SURFACE HEATING EFFECT ON LOCAL HEAT TRANSFER IN A ROTATING TWO-PASS SQUARE CHANNEL WITH 60° ANGLED RIB TURBULATORS

Y. M. Zhang, J. C. Han, and J. A. Parsons
Department of Mechanical Engineering
Turbine Heat Transfer Laboratory
Texas A&M University
College Station, Texas

C. P. Lee
General Electric Company
Cincinnati, Ohio

ABSTRACT

The influence of uneven wall temperature on the local heat transfer coefficient in a rotating, two-pass, square channel with 60° angled ribs was investigated for Reynolds numbers from 2,500 to 25,000 and rotation numbers from 0 to 0.352. Each pass, composed of six isolated copper sections, had a length-to-hydraulic diameter ratio of 12. The mean rotating radius-to-hydraulic diameter ratio was 30. Three thermal boundary condition cases were studied: (A) all four walls at the same temperature, (B) all four walls at the same heat flux, and (C) trailing wall hotter than leading with side walls unheated and insulated. Results indicate that rotating ribbed wall heat transfer coefficients increase by a factor of 2 to 3 over the rotating smooth wall data and at reduced coefficient variation from inlet to exit. As rotation number (or buoyancy parameter) increases, the first pass (outflow) trailing heat transfer coefficients increase and the first pass leading heat transfer coefficients decrease, whereas, the reverse is true for the second pass (inflow). The direction of the Coriolis force reverses from the outflow trailing wall to the inflow leading wall. Differences between the first pass leading and trailing heat transfer coefficients increase with rotation number. A similar behavior is seen for the second pass leading and trailing heat transfer coefficients, but the differences are reduced due to buoyancy changing from aiding to opposing the inertia force. The results suggest that uneven wall temperature has a significant impact on the local heat transfer coefficients. The heat transfer coefficients on the first pass leading wall for cases B and C are up to 70-100% higher than that for case A, while the heat transfer coefficients on the second pass trailing wall for cases B and C are up to 20-50% higher.

NOMENCLATURE

A heat transfer surface area
D hydraulic diameter; square channel width or height
e rib height
h heat transfer coefficient

INTRODUCTION

It is well known that turbine engine thermal efficiency can be improved by increasing the turbine inlet gas temperatures. This causes an increase of heat load to the turbine components. Highly
sophisticated cooling techniques such as film cooling and augmented internal cooling (shown in Figure 1) have been employed for turbine blades in order to maintain acceptable safety requirements under extreme operating conditions. However, it is important to understand the effect of blade rotation on local heat transfer coefficient distributions inside the serpentine coolant passages. This paper focuses on the influence of surface heating condition on local heat transfer coefficients in a rotating, two-pass, square channel with 60° rib-turbulated walls.

Previous investigations of turbine blade internal coolant passage heat transfer have concentrated on non-rotating models and have not accounted for the Coriolis force and the centrifugal buoyancy force effects on coolant motion and heat transfer (Han, 1984, 1988). However, some researchers reported the effect of rotation on the heat transfer characteristics in a straight channel with smooth walls and radial outward flow (Mori et al., 1971; Clifford et al., 1984; Haragama and Morris, 1988; Guidez, 1988; Morris and Ghavami-Nasr, 1991). Taslim et al. (1991a, 1991b) studied the effect of rotation on the heat transfer coefficients in a rectangular channel with ribbed walls and radial outward flow. Wagner et al. (1991a, 1991b, 1992) and Johnson et al. (1992) systematically investigated the effect of rotation on the heat transfer coefficient in a serpentine coolant passage (three-pass) with smooth and ribbed walls, respectively, for parameters similar to typical engine conditions. Prakash and Zerkle (1992) predicted the flow and heat transfer coefficients in a rotating smooth channel with radial outward flow and agreed within 10-30% with the data of Wagner et al. (1991a).

In summary, for the case of a multi-pass smooth wall channel, Wagner et al. (1991a, 1991b) reported that the rotating trailing surface heat transfer coefficient of the first coolant passage (radial outward flow) increased up to 3.5 times the non-rotating fully developed circular tube values (Rohsenow and Choi, 1961), but the leading surface heat transfer coefficient decreased to 40% of the fully developed circular tube values. However, the rotating trailing surface heat transfer coefficient of the second coolant passage (radial inward flow) decreased by 30% compared to the stationary values, while the leading surface heat transfer coefficient increased by 20% compared to the stationary results. For the case of a multi-pass, square channel with trips normal to the flow (90° ribs), Wagner et al. (1992) indicated that the maximum rotating heat transfer coefficient increased up to 4.0 times from the non-rotating fully developed circular tube values, which were slightly above the highest levels obtained with the rotating smooth wall model (~3.5 times). However, the minimum rotating heat transfer coefficients decreased to 80% of the stationary 90° ribbed wall model. For the case of a multi-pass square channel with trips skewed to the flow (45° ribs), Johnson et al. (1992) concluded that the maximum rotating heat transfer coefficient increased up to 5.0 times of the non-rotating circular tube values, while the minimum rotating heat transfer coefficient decreased to 40% of the stationary 45° ribbed model. They recommended that skewed trip strips (45° ribs) rather than normal trip strips (90° ribs) be used for the turbine blade.

Figure 1. Cooling concepts of a modern multipass turbine blade.

Figure 2. Conceptual view of a two-pass rotating coolant flow profile.
cooler coolant passage design.

Han et al. (1992, 1993) reported the effect of surface heating condition on local heat transfer coefficients in a one-pass and two-pass square channel with smooth walls (shown in Figure 2). Their rotating smooth-wall heat transfer results agreed with those of Wagner et al. (1991a, 1991b) for the case of uniform wall temperature conditions. However, Han et al. (1992, 1993) found that, for the uniform wall heat flux and simulated engine wall heating conditions, the rotating leading surface heat transfer coefficient of the first coolant passage (radial outward flow) as well as the rotating trailing surface heat transfer coefficient of the second coolant passage (radial inward flow) were up to 100% greater than to those for the uniform wall temperature conditions. Since wall heating condition significantly affects rotating smooth wall channel heat transfer, it is unknown how effects due to wall heating are changed when ribs are added to the channel walls. Therefore, the objective of this study is to investigate the effects of wall heating condition on local heat transfer coefficients in a rotating two-pass square channel with 60° angled ribs on the leading and trailing walls. Three wall heating conditions were tested: Case (A) four walls at the same temperature, Case (B) four walls at the same heat flux, and Case (C) the trailing wall hotter than the leading wall but with two side walls unheated and insulated (to simulate turbine engine operating conditions).

EXPERIMENTAL FACILITY

The test stand has been previously described and illustrated in Han et al. (1992, 1993). A shortened description of the test stand follows. Regulated compressor air flows from an orifice meter and passes through a hollow rotating shaft and a hollow rotating arm, which is perpendicular to the shaft. It then goes into the test model (a ribbed, two-pass, square channel) and is exhausted into the atmosphere at the opposite end of the rotating shaft. Slip ring units transfer thermocouple outputs to a data logger interfaced to a personal computer, and transfer variac transformer outputs to wire resistance heaters uniformly cemented in grooves on the backside of the copper plates. The rotating shaft speed is measured by a digital photo tachometer. It is better to have a test model for turbine cooling design that can determine the regionally averaged heat transfer coefficients in the channel streamwise flow direction. The two-pass square channel test model is divided into twelve short copper sections (see Figure 3). Each copper section is composed of four copper plates and has an inner cross section of 1.27 cm by 1.27 cm (1/2 in by 1/2 in). Thin Teflon strips are machined along the periphery contact surface between copper sections for insulation to prevent possible heat conduction. The channel length-to-hydraulic diameter ratio is 24, while each pass length-to-hydraulic diameter ratio (L/D) is 12. The ratio of the mean rotating arm radius to the channel hydraulic diameter (R/D) is 30. The ribbed trailing and leading surfaces were made by gluing brass ribs of square cross section to the copper plates in a required distribution and orientation. The thickness of conductive glue is less than 0.01 cm and creates a negligible thermal insulation effect between the ribs and the copper plates. For this study, the rib height-to-hydraulic diameter ratio (e/D) is 0.125, the rib pitch-to-height ratio (p/e) is 10, and the rib flow-attack-angle (the angle between the rib and coolant flow direction) equals 60°. The 60° ribs on both the leading and trailing walls are in-line and parallel to each other. Each wall has its own heater powered by a variac transformer for controllable heat flux. The smooth side walls are isolated from the leading and trailing walls to eliminate heat conduction. The entire heated test duct is insulated by Teflon material. The local wall temperature of the test model is measured by 48 copper-constantan thermocouples distributed along the length and around the perimeter of the copper channel. Two more thermocouples measure the inlet and outlet bulk air temperature. There is an unheated Teflon entrance channel (partially shown in Figure 3) that has the same cross section and length of one pass of the channel. This serves to establish hydrodynamically fully developed flow at the entrance to the heated channel.

DATA REDUCTION

The local net heat transfer coefficient is calculated from the local net heat transfer rate per unit surface area to the cooling air, the local wall temperature on each copper plate, and the local bulk mean air temperature as:

\[ h = \frac{q_{\text{net}}}{A(T_w - T_{a})} \]  

Local net heat transfer rate \( q_{\text{net}} \) is the electrical power generated from the heaters, which is calculated from heater voltage and current measurements minus the heat loss to the outside of the test section. Heat loss tests, done without air flow, determine heat loss for each test model wall for both stationary and rotating conditions. Several
are based on the average of the inlet and outlet bulk mean air temperatures. The uncertainty of the local heat transfer coefficient depends on the local wall-to-coolant (air) temperature difference and the net heat input to the air for each copper plate. This uncertainty increases for decreasing both the local wall-to-air temperature difference \((T_w - T_a)\) and the net heat input. Based on the method described by Kline and McClintock (1953), the typical uncertainty in the Nusselt number is estimated to be less than 8 percent for Reynolds numbers larger than 10,000. The maximum uncertainty, however, could be up to 20-25 percent for the lower heat transfer coefficient at the lowest Reynolds number tested \((Re = 2500)\).

**EXPERIMENTAL RESULTS AND DISCUSSION**

According to Wagner et al. (1991a, 1991b, 1992) and Han et al. (1992, 1993), the Nusselt number in a rotating channel is a function of the ratio of the rotating mean radius to channel hydraulic diameter, the ratio of the axial distance to channel hydraulic diameter, Reynolds number, Prandtl number, rotation number, wall-to-coolant temperature (density) difference ratio, flow direction (radial outward flow or radial inward flow), and the rib turbulator orientation, respectively. Their functional relationship can be expressed as:

\[
Nu = f\left(Ro, X/D, Re, Pr, Ro, \Delta \rho/\rho, flow\, direction, \, roughness\right) \tag{3}
\]

where, for the present study, \(Pr = 0.72\) and \(R/D = 30\). Tests in this study have the following parameter values: \(Re = 2500, 5000, 10,000\) and \(25,000\); \(\Omega = 0, 400\) and \(800\) rpm; combining to produce \(Ro = 0.0, 0.0176, 0.0352, 0.044, 0.088, 0.176\) and \(0.352\). The inlet wall-to-coolant density ratio \((\Delta \rho/\rho)\) has the following values: Case A = 0.11; Case B = 0.10, 0.07 and 0.08 for the first pass leading, trailing and side walls (the reverse is true for the second pass leading and trailing walls); and Case C = 0.10 and 0.08 for trailing and leading, respectively.

**Effect of Rotation Relative to Non-Rotation**

Figure 5 shows the effect of rotation on the local Nusselt number ratio \((Nu/Nu_0)\) for Case A. Note that the Nusselt number ratio is the ratio of the local Nusselt number to that of the non-rotating fully developed turbulent flow smooth tube value shown in equation (2). The results show that the local Nusselt number ratios on the trailing and leading ribbed walls are fairly uniform for non-rotation (around 3-4 through the entire two-pass channel for Reynolds numbers between 2500 and 25,000). In the first outflow pass \((0 < X/D < 12)\), the rotation significantly enhances the Nusselt number ratios on the trailing ribbed wall and greatly decreases the Nusselt number ratios on the leading ribbed wall. In the second inflow pass \((12 < X/D < 24)\), the rotation decreases the trailing wall Nusselt number ratios and increases the leading wall Nusselt number ratios as compared to that of the non-rotation values. This is because rotation induces the Coriolis forces that produce secondary cross-stream flows and thins out the first pass trailing and the second inflow pass leading boundary layers. It also thickens the first pass leading and the second pass trailing boundary layers (see the conceptual velocity profile in Figure 2). Therefore, the heat transfer coefficients on the first pass trailing and the second pass leading walls with rotation are higher than those without rotation, whereas the heat transfer coefficients on the first pass leading and the second pass trailing with rotation are lower than those without rotation. However, the rotation effect is reduced when the rotation number decreases from \(Ro = 0.352\) to \(Ro = 0.0352\) (see Figure 5).

\[
Nu/Nu_0 = (hD/k)[0.023 \, Re^{0.8} \, Pr^{0.4}] \tag{2}
\]

with \(Pr = 0.72\). Properties in the Nusselt and the Reynolds numbers...
Effect of Wall Heating Condition

Figure 6 shows the effect of varying the wall heating on the local Nusselt number ratio for rotation numbers $Ro = 0.0352$ and 0.352. Nusselt number ratios on the leading ribbed wall for Case B and Case C in the first pass ($0 < X/D < 12$) are 70% and 100% higher, respectively, than those for Case A at $Nu/Nu_0 = 1.5$. The Nusselt number ratios on the trailing ribbed wall for Case C are about 20% higher than those for Case A at $Nu/Nu_0 = 6.0$. The Nusselt number ratios on the trailing ribbed wall for Cases B and C in the second pass ($12 < X/D < 24$) are 20% and 50% higher, respectively, than for Case A at $Nu/Nu_0 = 2.0$, while the Nusselt number ratios on the leading ribbed wall for Case C are about 50% lower than those for Case A at $Nu/Nu_0 = 4.5$. However, the wall heating condition effect is reduced with decreased rotation number from $Ro = 0.352$ to 0.0352 (see Figure 6).

For Case B of four walls at the same heat flux, the first pass trailing wall temperature ($T_w = 50-55°C$ and $(\Delta p/\rho) = 0.07$) being lower than the leading ($T_w = 60-65°C$ and $(\Delta p/\rho) = 0.1$) and side walls ($T_w = 55-60°C$ and $(\Delta p/\rho) = 0.08$) can result in more cooler fluid near the trailing and side wall surfaces. It is assumed that the Coriolis-induced secondary cross-stream flows carry these cooler fluids from the trailing and side wall surfaces towards the leading surface. Therefore, the leading surface heat transfer coefficients (i.e., Nusselt number ratios) for Case B are higher than those for Case A due to rotation. Similarly, for Case C of trailing hotter than leading surface heat transfer coefficients for Case B are higher than those for Case A. Similarly, for Case C of trailing hotter than leading surface heat transfer coefficients for Case C are much higher than those for Case A.

Effect of Reynolds Number

Figure 7 shows the effect of Reynolds number on the Nusselt number ratio at a given rotation number of $Ro = 0.088$. The rotation number $Ro = \Omega D/V$ can be held constant with various combinations of rotation speed ($\Omega$) and axial flow velocity ($V$ or $Re$). The rotation number $Ro = 0.088$ is based on two combinations of $\Omega$ and $Re$: $\Omega = 400$, $Re = 5000$; and $\Omega = 800$, $Re = 10,000$. The results show that the Nusselt number ratio slightly increases with an increasing Reynolds number by holding the rotation number constant. However, the amount of Nusselt number ratio increase for the first pass leading wall is relatively large.

Effect of Rotation Number and Wall Heating Condition and Comparison

Figure 8 shows the effect of rotation number on the Nusselt number ratio at six selected channel axial locations for the three studied heating conditions. The experimental results from Johnson et al. (1992) for the case of uniform wall temperature and with 45° ribbed walls, and from Han et al. (1992, 1993) for the case of four
walls at the same temperature and smooth walls are also included for comparison. Results of Johnson et al. (1992) are based on the following conditions and locations: Ro calculated from Re = 25,000 and varying rotation speeds, (Ap/ρ) = 0.13, R/D = 49, the first pass X/D = 4.6, 8.5, 12.4 (no ribs for X/D < 3), and the second pass X/D = 21.6, 25.6 (or for X' starting at the second pass inlet, X'/D = 2.0, 6.0). These data are taken from Figure 6 of Johnson et al. (1992). The present data for the case of four walls at the same temperature are based on Ro calculated from Re = 800 rpm and varying Reynolds numbers from 2500 to 25,000, (Ap/ρ) = 0.11, R/D = 30, the first pass X/D = 5, 9, 11 (ribs start at X/D = 0), and the second pass X/D = 13, 15, 19 (or for X', X'/D = 1, 3, 7).

The previous smooth wall results (shown in Figure 8) indicate that the first pass trailing surface Nusselt number ratio increases with an increasing rotation number, while the first pass leading surface Nusselt number ratio decreases and then increases with an increasing rotation number. The smooth wall results also show that the difference between the leading and trailing surface heat transfer coefficients in the second pass is not as significant as that in the first pass. The leading and trailing surface heat transfer coefficients in the second pass are relatively independent of the rotation number as compared to that in the first pass.

The effect of rotation number on the 60° ribbed wall Nusselt number ratio shows similar trends as those on the smooth wall results except that the 60° ribs greatly enhance the surface Nusselt number ratios through the entire two-pass channel (Figure 8). In the first pass, the Nusselt number ratios on the trailing ribbed wall increase up to 6.0 while the Nusselt number ratios on the leading ribbed wall decrease to 1.5 as compared to the non-rotating ribbed wall Nusselt number ratios of around 3 to 3.5. The Nusselt number ratios on the leading ribbed wall in the second pass increase from 3.0 to 4.0 while the Nusselt number ratios on the trailing ribbed wall decrease from 3.0 to 2.0 when the rotation number changes from 0 (non-rotating) to 0.352 (higher rotation). The difference between the leading and trailing Nusselt number ratios in the second pass is smaller than that in the first pass.

As previously discussed, Figure 8 shows that the first pass Nusselt number ratios on the leading ribbed surface for Cases B and C are respectively higher than that for Case A. The second pass Nusselt number ratios on the trailing ribbed surface for Cases B and C are respectively higher than that for Case A. However, the second pass Nusselt number ratios on the leading ribbed surface for Case C are lower than that for Case A over the range of rotation numbers studied.

Figure 8 also shows the comparison between the present 60° rib data and Johnson et al. (1992) 45° rib results for the case of uniform wall temperature condition. In the first pass, the present Nusselt number ratios on the leading surface agree with those of Johnson et al., while the present Nusselt number ratios on the trailing surface are significantly higher than those of Johnson et al.
Effect of rotation number on channel-averaged Nusselt number ratio variation for Cases A, B, and C. Over the range of rotation numbers studied, in the second pass, the present data on the leading surface are higher; however, the present data on the trailing surface are lower than those of Johnson et al. The difference between these two studies may be explained as follows. The 45° ribs of Johnson et al. are semi-circular in cross section and have a rib height ratio of e/D = 0.10. The 60° ribs of this study are square in cross section and have a rib height ratio of e/D = 0.125. Since rotation creates a thinner boundary layer in the first pass trailing wall and a thicker boundary layer in the first pass leading wall, it is expected that the rib height, rib shape, and rib orientation/angle have a more significant effect on the thinner trailing wall boundary layer than on the thicker leading wall boundary layer. Therefore, the present Nusselt number ratios on the ribbed trailing wall are higher than those of Johnson et al., while the leading wall Nusselt number ratios are about the same for the two studies. Similarly, in the second pass, the rotation induces a thinner boundary layer on the leading wall and a thicker boundary layer on the trailing wall. Therefore, due to the sharper and taller ribs of this study, the present data on the leading surface are higher than that of Johnson et al. However, the present data on the trailing surface are lower than Johnson et al., particularly at the higher rotation numbers. This may be due to the different combined effects of rotation, rib orientation (60° versus 45°), and second pass entrance geometry (sharp 180° turn versus gradual 180° bend).

Channel-Averaged Results

Figure 9 shows the variation of the channel-averaged Nusselt number ratio with rotation number for Cases A, B, and C. The channel-averaged Nusselt number (Nu) for a wall is the average value of the entire wall local Nusselt number (Nu) from X/D = 1 to X/D = 11 for the first pass and from X/D = 13 to X/D = 23 for the second pass. The data are for Ro = 0.352, 0.176, 0.088 and 0.0352 (based on S2 = 800 rpm and Re = 2500, 5000, 10,000 and 25,000). The channel-averaged results show similar trends as those presented and discussed in Figure 8. The wall heating condition has a significant effect on the first pass leading ribbed wall as well as on the second pass trailing ribbed wall results, whereas the difference between leading and trailing Nusselt number ratios in the second pass is smaller than that in the first pass. Figure 9 shows additional information. The channel-averaged Nusselt number ratios on the smooth side walls are enhanced up to 2-3 for both the first and second pass. The smooth side wall Nusselt number ratios for Case B are slightly higher than that for Case A.

Effect of Buoyancy Parameter and Wall Heating Condition and Comparison

A buoyancy parameter \((\Delta p/p)(R/D)(Ro)^2\), also written as \((\Delta p/p)(\Omega R/V)(\Omega D/V)\), is used by Wagner et al. (1991a, 1991b, 1992), Johnson et al. (1992), and Han et al. (1992, 1993) to consider combined effects of Coriolis and buoyancy forces on heat transfer. The buoyancy parameter includes the effects of the local coolant-to-wall density ratio \(\Delta p/(\rho_w - \rho_b)/\rho_w\) related to buoyancy force.
secondary cross-stream flow (Coriolis force, related to Ro), and rotating radius-to-hydraulic diameter ratio (R/D, related to buoyancy force). Figure 10 shows the Nusselt number ratio variation with buoyancy parameter at selected axial locations for the present 60° rib data at wall heating conditions A, B and C. The results from Han et al. (1992, 1993) for smooth wall and from Johnson et al. (1992) for 45° rib (R/D = 49 and Ro between 0.0 and 0.34) are also included for comparison. In the present study, R/D = 30 and Ro is between 0.0 and 0.352. The buoyancy parameter effect on the Nusselt number ratios show similar trends as those presented and discussed in Figure 8. The wall heating condition has a significant effect on the first pass leading surface as well as on the second pass trailing surface. This is because the rotation induced secondary flows carry cooler fluid from the first pass trailing to leading and from the second pass leading to trailing, respectively, as discussed in the section "Effect of Wall Heating Condition." The present 60° rib data in the first pass trailing as well as in the second pass leading surface are higher than those of Johnson et al. 45° rib results. This is because, as previously discussed, the present taller and square-edged rib has more effect on the first pass thinner trailing wall boundary layer as well as on the second pass thinner leading wall boundary layer (due to rotation). However, the present 60° rib data in the second pass trailing surface are lower than those of Johnson et al. 45° rib results. As discussed above, this is due to the difference in turn geometry and rib orientation between two studies.

CONCLUDING REMARKS

The influence of uneven wall temperature on the local heat transfer coefficients in a rotating two-pass square channel with in-line 60° angled ribs on leading and trailing walls has been observed for rotating numbers from 0.0 to 0.352 and Reynolds numbers from 2500 to 25,000. The findings are:

1. The trailing wall Nusselt number ratios for the first pass (Case A) are higher than the leading wall Nusselt number ratios, and increase with increasing rotation numbers. The leading wall Nusselt number ratios decrease with an increasing rotation number. The rotating ribbed wall heat transfer coefficients are 2 to 3 times higher than their corresponding rotating smooth wall values. The difference between the leading and trailing wall Nusselt number ratios increases with increasing rotation number. This is because the rotation creates a thinner boundary layer on the trailing wall and a thicker boundary layer on the leading wall.

2. The leading wall Nusselt number ratios in the second pass (Case A) are higher than the trailing wall Nusselt number ratios due to the reversing of the Coriolis force direction. The leading wall Nusselt number ratios increase and the trailing wall Nusselt number ratios decrease with increasing rotation number. Again, the rotating ribbed wall heat transfer coefficients are higher than their corresponding rotating smooth wall values. The difference between the leading and trailing wall Nusselt number ratios in the second pass is smaller than that in the first pass because the rotation-induced buoyancy force opposes the inertia force in the second pass.

3. In the first pass, the Nusselt number ratios on the leading wall for Cases B and C are 70-100% higher than those for Case A. In the second pass, the Nusselt number ratios on the trailing wall for Cases B and C are 20-50% higher than those for Case A. This is because the rotation-induced secondary flows carry cooler fluid from the trailing wall towards the leading wall (in the first pass), as well as from the leading and side walls towards the trailing wall (in the second pass).

4. The trends of the Nusselt number ratio versus the rotation number for both the present 60° ribs and Johnson et al. (1992) 45° ribs agree for Case A. In the first pass, the Nusselt number ratios on the leading thicker boundary layer wall for 60° ribs are about the same as the 45° rib values, while the Nusselt number ratios on the trailing thinner boundary layer wall for 60° ribs are higher than those for 45° ribs. However, due to different rib angles and turn geometry under rotation, the second pass Nusselt number ratios for 60° ribs are higher on the leading wall but lower on the trailing wall as compared to those for 45° ribs.

5. The channel-averaged Nusselt number ratios on the first pass trailing and second pass leading for all Cases (A, B and C) increase with rotation number. However, the channel-averaged Nusselt number ratios on the first pass leading and second pass trailing decrease with rotation number. The channel-averaged Nusselt number ratios on the first pass and second pass smooth side walls increase from 2.0 to 3.0 over the range of rotation numbers studied.

ACKNOWLEDGEMENTS

This investigation was supported by General Electric-Aircraft Engines, and by the Texas Higher Education Coordinating Board (Energy Research in Application Programs, TEES 70730).

REFERENCES


