in good agreement with results of previous investigators.

Conclusions

Confined fully developed turbulent flow in horizontal annuli with/without roughened surfaces has been made of the heat transfer mechanism which occurs when water flows over a surface fitted with helical surface roughness. The effect of roughness height, its pitch, and Reynolds number on a fully developed heat transfer coefficient was determined, and an entrance effect was observed. The thermal performance is found to be dependent on both surface roughness spacing and height. The increase in heat transfer coefficient, relative to the coefficient of smooth annuli, was about 230–350% for all the cases studied. Two simple correlations of the thermal performance of the present study can be briefly expressed as

$$\mathrm{Nu} = 5.90(\mathrm{Re_H})^{0.53}\left(\frac{e}{D_H}\right)^{0.54} \quad \text{for } \frac{p}{e} = 1.44$$

$$\mathrm{Nu} = 5.23(\mathrm{Re_H})^{0.51}\left(\frac{e}{D_H}\right)^{0.33} \quad \text{for } \frac{p}{e} = 2.88$$

for $3000 \leqslant \mathrm{Re_H} \leqslant 30{,}000$, respectively, with an average deviation of $\pm 6\%$ and $\pm 9\%$, respectively, from the original data. The result may provide thermal design data for conventional heat exchangers with the same configuration roughened surfaces.

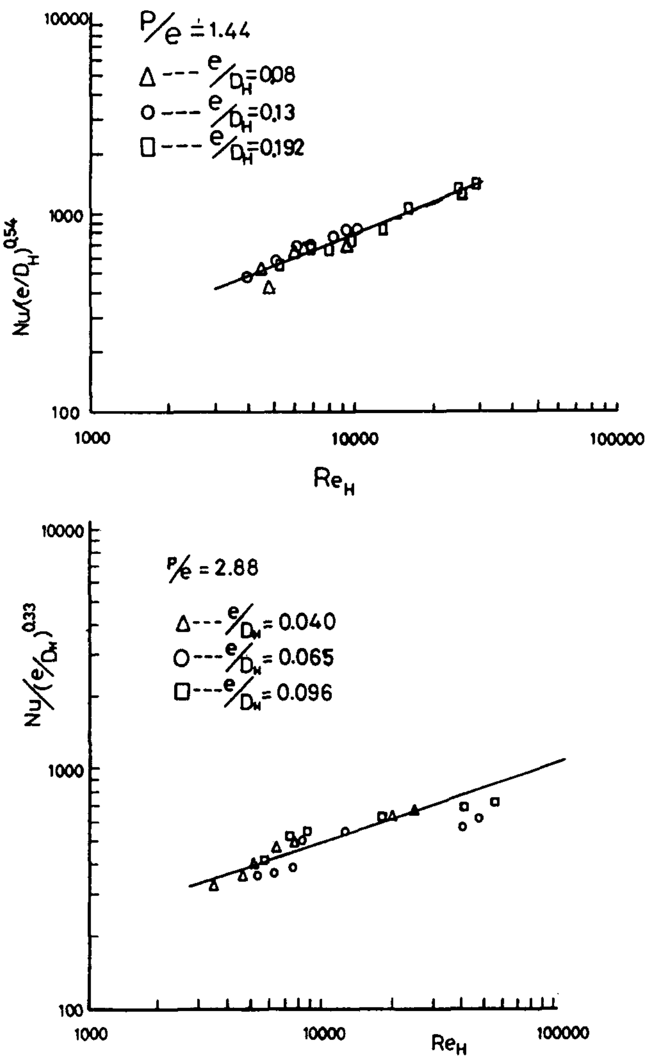

Figure 5 Plot of $\mathrm{Nu}/(e/D_H)$ versus $\mathrm{Re_H}$

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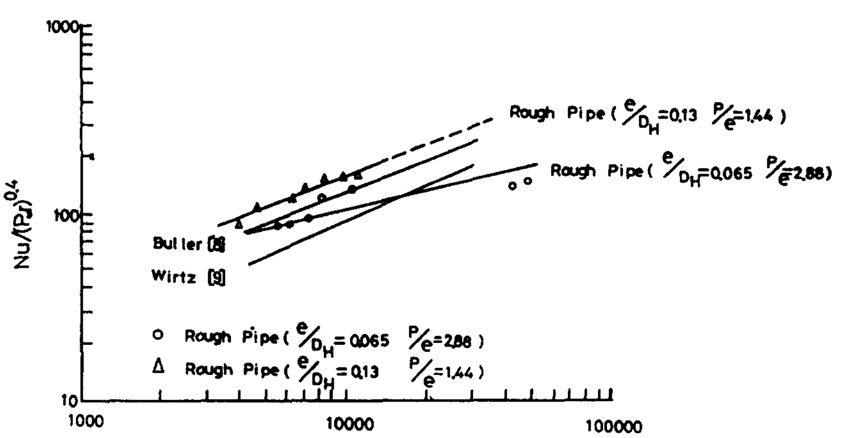

Figure 6 Comparisons of the present results with Buller[9] and Wirtz[10]