JOURNAL OF THERMOPHYSICS AND HEAT TRANSFER
Vol. 8, No. 3, July–Sept. 1994

Augmented Heat Transfer in Triangular Ducts with Full and Partial Ribbed Walls

Y. M. Zhang* and W. Z. Gu†
Chinese Academy of Sciences, Beijing 100080, People's Republic of China
and
J. C. Han‡
Texas A&M University, College Station, Texas 77843

Fully developed heat transfer coefficients and friction factors in triangular ducts with full and partial ribbed walls were experimentally obtained for Reynolds numbers between 10^4–10^5. The vertex corners of the triangular ducts were 90, 55, and 35 deg. Five rib configurations were tested: full and partial 90-deg transverse ribs, full and partial 45-deg crossed ribs on three-wall, and partial 45-deg crossed ribs on two-wall of the triangular duct. The results show that heat transfer coefficients were higher on the ribbed walls and the 90-deg vertex corner of the triangular duct than on the 55-deg vertex corner, and subsequently higher than those on the 35-deg vertex corner. The heat transfer coefficients and friction factors in triangular ducts with partial ribbed walls (90- or 45-deg ribs) were 10% higher than those with full ribbed walls. The results also show that heat transfer coefficients and friction factors were higher in triangular ducts with 90-deg transverse ribs (full or partial) than those with 45-deg crossed ribs. The semi-empirical friction and heat transfer correlations in ribbed triangular ducts were obtained.

Nomenclature

A_c = cross-sectional area of triangular channel
A_s = heat transfer surface area
C_p = specific heat of air
D = hydraulic diameter of triangular channel
e = rib height
e^+ = roughness Reynolds number
f = friction factor in ribbed triangular channel
f_s = friction factor in smooth triangular channel
f_0 = friction factor in fully developed smooth tube flow
G = mass flux ρV
$G(e^+)$ = heat transfer roughness function
g_c = conversion factor
h = heat transfer coefficient
k = thermal conductivity of stainless steel
L = length of triangular channel
p = rib pitch
p_r = Prandtl number of air
q = net heat transfer rate, Eq. (3), W
R = friction roughness function
Re = Reynolds number, $\rho DV/\mu$
St_r = Stanton number on ribbed flat walls and 90, 55, and 35 corners
St_s = Stanton number in smooth triangular duct
St_0 = Stanton number in fully developed smooth tube flow
T_b = local bulk mean temperature
T_i = inlet temperature of air
T_w = corrected local wall temperature
T_{w0} = outer local wall temperature
T_0 = average outlet temperature of air
V = mean velocity of air
X = distance from entrance of heated triangular channel
α = rib angle of attack
Δp = pressure drop across the entire test channel
δ = stainless steel thickness
μ = average dynamic viscosity of air
ρ = average density of air

Introduction

CASTING turbulence promoters/ribs inside cooling channels is an effective technique to augment heat transfer. The ribs penetrate the laminar sublayer and promote local wall turbulence due to flow separation from the ribs and reattachment between the ribs. Therefore, the heat transfer rate is enhanced. The effects of rib orientation and the vertex corner on heat transfer and pressure drop in triangular channels with two or three rib-roughened walls are investigated.

Turbulent heat transfer and friction in channels with rib turbulators have been extensively studied over the past 20 yr. Considerable data has been reported for rib-roughened heat transfer and pressure drop in flow channels of different cross-sectional areas: 1) flow in ribbed circular tubes,[1–3] 2) in ribbed rectangular ducts,[4–12] and 3) in ribbed annular ducts.[13,14] The results show that transverse ribs produce a slightly higher heat transfer rate than crossed ribs,[11,15] whereas angled ribs perform much better than transverse ribs.[2,6,7] Broken angled ribs give a slightly higher heat transfer rate than full angled ribs, while broken V-shaped ribs give the best heat transfer performance.[12] Based on previous studies, the effects of rib height, rib spacing, rib shape, and rib angle orientation on heat transfer coefficient and friction factor over a wide range of Reynolds numbers have been well established for circular, rectangular, and annual channels. Semi-empirical friction and heat transfer correlations have been developed for heat transfer designers. However, high-performance enhanced heat transfer surfaces and friction characteristics are very important in some applications. For example, the heat transfer enhancement from the leading edge cooling of a gas turbine blade (Fig. 1) is desirable on two or three walls and the vertex corners of the triangular channel. The heat transfer and fric-

Received July 2, 1993; revision received Nov. 19, 1993; accepted for publication Nov. 19, 1993.

*Assistant Professor, Institute of Engineering Thermophysics.
†Professor, Institute of Engineering Thermophysics.
‡HTRI Professor, Department of Mechanical Engineering. Senior Member AIAA.

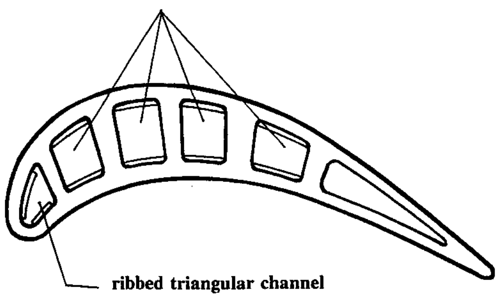
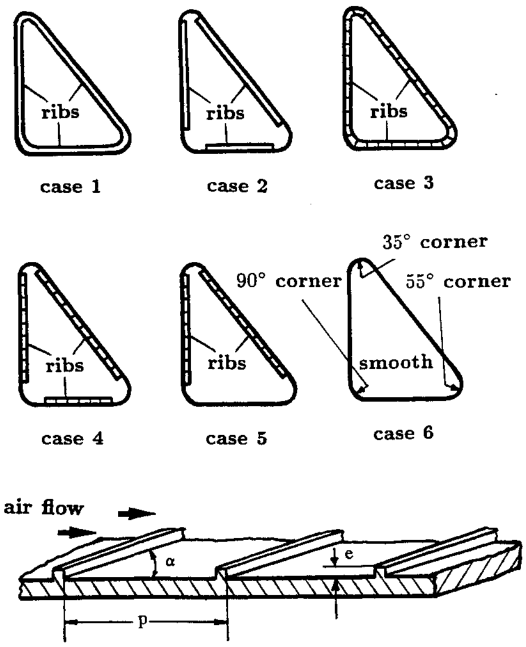


Fig. 1 Cross-sectional view of advanced gas turbine blade.



CASE NO.	WALL CONDITIONS	p/e	e/D	α
1	three-wall full ribs	9.2	0.012	90°
2	three-wall partial ribs	9.2	0.012	90°
3	three-wall full ribs	9.2	0.012	45°
4	three-wall partial ribs	9.2	0.012	45°
5	two-wall partial ribs	9.2	0.012	45°
6	smooth	-	-	-

Fig. 2 Schematic of triangular channels with full and partial ribbed walls.

tion characteristics in this kind of channel may be different from circular, rectangular, and annular channels. Metzger et al.[15] reported the results of heat transfer in equilateral triangular channels with angled rib roughness on two walls. The results show that heat transfer coefficients were low when the crossed ribs were applied on the two walls.

There is limited research work devoted to heat transfer enhancement in triangular channels with rib-roughened walls. Nevertheless, the channels of such geometries may be used to cool gas turbine blades, heat exchangers, and high-temperature heat transfer devices. Therefore, it is important to understand heat transfer and friction characteristics in triangular channels.

The objective of this study is to investigate the effect of vertex corners and full and partial ribbed walls on heat transfer and friction in triangular channels. Six triangular channels were tested. Figure 2 shows a cross section of the channels. Two of the channels have transverse (90 deg) rib roughness on the walls (case 1: full ribs on three flat walls and on three vertex corners; case 2: ribs are discontinued on the vertex corners). The remaining three channels are roughened by 45-deg crossed ribs on the walls (case 3: full ribs on three flat walls and on three vertex corners; case 4: ribs are discontinued on three vertex corners; case 5: ribs are discontinued on three corners and removed from one flat wall). The last one (case 6) is a smooth triangular channel that has the same shape and size as the five rib-roughened channels. The heat transfer coefficients on the ribbed flat walls and vertex corners of the triangular channels are obtained and the heat transfer vs pressure drop performances are compared for rib-roughened triangular channels. The heat transfer and friction similarity laws for flow in circular tubes have been extended to the triangular channels. The semi-empirical friction and heat transfer correlations are provided.

Experimental Apparatus and Data Reduction

Test Apparatus

Figure 2 shows the vertex corners of the triangular channel are 90, 55, and 35 deg. The hydraulic diameter is 22 mm for each test channel. The rib height-to-hydraulic diameter ratio e/D is 0.012 and the rib pitch-to-height ratio p/e is 9.2. Nine thermocouples are located on the vertex corners and nine on the flat walls in the thermally fully developed region (at 60, 70, and 80% of the channel length, or at X/D = 27, 32, and 36) to measure wall temperature (Fig. 3a). Figures 3b and 3c show the rib roughness distribution for 90-deg transverse ribs and 45-deg crossed ribs. All test channels are made of stainless steel plates. Each stainless steel plate is 1 m long (L = 1 m) and 2 mm thick (δ = 2 mm). To fabricate the test channel, the ribs are first machined on the stainless plate (e = 0.26 mm) in a desired pattern and then bent into a triangular shape and soldered along the gap. A foil heater is cemented on the outside surface of the test channel. The entire test channel is wrapped in fiberglass insulating material. Upstream of the test channel is an unheated smooth entrance section made of Plexiglas® (not shown) with the same cross section and length as the test channel. This entrance channel serves to establish hydrodynamically fully developed flow at the entrance of the heated triangular channel. Two pressure tapes measure static pressure drop across the test section.

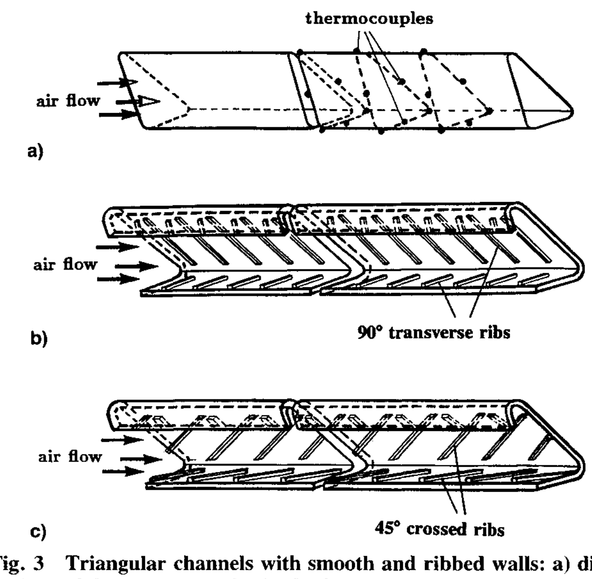


Fig. 3 Triangular channels with smooth and ribbed walls: a) distributions of the thermocouples in the fully developed region (case 6), b) configuration of the 90-deg transverse ribs in the triangular channel (case 2), and c) configuration of the 45-deg crossed ribs in the triangular channel (case 4).

Data Reduction

A micromanometer measured the pressure drop across the test section. The friction factor in fully developed channel flow is found by measuring the Δp and G of air. The friction factor can be calculated from

$$f = \Delta p/[4(L/D)(G^2/2\rho g_c)] \tag{1}$$

The friction factor f is based on isothermal conditions (test without heating). The uncertainties of Δp, L/D, and G were 5, 3, and 5%, respectively. The maximum uncertainty in the friction factor, based on the method of Kline and McClintock,[16] is estimated to be less than 7% for Reynolds numbers greater than 10^4. The friction factor f of the present study is normalized by the friction factor for fully developed turbulent flow in smooth circular tubes ($10^4 < Re < 10^6$) proposed by Blasius[17] as

$$f/f_0 = f/(0.046Re^{-0.2}) \tag{2}$$

The net heat transfer rate is calculated from

$$q = C_p GA_c(T_0 - T_i) \tag{3}$$

where T_0 is obtained by averaging 10 temperature readings from different locations at the same exit plane of the test channel. The local heat transfer coefficient is calculated from the net heat transfer rate per unit surface area to cooling air, the corrected local wall temperature on each measured section, and local bulk mean air temperature as

$$h = q/[A_s(T_w - T_b)] \tag{4}$$

T_{w0} is read from each cross-sectional thermocouple output. The inner T_w used in Eq. (4) is calculated by

$$T_w = T_{w0} - (q\delta/kA_s) \tag{5}$$

The axial and spanwise heat conductions are not considered in this study. Equation (4) is used for calculating the heat transfer coefficient on flat walls and vertex corners of the triangular channel. The A_s is based on the smooth surface area but does not include the area increases from ribs. The heat transfer coefficients are based on the average values of three axial locations in the thermally fully developed region of the triangular channel. The T_{w0} is kept between 120–125°C for different locations of the test channel. The local bulk mean air temperature used in Eq. (4) is calculated assuming a linear air temperature rise along the flow channel. The inlet bulk mean air temperature is 26–30°C depending on the test conditions. The maximum heat loss from the rib-roughened flat wall and vertex corner is estimated to be less than 5 and 6%, respectively, for Reynolds numbers greater than 10^4. The net heat input agrees with the air enthalpy rise along the flow channel. The uncertainties of q, A_s, and $(T_w - T_b)$ are 6, 3, and 5%, respectively. By using the uncertainty method of Kline and McClintock,[16] the maximum uncertainty in the heat transfer coefficient is estimated to be less than 9%, while the maximum uncertainty in the Stanton number is estimated to be less than 10% for Reynolds numbers larger than 10^4.

The Stanton number of the present study is normalized by the Stanton number for fully developed turbulent flow in smooth circular tubes correlated by McAdams[17] as

$$St_r/St_0 = (h/\rho VC_p)/(0.023Re^{-0.2}Pr^{-0.6}) \tag{6}$$

Friction and Heat Transfer Correlations

Webb et al.[1] applied friction and heat transfer similarity laws, which are derived from the wall law for flow over rough surfaces, to correlate the friction factors and heat transfer coefficients for turbulent flow in circular tubes with repeated rib roughness. If the same method can be applied to rib roughened triangular channels, the friction and heat transfer similarity laws are

$$R = (f/2)^{-1/2} + 2.5\ln(2e/D) + 3.75 \tag{7}$$

$$G(e^+) = R + [(f/2St_r) - 1]/(f/2)^{1/2} \tag{8}$$

$$e^+ = (e/D)Re(f/2)^{1/2} \tag{9}$$

where R is independent of e^+ in the fully rough condition. However, $G(e^+)$ is increased with increasing e^+. Equations (7) and (8) imply that f and St_r for any geometrically similar roughness family (i.e., any e/D ratio) may be correlated by R and $G(e^+)$. In addition, the effect of nongeometric similar factors such as rib angle of attack (i.e., α) on the R and $G(e^+)$ correlations can be experimentally determined.

Experimental Results and Discussions

Heat Transfer and Pressure Drop

Figure 4 shows the Stanton number and friction factor vs Reynolds number for the smooth triangular channel (case 6). The heat transfer coefficients on the flat walls and 90-deg corner are higher than those on the 55- and 35-deg corners; however, the 35-deg corner is the lowest. This may be due to lower velocity and turbulence and thicker laminar sublayer in the 35-deg corner region. The channel-averaged Stanton numbers and friction factors (black symbols) have good agreement with those of the smooth tube correlations.

Figure 5 shows the Stanton number vs Reynolds number for five rib-roughened triangular channels. The results show the Stanton number decreases slightly with increasing Reynolds number. The heat transfer coefficients on the flat ribbed walls and 90-deg corners are higher than those on the 55-deg corners and, subsequently, higher than those on the 35-deg corners. The Stanton numbers for cases 2 and 4 are about 10% higher than those for cases 1 and 3, respectively. This is because the three-wall partial ribs induce higher turbulence and reduce the laminar sublayer effect near corners.

In general, the heat transfer coefficients for 90-deg transverse ribs (cases 1 and 2) on the flat walls and vertex corners are higher than those for 45-deg crossed ribs (cases 3 and 4) on the corresponding locations. This occurs when the 45-deg ribs are originally machined on the flat stainless steel plate

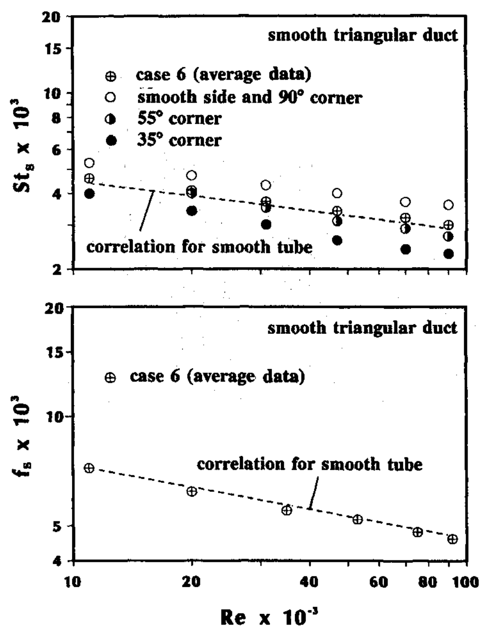


Fig. 4 Stanton numbers and friction factors vs Reynolds number for the smooth triangular channel.

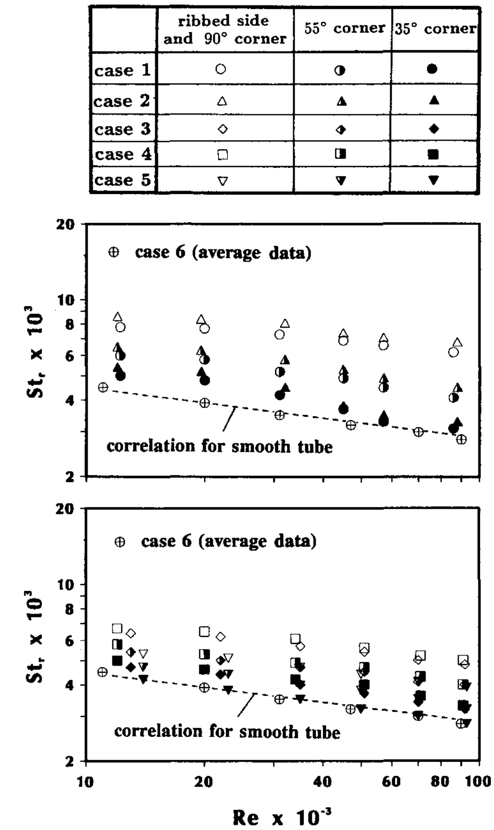


Fig. 5 Stanton numbers vs Reynolds numbers for five ribbed triangular channels.

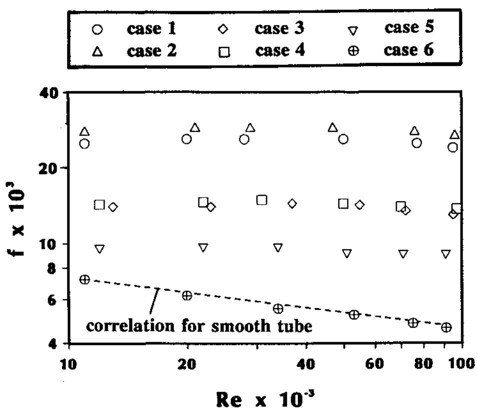


Fig. 6 Friction factors vs Reynolds numbers for five ribbed triangular channels.

and bent into a triangular shape. The orientations of the ribs on each wall cross each other in the triangular channel (see Fig. 3c). Zhang et al.,[9] Han et al.,[11] and Metzger et al.[15] have reported on low-heat transfer coefficients with crossed ribs, which may happen because secondary flow induced by the 45-deg ribs on each wall may counteract each other and, therefore, reduce the heat transfer coefficients. Figure 5 also shows that the heat transfer coefficients are lowest when the ribs are removed from one flat wall of the triangular channel (case 5). This is because the two-wall ribs have less turbulence effect than the three-wall ribs.

Figure 6 shows friction factors for two 90-deg rib configurations and three 45-deg crossed rib configurations. The pressure drops across the test section are measured at the unheated flow conditions. The friction factor correlations for fully developed turbulent flow in smooth circular tubes is also included for comparison. The 90-deg transverse rib channels provide higher friction factors than the 45-deg crossed rib channels over the range of Reynolds numbers studied. This is because the 90-deg transverse ribs may produce higher form drag than the 45-deg crossed ribs. The friction factors for the three-wall partial ribs are slightly higher than those for the three-wall full ribs of the test channel, which may occur from the discontinued ribs creating more pressure drop than the continued ribs near corners. Figure 6 also shows the friction factors for case 5 with two-wall partial ribs lower than those for case 4 with three-wall partial ribs. This is because the two-wall ribs have a lower turbulence effect than the three-wall ribs.

Heat Transfer Performance Comparison

Figure 7 represents the Stanton number ratio vs the friction factor ratio for the rib roughened triangular channels over the range of Reynolds numbers between 10^4–10^5. The results show that, in general, the Stanton number ratio (heat transfer augmentation) increases while increasing the friction factor ratio (pressure drop increment). The Stanton number ratio decreases only slightly with increasing Reynolds number. However, the friction ratio greatly increases with the Reynolds number. The heat transfer augmentation (St_r/St_0) for the three-wall partial ribs is about 10% higher than those for the three-wall full ribs. The 90-deg transverse rib roughness provides 2.0–2.3 times the heat transfer augmentation on flat walls and 90-deg corners with 3.6–6.6 times the pressure drop penalty in the triangular channels. The 45-deg crossed rib roughness provides 1.6–1.8 times the heat transfer augmentation on flat walls, and 90-deg corners with 1.6–2.5 times the pressure drop penalty in the triangular channels. The Stanton number ratio on 55-deg corners for the 90-deg transverse rib ducts is 1.4–1.7 times, whereas the St_r/St_0 on 55-deg corners for the 45-deg crossed rib ducts is 1.2–1.5 times.

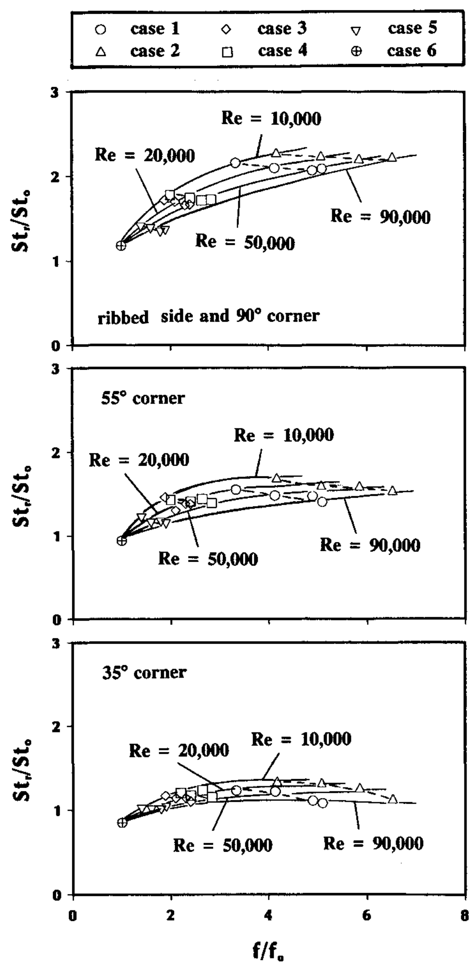


Fig. 7 Stanton number ratio vs friction factor ratio for five ribbed triangular channels.

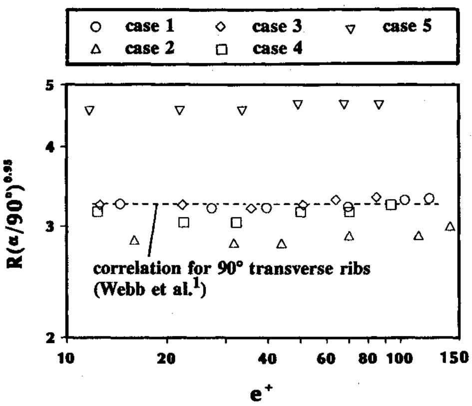


Fig. 8 Friction factor correlation.

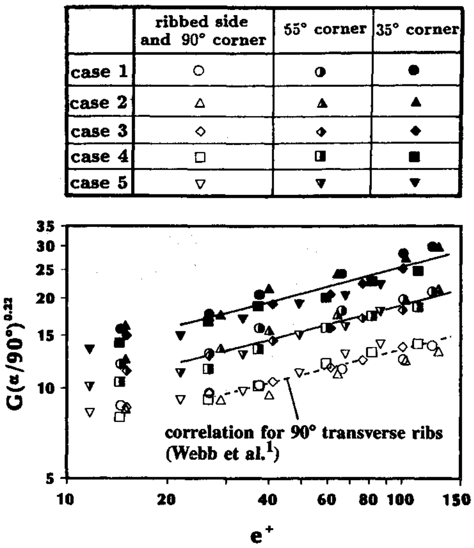


Fig. 9 Heat transfer correlation.

The Stanton number ratio on 35-deg corners is quite low (1.1–1.4 times) for both the 90-deg transverse rib ducts and 45-deg crossed rib ducts.

Heat Transfer and Friction Correlations

Figure 8 shows the R vs the e^+ for rib roughened triangular channels over the range of Reynolds numbers studied. Based on the rib roughened channel analysis discussed earlier, the wall similarity laws correlate the friction and heat transfer data for fully developed turbulent flow in the triangular channels of the present study. According to the friction similarity law derived in Eq. (7), the measured f, the rib height-to-hydraulic diameter ratio e/D, and the Reynolds number can be correlated with R. Figure 8 shows that the friction roughness function for 90-deg full ribs (case 1) is very close to the previous correlation for 90-deg transverse ribs in the circular tubes, which was reported by Webb et al.[1] The values of R for three-wall partial ribs (cases 2 and 4) are lower than those for three-wall full ribs (cases 1 and 3). This is because cases 2 and 4 have higher friction factors than those of cases 1 and 3 [seen in Eq. (7)]. The effect of the rib angle on R can be expressed as

$$R = 3.2(\alpha/90°)^{-0.95} \tag{10}$$

for three-wall full ribs.

Equation (10) is valid for $15 \le e^+ \le 150$, $P/e = 9.2$, $\alpha =$ 45 and 90 deg, and $10^4 \le Re \le 10^5$. The deviation of Eq. (10) is $\pm 5\%$. After R is experimentally correlated from Eq. (10), f can be predicted by Eqs. (7) and (9) for the given e/D, p/e, and Re. Note that R in Eq. (10) is independent of e^+, which implies that the friction factor is almost independent of the Reynolds number. R for case 5 (two-wall partial ribs) is higher than the three-wall partial ribs. This is because the friction factor for case 5 is lower.

Figure 9 shows the averaged $G(e^+)$ vs e^+ for rib roughened walls over the range of Reynolds numbers studied. According to the heat transfer similarity law derived in Eq. (8), St_r, f, and R can be correlated with $G(e^+)$. $G(e^+)$ increases with e^+ for all rib configurations studied. The heat transfer roughness function for the ribbed side and 90-deg corner is very close to the previous correlation developed for 90-deg transverse ribs in circular tubes.[1] The values of G for 55- and 35-deg corners are much higher than the ribbed side and 90-deg corner, which implies that the ribbed side and 90-deg corner have a higher Stanton number than the 55- and 35-deg corners [seen in Eq. (8)]. $G(e^+)$ is expressed as

for ribbed side and 90-deg corner

$$G(e^+) = 3.70(e^+)^{0.28}(\alpha/90°)^{-0.22} \tag{11}$$

for 55-deg corner

$$G(e^+) = 5.02(e^+)^{0.28}(\alpha/90°)^{-0.22} \tag{12}$$

for 35-deg corner

$$G(e^+) = 6.67(e^+)^{0.28}(\alpha/90°)^{-0.22} \tag{13}$$

Equations (11–13) are valid for $30 \le e^+ \le 150$, $P/e = 9.2$, $\alpha = 45$ and 90 deg, and $10^4 \le Re \le 10^5$. The deviation of Eqs. (11–13) is ± 10, ± 13, and $\pm 15\%$, respectively. After $G(e^+)$ is experimentally correlated from Eqs. (11–13), St_r can be predicted by combining Eqs. (7–9) for a given e/D, p/e, and Re.

Concluding Remarks

The effects of rib turbulators on friction factors and heat transfer coefficients in triangular channels with full and partial ribbed walls have been investigated. The main findings are as follows:

1) The heat transfer coefficients on the 90-deg corner are the same as those on the flat ribbed walls of the triangular channels. The heat transfer coefficients on the ribbed flat walls and 90-deg corner are 30% higher than those on the 55-deg corner, and 50% higher than those on the 35-deg corner of the triangular channel.

2) The heat transfer coefficients are enhanced about 10% if the ribs are removed from the corners of the triangular channel (partial ribs) for both the 90-deg transverse ribs and 45-deg crossed ribs on the flat walls. The friction factors also increase about 10%.

3) The 90-deg rib roughened wall enhances heat transfer about 2.0–2.3 times, whereas the 45-deg crossed rib roughened wall enhances heat transfer 1.6–1.8 times on the flat wall and 90-deg corners of the triangular channels. The 90-deg rib roughened wall gives a pressure drop penalty of 3.6–6.6 times, and the 45-deg crossed rib roughened wall gives a pressure drop penalty of 1.6–2.5 times.

4) The heat transfer coefficients on the 55- and 35-deg corners of the triangular channel are insensitive to both the Reynolds number and rib configuration. The heat transfer coefficients on the 55- and 35-deg corners are enhanced 1.4–1.7 times and 1.2–1.5 times, respectively.

5) Semi-empirical friction and heat transfer correlations based on the rib channel analysis are obtained for the triangular channels with full and partial rib walls. The correlations are valid for $30 \le e^+ \le 150$, $P/e = 9.2$, $\alpha = 45$ and 90 deg, and $10^4 \le Re \le 10^5$.

Acknowledgments

This project is sponsored by the China National Science Foundation and performed at the Institute of Engineering Thermophysics, Chinese Academy of Sciences, Beijing, China.

References

[1]Webb, R. L., Eckert, E. R. G., and Goldstein, R. J., "Heat Transfer and Friction in Tubes with Repeated-Rib Roughness," *International Journal of Heat and Mass Transfer*, Vol. 14, No. 4, 1971, pp. 601–617.

[2]Gee, D. L., and Webb, R. L., "Forced Convection Heat Transfer in Helically Rib-Roughened Tubes," *International Journal of Heat and Mass Transfer*, Vol. 23, No. 8, 1980, pp. 1127–1136.

[3]Sethumadhavan, R., and Raja Rao, M., "Turbulent Flow Heat Transfer and Fluid Friction in Helical-Wire-Coil-Inserted Tubes," *International Journal of Heat and Mass Transfer*, Vol. 26, No. 12, 1983, pp. 1833–1844.

[4]Burggraf, F., "Experimental Heat Transfer and Pressure Drop with Two-Dimensional Turbulence Promoter Applied to Two Opposite Walls of a Square Tube," *Augmentation of Convection Heat and Mass Transfer*, edited by A. E. Bergles and R. L. Webb, American Society of Mechanical Engineers, New York, 1970, pp. 70–79.

[5]Han, J. C., "Heat Transfer and Friction in Channels with Two Opposite Rib-Roughened Walls," *Journal of Heat Transfer*, Vol. 106, No. 4, 1984, pp. 774–781.

[6]Han, J. C., Park, J. S., and Lei, C. K., "Heat Transfer Enhancement in Channels with Turbulence Promoters," *Journal of Engineering Gas Turbines Power*, Vol. 107, No. 3, 1985, pp. 629–635.

[7]Han, J. C., and Park, J. S., "Developing Heat Transfer in Rectangular Channels with Rib Turbulators," *International Journal of Heat and Mass Transfer*, Vol. 31, No. 1, 1988, pp. 183–195.

[8]Han, J. C., "Heat Transfer and Friction Characteristics in Rectangular Channels with Rib Turbulators," *Journal of Heat Transfer*, Vol. 110, No. 2, 1988, pp. 321–328.

[9]Zhang, Y. M., Gu, W. Z., and Xu, H. Q., "Enhancement of Heat Transfer and Flow Characteristics in Rib Roughened Rectangular Channels," *Journal of Engineering Thermophysics*, Vol. 5, No. 3, 1984, pp. 275–280.

[10]Zhang, Y. M., Gu, W. Z., and Liu, C. C., "Enhancement of Heat Transfer in the Cooling of Gas Turbine Blades," *Journal of Aerospace Power*, Vol. 2, No. 4, 1987, pp. 355–357.

[11]Han, J. C., Zhang, Y. M., and Lee, C. P., "Augmented Heat Transfer in Square Channels with Parallel, Crossed, and V-Shaped Angled Ribs," *Journal of Heat Transfer*, Vol. 113, No. 3, 1991, pp. 590–596.

[12]Han, J. C., and Zhang, Y. M., "High Performance Heat Transfer Ducts with Parallel Broken and V-Shaped Broken Ribs," *International Journal of Heat and Mass Transfer*, Vol. 35, No. 2, 1992, pp. 513–523.

[13]Donne, M. D., and Meyer, L., "Turbulent Convective Heat Transfer from Rough Surfaces with Two-Dimensional Rectangular Ribs," *International Journal of Heat and Mass Transfer*, Vol. 20, No. 6, 1977, pp. 583–620.

[14]Meyer, L., "Thermohydraulic Characteristics of Single Rods with Three-Dimensional Roughness," *International Journal of Heat and Mass Transfer*, Vol. 25, No. 7, 1982, pp. 1043–1058.

[15]Metzger, D. E., and Vedula, R. P., "Heat Transfer in Triangular Channels with Angled Roughness Ribs on Two Walls," *Journal of Experimental Heat Transfer*, Vol. 1, No. 1, 1987, pp. 31–44.

[16]Kline, S. J., and McClintock, F. A., "Describing Uncertainties in Single-Sample Experiments," *Mechanical Engineering*, Vol. 75, Jan. 1953, pp. 3–8.

[17]Rohsenow, W. M., and Choi, H., *Heat, Mass and Momentum Transfer*, Prentice-Hall, Englewood Cliffs, NJ, 1961, pp. 56–59, 192, 193.