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ABSTRACT

Experiments were conducted to determine the effects of buoyancy and Coriolis forces on heat transfer in turbine blade internal coolant passages. The experiments were conducted with a large scale, multi–pass, heat transfer model with both radially inward and outward flow. Trip strips on the leading and trailing surfaces of the radial coolant passages were used to produce the rough walls. An analysis of the governing flow equations showed that four parameters influence the heat transfer in rotating passages: coolant–to–wall temperature ratio, Rossby number, Reynolds number and radius–to–passage hydraulic diameter ratio. The first three of these four parameters were varied over ranges which are typical of advanced gas turbine engine operating conditions. Results were correlated and compared to previous results from stationary and rotating similar models with trip strips. The heat transfer coefficients on surfaces, where the heat transfer increased with rotation and buoyancy, varied by as much as a factor of four. Maximum values of the heat transfer coefficients with high rotation were only slightly above the highest levels obtained with the smooth wall model. The heat transfer coefficients on surfaces, where the heat transfer decreased with rotation, varied by as much as a factor of three due to rotation and buoyancy. It was concluded that both Coriolis and buoyancy effects must be considered in turbine blade cooling designs with trip strips and that the effects of rotation were markedly different depending upon the flow direction.

NOMENCLATURE

A	Area of passage cross–section
D	Hydraulic diameter
e	Trip height
Gr	Rotational Grashof number
h	Heat transfer coefficient
k	Thermal conductivity
m	Mass flowrate
Nu	Nusselt number, hD/k
P	Trip spacing, i.e. pitch
R	Radius
Re	Reynolds number, $(mD)/(\mu A)$
Ro	Rotation number, $\Omega D/V$
T	Temperature
V	Mean coolant velocity
x	Streamwise distance from inlet
μ	Absolute viscosity
ν	Kinematic viscosity
ρ	Coolant density
$\Delta\rho/\rho$	Density ratio, $(\rho_b - \rho_w)/\rho_b$
Ω	Rotational speed

Subscripts:

b	Bulk property
f	Film property
i	Inlet to model
w	Heated surface location
∞	Fully developed, smooth tube

Superscripts:

–	Average
'	Distance from beginning of second passage
''	Distance from beginning of third passage

INTRODUCTION

Advanced gas turbine airfoils are subjected to high heat loads that require escalating cooling requirements to satisfy airfoil life goals. The efficient management of cooling air dictates detailed knowledge of local heat load and cooling air flow distribution for temperature and life predictions. However, predictions of heat transfer and pressure loss in airfoil coolant passages currently rely primarily on correlations derived from the results of stationary experiments. Adjustment factors are usually applied to these correlations to bring them into nominal correspondence with engine experience. This is unsatisfactory when blade cooling conditions for new designs lie outside the range of previous experience.

Knowledge of the local heat transfer in the cooling passages is extremely important in the prediction of blade metal temperatures, i.e. blade life. Rotation of turbine blade cooling passages gives rise to Coriolis and buoyancy forces which can significantly alter the local heat transfer in the internal coolant passages due to the development of cross stream (Coriolis), as well as, radial (buoyant) secondary flows. Buoyancy forces in gas turbine blades are substantial because of the high rotational speeds and coolant temperature gradients. Earlier investigations (e.g. Eckert et al., 1953) with single pass co– and counter–flowing stationary coolant passages indicated that there can also be substantial differences in the heat transfer when the buoyancy forces are aligned with or counter to the forced convection direction. A better understanding of Coriolis and buoyancy effects and the capability to predict the heat transfer response to these effects will allow the turbine blade designer to achieve cooling configurations which utilize less flow and which reduce thermal stresses in the airfoil.

An extensive analytical and experimental program was originated and sponsored by NASA at the Lewis Research Center, Cleveland, Ohio, as part of the Hot Section Technology (HOST) program. The objectives of this program were (1) to gain insight on the effect of rotation on heat transfer in turbine blade passages, (2) to develop a broad data base for heat transfer and pressure drop in rotating coolant passages, and (3) to improve computational techniques and develop correlations that can be useful to the gas turbine industry for turbine blade design. The attainment of these objectives become even more critical with the advent of the Integrated High Performance Turbine Engine Technology (IHPTET) initiative. As part of the IHPTET goal, the turbine would operate at near stoichiometric (3500–4000F) inlet temperatures, maintain efficiencies in the 88–94% range, and require total coolant flows of only 5% of the engine air flow rate. To attain these ambitious goals, a thorough understanding on the rotational effects of heat transfer and flow in turbine blade passages is mandatory.

Previous Studies

Heat transfer experiments in multiple–pass coolant passages with normal trips have been conducted in stationary models by several investigators to obtain a data base for the thermal design of gas turbine airfoils, e.g. Boyle (1984), Han et al. (1986), Metzger et al. (1988). These data bases are directly applicable to the cooling designs of stationary vanes. However, the effects of Coriolis forces and buoyancy, due to the large rotational gravity forces (up to 50,000 g), are not accounted for.

The complex coupling of the Coriolis and buoyancy forces has prompted many investigators to study the flow field generated in unheated, rotating circular and rectangular passages without the added complexity of buoyancy, i.e., Hart (1971), Wagner and Velkoff (1972), Moore (1967) and Johnston et al. (1972). The effects of rotation on the location of flow reattachment after a backward facing step presented by Rothe and Johnston (1979) is especially helpful in understanding the effects of rotation on heat transfer in passages with trips. These investigators have documented strong secondary flows and have identified aspects of flow stability which produce streamwise oriented, vortex–like structures in the flow of rotating radial passages.

The effects of buoyancy on heat transfer without the complicating effects of Coriolis generated secondary flow have been studied in vertical stationary ducts. Effects of buoyancy on heat transfer were reported by Eckert et al. (1953), Metais and Eckert (1964) and Brundrett and Burroughs (1967). Flow criteria for forced–, mixed– and free–convection heat transfer was developed for parallel flow and counter flow configurations by Eckert et al. (1953) and Metais and Eckert (1964). Based on these experimental results, buoyancy forces would be expected to cause significant changes in the heat transfer in turbine blade coolant passages and to be strongly dependent on flow direction (radially inward vs. radially outward).

The combined effects of Coriolis and buoyancy forces on heat transfer has been studied by a number of investigators. Heat transfer in rotating models has been reported by Wagner et al. (1989 and 1990) Taslim et al. (1989), Guidez (1988), Clifford (1985), Iskakov and Trushin (1983), Morris (1981), Morris and Ayhan (1979), Lokai and Gunchenko (1979), Johnson (1978), and Mori et al. (1971). With the exception of Taslim and Clifford, all of the aforementioned work was conducted with smooth–wall models. Large increases and decreases in local heat transfer were found to occur by some investigators under certain conditions of rotation while other investigators showed lesser effects. Analysis of these results do not show consistent trends. The inconsistency of the previous results is attributed to differences in the measurement techniques, models and test conditions.

Objectives

Under the NASA HOST program, a comprehensive experimental project was formulated in 1982 to identify and separate effects of Coriolis and buoyancy forces for the range dimensionless flow parameters encountered in axial flow, aircraft gas turbines. The specific objective of this experimental project was to acquire and correlate benchmark–quality heat transfer data for a multi–pass, coolant passage under conditions similar to those experienced in the blades of advanced aircraft gas turbines. A comprehensive test matrix was formulated, encompassing the range of Reynolds numbers, rotation numbers, and heating rates expected in a modern gas turbine engine.

The results presented in this paper are from the second phase of a three phase program directed at studying the effects of rotation on a multi–pass model with smooth and rough wall configurations. The first phase utilized the smooth wall configuration. Initial results for outward flow in the first passage were previously presented by Wagner, Johnson and Hajek (1989). The effects of flow direction and buoyancy with smooth walls were presented by Wagner, Johnson and Kopper (1990). The present paper covers the phase with surface roughness elements oriented at 90 degrees to the flow direction. Comparisons will be made with the results for smooth walls in the same model and with previous rotating and stationary experiments employing trips 90 degrees to the flow direction. Results from the remaining phase of the program with trips oriented 45 degrees to the flow direction will be discussed in a subsequent paper.

The facility, data acquisition and data reduction techniques employed in this experiment were discussed in the Wagner et al. (1989) paper and will not be repeated. However, the description of the model will be repeated for the convenience of the reader.

DESCRIPTION OF EXPERIMENTAL EQUIPMENT

Heat Transfer Model

The heat transfer model was designed to simulate the internal multi-passage geometry of a cooled turbine blade (Figure 1). The model consists of three straight sections and three turn sections which were instrumented followed by one uninstrumented straight section, as shown in Figure 2. Data presented herein were obtained in the first, second and third passages with radially outward, inward and outward flow, respectively. The model passages are approximately square with a characteristic dimension of 0.5 in. (12.7 mm). Four elements form the walls of the square coolant passage at each streamwise location. The heated length of the first passage is 14 hydraulic diameters and is comprised of sixteen heated copper elements at four streamwise locations. The heated copper elements at the first streamwise location were all smooth walls and were used as guard heaters. The two cross-section views shown in the figure show the orientation of the leading, trailing and sidewall surfaces. Each copper element is heated on the side opposite the test surface with a thin film, 0.003 in. (0.1 mm), resistance heater. Each element is 0.150 in. (3.8 mm) thick and is thermally isolated from surrounding elements by 0.060 in. (1.5 mm) thick fiberglass insulators. The insulating material separating the copper elements at each streamwise location resulted in a 0.04 in. (1.0 mm) chamfer in the corners, which yielded a hydraulic diameter, D, in the straight sections of 0.518 in. (13.2 mm). The radius at the center of the heat transfer test sections with trips, i.e., average model radius, was 26.1 in. (663 mm). The power to each element was adjusted to obtain an isothermal wall boundary condition. In practice, temperature differences less than 2F (1C) were achieved. The heat flux between elements with a 2F (1C) temperature difference was estimated to be less than 2 percent of a typical stationary heat flux.

Trip strips were machined in a staggered pattern on the leading and trailing surfaces of the 6 inch (152.4 mm) straight length of each passage. No trips were on the guard elements ($x/D < 3$) in the first passage. The height, ($e/D = 0.1$), shape (circular) and spacing ($P/e = 10$) of the trips are shown in Figure 3. These geometrical parameters are typical of the trips cast on the coolant passage walls of turbine blades.

Testing was conducted with air at dimensionless flow conditions typical of advanced gas turbine designs. The required dimensionless rotation numbers were obtained with rotation rates of 1100 RPM or less by operating the model at a pressure of approximately 10 atmospheres. The model inlet air temperature was typically 80F (27C) and the copper elements were held at 120F, 160F, 200F and 240F (49C, 71C, 93C and 116C) for coolant-to-wall temperature differences of 40F, 80F, 120F and 160F (22C, 44C, 67C and 89C). Temperatures of the copper elements were measured with two chromel-alumel thermocouples inserted in drilled holes of each element. Heat transfer coefficients were determined by performing an energy balance on each copper element to obtain the convected heat flux and the local coolant bulk temperature. The heat transfer coefficients were based on the projected area rather

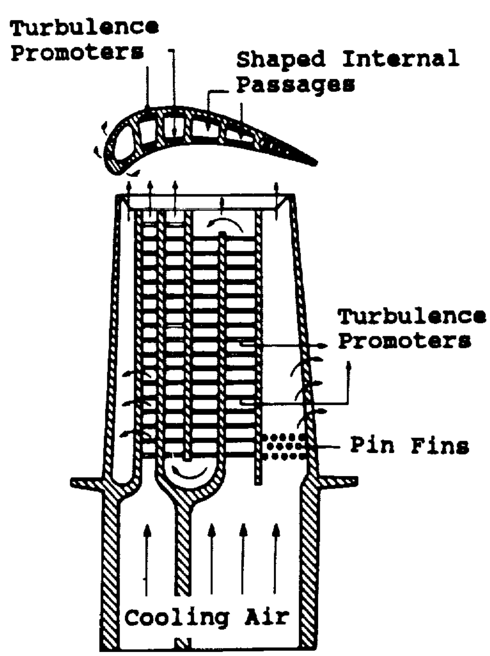

Fig. 1 Typical Turbine Blade Internal Convection Cooling Configuration (from Han et. al. [1986]).

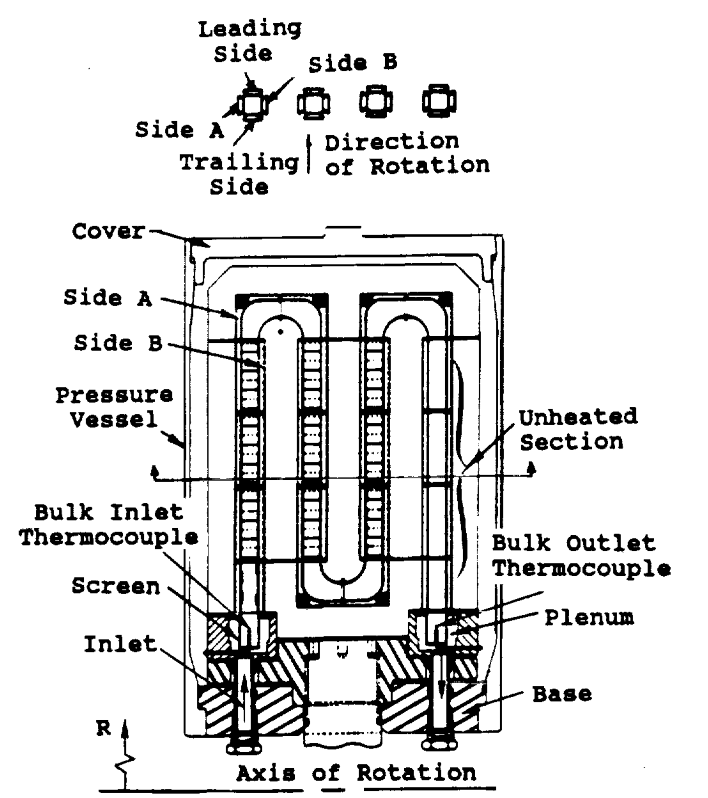

Fig. 2 Cross Sectional Views of Coolant Passage Heat Transfer Model Assembly.

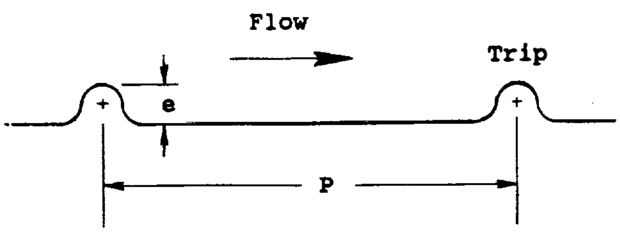

Fig. 3 Cross Sectional View of Normal Trip Layout.

than the total heat transfer surface area due to trip geometry. (The total heat transfer surface area was 1.11 times the projected area.) See Wagner et al. (1989) for additional information about the data reduction procedure.

Nusselt numbers and Reynolds numbers were calculated for each element. The fluid properties in the Nusselt and Reynolds numbers were evaluated at the film temperature, i.e., $T_f = (T_w + T_b)/2$. All of the heat transfer results presented herein have been normalized with a correlation for fully developed, turbulent flow in a smooth tube. The constant heat flux Colburn equation, adjusted for constant wall temperature was used to obtain the Nusselt number for fully developed, turbulent flow in a smooth tube (Kays and Perkins (1973)). The resulting equation for the constant wall temperature condition with a Prandtl number equal to 0.72 is as follows.

$$Nu_{\infty} = 0.0176\, Re^{0.8}$$

An uncertainty analysis of the data reduction equations using the methods of Kline and McClintock (1953) showed that approximately 3/4 of the estimated uncertainty in calculating heat transfer coefficient was due to the measurement of temperatures in the model. The uncertainty of the heat transfer coefficient is influenced mainly by the wall–to–coolant temperature difference and the net heat flux from each element. Uncertainty in the heat transfer coefficient increases when either the temperature difference or the net heat flux decreases. For increasing x/D, the uncertainty increases because the wall–to–coolant temperature difference decreases. For low heat fluxes (i.e. low Reynolds numbers and on leading surfaces with rotation) the uncertainty in the heat transfer increased. Estimates of the error in calculating heat transfer coefficient typically varied from approximately ±6 percent at the inlet to ±30 percent at the exit of the heat transfer model for the baseline stationary test conditions. The uncertainty in the lowest heat transfer coefficient on the leading side of the third passage with rotation is estimated to be 40 percent, primarily due to the uncertainty in the calculated bulk temperature. Although the uncertainty analysis was useful in quantifying the maximum possible uncertainty in calculating heat transfer coefficient, multiple experiments at the same test condition were repeatable within ranges smaller than those suggested by the analysis.

RESULTS

Forward

Heat transfer in stationary experiments with augmentation devices on the passage walls is primarily a function of the Reynolds number (a flow parameter), the streamwise distance from the inlet, x/D (a geometric parameter), and the geometry of the augmentation device. However, when rotation is applied, the heat transfer is also strongly influenced by the coupled effects of Coriolis and buoyancy and becomes asymmetric around the passage. An unpublished analysis of the equations of motion by Suo (1980), similar to that of Guidez (1988), showed that the basic dimensionless fluid dynamic parameters governing the flow in a radial coolant passage were the Reynolds number, the rotation number, Ro, the fluid density ratio, $\Delta\rho/\rho$, and the geometric parameter, R/D. The same analysis of the equations of motion produces the rotational Reynolds number, $J = \Omega D^2/\nu$ as an alternate governing parameter. Note also that Ro equals J/Re. Note that the rotation parameter is the reciprocal of the Rossby number, $V/\Omega D$, and governs the formation of cross–stream secondary flow. The rotation number, Ro, the fluid density ratio, $\Delta\rho/\rho$, and the geometric parameter, R/D, appear in the governing equation as a buoyancy parameter. This buoyancy parameter, $(\Delta\rho/\rho)(R/D)(\Omega D/V)^2$, is similar to Gr/Re^2 for stationary heat transfer. The difference between our rotational buoyancy parameter and the stationary Gr/Re^2 is that $\Delta\rho/\rho = (T_w - T_b)/T_w$ rather than $\beta\Delta T = (T_w - T_b)/T_b$. The difference between the parameters decreases as T_w approaches T_b. Thus, with rotation, the heat transfer is a function of three geometric parameters (surface roughness geometry, x/D and surface orientation relative to the direction of rotation) and three flow parameters (Reynolds number, rotation number and the buoyancy parameter).

Due to the vector nature of the equations of motion, it can also be expected that flow direction can also have a significant effect on the coolant flow. In the parallel flow case, the flow is radially inward, coincident with buoyancy driven flow for heated walls. For the counter–flow case the flow is radially outward, opposite to the direction of the buoyancy driven flow. Flow direction (i.e. radially inward or outward) and a fixed radially outward directed force field, created by the rotating reference frame, establish the potential for parallel and counter flow situations as observed by Eckert et al. (1953) in their vertical tube experiments.

The references used in the text for low and high pressure surfaces are consistent with the leading to trailing side, Coriolis–generated, pressure gradients. In general, high pressure surfaces are expected to have normal components of flow towards the surface while low pressure surfaces are expected to have normal components of flow away from the surface. Therefore, trailing surfaces in the first passage with outward flow are on the high pressure side of the passage. Similarly, leading surfaces in the second passage with inward flow are on the high pressure side. In terms of turbine airfoils, the leading surfaces of the coolant passage are adjacent to the suction side of the airfoil and the trailing surfaces of the coolant passage are adjacent to the pressure side of the airfoil.

The format of this paper is to show the effects of each of the primary variables (x/D, rotation number, density ratio) on the heat transfer about a baseline flow condition to develop an understanding of the cause/effect relationships. The entire body of experimental results are then examined to determine the effects of the buoyancy parameter on the heat transfer in selected locations of the coolant passage.

Baseline Experiments

Two baseline experiments, one stationary and one rotating, were conducted to obtain data for comparison with all other data generated in this program. The stationary and rotating baseline experiments had dimensionless flow conditions which consisted of a Reynolds number of 25,000 and an inlet density ratio, $(\Delta\rho/\rho)_i = (T_w - T_b)/T_w$, of 0.13. The rotating baseline experiment had a rotation number, $\Omega D/V$, of 0.24 and a radius ratio at the average model radius, R/D, of 49. These values were selected because they are in the central region of the operating range of current large aircraft gas turbine engines.

Stationary. Streamwise variations of Nusselt number for the stationary baseline test are shown in Figure 4. The Nusselt number for fully developed, turbulent flow in a smooth tube with constant wall temperature and the results from the previous (Wagner et al. 1990) smooth wall experiments are shown for comparison.

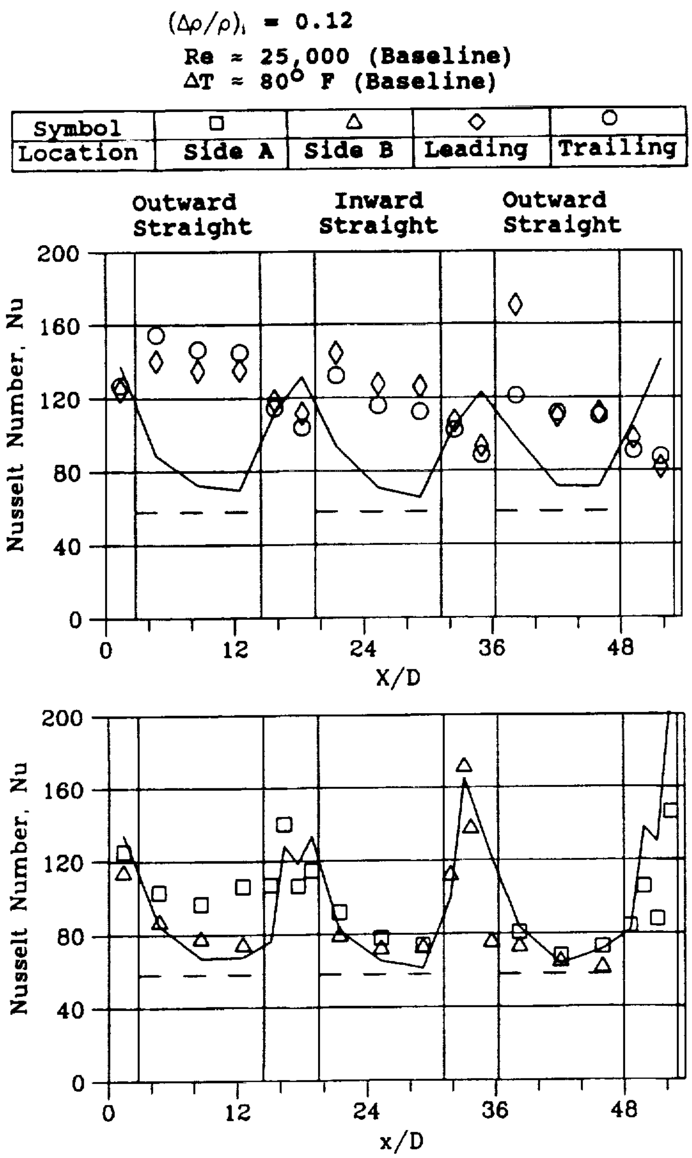

Fig. 4 Heat Transfer Results for Stationary Flow Condition; --- correlation for smooth-wall fully developed flow, —— smooth-wall experiments (Wagner, et. al. 1990).

The heat transfer from the walls with trips (denoted leading and trailing) in the first outward straight (3 < x/D < 14) passage has heat transfer coefficients more than twice that from the fully–developed, smooth–wall correlation. Note that the heat transfer coefficients for the normal trips do not decrease significantly with x/D in each passage as they did for the smooth wall in the same model. Some differences in heat transfer are observed between the leading and trailing surfaces for this stationary baseline condition. The exact cause of the difference is not known but may be due to the staggering of the trips on the two surfaces. The heat transfer coefficients measured in the remaining two passages (i.e., 20 < x/D < 31 and 36 < x/D < 48) show similar characteristics. However, the greatest increase in heat transfer from the trips was less (i.e. 10 and 20 percent, respectively) than that obtained in the first outward straight section. This general reduction in heat transfer was attributed to the increased uncertainty in the bulk temperature for the model with the normal trips. The increased heat transfer compared to the smooth wall model causes the difference between bulk temperature and the wall temperature to decrease and hence the uncertainty of the heat transfer coefficient determined to increase.

The heat transfer in the turn regions was generally less for the present experiment than for the previous smooth wall experiments. These changes on the leading and trailing surfaces of the turn sections are attributed in part to the differences in the velocity profiles expected at the entrance to the turn regions. For the smooth wall flow condition, the velocities are expected to be high in the corners of the duct (e.g. Schlictling, 1968). For flow over normal trips, the velocity can be expected to be peaked in the center of the channel due to the large momentum losses at each trip. The changes in heat transfer on the sides A & B (outside walls of turn sections) attest to the complexity of the flow structure in the turns and is not yet explained.

The results from the first outward straight coolant passage are compared with results from Boyle (1984) and Han et al. (1986) in Figure 5. The present results in the region with trips, 3 < X/D < 14, are almost identical with those from Boyle. The Boyle results were obtained for a constant heat flux boundary condition and sharp cornered

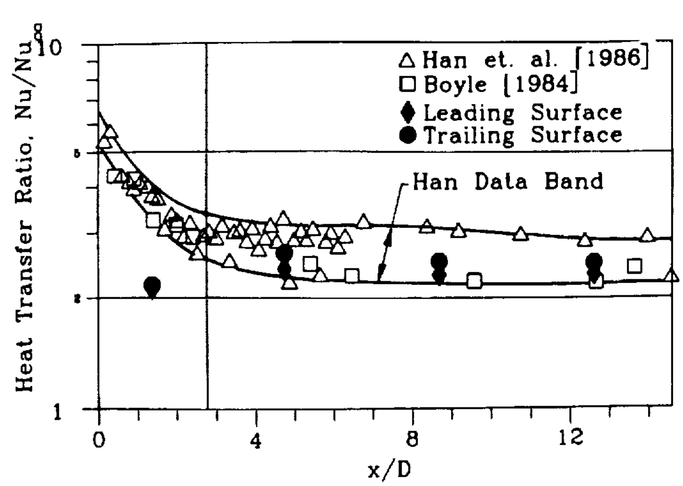

Fig. 5 Comparison of Stationary Heat Transfer Results From Leading and Trailing Surfaces With Han et. al. (1986).

trips which are modest variations from the present experiment. Heat transfer ratios from the surfaces with trips are generally consistent with the data band for Han's measurements. Note that the heat transfer results from the present program for x/D < 3 are from the smooth wall surfaces near the inlet of the first passage. However, in general, the levels of heat transfer augmentation due to the presence of the trips are consistent with those of Boyle and Han et al.

Rotating. The streamwise distributions of heat transfer ratio for the rotating baseline condition for the first two coolant passages are shown in Figure 6. These results and those discussed in the following sections are shown as heat transfer ratio, Nu/Nu_∞. Nu_∞ is that expected from the Kays and Perkins (1973) correlation for fully developed, turbulent flow. The results will be shown in this manner to minimize effects of Reynolds number variations from test to test.

The most important feature of these results is the decrease in heat transfer on the "low pressure" sides shown for the leading surfaces for flow outward (x/D < 14) and the trailing surfaces for flow inward (x/D < 31). The lowest values of Nu/Nu_∞ are less than one-half the nonrotating values. The heat transfer on the high pressure side of the coolant passage with flow outward (i.e., the trailing surfaces) increases about 50 percent compared to the stationary case. However, the heat

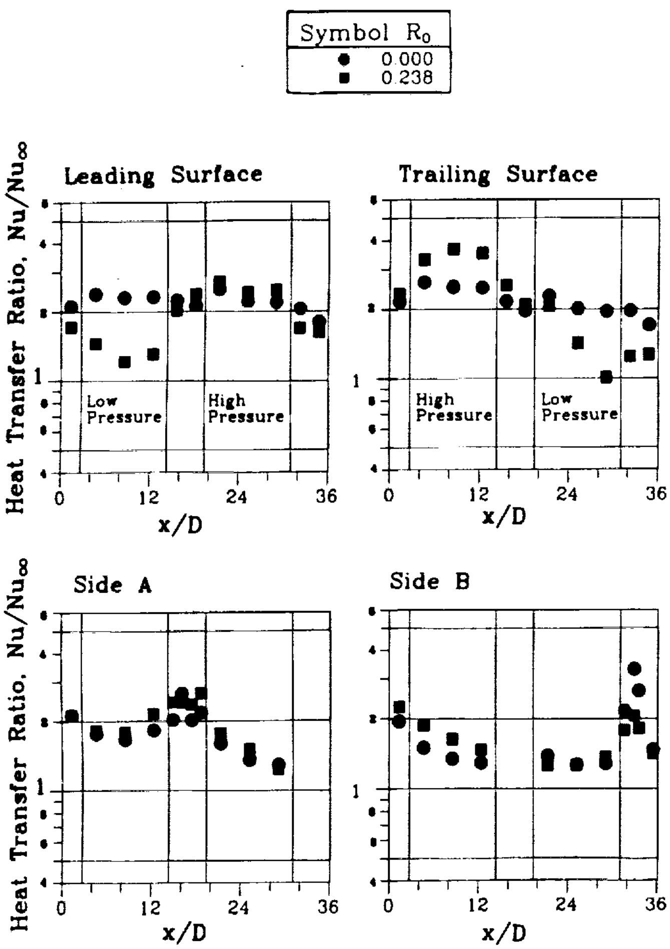

Fig. 6 Variation of Heat Transfer Ratio with Streamwise Location for "Rotating" Baseline Flow Condition; Re=25000, R_o=0.24, $(\Delta\rho/\rho)_i$=0.13.

transfer on the leading surface for flow inward does not increase noticeably. These results are qualitatively similar to those obtained for the smooth wall model. Further comparison with the smooth wall model results will be made in a later section.

The baseline results with rotation showed significant changes in the heat transfer in the first passage on the leading, trailing, and turn surfaces but relatively smaller changes on the sidewall surfaces. Therefore, the following discussion will focus on the heat transfer results from only the leading and trailing surfaces in the straight sections of the coolant passage with both inward and outward flow and will focus on the differences between inward vs. outward flow. Discussion of effects of rotation on the heat transfer in the turn regions of the coolant passage are deferred to a subsequent paper.

Varying Rotation Number

The rotation number, $\Omega D/V$, was varied from 0 to 0.35 for this series of flow conditions. The Reynolds number, inlet density ratio and radius ratio were held constant at the nominal values of 25,000, 0.13 and 49, respectively.

High Pressure Surfaces. Increasing the rotation rate causes significant increases in heat transfer on the trailing surfaces (Figure 7a) of the first passage but relatively small increases occurred on the leading surfaces in the second passage (Figure 7b). Heat transfer in the first passage increased by more than 60 percent for the largest value of rotation parameter (0.35) compared to stationary heat transfer values. The substantial increases in heat transfer in the first passage are consistent with the results of Rothe and Johnston (1979). They found that as rotation rate was increased, the reattachment length after a step decreased. For the trip spacing of the present program (P/e = 10), this would translate into an increase in the effective heat transfer area between the trips with attached, turbulent flow, thereby, causing an increase in the heat transfer. Compared to the stationary results, the heat transfer on the leading, high pressure side of the second passage increased approximately 10 percent. The effects on heat transfer due to Coriolis generated secondary flows and flow reattachment might be expected to be approximately the same for the first and second passages. The differences in heat transfer between the outward and inward flowing passages are therefore attributed to the different effects of buoyancy in the counter-flowing first passage (radially outward flow) and the co-flowing second passage (radially inward flow). In general, the trends noted above are compatible with those obtained for the smooth wall test surfaces in the same model (Wagner et al. 1990).

The small increase in the heat transfer ratio on the high pressure side of the second passage relative to the first passage is attributed to a reduction in the generation of near-wall turbulence. In the first passage, the near-wall buoyancy driven flow was inward toward the axis of rotation and the coolant flow was outward. This counter flow is expected to generate additional near-wall turbulence due to the strong shear gradient. The large increases in heat transfer in the first passage are attributed to the destabilizing effects of the shear flow combined with the cross stream secondary flows generated by Coriolis forces. However, when the flow and the buoyancy driven near-wall flows are coincident, as in the second passage, the generation of near-wall turbulence may be diminished because of the relatively weaker near-wall shear layer. The expected lower near-wall turbulence and weaker shear flows may also contribute to increases in reattachment lengths following the trips. Therefore, the reduced effects of the buoyant and the cross stream secondary flows coupled with possible increases in reattachment lengths in the second passage may have resulted in lesser changes in heat transfer. The magnitude of the buoyancy effect on the heat transfer is unclear in that the buoyancy effect on the heat transfer in the second passage may be zero (which implies a modest Coriolis dominated heat transfer increase) or negative (which implies a larger Coriolis dominated heat transfer increase which is offset by a reduction due to buoyancy). Future results from concurrent numerical simulations of these flow conditions are expected to assist in the understanding of this complex flow field.

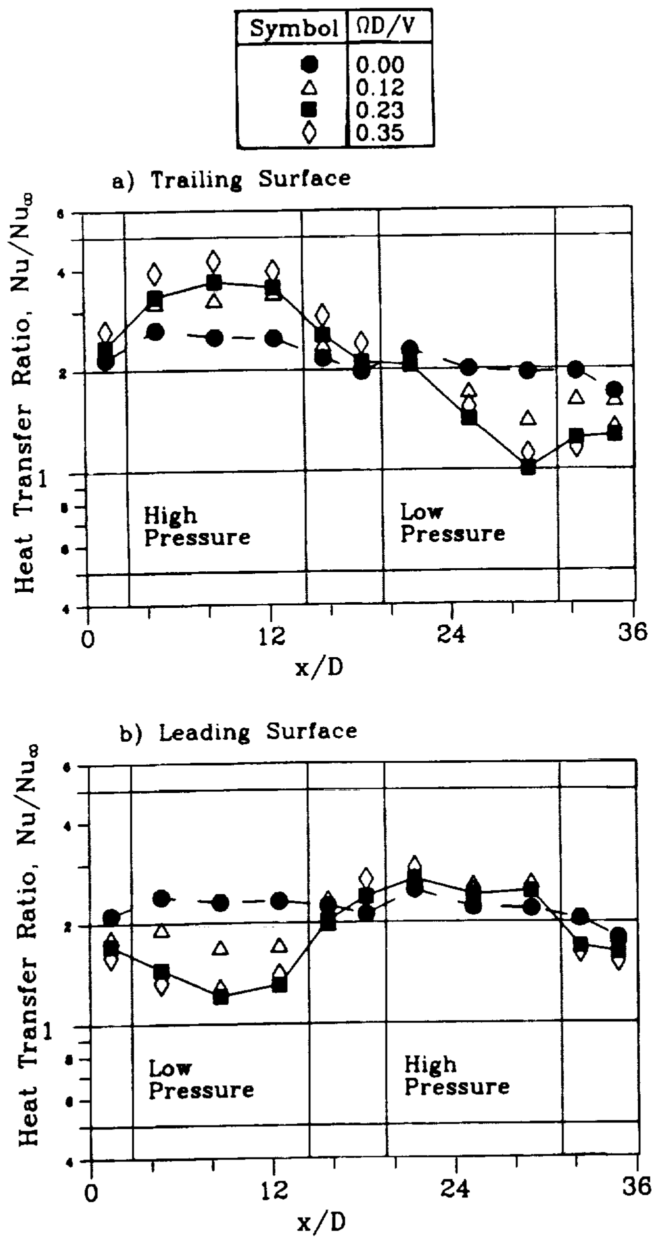

Fig. 7 Effect of Rotation Number on Heat Transfer Ratio; Re=25000, $(\Delta\rho/\rho)_i$=0.13, R/D=49.

Low Pressure Surfaces. In contrast to the continual increase in heat transfer with increasing rotation number on the trailing side, the heat transfer ratio decreases with increasing rotation number on the leading side of the passage near the inlet, i.e. $x/D < 6$. For all of the remaining locations on the leading side of the passage, the heat transfer ratio decreases and then increases again with increasing rotation number. Heat transfer from the trailing, low pressure surfaces of the second passage also had large decreases in heat transfer. Heat transfer in the first and second passages decreased to almost 50 percent of the stationary heat transfer levels. In both passages, the heat transfer decreased and then subsequently increased again as the rotation rate was increased.

The decreases in the heat transfer ratio are attributed to the cross-stream flow patterns as well as the stabilization of the near-wall flow on the leading side of the passage, e.g, Johnston et al. (1972). The cross-stream flows cause heated, near-wall fluid from the trailing and sidewall surfaces to accumulate near the leading side of the coolant passage resulting in reduced heat transfer. In addition, as described by Rothe and Johnston (1979), it can be expected that flow reattachment after trips on low pressure surfaces occurs at larger distances from the trips with increasing rotation number. Longer reattachment lengths, due to the stabilizing effects, will decrease the effective heat transfer area between trips, thereby, further reducing the turbulent transport of heat. The increase in the heat transfer ratio in the latter half of the coolant passage for the larger rotation numbers is attributed to buoyancy effects, possibly caused by buoyancy enhanced flow in the recirculation cells downstream of the trips. Similar effects of rotation are noted for the low pressure surfaces in both the first and second passages, with flow radially outward and radially inward, respectively. These results suggest that the decrease in heat transfer on low pressure surfaces with trips is dominated by Coriolis generated cross-stream flows which cause a stabilization of the near-wall flows and that the heat transfer on the high pressure surfaces is affected by a combination of Coriolis and buoyant effects. Therefore, it can be expected that the correlations of local heat transfer data may be substantially different, depending on local flow conditions (i.e. due to differing near-wall shear gradients).

Varying Density Ratio

The inlet density ratio, $(\Delta\rho/\rho)_i$, was varied from 0.07 to 0.22 for this series of flow conditions. The Reynolds number, rotation number and radius ratio were held constant at the baseline values of 25,000, 0.24 and 49, respectively. Heat transfer was obtained at a fixed rotation number and, therefore, conclusions can be obtained regarding the effects of buoyancy for flow conditions near the rotating baseline flow conditions.

Increasing the inlet density ratio (i.e., the wall-to-coolant temperature difference) from 0.07 to 0.22 causes the heat transfer ratio in the first passage to increase on all trailing surfaces by as much as 25 percent (Figure 8a) and on the leading surfaces by as much as 20 percent (Figure 8b). The exception to the general increase in heat transfer with increasing density ratio occurred near the inlet of the first passage on the leading side, where the heat transfer ratio is observed to be relatively unaffected by varying density ratio. Heat transfer in the second, inward flowing passage on the low pressure side increased as much as 70 percent with increases in the temperature difference (Figure 8a). (Larger effects of density ratio were obtained for a rotation number of 0.35.)

Varying Rotation Number and Density Ratio

Additional data from parametric variations of density ratio and rotation parameter were necessary to determine the effects of rotation and buoyancy over the range of interest. The inlet density ratio was varied from 0.07 to 0.23 for selected rotation numbers. Heat transfer results from these experiments were plotted vs. inlet density ratio with rotation number as a secondary variable. The variation of heat transfer ratio with density ratio (not shown) was extrapolated for each value of

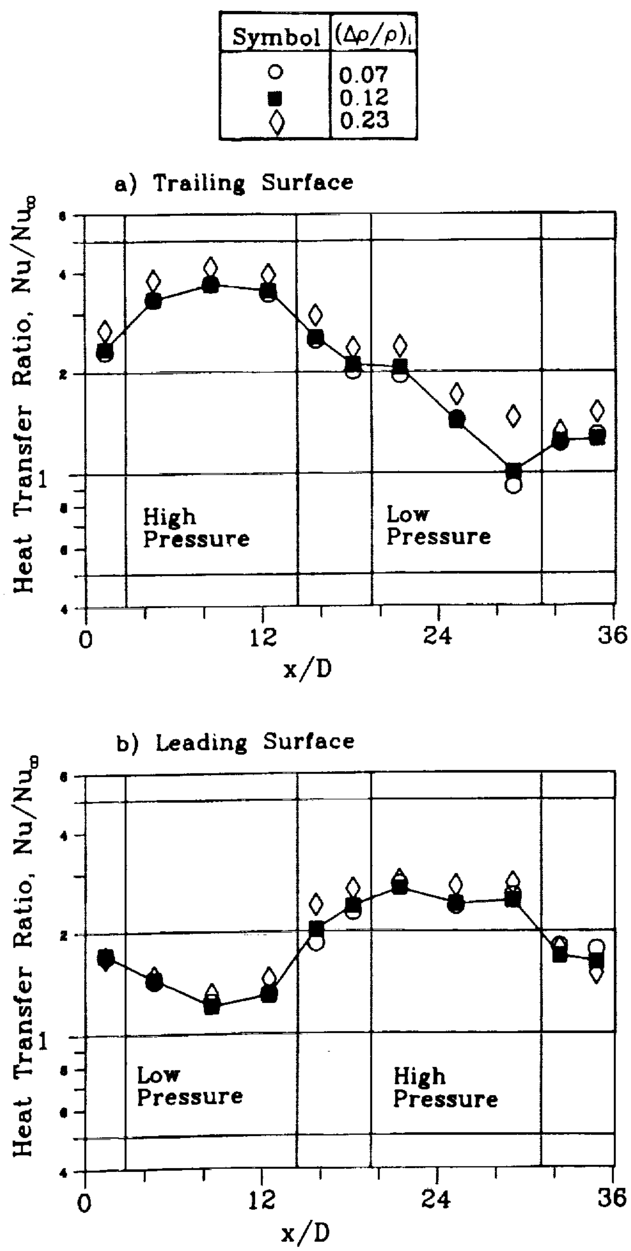

Fig. 8 Effect of Wall-to-Coolant Density Difference on Heat Transfer Ratio; Re=25000, R_o=0.24, R/D=49.

the rotation number to obtain the value of the heat transfer ratio at a density ratio of 0.0 (i.e., limit as ΔT approaches 0.0). The heat transfer results obtained from the experiments plus the extrapolated values for a density ratio of 0.0 (dashed lines) are presented in Figure 9 as the variation of heat transfer ratio with the rotation number with the density ratio as the secondary variable for three streamwise locations for the first and the second passage. The following discussion will concentrate on the differences in the heat transfer from the first and second passages.

High Pressure Surfaces. Heat transfer results from the high pressure side of the first and second passages is shown in Figure 9a and b for ranges of rotation number and density ratio. Note that no effect of density ratio on the heat transfer ratio was expected (e.g. Wagner et al. 1990) for a rotation number of 0 when film properties are used for the dimensionless heat transfer and flow parameters. Increasing the rotation number causes local increases in the heat transfer in the first passages by as much as 75 percent compared to the heat transfer for a rotation number of 0. Whereas the heat transfer ratios for the high pressure surfaces in the first passage increase sharply with increases in either the density ratio or the rotation number, the heat transfer ratios in the second passage are less affected (increases of 30 to 35 percent) by variations of either parameter.

Low Pressure Surfaces. The heat transfer from the low pressure surfaces from the first and second passages (Figure 9a and b) is more complex than that from the high pressure surfaces. The heat transfer ratio in the first passage decreases with increasing rotation number for low values of rotation number (i.e., $\Omega D/V < 0.25$ at the downstream location) and then increases with increases in rotation for larger values of rotation number depending on density ratio. The heat transfer ratio increases with increases in the density ratio, similar to the results obtained for the trailing surface of the first passage.

The effects of density ratio on the heat transfer ratio are larger in the second passage with radially inward flow than in first passage, (a factor of three for the second passage compared to a factor less than two for the first passage) for inlet density variations from 0.07 to 0.23. Note that the local density ratios in the second passage will be about half of the inlet values.

The more complicated heat transfer distributions on the low pressure surfaces of the coolant passages are attributed to 1) the combination of buoyancy forces and the stabilization of the near-wall flow for low values of the rotation number and 2) the developing, Coriolis driven secondary flow cells and 3) the increases in flow reattachment lengths after trips for the larger values of the rotation number. It is postulated that the relatively small effects from variations in density ratio near the inlet of the second passage and the large effects near the end of the second passage are due to the development of the near-wall thermal layers (i.e. thickening for the normal trip model compared to thinning for the smooth wall model). Near the inlet of the second passage, the thermal layers are postulated to be thin because of the strong secondary flows in the first turn region. With increasing x/D, the turn dominated secondary flows diminish and the counteracting effect of buoyancy and the Coriolis generated secondary flow increases.

CORRELATING PARAMETERS

The analysis of the equations of motion for flow in rotating radial passages by Suo (1980), discussed above, showed that 1) the cross–stream flows will be proportional to the rotation number, $\Omega D/V$, and 2) the buoyant flows will be proportional to the buoyancy parameter, $(\Delta\rho/\rho)$ $(R/D)(\Omega D/V)^2$. The combined effect of the cross–stream flows and the buoyant flows is not easily ascertained from the equations of motion. The preceding discussions indicate that the combined effects are quite complex and are a strong function of flow direction. Therefore, the flow direction is also considered in the following paragraphs.

The buoyancy parameter, discussed previously, is similar to the ratio of the Grashof number (with a rotational gravitation term, $R\Omega^2$) to the square of the Reynolds number and has previously been used to characterize the relative importance of free– and forced–convection in the analysis of stationary mixed–convection heat transfer. Guidez (1988) used a similar analysis to establish appropriate flow parameters for the presentation of his results. These parameters, $\Omega D/V$ and $(\Delta\rho/\rho)$ $(R/D)(\Omega D/V)^2$, will also be used in the present discussion of the effects of Coriolis and buoyancy forces on the heat transfer for inward and outward flow directions.

The data was analyzed to determine the effects of flow direction (radially inward or radially outward) on the heat transfer characteristics and to determine the differences between the first passage with outward flow downstream of an inlet, the second passage with inward flow downstream of a 180° turn and the third passage with outward flow downstream of a 180° turn. The variations of heat transfer ratio with buoyancy parameter for the heated surface at the most downstream location from the inlet or a turn for each of the three passages are shown in Figure 10 with heat transfer ratios obtained in the same model with smooth surfaces.

The data presented in Figure 9 showed that the effects of Coriolis and buoyancy forces are coupled in the first two passages through the entire operating range investigated. The results from Figure 9 are presented in Figure 10 as the variation of the heat transfer ratio with the buoyancy parameter based on the local density ratio and radius, R. Thus, the range of the buoyancy parameter decreases with increasing values of x/D (i.e. decreasing temperature difference with increasing x). The temperature differences, T_b–T_w, at the end third passage were only one–third of the inlet value.

Heat transfer distributions from the low pressure surfaces of each of the three passages exhibit a similar relationship with the buoyancy parameter. Heat transfer for all values of $(\Delta\rho/\rho)_i$ decreases with increasing values of buoyancy between 0.0 and 0.15. Heat transfer

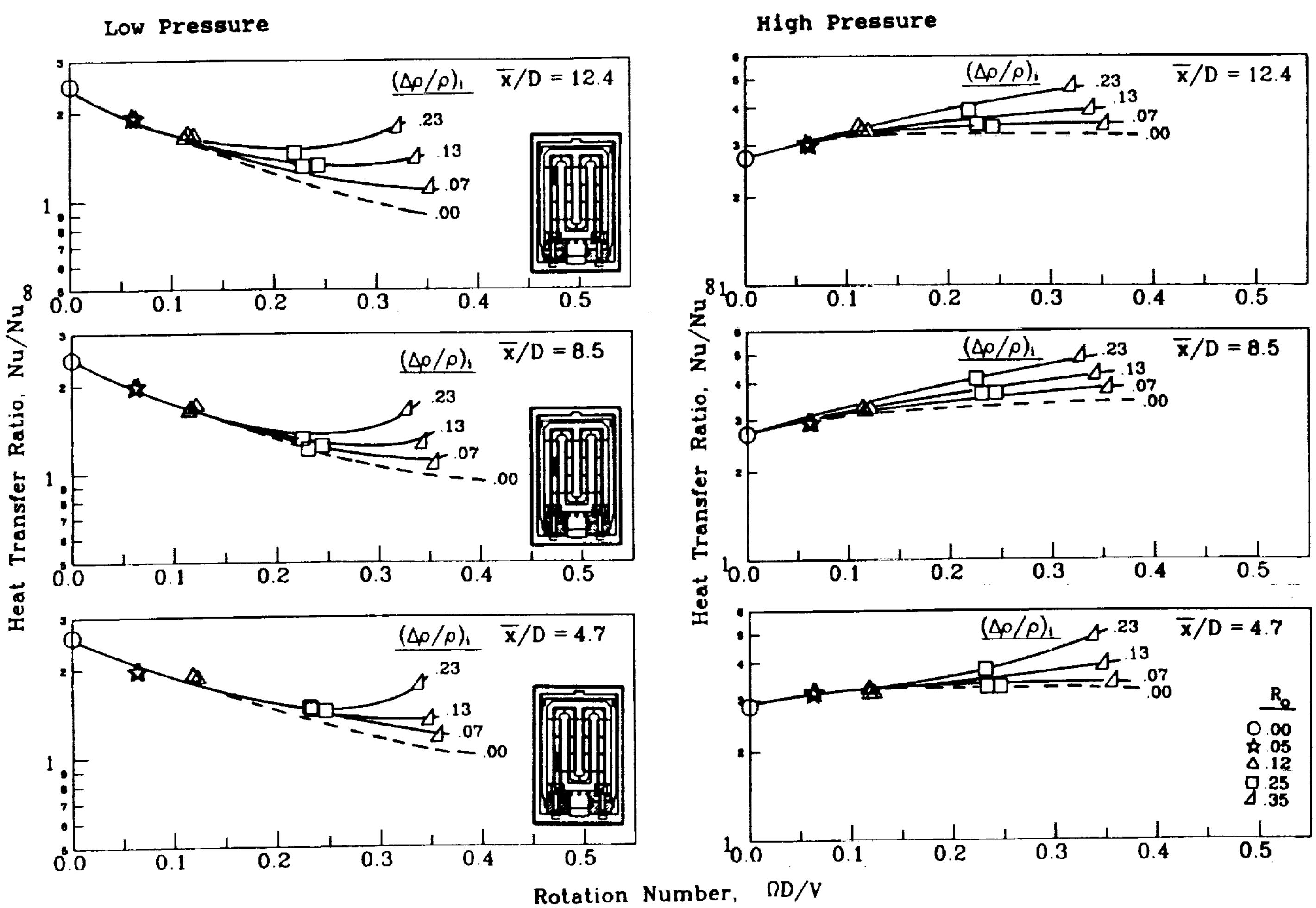

Fig. 9a Effect of Rotation Number and Density Ratio on Heat Transfer Ratios in the First Passage; Re=25000, R/D=49.

subsequently increases again with increasing values of buoyancy. Heat transfer on the low pressure surfaces of rotating coolant passages is governed by complex relationships of streamwise location, rotation number and buoyancy parameter.

The heat transfer results from the high pressure surfaces in the first passage are better correlated by the buoyancy parameter. The second passage with radially inward flow had different heat transfer characteristics than the first and third passages with radially outward flow. Whereas the heat transfer ratios for conditions of large density ratios for the high pressure surfaces of the first and third passages generally increased with the buoyancy parameter, the heat transfer in the second passage was relatively less affected by buoyancy parameter for values of buoyancy greater than 0.05. These results for co-flowing and counter-flowing buoyancy effects on the high pressure surfaces are generally consistent with the stationary combined free- and forced-convection experiments of Eckert et al. (1953). They measured decreased levels of heat transfer for the co-flowing condition (i.e. similar to that of radially inward flow in rotating systems).

The heat transfer results for surfaces with trips show trends which are similar to those observed for the same model with smooth surfaces. It is also interesting to note that the levels of heat transfer augmentation obtained in the first passage of the model with trips are only 10 to 30 percent greater than those for the smooth model for values of the buoyancy parameter greater than 0.4. The difference would be even less if the heat transfer coefficient were based on the total surface area (i.e., including trip area) instead of the projected surface area.

COMPARISON WITH PREVIOUS ROTATING EXPERIMENTAL RESULTS

Results from this study have shown that rotational and buoyancy forces strongly influence turbulent heat transfer in rotating passages with trips normal to the flow for conditions found in gas turbine blades. The heat transfer results from stationary models with similar geometries agree quite well with the present work, i.e., Boyle (1984), Han et al. (1986) and Metzger et al. (1988). The heat transfer results from rotating models are more difficult to compare because of differences in the geometries and the boundary conditions. However, the heat transfer results of Clifford (1985) and Taslim et al. (1989) obtained with rotation will be related to the present results.

Clifford (1985) obtained heat transfer coefficients in a multi-pass model with trips normal to the flow using transient measurement techniques. Direct comparison with Clifford's results is not possible

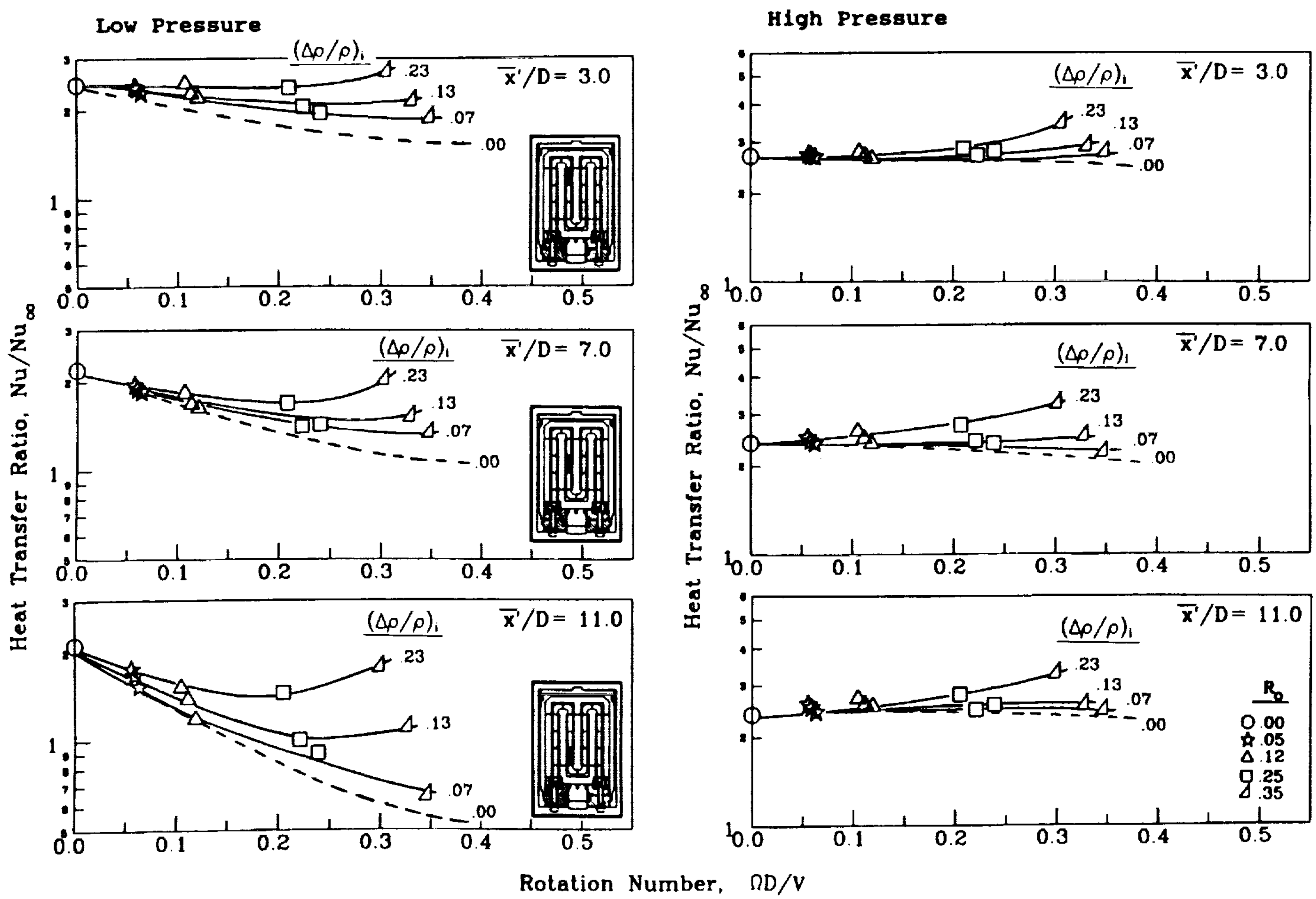

Fig. 9b **Effect of Rotation Number and Density Ratio on Heat Transfer Ratios in the Second Passage; Re=25000, R/D=49.**

due to the lack of specific model geometry and precise test conditions. Clifford observed increases in heat transfer of 36 percent on the pressure side of the model and decreases of 24 percent on the suction side of the first passage. Clifford's trends are in general agreement with the present results. However, the effects of rotation measured by Clifford are somewhat less than those measured in the present experiment. Clifford's heat transfer data from the second, inward flowing passage, was generally consistent with the present results.

Taslim et al. (1989) also obtained heat transfer results in a rotating square passage with trips normal to the flow for several trip heights. Trips were square-edged and were mounted on two opposing walls (one heated). The remaining smooth walls and one of the walls with trips were unheated. Although all of the heat transfer results with rotation measured by Taslim were greater than the stationary value for Re_d = 24800 and e/D = 0.133, the leading side heat transfer coefficients with rotation decreased with increasing rotation rate. This effect is similar to that observed by Clifford and in the present results. Taslim also measured increases in heat transfer, for most Reynolds numbers, on the trailing side of the model with increases in rotation rate for low values of rotation rate followed by relative decreases for further increases in rotation. The observations of Taslim on the trailing side of the passage are inconsistent with the present experiment where heat transfer was observed to increase with increases in rotation rate for a similar range of rotation number. The differences in the measured effects of rotation on the trailing side heat transfer are attributed to the differences in trip geometry (e/D = 0.1 and round trips for the present work and e/D = 0.133 and square trips for Taslim) and to the differences in the wall boundary conditions (T_w = constant for the present work and q_w = constant on one wall for Taslin). Additional work is necessary to determine the effects of model geometry and thermal boundary conditions with rotation.

CONCLUDING COMMENTS

Results from the present experiments with normal trips in rotating, radial, square coolant passages show that Coriolis forces and buoyancy effects can strongly influence heat transfer. The heat transfer coefficients on surfaces with trips were especially sensitive to rotation and buoyancy, decreasing as much as to one-third the stationary value due to rotation and increasing by a factor of 2.5 due to buoyancy. These effects were greater than measured previously for a smooth wall model. The maximum heat transfer coefficients on the pressure side of the coolant passage at highest values of the buoyancy parameter were not

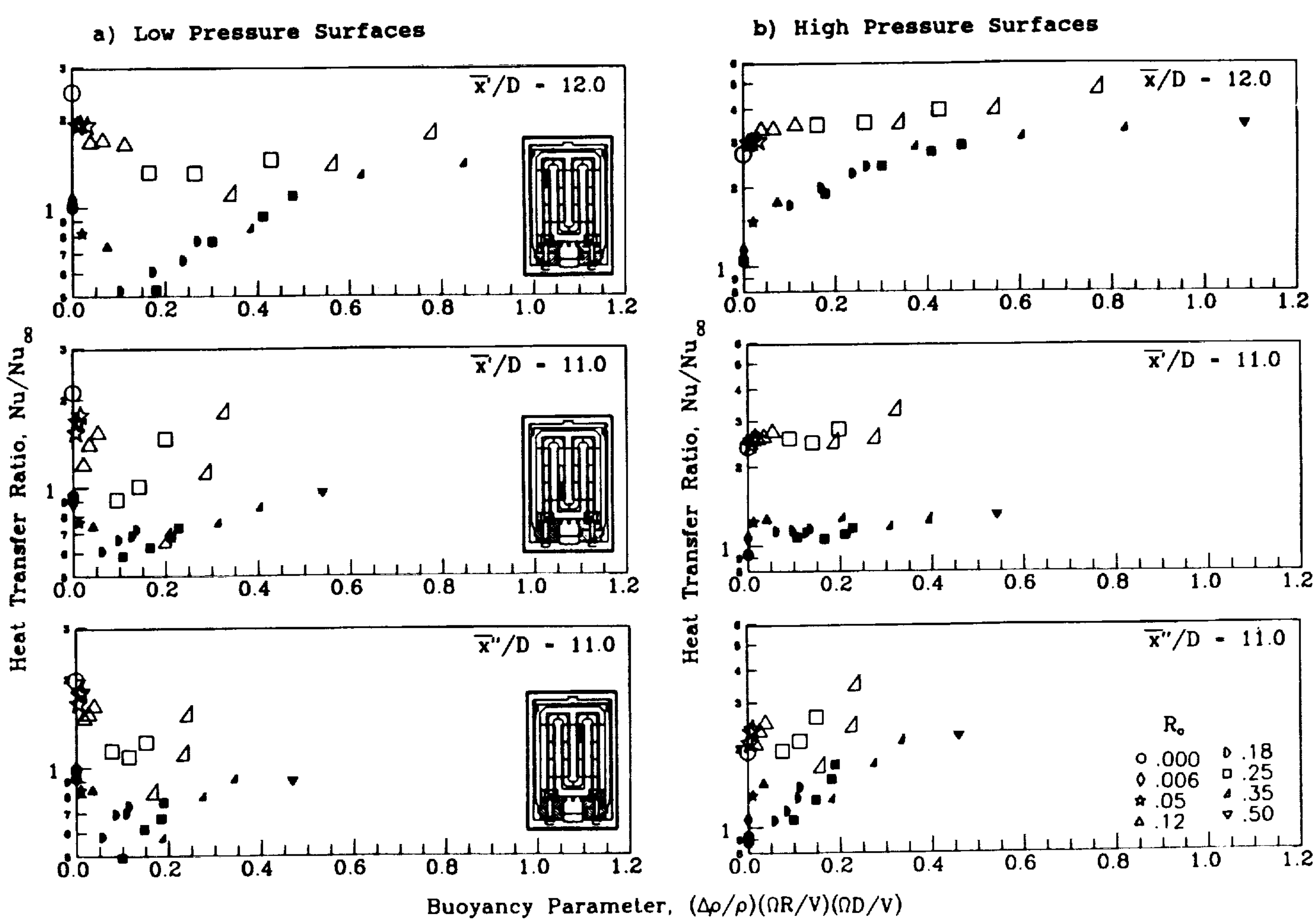

Fig. 10 Comparison of Heat Transfer Ratios of the First, Second and Third Passages; open symbols - normal trip data, solid symbols - smooth wall data.

much greater than obtained for a smooth wall model. The conclusion from this second observation is that some trips in the coolant passage can be relatively ineffective for certain combinations of coolant passage geometries and rotating flow conditions.

The comparison of results from the present experiments with previous results show that flow and heat transfer in rotating coolant passages can be complex, especially when no single flow mechanism dominates the heat transfer process. The present results were obtained for normal trips with values of trip pitch to trip height (P/e = 10) and trip height to coolant passage width (e/D = 0.1), typical of those used in coolant passages. This trip geometry generally produces heat transfer coefficients two times those obtained for smooth wall passages. The wide range of heat transfer coefficients obtained (0.65 to 4.5 times the values for fully developed flow in smooth passages) indicates that it is prudent to have a data base available for the design of specific coolant passages used in rotating turbine blades.

This paper has presented an extensive set of experimental data from heat transfer experiments in a rotating square passage with trips normal to the flow direction. Following are observations regarding the effects of forced convection, Coriolis forces, buoyancy and flow direction on the heat transfer:

1. Changes in either the density ratio or the rotation number caused large changes in the heat transfer coefficients in passages with trips for flow radially outward or for flow radially inward.

2. The heat transfer ratio is a complex function of buoyancy parameter and density ratio on the low pressure surfaces of the coolant passages, regardless of flow direction.

3. The heat transfer ratio on the high pressure surfaces was significantly affected by flow direction. The heat transfer was a strong function of the buoyancy parameter for the high pressure surfaces in the first and third passages with flow radially outward. However, the heat transfer was relatively unaffected by the buoyancy parameter for flow radially inward.

4. Increasing the density ratio with high rotation numbers generally caused an increase in heat transfer. However, the increase in heat transfer for the inward flowing passage was generally greater than that for outward flow.

5. The maximum increase in heat transfer in passages with normal trips with increases in density ratio were greater than the maximum increases measured from the same model with smooth surfaces.

6. Heat transfer ratios from rotating passages with normal trips at the highest rotation numbers and buoyancy parameters were not significantly greater than the heat transfer ratios measured in the same model with smooth surfaces for the same parameters.

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16. Abstract

Experiments were conducted to determine the effects of buoyancy and Coriolis forces on heat transfer in turbine blade internal coolant passages. The experiments were conducted with a large scale, multi-pass, heat transfer model with both radially inward and outward flow. Trip strips on the leading and trailing surfaces of the radial coolant passages were used to produce the rough walls. An analysis of the governing flow equations showed that four parameters influence the heat transfer in rotating passages: coolant-to-wall temperature ratio, Rossby number, Reynolds number and radius-to-passage hydraulic diameter ratio. The first three of these four parameters were varied over ranges which are typical of advanced gas turbine engine operating conditions. Results were correlated and compared to previous results from stationary and rotating similar models with trip strips. The heat transfer coefficients on surfaces, where the heat transfer increased with rotation and buoyancy, varied by as much as a factor of four. Maximum values of the heat transfer coefficients with high rotation were only slightly above the highest levels obtained with the smooth wall model. The heat transfer coefficients on surfaces, where the heat transfer decreased with rotation, varied by as much as a factor of three due to rotation and buoyancy. It was concluded that both Coriolis and buoyancy effects must be considered in turbine blade cooling designs with trip strips and that the effects of rotation were markedly different depending upon the flow direction.

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