THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47 St., New York, N.Y. 10017

An Experimental Investigation of Heat Transfer in an Orthogonally Rotating Channel Roughened with 45 Degree Criss-Cross Ribs on Two Opposite Walls

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ABSTRACT

Turbine blade cooling is imperative in advanced aircraft engines. The extremely hot gases that operate within the turbine section require turbine blades to be cooled by a complex cooling circuit. This cooling arrangement increases engine efficiency and ensures blade materials a longer creep life. One principle aspect of the circuit involves serpentine internal cooling passes throughout the core of the blade. Roughening the inside surfaces of these cooling passages with turbulence promoters provides enhanced heat transfer rates from the surface.

The purpose of this investigation was to study the effects of rotation, aspect ratio, and turbulator roughness on heat transfer in these rib-roughened passage. The investigation was performed in an orthogonally rotating setup to simulate the actual rotation of the cooling passages. Single pass channels, roughened on two opposite walls, with turbulators positioned at a 45° angle to the flow, in a criss-cross arrangement, were studied throughout this experiment. The ribs were arranged such that their pitch-to-height ratio remained at a constant value of 10. An aspect ratio of unity was investigated under three different rib blockage ratios (turbulator height/channel hydraulic diameter) of 0.1333, 0.25, and 0.3333. A channel with an aspect ratio of 2 was also investigated for a blockage ratio of 0.25. Air was flown radially outward over a Reynolds number range of 15000 to 50000. Rotation number was varied from 0 to 0.3. Stationary and rotating cases of identical geometries were compared. Results indicate that rotational effects are more pronounced in turbulated passages of high aspect and low blockage ratios for which a steady increase in heat transfer coefficient is observed on the trailing side as rotation number increases while the heat transfer coefficient on the leading side shows a steady decrease with rotation number. However, the all-smooth-wall classical pattern of heat transfer coefficient variation on the leading and trailing sides is not followed for smaller aspect ratios and high blockage ratios when the relative artificial roughness is high.

NOMENCLATURE

a	test section width (see Figure 2)
A_h	total area of each heater
AR	aspect ratio of passage, a/b
b	test section height
C_p	specific heat at constant pressure
D_h	hydraulic diameter of passage
e	turbulator (rib) height
Gr	Grashof number $(= Ra/Pr = (r/D_h)J_{D_h}RoRe\beta\Delta T)$
h_t	heat transfer coefficient on turbulated wall
J_{D_h}	Rotational Reynolds number based on hydraulic diameter $(= \Omega {D_h}^2/\nu)$
Nu_o	Nusselt number in a smooth channel, stationary case, from Dittus-Boelter $(= 0.023Re^{0.8}Pr^{0.4})$

*Presented at the Gas Turbine and Aeroengine Congress and Exposition—June 11-14, 1990—Brussels, Belgium
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1990

Nu_s	Nusselt number on the turbulated wall for stationary case
Nu_r	Nusselt number on the turbulated wall for rotating case
P	perimeter
Pr	Prandtl number
$\dot{q}''$	heat flux generated by electric heater
$\dot{q}''_b$	heat flux lost through back of test section
$\dot{q}''_r$	heat flux lost by radiation
r	radial distance from the point under investigation to the axis of rotation
Ra	Rotational Rayleigh number : $r\Omega^2(T_s - T_m)\beta {D_h}^3 Pr/\nu^2 = (r/D_h)J_{D_h} RoRePr\beta\Delta T$
Re	Reynolds number based on test section hydraulic diameter
Ro	Rotation number ($=\Omega D_h/U_m = J_{D_h}/Re$)
$Ross$	Rossby number ($=1/Ro$)
S	turbulator (rib) pitch
St	Stanton number ($h/\rho U_m C_p$)
ΔT	$(T_s - T_m)$
T_f	film temperature, $(T_m + T_s)/2$
T_m	air mixed mean temperature
T_s	surface temperature
U_m	mean velocity of air
X	distance from the turbulated section entrance to the point under investigation
α	angle of attack
β	coefficient of thermal expansion
ν	kinematic viscosity of air
ρ	density
Ω	angular velocity
μ	dynamic viscosity of air

INTRODUCTION

Modern turbine blades are designed with internal cooling passages arranged in a serpentine pattern. These passages are often roughened with rib shape turbulence promoters called turbulators. The turbulators which are usually repeated along one wall or two opposite walls within the cooling path significantly affect the flow pattern and heat transfer.

The ability to analytically predict the effects of rib roughness on the flow field and heat transfer, specially in a rotating frame, is limited. Therefore, many experimental investigations have been conducted in order to determine those configurations which produce the optimum results in terms of both heat transfer and pressure drop. The pertinent geometric parameters involved in these experimental investigations are passage aspect ratio, AR; turbulator angle of attack, α; pitch-to-height ratio, S/e; blockage ratio, e/D_h and the manner by which turbulators are positioned with respect to each other. Interested reader is referred to the work done by Burggraf (1970), Han et al. (1978,1984,1985), Metzger et al. (1983,1989) , Sparrow and Tao (1982), Taslim and Spring (1987, 1988) and Webb et al. (1971). However, in a great majority of these investigations, the test section is stationary thus the rotational effects which play an important role in establishing the flow field and heat transfer behavior in the passage are not taken into consideration.

In the presence of rotation, the Coriolis and centripetal accelerations generate secondary flows that, specially in a rib-roughened passage of high blockage ratio, can change the flow structure and heat transfer behavior significantly. Orthogonal mode of rotation in which the passage's longitudinal axis is perpendicular to the axis of rotation describes the configuration of an internal cooling passage of a turbine blade. In such a mode, the secondary flows induced by Coriolis accelerations and centripetal buoyancy have a significant effect on the flow field and heat transfer. In addition to those pertinent geometrical parameters for the stationary case mentioned above, the following parameters play an important role in the study of an orthogonally rotating test section: Rotational Reynolds number, $J_{D_h} = \Omega D_h^2/\nu$, Rotational Rayleigh number, $Ra = \left(\frac{r}{D_H}\right) J_{D_h} RoRePr\beta\Delta T$, Rotation number, $Ro = \Omega D_h/U_m$, Grashof number, $Gr = Ra/Pr$, and centripetal buoyancy parameter, Gr/Re^2. The studies described below present the investigations conducted in this area.

Mori et al. (1971) investigated a straight circular pipe, both theoretically and experimentally, under the influence of orthogonal rotation using the naphtalene sublimation technique. The investigations were performed in both laminar and turbulent regimes. First, by assuming a boundary layer along the wall, an analytical study was performed for a turbulent flow in the hydrodynamically and thermally fully developed region (theoretical analysis of a fully-developed laminar flow was previously completed in their first report 1968). Second, using air as the working fluid, experimental data were taken in a straight circular tube of 9-mm-diameter for a range of Reynolds number from 1000 to about 10800, rotational speed of 0 to 1000 RPM and T_{wall} of 25°C to 45°C. It was concluded that Nusselt numbers did not increase for the turbulent case as much as they did for the laminar case when rotation was introduced. Experimental and analytical results were in agreement for the Nusselt number for both laminar and turbulent cases.

Johnston et al. (1972) studied the effects of spanwise rotation in a rectangular duct for a fully developed flow. Water was used as the working fluid and dye was injected for flow visualizations. Reynolds number ranged from 2500 to 36000 and Rotation number from 0.01 to 0.25. Their conclusion was that turbulence increased on the trailing side and decreased on the leading side. The changes in flow structure were attributed to the Coriolis forces due to radial motion and spanwise rotation. Zysina-Molozhen et al. (1977) investigated a radially outward turbulent flow. Their results showed that flow took on a laminar-like pattern under the presence of rotation. No significant rotational effects were noticed for Reynolds numbers greater than $2.5x10^4$. Morris and Ayhan (1979) tested two passages with diameters of 4.85 mm and 10 mm. Air was flown radially outward at Reynolds numbers of 5000, 10000 and 15000, and rotational speeds of 0, 1000 and 2000 RPM. Results showed that for fixed Reynolds and Rayleigh numbers, heat transfer increased with Rossby number. With a 15% mean scatter band, all rotating data could be correlated by

$$Nu_m = 0.022 Re^{0.8} \left[\frac{Ra}{Re^2}\right]^{-0.186} Ross^{0.33}$$

Morris and Ayhan (1981) studied turbulent local and mean heat transfer in a tube of circular cross section in an orthogonal mode of rotation. They examined radially outward and inward flows. Results showed that, for a radially inward flow, mean heat transfer coefficient increased with Rotational Rayleigh number and decreased with Rossby number for a fixed rotational buoyancy. Under the condition of radially outward flow, the Coriolis acceleration improved heat transfer but centripetal buoyancy adversely affected it. Wagner et al. (1986) investigated heat transfer phenomena in rotating, two-pass serpentine, turbulated passages. Two models were tested : a two-leg model with a smooth 180° bend; and an engine-scale model of a typical multi-pass cooled turbine airfoil. The two-leg model had an aspect ratio of 0.25 and Reynolds number was varied from 15000 to 30000 for rotational speeds of up to 700 RPM. The engine-scale model was tested over a range of Rotation number from 0 to 0.09. It was concluded that average heat transfer coefficients on the leading and trailing surfaces can significantly change depending upon flow direction and Rossby number.

Harasagama and Morris (1987) investigated three test sections. The first was a triangular duct with data taken on the leading side only and flow traveling radially inward. The second was a square duct with a radially inward flow while the third was a circular tube with radially outward flow. For the last two cases, data were taken on both trailing and leading surfaces. Reynolds number ranged from 7000 to 25000 and Rotation number varied from 0 to about 0.1. It was concluded that radially inward flows produced a higher mean Nusselt number due to rotation compared with the stationary case. Increased centripetal buoyancy further increased heat transfer on the leading side and reduced it on the trailing side. When Rotational Reynolds number increased, the heat transfer coefficient decreased on the leading side. Radially outward flows produced a reduction in average Nusselt number on the leading side in comparison to the stationary case.

Morris et al. (1988) measured heat transfer coefficients on the leading and trailing surfaces of a square rotating duct. With flow traveling radially outward, Rotation number ranged from 0 to 0.08 and Reynolds number from 7200 to 44000. Results indicated that the Coriolis induced secondary flows increased local heat transfer coefficients on the trailing surface when compared to the stationary case and had little effect on the leading surface. Wagner et al. (1989) investigated a four-pass serpentine smooth duct of square cross section. Rotation number was varied from 0 to 0.48 and Reynolds number was held at a constant value of 25000. Buoyancy effects, by varying the inlet density, were studied on the heat transfer coefficient of flow traveling outward on the first leg of the test section. Results concluded that heat transfer enhancement significantly varied on both the leading and trailing surfaces from the inlet to a location of $X/D_h \leq 12$. Enhancement in heat transfer varied on the different surfaces due to changes in density. As density and the buoyancy parameter, Gr/Re^2, were increased, heat transfer was always enhanced.

Taslim et al. (1989) measured heat transfer coefficients in a spanwise rotating passage with two opposite rib-roughened walls, where $S/e = 10$, $AR = 1$, and the rib angle of attack, $\alpha = 90°$. Three blockage ratios were tested (0.133, 0.25, 0.333). Reynolds number ranged from 15000 to 50000 while the rotational Reynolds number varied between 300 and 1400 and Rotation number varied from 0. to 0.08. On the trailing surface, the smallest rib blockage measured an enhancement (relative to non-rotating) up to 46% where as the highest blockage experienced insignificant enhancement of only 6%. The same trend was observed on the leading side. After an initial high enhancement at low Rotation number, there was a continued decrease in the heat transfer coefficient with increasing Rotation number on both leading and trailing surfaces for the two lowest blockages. This trend is opposite to the observed trend for trailing smooth surfaces. The initial high enhancement on the leading surface at low Rotation number is a noted difference from smooth leading surfaces.

Taslim et al. (1989) continued the investigation of rotational effects on heat transfer in spanwise rotating channels. The test sections were turbulated on two opposite walls with square turbulators at an angle of attack of 45° in a criss-cross arrangement. Two aspect ratios were investigated, 2 and 0.5. All test sections had a constant pitch-to-height ratio of 10 with varying blockage ratios. Reynolds number ranged from 11000 to 55000 and Rotation number from 0 to 0.3. Results showed that rotational effects were more pronounced in channels with low blockage and high aspect ratios. As Rotation number increased, the heat transfer coefficient increased on the trailing surface and decreased on the leading surface.

The goal of this investigation was to determine the influence of rotation, rib roughness, and aspect ratio on heat transfer, in a typical internally cooled gas turbine blade. The working fluid chosen was air, running through a single-pass cooling passage which was subjected to a constant heat flux boundary condition. Only two opposite walls of the passage were rib roughened. Constant heat flux was applied to one of the two turbulated walls. This arrangement is closer to the actual turbine blade since in multiple cooling passage blades there exists a large temperature gradient between the adjacent walls. It is, however, planned to run tests with four heated walls under otherwise identical conditions and compare the results. The air flow was radially outward in the test sections, which were rotated in a manner similar to actual turbine blades.

EXPERIMENTAL APPARATUS AND PROCEDURE

Channels, roughened with turbulators positioned at an angle of 45° with flow direction and a constant pitch-to-height ratio, S/e, of 10 were investigated for two aspect ratios. The turbulators were located on opposite walls in a criss-cross configuration. The measurement surface was constructed of 1.02-cm-thick white pinewood on which four custom made etched foil heaters were affixed. A liquid crystal sheet was placed over these heaters to display isotherms. Photographs of these isotherms were taken for heat transfer coefficient measurement purposes. Liquid crystals display areas of constant temperature compared to thermocouples that only measure temperature at discrete points. This feature provided a more detailed and accurate temperature distribution measurement. The test

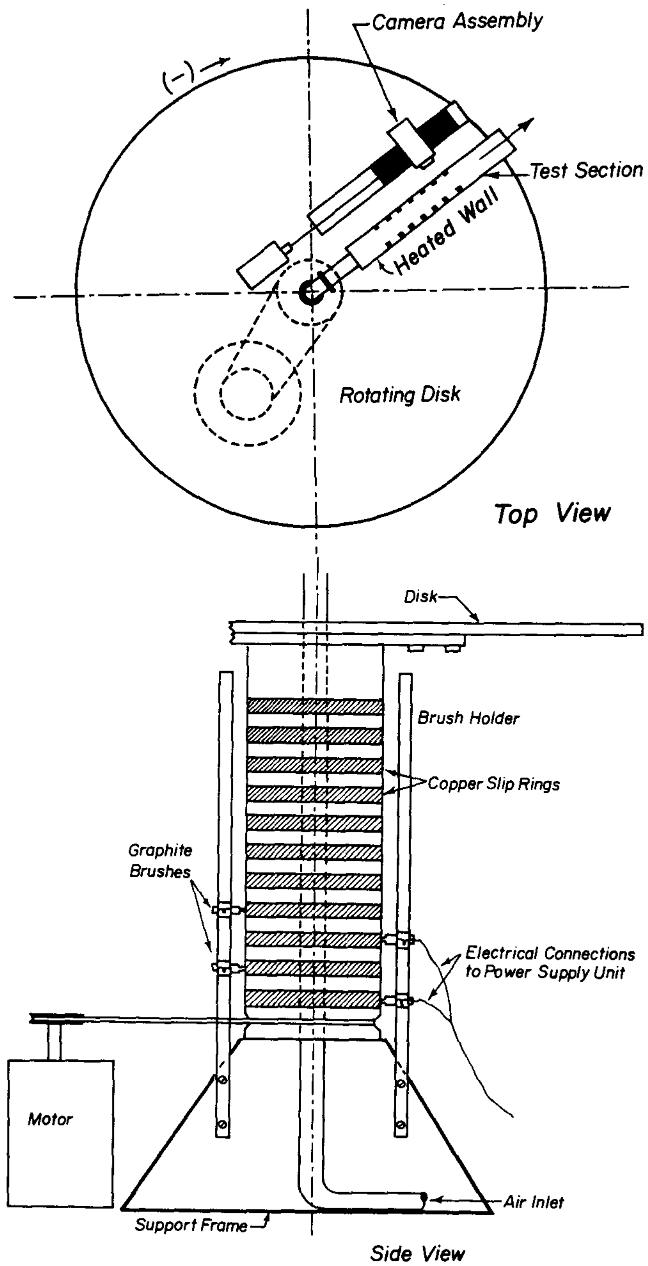

Fig. 1 Experimental Apparatus

sections were constructed of 1.27-cm-thick plexiglas on three walls. The fourth wall was the wooden measurement section shown in Fig. 2. All test sections were 1.17 m in length. Three test sections at an aspect ratio of unity were constructed with a cross section of 3.81 cm x 3.81 cm. All turbulators had a square cross-sectional area and

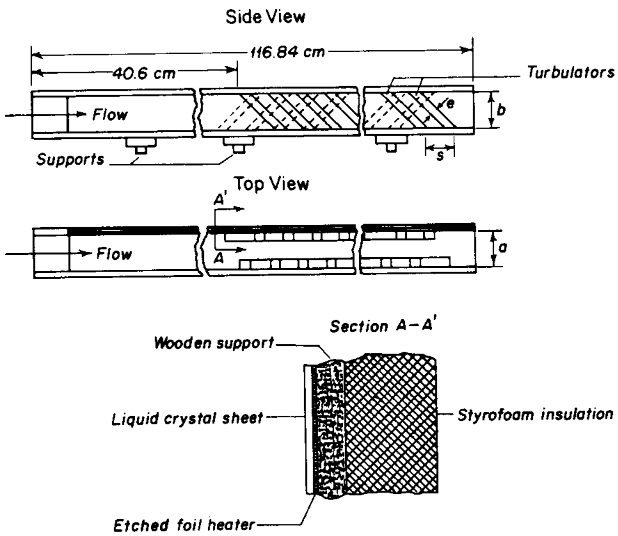

Fig. 2 A Typical Test Section

were constructed of plexiglas. These three test sections had turbulator heights, e, of 0.508 cm, 0.953 cm, and 1.27 cm corresponding to three different blockage ratios, e/D_h, of 0.133, 0.250, and 0.333, respectively. The fourth test section had an aspect ratio of 2, with a cross section of 3.81 cm x 7.62 cm, and a turbulator height of 1.27 cm, corresponding to a 0.25 blockage ratio. 5.08 cm of styrofoam insulation covered the test section on all four sides to minimize heat losses to the environment except for a small window at the camera location. The rotating test apparatus consisted of a horizontally positioned 2.438-m-diameter circular disc driven by a 5 HP reversible motor. Test sections were mounted radially on the disc. Air was directed into a plenum from the center of the disc through a rotary joint and into the test section. The electric power was supplied to the test section through a slip-ring assembly. Color photographs of the isotherms were obtained from a camera mounted on a track positioned parallel to the test section. An X/D_h well beyond the turbulated section entrance (for AR=2 : X/D_h = 6.4 and for AR=1 : X/D_h = 9) was chosen for the camera position. The heaters were powered using a custom design control pannel and each heater was independently controlled with a variable transformer.

Experimental data were taken on the turbulated walls of the four test sections in both rotating and stationary modes. Rotational speeds of 50, 100, 150, and 200 RPM were set counterclockwise (positive rotation) and clockwise (negative rotation) directions shown in Fig. 1. The two rotational directions allowed data to be taken on both the trailing and leading surfaces of the test section. Reynolds numbers ranged from 15000 to 50000.

Prior to any testing, the liquid crystal sheet was calibrated. Calibration procedure is explained in detail in Bondi (1989). A shade of green corresponding to a temperature of 37°C which was most distinct was chosen as the reference color.

For a given mass flow rate and angular velocity, after the system reached thermal equilibrium i.e. the isochromes on the liquid crystal sheet remained stagnant, a photograph was taken of the entire area between two adjacent ribs. The heater power supply was then increased forcing isotherms to move so that another segment of the region of interest could be recorded on film. The above procedure was repeated until the reference isotherm had covered the entire area between two turbulators. An average of 15 photographs were required to cover the entire area of interest with the reference temperature. This concluded a set of data for a single Reynolds number. Then the flow rate was increased and the above process was repeated for all remaining Reynolds numbers and angular velocities. Data was archived and organized using an interactive computer program on a VAX 8650 main frame. The heat transfer coefficient corresponding to each picture was then calculated from:

$$h = \frac{\dot{q}'' - \dot{q}''_b - \dot{q}''_r}{T_s - T_m}$$

where T_s is the surface temperature and T_m is the air mixed mean temperature calculated from the energy balance between the test section entrance and the camera location. $\dot{q}''_b$ is the total heat loss through the back of the test section and $\dot{q}''_r$ is the radiational loss from the heated wall to the other three unheated walls of the test duct. Air properties were evaluated at the film temperature, T_f .

The second step in the process of data reduction was digitizing the reference color from the photographs taken to use in the calculation of the area-weighted average heat transfer coefficient. A dedicated software package with an active tablet and a magnetic field mouse was utilized for digitization. The area-weighted average heat transfer coefficient was then calculated from:

$$h_t = \sum_{1}^{np} \left(\frac{h_1 a_1 + h_2 a_2 + \cdots + h_{np} a_{np}}{a_1 + a_2 \cdots + a_{np}} \right)$$

where np is the total number of pictures taken for a given Reynolds number, a_1 through a_{np} are the areas covered with the reference color and h_1 through h_{np} are the measured heat transfer coefficients corresponding to those areas.

RESULTS AND DISCUSSION

The buoyancy parameter, $Gr/Re^2 = (r/D_h)Ro^2\beta\Delta T$, ranged from $1.3x10^{-5}$ to 0.072. The effect of centripetal buoyancy on Nusselt number was not investigated since the liquid crystal reference temperature selected required relatively small temperature differences between the wall and fluid of approximately 14° C. Results for the leading and trailing sides of the test sections are plotted together (in order to conveniently compare them) and are identified by the sign of Rotation and Rotational Reynolds numbers. To an observer looking at the disk from the top, (see Fig. 1) negative Rotation values represent clockwise rotation with the heated wall on the leading side of the test section. Similarly, positive values denote counterclockwise rotation with the trailing side under investigation. Heat transfer coefficients are measured only on the turbulated wall between ribs. Rib fin-type effects were not investigated. Experimental results presented here are all for one region between a pair of turbulators, well beyond the test section entrance. Uncertainty in the heat transfer coefficient measurements, following the method of Kline and McClintock (1953), is estimated to be about 6%. Details of uncertainty analysis are presented by Bondi (1989).

Figure 3 demonstrates the variation of Nusselt number with Re for the stationary test sections. Nusselt number shows a substantial increase with increasing rib blockage ratio. This increase is due to separation, reattachment and recirculation of the flow as it passes over the turbulated surface. Significant mixing rates between the high temperature air near the heated wall and the low temperature air away from it results in this strong behavior. These results will be further used to compare the rotating versus stationary cases for identical test conditions. Two data points from Metzger et al. (1989) are plotted for angles of attack of 30° and 60° in a criss-cross arrangement. The 60° case at the same X/D_h, although having a geometry of a lower pitch-to-height ratio and blockage ratio, compares favorably with the present data. The much lower Nusselt number for the

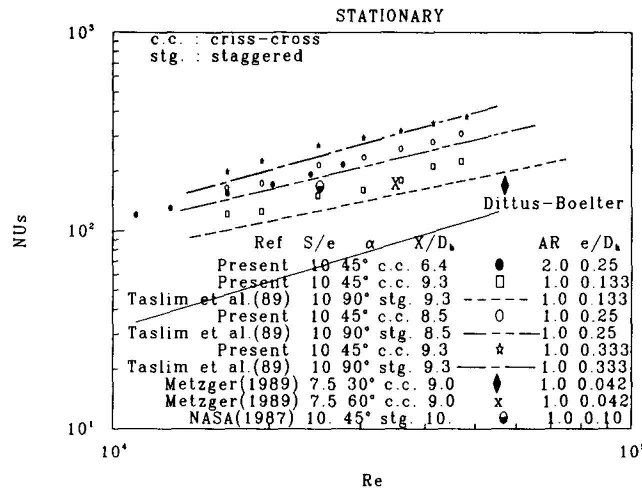

Fig. 3 Variation of Nu with Re for Stationary Cases

30° criss-cross case is contributed to its much lower blockage ratio, criss-cross arrangement and angle of attack. The NASA (1987) 45° data, although staggered with slightly lower blockage ratio, agrees well with the present stationary data in Fig. 3 and rotational data in Fig. 7. The Dittus-Boelter correlation is also plotted to show the effects of rib-roughness on heat transfer enhancement. A comparison between the present data for an aspect ratio of unity and the data published previously for an aspect ratio of 2, both for a blockage ratio $e/D_h = 0.25$, indicates a slight increase in heat transfer coefficient for a decrease in passage aspect ratio. Figure 3 also shows the present 45° criss-cross data compared with 90° staggered data reported by Taslim et al. (1989) at the same blockage aspect ratios. The 45° criss-cross configuration indicates superior heat transfer characteristics at all wall roughnesses as also reported by Metzger(1989).

The variation of Nusselt number versus Rotation number is shown in figures 4 through 11. Results are also shown from Taslim et al. (1989) for AR=2, with e/D_h = 0.1 and 0.1875. Each figure is for a specific rotational Reynolds number, J_{D_h}, as a result of a constant angular velocity, Ω. As discussed previously, positive Rotation numbers refer to the trailing surface measurements and negative numbers refer to measurements on the leading surface. In all cases tested, for a fixed J_{D_h}, Nusselt number decreases as Rotation number increases. This behavior coincides with that of the stationary pattern since as a result of constant angular velocity, higher Rotation numbers correspond to lower air mean velocity, U_m, which results in a lower Nusselt number. On both the leading and trailing surfaces, Nusselt numbers increase with increasing blockage ratios, for all three geometries tested for the aspect ratio of 1. Aspect ratio of 2, however, shows less sensitivity to the blockage ratio in general. Higher blockage ratios exhibit higher Nu on the leading side but as J_{D_h} increases, Nusselt number shows less sensitivity to the blockage ratio. On the trailing side, however, at high rotational Reynolds numbers, the smaller blockage ratio produces higher Nu. It seems that, at high rotational Reynolds numbers, the Coriolis forces suppress the mixing effects of high blockage ratios.

The rotational effects on heat transfer are presented by plots of the ratio of Nusselt number for the rotating case to that of stationary case, Nu_r/Nu_s, versus Rotation number for all J_{D_h} values. It is apparent that Nu_r/Nu_s values greater than unity imply heat transfer enhancement as rotation occurs. On figures 12 through 19, it is evident that rotation has a significant effect on the Nusselt number ratio. For the case of a rotating all-smooth-wall test section, it is shown (see Morris (1981)) that the heat transfer coefficient on the trailing side increases with rotation compared to that of the stationary case while a decrease in heat transfer coefficient is observed on the leading side. Such behavior is due to the fact that, in a spanwise rotating channel, Coriolis forces cause a tendency for the flow to separate from the leading side and push against the trailing side. However, for a rib-roughened passage, this behavior is only noticed for low blockage and high aspect ratios (AR=2 and $e/D_h = 0.1$, Taslim et al. (1989)) as shown in figures 16 thru 19. Aspect ratio of 2 with higher blockage ratios (0.1875 and 0.25) and all three blockage ratios

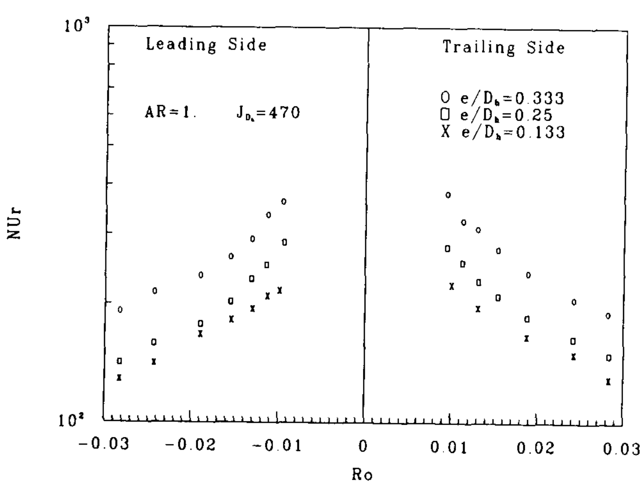

Fig. 4 Nu_r **versus** Ro **for** $AR = 1$ **at** $J_{D_h} = 470$

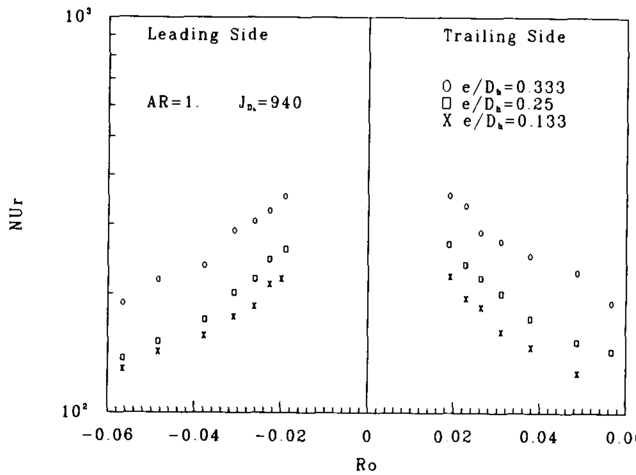

Fig. 5 Nu_r **versus** Ro **for** $AR = 1$ **at** $J_{D_h} = 940$

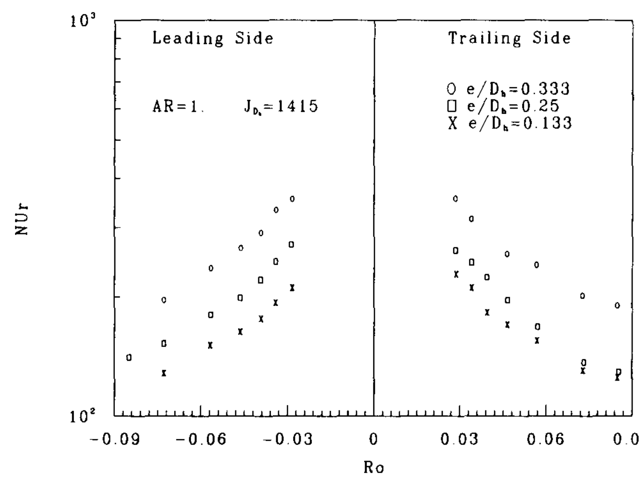

Fig. 6 Nu_r **versus** Ro **for** $AR = 1$ **at** $J_{D_h} = 1415$

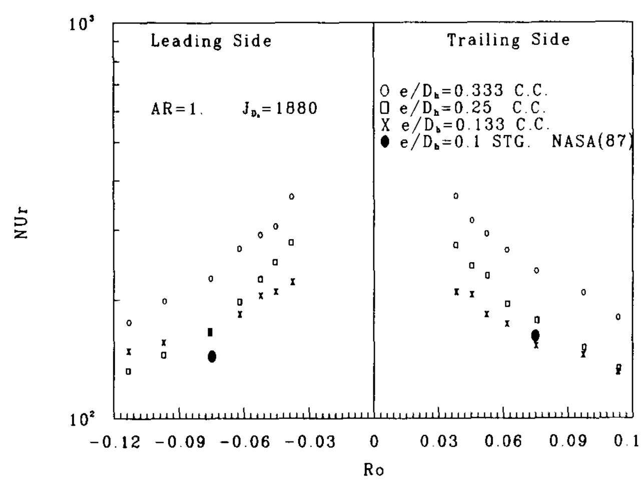

Fig. 7 Nu_r **versus** Ro **for** $AR = 1$ **at** $J_{D_h} = 1880$

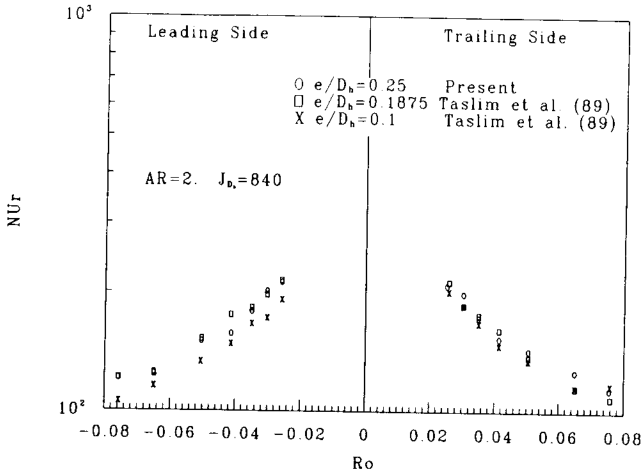

Fig. 8 Nu_r versus Ro for $AR = 2$ at $J_{D_h} = 840$

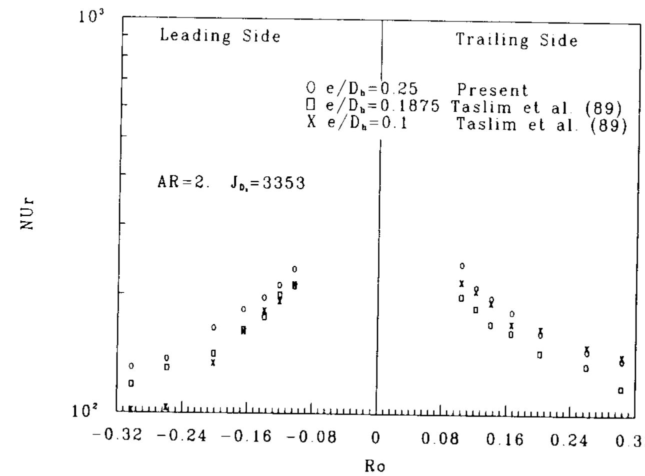

Fig. 11 Nu_r versus Ro for $AR = 2$ at $J_{D_h} = 3353$

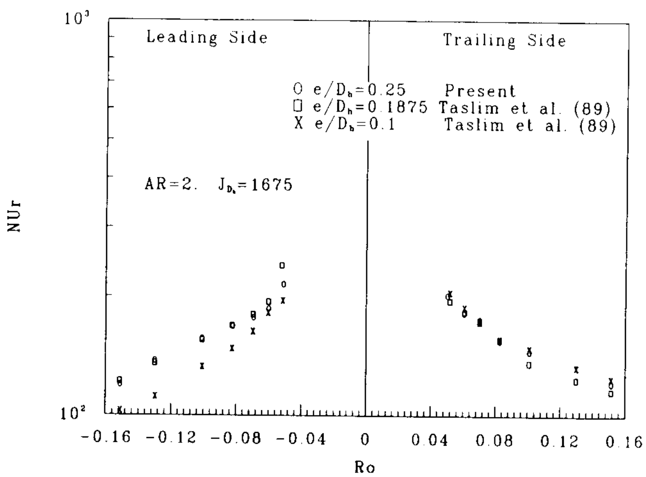

Fig. 9 Nu_r versus Ro for $AR = 2$ at $J_{D_h} = 1675$

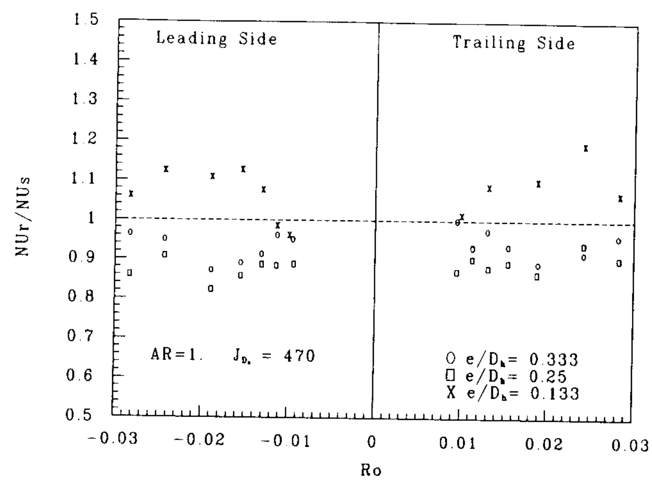

Fig. 12 Nu_r/Nu_s versus Ro for $AR = 1$ at $J_{D_h} = 470$

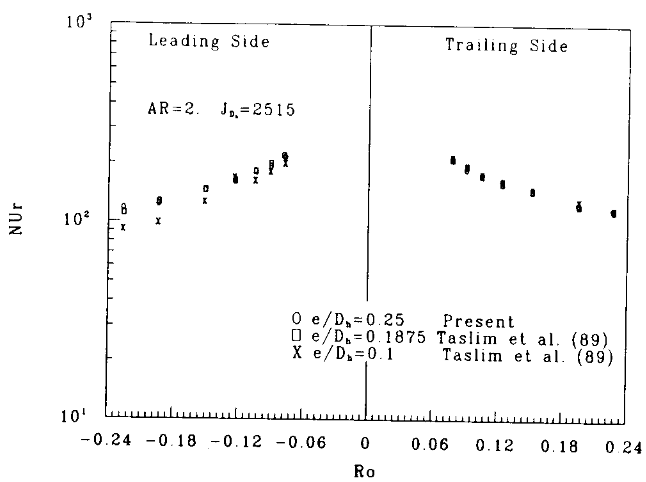

Fig. 10 Nu_r versus Ro for $AR = 2$ at $J_{D_h} = 2515$

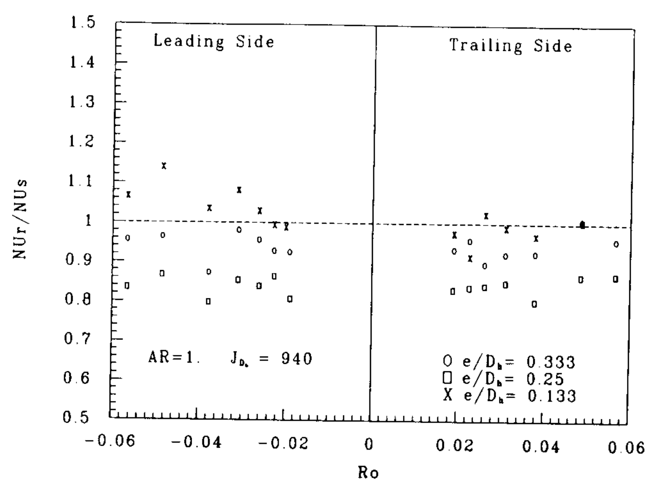

Fig. 13 Nu_r/Nu_s versus Ro for $AR = 1$ at $J_{D_h} = 940$

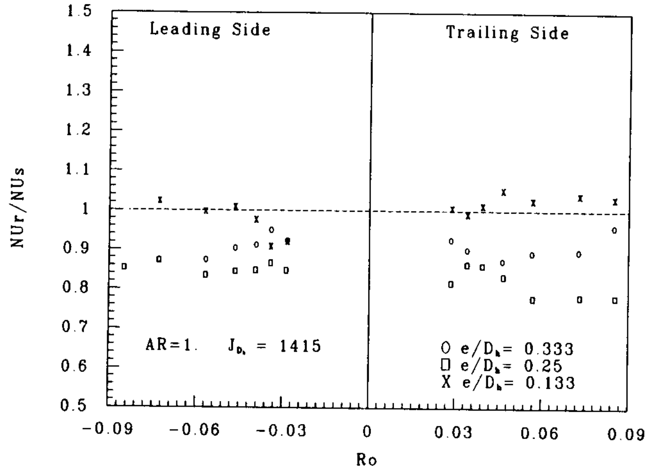

Fig. 14 Nu_r/Nu_s versus Ro for $AR = 1$ at $J_{D_h} = 1415$

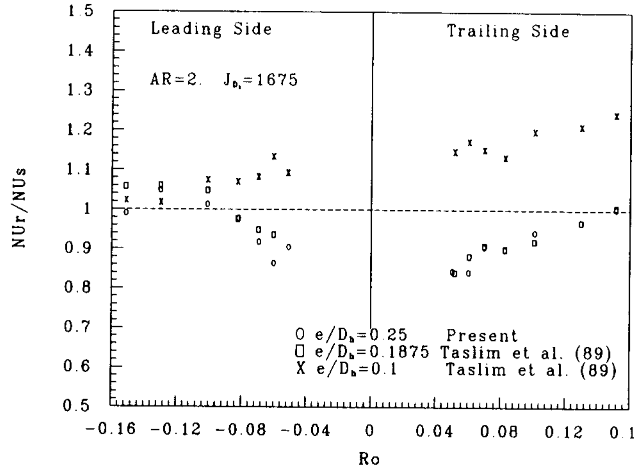

Fig. 17 Nu_r/Nu_s versus Ro for $AR = 2$ at $J_{D_h} = 1675$

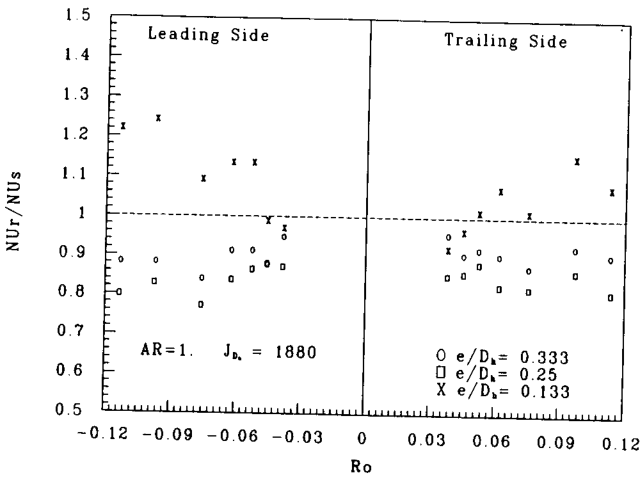

Fig. 15 Nu_r/Nu_s versus Ro for $AR = 1$ at $J_{D_h} = 1880$

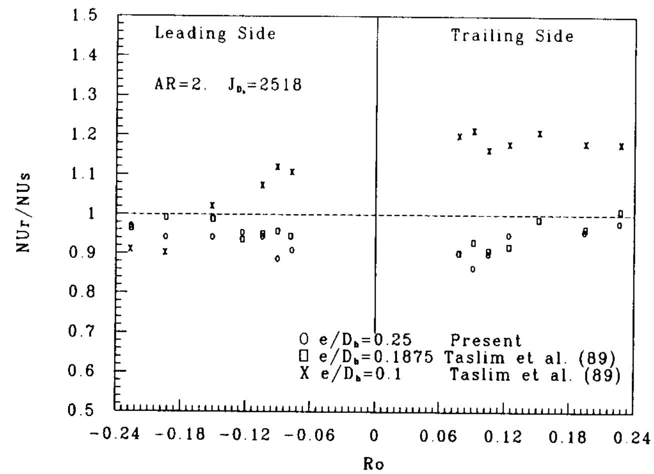

Fig. 18 Nu_r/Nu_s versus Ro for $AR = 2$ at $J_{D_h} = 2518$

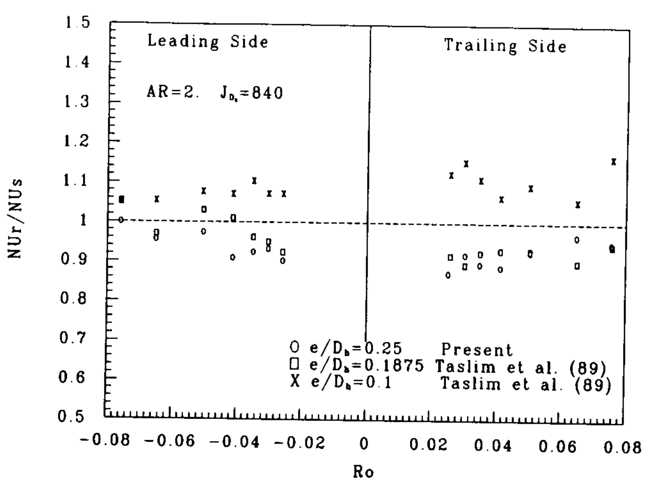

Fig. 16 Nu_r/Nu_s versus Ro for $AR = 2$ at $J_{D_h} = 840$

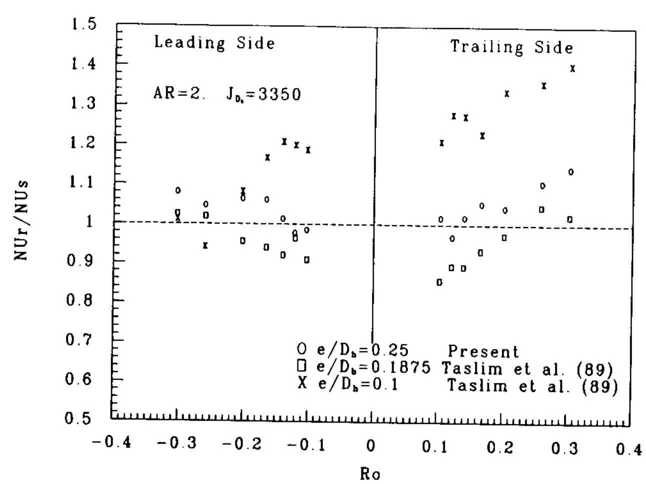

Fig. 19 Nu_r/Nu_s versus Ro for $AR = 2$ at $J_{D_h} = 3350$

associated with aspect ratio of unity do not follow this pattern. For these cases, first, variation of Nusselt number with Rotation number is not as pronounced and, second, the Nusselt number ratios remain below unity for most Rotation numbers tested. This behavior may be due to the limited relative space between turbulated walls, specially for AR=1, inadequate for the flow to separate from the leading side and push against the trailing side. A maximum increase of about 40% in heat transfer coefficient due to rotation for the aspect ratio of 2 is observed on the trailing surface for the blockage ratio of 0.1 (Fig. 19). The maximum decrease is about 15% for the blockage ratio of 0.25 on the trailing surface. For the aspect ratio of unity a maximum enhancement of about 25% due to rotation is observed for the small turbulators, $e/D_h = 0.1333$, on the leading surface of the test section. A maximum decrease of about 23% on the leading surface resulted from the medium sized turbulators, $e/D_h = 0.25$.

CONCLUSION

Based on the results presented here, it can be concluded that rotational effects are more pronounced in turbulated passages of higher aspect and lower blockage ratios for which a steady increase in heat transfer coefficient is observed on the trailing side as Rotation number increases while the heat transfer coefficient on the leading side shows a steady decrease with Rotation number. The all-smooth-wall classical pattern of heat transfer coefficient variation on the leading and trailing sides is not entirely followed for smaller aspect ratios and higher blockage ratios i.e. as the Rotation number increases, on the trailing side a rotational enhancement is associated with low blockage ratios while on the leading side any general decrease is associated with high blockage ratios.

ACKNOWLEDGEMENT

Financial support of the General Electric Company, Aircraft Engines, Lynn, Massachusetts is hereby gratefully acknowledged. The authors would also like to thank Mr. R.E. Gladden for his valuable suggestions during the course of this investigation.

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