those by Hwang and Liou.[4] The ribs with relatively large holes and A_r values up to 0.14 of this study do not perform any better than solid ribs. These ribs are not recommended.

Conclusions

Ribs with relatively large holes are not as effective as solid ribs in enhancing overall heat transfer. For the rib configurations with small hole area-to-rib frontal area ratios considered in this study, increasing the size or the number of holes, or the total hole area, lowers the overall heat transfer. Ribs with holes cause lower pressure drop, and therefore, lower required pumping power than solid ribs, but they have about the same thermal performance as solid ribs.

Acknowledgment

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Heat Transfer and Friction in Segmental Turbine Blade Cooling Channels

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Introduction

MANY researchers have examined the effects of varying the rib geometry, the rib configuration, and the channel wall boundary conditions on the heat transfer and friction in stationary models of rib-roughened internal cooling channels in turbine blades. Metzger and Vedula[1] and Zhang et al.[2] conducted experiments with rib-roughened triangular ducts that modeled the internal cooling passages at the leading edges of turbine blades. Taslim et al.[3] tested six shaped channels roughened with ribs of nine different geometries. Their channels also simulated the leading-edge cooling cavities of turbine blades.

In this investigation, the leading-edge cooling passages are modeled as three straight channels with segmental cross sections. The rib-roughened curved wall and the smooth flat wall of each channel are subjected to various heat fluxes, and the results consist of the regionally averaged heat transfer and the overall pressure drop.

Experimental Apparatus

The test apparatus was an open-flow loop. The three test channels had nominal inner diameters of 2.51, 4.13, and 2.86 cm, respectively. Their segmental flow cross-sectional areas were 34, 50, and 80% of corresponding circular cross-sectional areas (A_r = 0.34, 0.50, and 0.80), such that the hydraulic diameters of the test channels were all within 1.0% of 2.5 cm.

Each test channel had 12 separate 9.37-cm-long aluminum wall segments: six curved segments along the top wall and six flat segments along the bottom wall. Adjacent curved or flat segments were separated by a 1.59-mm-thick rubber gasket to minimize streamwise conduction. The curved wall and the flat wall were insulated from each other with balsa wood strips.

Transverse ribs were machined on the inner surface of each curved segment of the test section, whereas the inner surface of each flat segment was smooth. The ribs were square in cross section and had a height equal to 0.0625 times the channel hydraulic diameter. The distance between adjacent ribs (or the pitch) was equal to 10 times the height of the ribs. In most of the prior ribbed channel heat transfer studies, ribs were attached to the channel walls with adhesive.

The top curved wall and the bottom flat wall were heated separately, each with a flexible strip heater. Twenty-four 30-gauge thermocouples were installed every three rib pitches along the axial centerlines of the top curved wall and the bottom flat wall to measure the streamwise temperature variation of each wall. Four other thermocouples at off-center locations checked the spanwise variation of the wall temperature. Thermocouples also measured the inlet air temperature and the exit air temperature. These thermocouples, along with the data acquisition system, were calibrated with a NIST-certified mercury thermometer with a 0.1°C resolution.

A long, polyvinyl chloride, smooth entrance channel with the same flow cross section as the test section supplied a hydrodynamically fully developed flow of air to the inlet of the test section. An exit channel that was identical to the entrance channel connected the downstream end of the test section to a settling chamber.

Static pressures were measured at two axial locations that were 2.54 cm upstream of the test section entrance in the entrance channel and 7.62 cm downstream of the test section outlet in the exit channel. The pressures at the orifice of a calibrated orifice flow meter were measured to determine the mass flow rate of air through the test section during an experiment.

Experiments were conducted with ribbed-wall-to-smooth-wall heat flux ratios Q_r of 0.0, 1.0, 2.0, 4.0, 6.0, and ∞, and seven Reynolds numbers between 1×10^4 and 7×10^4. In all, there was a total of 126 test runs, 42 runs with each test section. Interested readers are referred to Spence[4] for a detailed description of the test apparatus and procedure.

Data Reduction

Both the Reynolds number and the friction factor, Re and f, respectively, are based on the hydraulic diameter and are defined as $\bar{u}D_h/\nu$ and $(\Delta p/\Delta x)D_h/[(\frac{1}{2})\rho\bar{u}^2]$, respectively. The friction factor is normalized with the corresponding friction factor for fully developed turbulent flow in a smooth tube, $f_{\mathrm{ref}} = (0.79\ \ell n\ Re - 1.64)^{-2}$.

The heat transfer coefficient h is calculated from the net heat transfer rate per unit surface area (the projected surface area

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in the case of the ribbed curved wall) of a segment, the average wall temperature of the segment, and the local average bulk temperature. The net heat transfer rate is the electrical power generated by the strip heater minus the extraneous conduction and radiation heat losses and the net axial conduction heat loss.

The segmental Nusselt number is also based on the hydraulic diameter and is defined as hD_h/k. The overall average Nusselt number of the test section $\overline{Nu}$ is the average of the segmental Nusselt numbers of the second through the sixth segments of each wall. The overall average Nusselt number is normalized with the corresponding Nusselt number for fully developed turbulent flow in a smooth tube

$$Nu_{\rm ref} = [(f_{\rm ref}/8)(Re - 1000)Pr]/[1 + 12.7(f_{\rm ref}/8)^{1/2}(Pr^{2/3} - 1)]$$

The relative thermal performance TP is defined as $(\overline{Nu}/Nu_{\rm ref})(f/f_{\rm ref})^{-1/3}$. This ratio compares the heat transfer per unit pumping power for the ribbed segmental channel with that for a tube.

The kinematic viscosity ν, thermal conductivity k, and Prandtl number Pr of air are evaluated at the average of the bulk temperatures at the inlet and exit of the test section.

The maximum uncertainties of the segmental Nusselt numbers for the ribbed wall and the smooth wall are estimated to be ± 10.4 and $\pm 15.0\%$, respectively.[5] The maximum uncertainties of the Reynolds number and the friction factor are ± 4.5 and $\pm 15.3\%$, respectively, for test runs at $Re \geq 1.4 \times 10^4$.

Presentation of Results

The variations of the normalized overall Nusselt numbers with the Reynolds number are shown in Figs. 1–3. Comparing these figures, it may be concluded that the geometry of a segmental channel with an area ratio between 0.34–0.80 only affects the overall heat transfer moderately. The average of all ribbed wall Nusselt number ratios for the channel with A_r = 0.80 is about 12 and 13% higher than those for the channels with A_r = 0.50 and 0.34, respectively. The average of all smooth wall Nusselt number ratios for the channel with A_r = 0.80 is about 11% higher than that for the semicircular channel, which, in turn, is about 8% higher than that for the channel with A_r = 0.34.

The ribbed wall Nusselt number ratio $\overline{Nu_R}/Nu_{\rm ref}$ decreases with increasing Reynolds number, and increases slightly with increasing wall heat flux ratio for the six heat flux ratios studied. The smooth wall Nusselt number ratio $\overline{Nu_S}/Nu_{\rm ref}$, however, decreases with increasing wall heat flux ratio. The effects of Q_r on both $\overline{Nu_R}/Nu_{\rm ref}$ and $\overline{Nu_S}/Nu_{\rm ref}$ lessen gradually with increasing Reynolds number. The semicircular channel results in Fig. 2 show that for $Re \approx 1 \times 10^4$, $\overline{Nu_R}/Nu_{\rm ref}$ increases by 11% when Q_r is increased from 1.0 to ∞. For $Re \geq 4 \times 10^4$, there is very little change of $\overline{Nu_R}/Nu_{\rm ref}$ with Q_r. At $Re \approx 1 \times 10^4$ and 7×10^4, $\overline{Nu_S}/Nu_{\rm ref}$ decreases by 65 and 29%, respectively, when Q_r is increased from 0.0 to 6.0.

The results of this study clearly show that changing the heat flux ratio affects the smooth flat wall heat transfer more than the ribbed curved wall heat transfer. This trend is consistent with that of the results presented in Han et al.[6] on the effect of the wall heat flux ratio on the heat transfer in a square channel with two ribbed walls and two smooth walls. In this study, when Q_r is increased, the rate of heat transfer from the ribbed wall is increased or that from the smooth wall is decreased. For a given heat flux on the smooth wall, when the

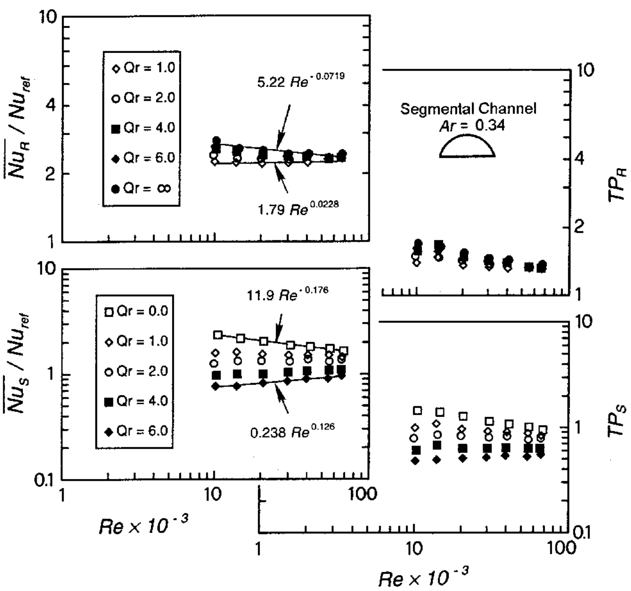

Fig. 1 Effects of wall heat flux ratio on normalized overall Nusselt number and thermal performance in a segmental channel with an area ratio of 0.34.

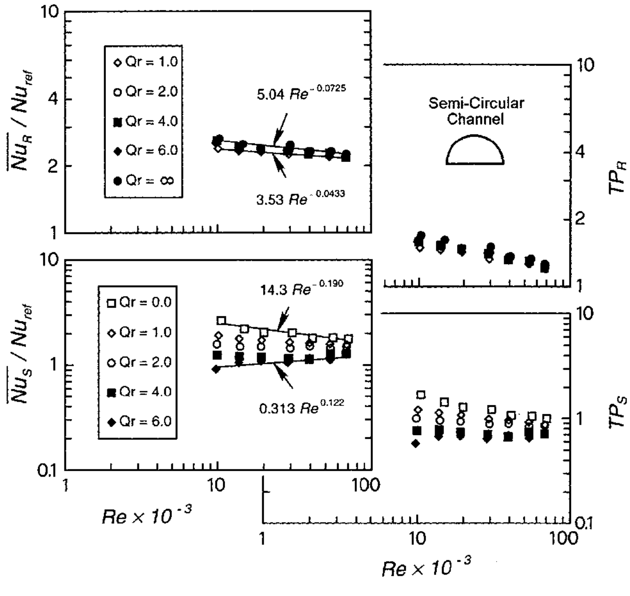

Fig. 2 Effects of wall heat flux ratio on normalized overall Nusselt number and thermal performance in a semicircular channel.

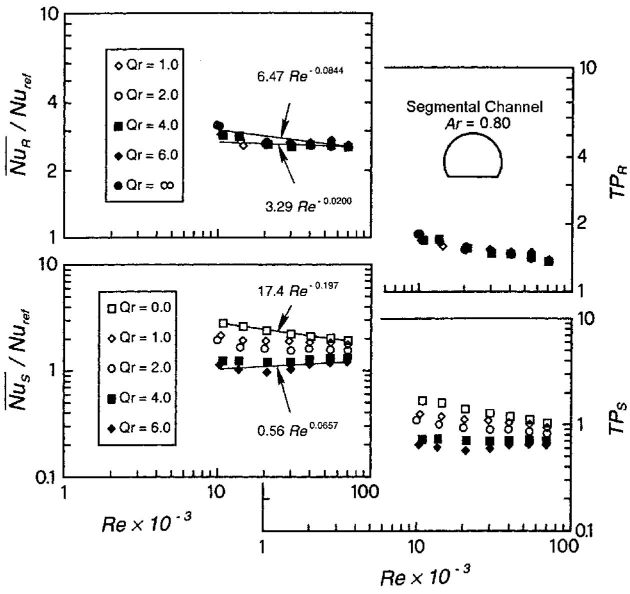

Fig. 3 Effects of wall heat flux ratio on normalized overall Nusselt number and thermal performance in a segmental channel with an area ratio of 0.80.

heat flux on the ribbed wall is high, the smooth wall is cooled by air that is warmer than the air in the low ribbed wall heat flux case. As a result, the overall heat transfer coefficient on the smooth wall is low. For a given heat flux on the ribbed wall, lowering the heat flux on the smooth wall decreases the bulk air temperature, causing the higher heat transfer coefficient on the ribbed wall. Over the range of Q_r studied, changing the heat flux on the smooth wall affects the bulk air temperature much less than changing the heat flux on the ribbed wall. Therefore, the ribbed wall heat transfer coefficient (or $\overline{Nu_R}/Nu_{\mathrm{ref}}$) is less sensitive to any change of Q_r than the smooth wall heat transfer coefficient (or $\overline{Nu_S}/Nu_{\mathrm{ref}}$).

By curve fitting a least-squares straight line through each set of the Nusselt number data (for a given Q_r), $\overline{Nu_R}/Nu_{\mathrm{ref}}$ and $\overline{Nu_S}/Nu_{\mathrm{ref}}$ may be determined as power functions of Re (see Figs. 1–3).

The $\overline{Nu_R}/Nu_{\mathrm{ref}}$ results of this study are comparable to those of Han et al.[6] and are only slightly lower than the results of Metzger and Vedula,[1] despite differences in the channel and rib geometries in these studies. The relatively large ribs in Taslim et al.[3] caused the ribbed wall heat transfer to be higher than that in this study.

For all three segmental channels, the friction factor ratio generally increases with increasing Reynolds number and varies only very slightly with Q_r. The f/f_{ref} value ranges from about 3.2 to about 6.6. The ribs on the curved wall of each test channel significantly increase the pressure drop across the channel, and therefore, the required pumping power. The average of all the friction factor ratios for the channel with $A_r = 0.80$ is about 14% higher than those for the other two channels. It is found that by curve fitting a least-squares straight line through each set of the friction factor data, the f/f_{ref} data for the three channels with $A_r = 0.34$, 0.50, and 0.80 may be correlated by the power functions, $0.681Re^{0.187}$, $0.867Re^{0.165}$, and $1.057Re^{0.158}$, respectively.

Figures 1–3 also give the relative thermal performance results that show that the relative thermal performance is not significantly affected by the channel cross-sectional geometry for the three segmental channels studied. The ribbed wall relative thermal performance value ranges from 1.21 to 1.81. The two to three times increase of the heat transfer on the ribbed curved wall is offset by the large increase (up to 6.6 times) of the friction factor. The average of all the ribbed wall relative thermal performance values for the channel with $A_r = 0.80$ is about 7 and 9% higher than those for the channels with $A_r = 0.50$ and 0.34, respectively. Although the ribs on the curved wall increase the heat transfer on the smooth flat wall, most of the smooth wall relative thermal performance values (except those for $Q_r = 0.0$ and 1.0) are below 1.0, because of the large increase of the friction factor.

Conclusions

For the three segmental channels studied, the geometry of the channel only affects the overall heat transfer and thermal performance moderately. For the heat flux ratios examined, the Nusselt number ratio for the ribbed wall of each channel increases only slightly when the heat flux ratio is increased; whereas the smooth wall Nusselt number ratio decreases significantly when the heat flux ratio is increased, especially at low Reynolds numbers. Over the Reynolds number range studied, the ribbed wall thermal performance values are between 1.21–1.81 times that for fully developed turbulent flow in a smooth tube.

Acknowledgment

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