[5]Nigen, J. S., and Amon, C. H., "Effect of Material Composition and Localized Heat Generation on Conjugate Heat Transport in Grooved Channels," American Society of Mechanical Engineers, Paper 93-WA/HT-15, Nov. 1993.

Effect of Rib Profiles on Turbulent Channel Flow Heat Transfer

Pankaj R. Chandra* and Michael L. Fontenot*
McNeese State University,
Lake Charles, Louisiana 70609
and
Je-Chin Han*
Texas A&M University, College Station, Texas 77843

Introduction

REPEATED ribs have been used as the promoters of turbulence to enhance the heat transfer to the flow of coolants in channels. Several publications have addressed the comprehensive review of turbine blade cooling and the analysis of heat transfer and friction characteristics of flow in channels with two opposite rib-roughened walls. The effects of flow Reynolds number and rib geometry (rib height e, rib spacing P, rib angle of attack α, and rib orientation) on heat transfer and pressure drop in the fully developed region of square, rectangular, and triangular channels have been investigated.[1-4] Semiempirical correlations over a wide range of rib geometry for the friction and heat transfer design calculations are derived from the law-of-the-wall similarity for flow over rough surfaces. A detailed analysis of the application of the similarity laws in the case of rectangular channels is presented by Han.[1,2] Also, studies of the effect of a number of channel-ribbed walls on heat and friction characteristics have been reported.[5,6]

Experiments with some nonrectangular rib shapes have been reported.[7,8] The previously mentioned studies were limited in the number of rib shapes and were for different conditions (rib height to channel hydraulic diameter ratio e/D_h, and Reynolds number Re). This study focuses on the effect of different rib profiles on turbulent channel flow heat transfer and friction characteristics.

Experimental Setup and Procedure

The test channel is a 5.08 × 5.08 cm square cross section (Fig. 1a) and is 101.6 cm long. The test channel is made of 10 10.16-cm-long sections of 0.64-cm-thick copper plates, separated by 0.08-cm-thick balsa wood to reduce both the streamwise and circumferential heat conduction effects. Copper ribs with a height of 0.64 cm ($e/D_h = 0.125$), and equally spaced at 6.4 cm ($P/e = 10$), are glued with silicone adhesive onto the two opposite walls of the channel (Fig. 1b). Rib profiles are shown in Fig. 1c. The walls are heated individually with electric strip heaters, which are embedded and flatly placed between the copper and wood plates to ensure good contact. Heaters are independently controlled by a transformer and provide a constant heat flux to the walls. Each section of the walls has a thermocouple in the center. The test section is enclosed by 3.81-cm-thick styrofoam insulation.

Received May 17, 1996; presented as Paper 96-1840 at the AIAA 31st Thermophysics Conference, New Orleans, LA, June 17–20, 1996; revision received July 11, 1997; accepted for publication July 11, 1997.

*Professor, Department of Mechanical Engineering.

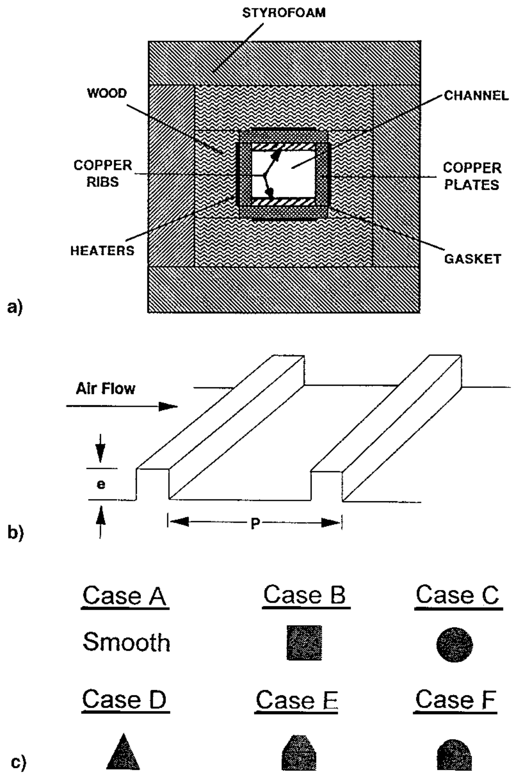

Fig. 1 a) Cross section of test channel, b) ribbed wall geometry, and c) cases with varying rib profiles.

Temperatures of the air entering and leaving the test channel are the average values of the readings recorded by a traversing probe. When thermal steady state is reached and the temperature of the four sections at any cross section of the channel is about the same, temperature and pressure data are recorded. Details are available in Refs. 5 and 6. The same procedure is repeated for the range of Reynolds numbers and for all rib configurations.

Data Reduction

Reference 6 may be referred to for the equations to calculate friction factor f, and the local heat transfer coefficient based on the area of a smooth channel h. The friction factor and Nusselt number are normalized by the respective values of the friction factor f_0, and Nusselt number Nu_0, for fully developed smooth pipe turbulent flow.

The maximum uncertainties in the heat transfer coefficient and friction factor are estimated to be ±7 and ±8%, respectively, using the uncertainty estimation method of Kline and McClintock.[9] The maximum heat loss to the atmosphere is about 6% of the total heat supplied to the channel walls.

For turbulent flow in square channels, f can be expressed as a weighted average of the four-sided smooth channel friction factor f_{ss}, and the four-sided ribbed channel friction factor f_{rr}. On the basis of the previous assumption, the relationship between three friction factors, f, f_{ss}, and f_{rr}, is given by

$$f = (f_{rr} + f_{ss})/2 \tag{1}$$

The friction roughness function $R(e^+)$, and heat transfer roughness function $G(e^+)$, can be experimentally determined and correlated by f, and the ribbed wall Stanton number St_r, for fully developed turbulent flow in a square channel with nonrectangular ribbed walls.

According to Han,[2] the laws of the wall can also be applied to fully developed turbulent flow in rectangular channels with repeated-rib rougheners and with different channel aspect ratios. Thus, the friction and heat transfer similarity laws in square channels with four-sided ribbed walls can be expressed by Eqs. (2–4):

$$R(e^+) = (f_{rr}/2)^{-1/2} + 2.5\,\ell n(2e/D_h) + 2.5 \qquad (2)$$

$$G(e^+) = [(f_{rr}/2)^{1/2}]/St_r + 2.5\,\ell n(2e/D_h) + 2.5 \qquad (3)$$

$$e^+ = (e/D_h)Re(f_{rr}/2)^{1/2} \qquad (4)$$

Results and Discussion

Channel-Average Heat Transfer and Friction

Figure 2 shows the average Nusselt number ratio (heat transfer enhancement) vs Reynolds number for cases A to F. The ribbed-sided and smooth-sided Nusselt number ratios Nu_r/Nu_0 and Nu_s/Nu_0, respectively, are the average values of the sectional ribbed wall smooth wall Nusselt number ratios for the fully developed flow ($X/D_h = 6–18$; X: axial distance from the channel entrance). The heat transfer coefficient is based on the actual wall area exposed to the flow. The channel Nusselt number ratio decreases as the Reynolds number increases.

Case B, the channel with square ribs, gives the highest Nusselt number ratio, and cases C and F, with full-circular and semicircular ribs, respectively, yielded the lowest Nusselt number ratios. The channel with square ribs exhibits 24% higher heat transfer than that with circular ribs. Figure 2 compares the friction factor ratio for all of the rib shapes studied. The ratio increases with increasing Reynolds number. Square ribs create 33% more pressure drop than that created by circular ribs, whereas triangular ribs are creating more pressure drop than slant-edged ribs.

These graphs illustrate the well-known concept that turbulators of increasingly rough surface (greater number of sharp corners) yield increasingly higher heat transfer coefficient and friction factor.

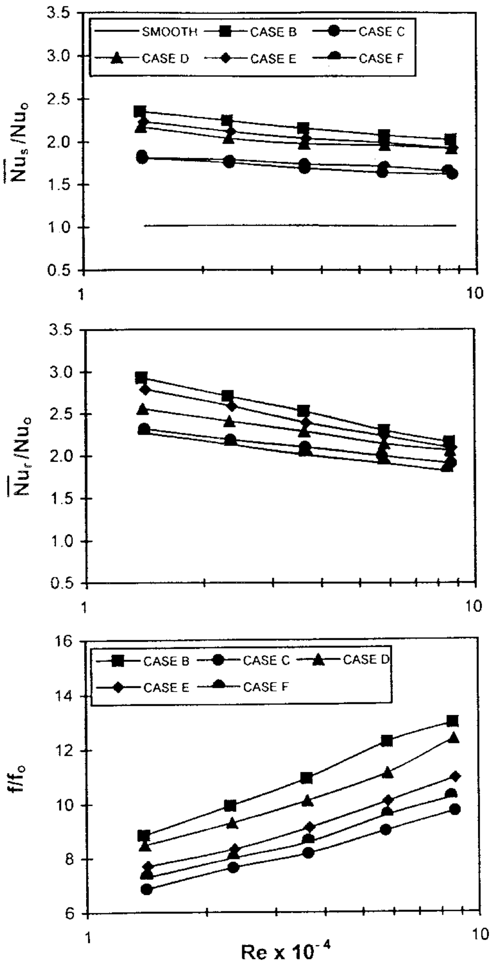

Fig. 2 Average Nusselt number ratio and friction factor ratio vs Reynolds number.

Heat Transfer Performance

Figure 3 shows that the square ribs produce the higher heat transfer augmentation for a given friction factor than any other rib profile. This means that the heat transfer performance decreases with increasing Reynolds number. There are extremely small differences in the performances of triangular and slant-edged ribs and the performances of circular and semicircular ribs. However, as expected, there is a wide spread between the two families of rib shapes, with the edged family of ribs naturally performing more closely to square ribs than the curved family of ribs.

Friction and Heat Transfer Correlations

The wall similarity laws were employed to correlate the friction and heat transfer data for fully developed turbulent flow. The plots of $R(e^+)$ and $G(e^+)$ vs the roughness Reynolds number e^+, are shown in Fig. 4. The friction roughness function

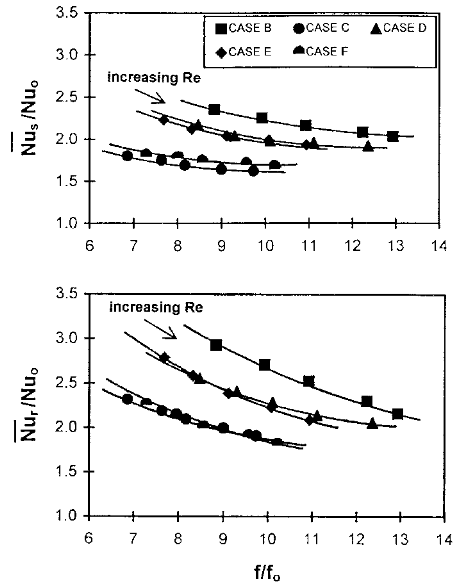

Fig. 3 Nusselt number ratio vs friction factor ratio.

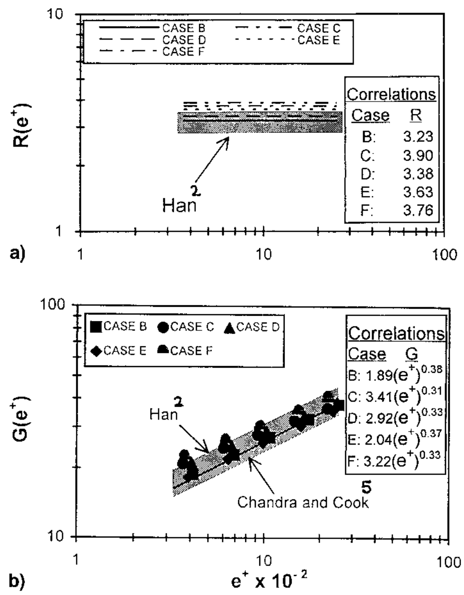

Fig. 4 Ribbed channel a) friction and b) heat transfer correlations.

correlation for the channel with square ribs (case B), is given by

$$R(e^+) = 3.23, \qquad 2541 \underline{>>} e^+ \underline{>>} 415 \tag{5}$$

The channel with full circular ribs (case C) yields the highest R value of 3.9. It is evident that the friction roughness function is independent of the roughness Reynolds number for all of the cases studied. From the experimental values of $R(e^+)$, f can be predicted using Eqs. (1), (2), and (4). The correlated value of the function $R(e^+)$, of the present study, is less than 1% higher than the correlated value of Han[2] for the channel with two square ribbed walls and $P/e = 10$.

The heat transfer roughness function increases with increasing roughness Reynolds number for the rib profiles. The correlation of $G(e^+)$ for the case with square ribs of the present investigation with a deviation of ±2% is given by Eq. (6):

$$G(e^+) = 1.891(e^+)^{0.381}, \qquad 2541 \underline{>>} e^+ \underline{>>} 415 \tag{6}$$

These results compare well with Han[2] for a channel with square ribs on two ribbed walls, and with Chandra and Cook[5] for a similar channel, but $P/e = 8$. For a given rib configuration and flow Reynolds number, the heat transfer coefficient can be predicted from the experimental values of $G(e^+)$ and Eqs. (1), (3), and (4).

Conclusions

1) The regionally averaged Nusselt number ratio decreases with increasing Reynolds number and is almost a constant value in the fully developed region of the channel.

2) The heat transfer enhancement increases with surface roughness, 2.53 with square ribs to 2.01 (a 26% decrease) with semicircular ribs, for $Re = 3.6 \times 10^4$.

3) The friction factor ratio increases with increasing Reynolds number. The minimum friction occurs with full circular ribs, and the maximum with square ribs. The heat transfer performance decreases with increasing Reynolds number.

4) $R(e^+)$ is independent of e^+, and the value with square ribs compares well with Han.[2]

5) $G(e^+)$ increases with increasing e^+, and decreases for turbulators with sharp corners. The channel with slant-edged ribs has the lowest heat transfer roughness function. The correlation of the function compares well with Han[2] and Chandra and Cook.[5]

6) The experimental data may be applied to the design of equipment that require internal cooling channels with two ribbed walls with nonrectangular rib profiles.

References

[1]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.

[2]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.

[3]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.

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

[5]Chandra, P. R., and Cook, M. M., "Effect of the Number of Channel Ribbed Walls on Heat Transfer and Friction Characteristics of Turbulent Flows," *Proceedings of the AIAA/ASME 6th Thermophysics and Heat Transfer Conference*, HTD-Vol. 271, AIAA, Washington, DC, 1994, pp. 201–209.

[6]Chandra, P. R., Niland, M. E., and Han, J. C., "Turbulent Flow Heat Transfer in a Rectangular Channel with Varying Number of Ribbed Walls," *Journal of Turbomachinery*, Vol. 119, No. 2, 1997, pp. 374–380.

[7]Liou, T. M., and Hwang, J. J., "Effect of Ridge Shapes on Turbulent Heat Transfer and Friction in a Rectangular Channel," *International Journal of Heat and Mass Transfer*, Vol. 36, No. 4, 1993, pp. 931–940.

[8]Taslim, M. E., and Spring, S. D., "Effect of Turbulator Profile and Spacing on Heat Transfer and Friction in a Channel," *Journal of Thermophysics and Heat Transfer*, Vol. 8, No. 3, 1994, pp. 555–562.

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

View-Factor Evaluation by Quadrature over Triangles

C. K. Krishnaprakas*
ISRO Satellite Centre, Bangalore 560 017, India

Introduction

THE need to evaluate radiation view factors often arises in thermal, illumination, and optical engineering practices. View factor between two finite surfaces is obtained as the solution of a double area integral. In the case of simple geometries, the solution can be obtained through analytical integration, and formulas are presented in Refs. 1–4. Howell[5] presented a catalog that contains view factors in terms of closed-form expressions for many geometries. However, complex geometries do not permit an analytical solution and, hence, the view-factor integral or its contour–integral representation should be evaluated by numerical means or statistical sampling techniques.[1–4] Chung and Kim[6] demonstrated the application of the finite element method (FEM) in the numerical estimation of view factors. Ambirajan and Venkateshan[7] presented a general method based on a contour–integral representation using the trapezoidal rule of numerical integration in conjunction with the Romberg extrapolation technique to get an accurate result. Ambirajan and Venkateshan[7] also resolved the problem of logarithmic singularity encountered along the common edge of two surfaces by analytically evaluating the integral. In a more recent paper, Rao and Sastri[8] extended the contour–integral technique to treat curved surfaces with the aid of a nonlinear transformation to map the boundaries and used the Gaussian quadrature for an accurate evaluation of the integral. This Note describes a method of numerical evaluation of view factors using the Gaussian quadrature over triangles. The advantage is that surfaces of more general shape can be better approximated by a set of triangles rather than by any other shaped area elements. Moreover, quadrature over triangles is also easier to use.

Radiation View Factor

The radiation view factor between two finite surfaces i and j is defined as the fraction of diffusely radiated energy leaving surface i that is directly incident on surface j, and is mathematically expressed as

$$F_{ij} = \frac{1}{A_i}\int_{A_i}\int_{A_j}\frac{\cos\theta_i \cos\theta_j \,\mathrm{d}A_i\,\mathrm{d}A_j}{\pi r^2} \tag{1}$$

Where θ_i and θ_j are the angles between the surface normals and the line connecting two points, respectively, on the two

Received March 5, 1997; revision received June 9, 1997; accepted for publication Sept. 18, 1997.

*Engineer, Thermal Systems Group.