NASA Technical Memorandum 106126

Heat Transfer in Rotating Serpentine Passages With Selected Model Orientation for Smooth or Skewed Trip Walls

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Prepared for the
38th International Gas Turbine and Aeroengine Congress and Exposition
sponsored by the American Society of Mechanical Engineers
Cincinnati, Ohio, May 24–27, 1993

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ABSTRACT

Experiments were conducted to determine the effects of model orientation as well as buoyancy and Coriolis forces on heat transfer in turbine blade internal coolant passages. Turbine blades have internal coolant passage surfaces at the leading and trailing edges of the airfoil with surfaces at angles which are as large as +/-50 to 60 degrees to the axis of rotation. Most of the previously- presented, multiple-passage, rotating heat transfer experiments have focused on radial passages aligned with the axis of rotation. The present work compares results from serpentine passages with orientations 0 and 45 degrees to the axis of rotation which simulate the coolant passages for the midchord and trailing edge regions of the rotating airfoil. The experiments were conducted with rotation in both directions to simulate serpentine coolant passages with the rearward flow of coolant or with the forward flow of coolant. The experiments were conducted for passages with smooth surfaces and with 45 degree trips adjacent to airfoil surfaces for the radial portion of the serpentine passages. At a typical flow condition, the heat transfer on the leading surfaces for flow outward in the first passage with smooth walls was twice as much for the model at 45 degrees compared to the model at 0 degrees. However, the differences for the other passages and with trips were less. In addition, the effects of buoyancy and Coriolis forces on heat transfer in the rotating passage were decreased with the model at 45 degrees, compared to the results at 0 degrees. The heat transfer in the turn regions and immediately downstream of the turns in the second passage with flow inward and in the third passage with flow outward was also a function of model orientation with differences as large as 40 to 50 percent occurring between the model orientations with forward flow and rearward flow of coolant.

NOMENCLATURE

A	Area of passage cross-section
d	Hydraulic diameter
e	Trip height
h	Heat transfer coefficient
k	Thermal conductivity
m	Mass flowrate
Nu	Nusselt number, hd/k
Nu_∞	Nusselt number for fully developed flow in smooth tube with Pr = 0.72, $Nu_\infty = 0.0176\ Re^{0.8}$
R	Local radius
Re	Reynolds number $(md)/(\mu A)$
Ro	Rotation number, $\Omega d/V$
V	Mean coolant velocity
X	Streamwise distance from inlet
μ	Absolute viscosity
ρ	Coolant density
$\Delta\rho/\rho$	Density ratio $(\rho_b - \rho_w)/\rho_b$
Subscripts:	
b	Bulk property
f	Film property
i	Inlet to model
w	Heated surface location
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.

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 (Eckert et al., 1953) with stationary, single pass, co- and counter-flowing 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 as part of the Hot Section Technology (HOST) program. The objectives of this program were (1) to gain insight regarding 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 became 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, i.e., 2200–2500 K, (3500–4000F) inlet temperatures, maintain efficiencies in the 88–94% range, and require total coolant flows of only 4 to 6% of the engine air flow rate (Ref. IHPTET Brochure, Circa 1984). To attain these ambitious goals, a thorough understanding of the rotational effects of heat transfer and flow in turbine blade coolant passages is mandatory.

Previous Studies

Heat transfer in rotating radial internal coolant passages, typical of turbine airfoils of large gas turbine aircraft engines, has been investigated experimentally and analytically for the past ten to fifteen years. The experimental studies have been sponsored by national and private laboratories (e.g. USA/NASA, USSR, UK/RAE, France, Germany/DLR, Japan, Taiwan and USA/EPRI) and the large gas turbine manufacturers (e.g. PW, GE, and RR). The pioneering studies were reported by Morris (1981). More recent studies up to 1991, with a wider range of flow and geometric parameters, are reported in the authors' previous papers by Wagner et al. [1991] and Johnson et al. [1992], and in NASA contractors reports, Hajek et al. [1991], and Johnson et al. [1993]. Other recent references include Han & Lee [1992], El-Husayni et al. [1992] and Mochizuki et al. [1992] and contain most references from the studies sponsored by GE, EPRI and RAE. The results from these studies are bringing an understanding to the turbine blade durability designer of the many phenomena in rotating radial coolant passages including the flow parameters: Reynolds number, rotation wall-to-bulk density ratio, buoyancy parameter and the many geometric parameters including trip geometry, passage aspect ratio and inward or outward flow direction. One important aspect of flow and heat transfer which has not been explored is the effect of multiple coolant passage orientations with respect to the axis of rotation.

Objectives

Under the NASA HOST program, a comprehensive experimental project was formulated to identify and separate effects of Coriolis and buoyancy forces for the range of 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 density ratios expected in modern gas turbine engines.

The results presented in this paper were obtained during the first and third 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, et al. (1991a). The effects of flow direction and buoyancy with smooth walls were presented by Wagner, et al. (1991b). The second phase utilized a configuration with normal trips on the leading and trailing surfaces of the straight passages and were presented by Wagner, et al. (1991c). Results from the third phase had skewed surface roughness elements oriented at 45 degrees to the flow direction, Johnson et al. [1992]. Only a cursory discussion of the effects of rotating the plane of the serpentine passages was included in one previous paper.

The present work is focused on the effects of the orientation of the plane of model passages on heat transfer in rotating, near-radial coolant passages. The results in this paper will be related to previous results from the NASA HOST/UTC experiments and to design consideration for the internal cooling passages at the leading or trailing regions of rotating blades.

MODEL

Sketches of two multiple-pass coolant passage configurations for turbine blades (Han et al. [1986] and Johnson et al. [1992]) are shown in Fig. 1. In the Fig. 1a configuration, the coolant flows radially outward through the center passage (B) and radially inward through the third passage (C), discharging through a fourth partial passage (D) and an array of pedestals. In the Fig. 1b configuration the coolant in the multiple pass portion of the blade flows radially outward in passage E, forward toward the leading portion of the blade and radially inward in passage D and further forward and radially outward in passage C. In the Fig. 1b configuration, the coolant from the multipass array leaves the blade through the tip. The coolant could also be discharged from passage C through film cooling holes.

Experiments were conducted with the plane of the coolant passage centerlines through the axis of rotation ($\alpha = 0$) and with the plane at a 45 degree angle to the axis of rotation ($\alpha = 45$ degrees) as shown in Fig. 2b. The model was rotated forward ($+\Omega$) and backward ($-\Omega$) as shown in Fig. 2b. When the model is rotated forward with $\alpha = 45$ degrees, the model passages correspond to the blade coolant passages shown in Fig. 2a. Passages 6, 5 and 4 (Fig. 2a) form a three-legged serpentine coolant path in the blade. The coolant flows outward in Passage 6, inward in Passage 5 and outward in Passage 4. This set of passages corresponds to the forward flow of coolant in Passages E, D & C (Fig. 1b) and the first, second and third leg of the serpentine heat transfer model shown in Figs. 2b and 3. When the model is rotated backward ($-\Omega$) with $\alpha = 45$ degrees, the model passages correspond to the blade coolant passage shown in Fig. 2c. Passages 4, 5 and 6 (Fig. 2c) form a rearward flowing coolant path in the blade corresponding to Passages B, C and D of Fig. 1a. Although the flow and heat transfer in the developed portions of each coolant passage (Locations D, I and N of Fig. 3) are not expected to be affected by the change in the direction of model rotation, the flow and heat transfer in the turn regions (Locations E, F, J and K of Fig. 3) and the regions immediately downstream of the turns (Locations G and L of Fig. 3) will be affected to some degree.

The smooth wall and skewed trip heat transfer models employed in the study were those described by Wagner et al. [1991] and Johnson et al. [1992], respectively. The model consisted of 0.5 in. square (with 0.045 in. chamfers in the corners)

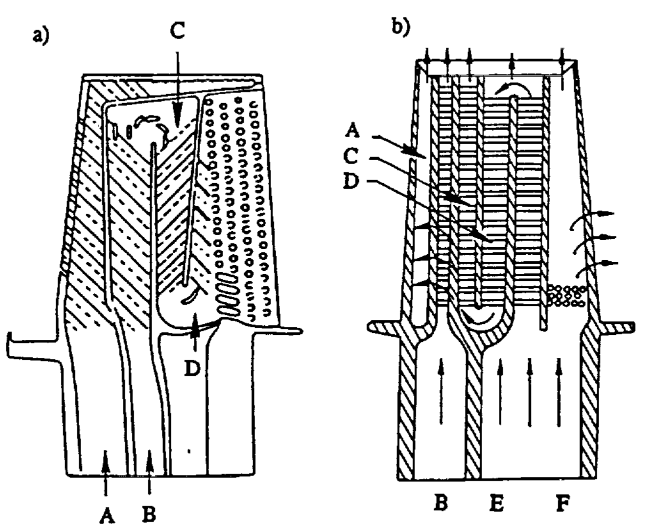

Figure 1. Typical Turbine Blade Internal Coolant Configurations: a) rearward flow of coolant, (Johnson et. al. [1992]), b) forward flow of coolant (Han et. al. [1986])

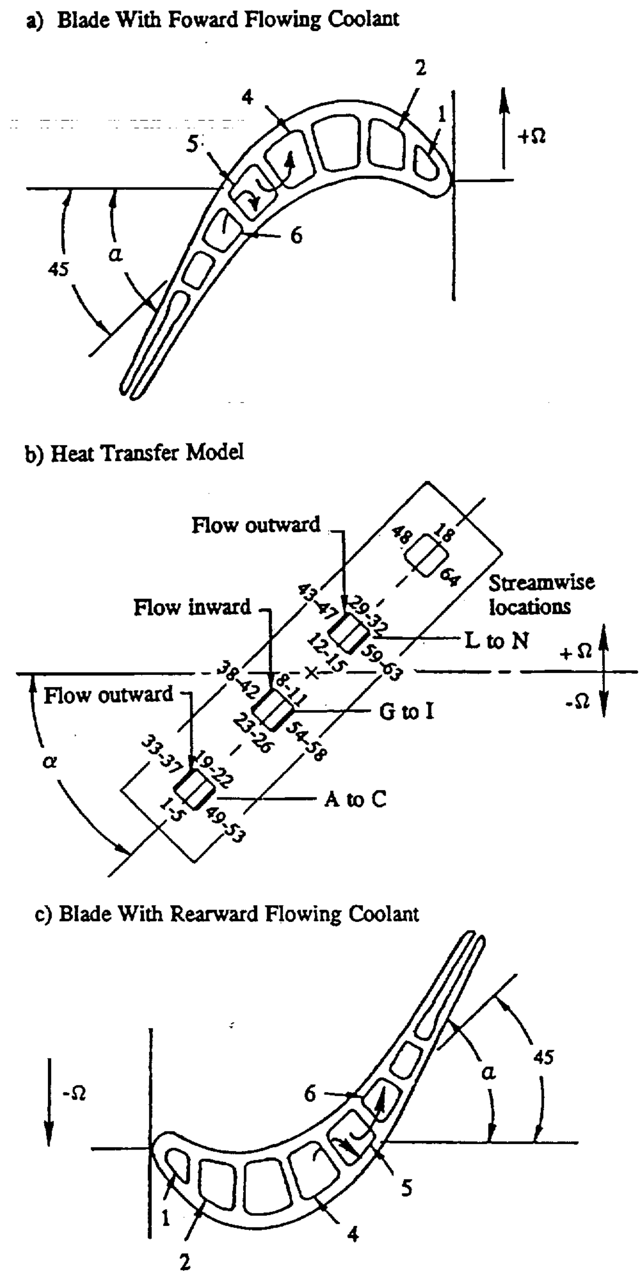

Figure 2. Sketches of Model and Blade Coolant Passage Cross-Sections Viewed From Root of Blades and Models. Coolant passages superimposed Upon UTRC LSRR Turbine Blade (Dring et. al. [1981]).

coolant passages with straight sections 6.0 inches long (e.g. combination of sections B, C and D) as shown in Fig. 3. Each test surface section (64 total for each model) was machined from a copper bar, was heated with an individually controlled and metered power supply, and had two thermocouples installed. The test surfaces were thermally isolated from each other with 0.064 in. rigid fiberglass strips. The test section streamwise identification, A through R (Fig. 3) and the wall and test section wall identification (Fig. 2b) will be used to identify the location of each heat transfer test section. Note that the heat transfer results from each copper test section segment are the average values over the identified test region.

The experimental procedures and uncertainties for the models with smooth walls and with skewed trips are the same as described by Wagner et al. [1991] and Johnson et al. [1992].

RESULTS

Baseline Flow Conditions

A set of parametric experiments were conducted with the models described in Wagner et al. [1991a], Wagner et al. [1991c] and Johnson et al. [1992]. The rotating baseline flow condition for all three models included a Reynolds number of 25,000, a rotation number of 0.24, a density ratio of 0.13, a geometric ratio ($\bar{R}/d$) of 52 and a model orientation, α, of 0 degrees. In our previous experiments, the use of a heat transfer ratio, Nu/Nu_∞, showed excellent correlation of the Reynolds number effects for Re = 25,000 and higher. Consequently, most of the previous parametric studies were conducted with Re = 25,000 and focused

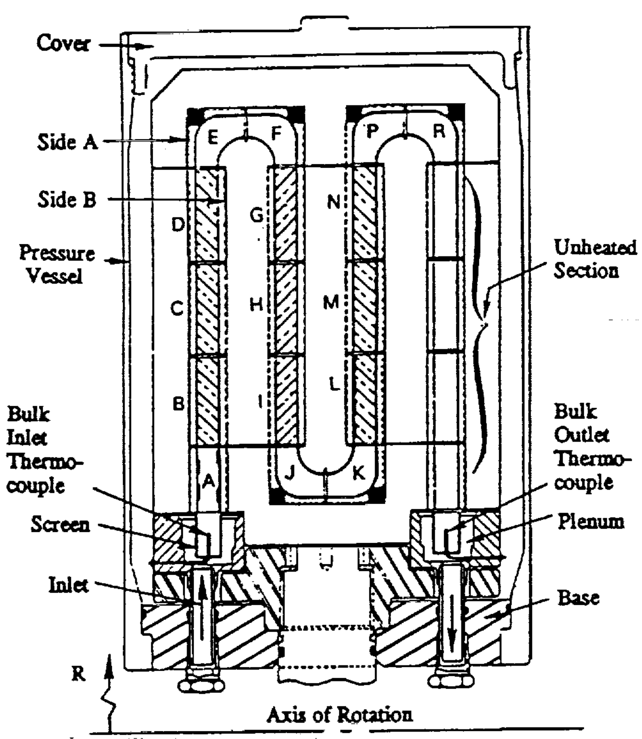

Figure 3. Cross Sectional View of Coolant Passage Heat Transfer Model Assembly With Skewed Trip Rough Walls: view through center of model toward leading surfaces with $\Omega > 0$, dotted ribs show locations on the trailing surfaces.

on the effects of the remaining flow parameters, i.e. Ro, $(\Delta\rho/\rho_b)_i$ and flow direction. The test conditions for the Baseline Flow Condition were : Ω = 550 rpm; P_{in} = 148.5 psia; m = 0.013 lb/sec; T_{in} = 80 F; T_{wall} = 160 F.

For the present study of the effects of model orientation, the Reynolds number was fixed at 25,000. The study was conducted with the smooth wall and skewed trip models at two orientations, α = 0 and 45 deg orientation. The model was rotated in the forward (+Ω) and backward (-Ω) direction with the smooth wall model and in the forward direction with the skewed trip model. The radius ratio had a constant value, $\bar{R}/d$ = 49 with d = 0.52 in. (due to chamfered corners), for this study. The rotation number and the inlet density ratio were varied and the effects of flow direction were observed.

Comparisons for α = 0 and 45 Degrees

A comparison of the heat transfer ratios for the smooth and skewed trip models and α = 0 and +45 degrees (Fig. 4) shows the differences for the two model orientations. For the smooth wall model, the largest differences occur in first leg where the flow patterns are governed by Coriolis and buoyancy forces and less by the secondary flow from the turn regions. Note that for the leading segment defined for α = 0 as "adjacent to the turbine blade airfoil suction surface", the minimum heat transfer ratio increased from 0.42 to 0.9 a factor of two. When the α = 0 side wall (Side B; Htr 19-32) is rotated and becomes a co-leading surface (Fig. 4b), the heat transfer ratio decreases from 1.5 to 0.8, a factor of almost two. For the smooth wall model, the effects of model orientation are less severe in the second and third legs after the turn regions.

The absolute changes in the heat transfer ratio, i.e. $\Delta Nu/Nu_\infty$, for the skewed trip model are as large as for the smooth model. However, the percentage change is considerably less due to the higher values of heat transfer for α = 0. The heat transfer ratio on the (α = 0) trailing surfaces of the first leg decreases from 4 to 3.2, approximately 20 percent. The same percentage decrease occurs in the third leg with flow also outward. The effect of model orientation has little affect in the second leg where the flow is inward. The absence of effects due to model orientation in the second leg is compatible with the previous studies for α = 0 where the heat transfer in the second leg was also relatively insensitive to rotation and buoyancy effects.

Effects of Model Orientation and Rotation Direction for Smooth Wall Model

Experiments were conducted with the smooth wall model at α = 0 and 45 degrees to determine the model symmetry and the effects of the serpentine model orientation (Fig. 2). As expected, the differences in the heat transfer (Fig. 5) in the first leg for a constant α between Ω = +550 and -550 rpm are small, of order 10 percent, because the flow has not developed asymmetries due to the turn regions and the model is essentially symmetric. The differences, both for α = 0, Ω = +/-550 rpm and for α = 45, Ω = +/-550 rpm, grow to 20 to 30 percent on the trailing side of the second leg and to 40 to 50 percent on both the leading and trailing side of the third legs. These differences are attributed to the differences in the secondary flow interactions through the first turn (outer turn) at 12 < x/D < 19 and the second (inside turn) at 31 < x/D < 36.

Effects of Rotation Number

The effects of rotation on heat transfer from the surfaces which would be adjacent to the leading and trailing surfaces of the airfoil for α = 45 are presented in Fig. 6. For the smooth wall model (Figs. 6a & b), the largest effects occur on the high pressure side of the coolant passage, i.e. trailing side for flow outward in the first and third legs and leading surface with flow inward in the second passage. For the skewed trip model (Figs. 6c & d), the largest effects of rotation occur on the leading side in the first passage. Note that for the highest value of the rotation number, Ro = 0.35, the heat transfer ratio is increased in all three legs on the trailing surface. These larger effects could be due to the increased influence of the buoyancy parameter which tended to dominate at higher values of rotation for the skewed wall model compared to the smooth wall model, e.g. Johnson et al. (1992).

The results from the first leg of the smooth model at three rotation numbers (Figs. 7a & c) show symmetry as expected. Note that the heat transfer on the α = 0 degrees trailing surface or the α = 45 degrees trailing or trailing side wall surfaces are approximately the same and symmetric on either side of the α = 0 results. Additionally, the average leading and leading-side-wall surfaces for α = 45 degrees have average heat transfer coefficients approximately the same as the Ro = 0 value.

The results from the first leg of the skewed trip model (Figs. 7b & d) are connected as shown because smooth (α = 0 sidewalls) and skewed trip walls (α = 0 leading and trailing walls) are adjacent to each other. The heat transfer on the leading surface (α = 0) is decreased by a factor of two due to rotation (Johnson et al. 1992) for both the α = 0 and 45 orientation. The heat transfer from the leading surface does not increase appreciably for the α = 45 orientation as did the smooth wall model. At this time it is not known if an alternate skewed trip strip orientation (i.e. skewed the other direction) would alter this result.

Symbol	α	Trips
●	0	Smooth
○	45	Smooth
■	0	Skewed
□	45	Skewed

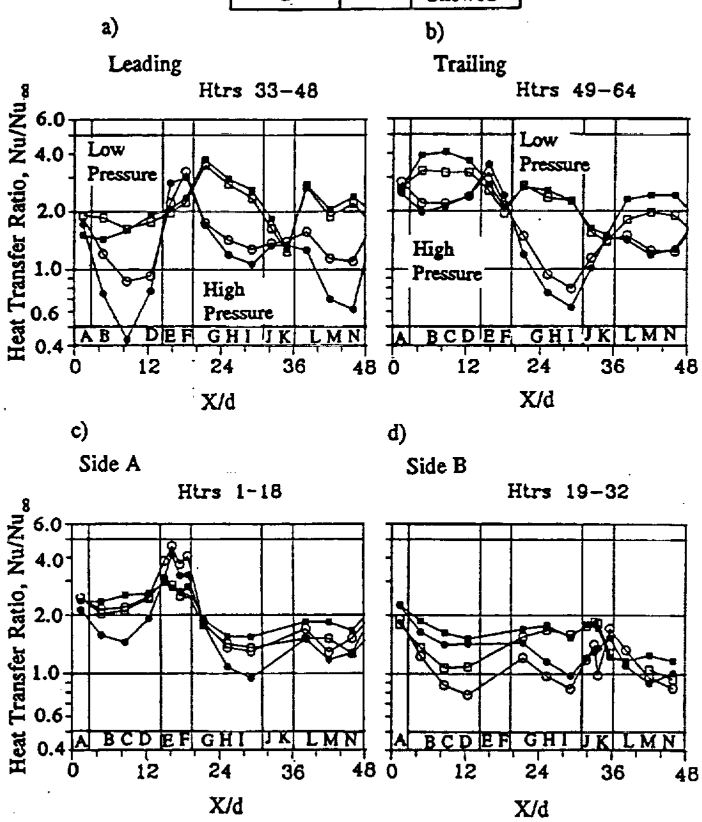

Figure 4. Comparison of Heat Transfer Results for Smooth and Skewed Trip Walls at $\alpha = 0$ and 45 deg for Baseline Flow Conditions; Re=25000, Ro=0.24, $(\Delta\rho/\rho)_i$=0.13, R/d=49.

Symbol	●	■	○	□
α	0	0	45	45
Ω	+550	-550	+550	-550
Ro	0.24	-0.23	0.24	-0.23

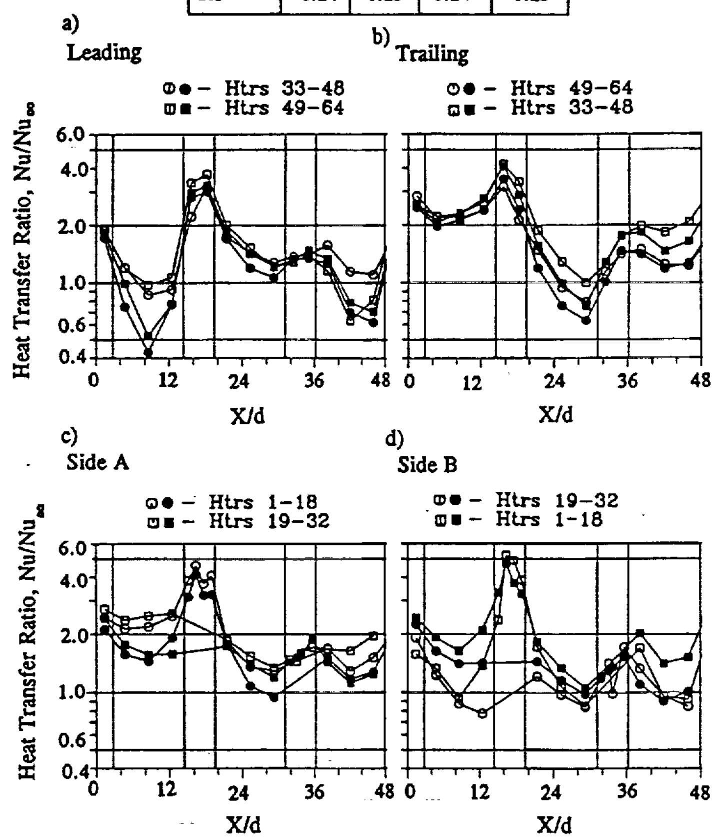

Figure 5. Effects of Orientation and Rotation Direction on Heat Transfer Ratio for Smooth Wall Model; Re=25000, $\Delta T=80^{o}F$, $(\Delta\rho/\rho)_i$=0.13.

Effects of Rotation and Inlet Density Ratio

The heat transfer ratio from the nominal leading and trailing downstream heat transfer surfaces are shown in Fig. 8 as a function of local rotation number with the inlet density ratio for each symbol noted. With the smooth wall model, the heat transfer ratio shows less effect of inlet density ratio for $\alpha = +45$ degrees than for $\alpha = 0$ on the low pressure surfaces (leading surface for flow out; trailing surface for flow in; Figs. 8a, c, e). As previously discussed for the smooth wall model, the most noticeable effects in terms of percentage change due to changes in rotation numbers for $\alpha = 0$, occur on the low pressure surfaces.

In previous papers from this series of NASA/HOST/UTC experiments, the results were also as correlated as a function of a buoyancy parameter. In the present study, the effects of buoyancy are less noticeable and the presentation does not appear warranted.

CONCLUDING COMMENTS

Experimental results for smooth wall and skewed trip models with the plane through the center of the serpentine coolant passages orientated at 45 degrees to the axis of rotation were related to previous results (Wagner et al. 1991a & b; Johnson et al. 1992). These results are directly applicable to airfoil coolant passage geometries where the coolant passages walls adjacent to the airfoil surface are not parallel to the axis of rotation (Figs. 1 and 2).

The following are the principal results and conclusions from this study:

- The largest fractional change in the heat transfer ratio due to model orientation occurred on the low pressure side of the smooth wall model where the average heat transfer coefficient for each section was less sensitive to rotation at $\alpha = 45$ degrees.
- The average heat transfer ratios for the developed–flow sections with skewed trips at $\alpha = +45$ degrees were within 15 percent of those for the $\alpha = 0$ orientation.

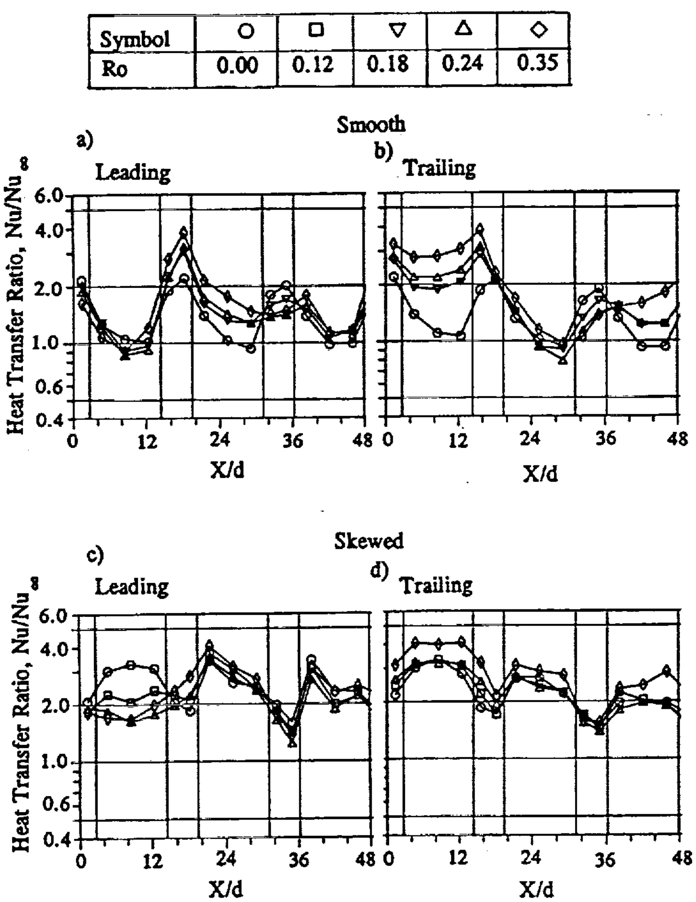

Figure 6. Effects of Rotation Number on Heat Transfer of Nominal Leading and Trailing Surfaces for Smooth and Skewed Trip Walls for Alpha Equal 45 Degrees; Re=25000, $\Delta T=80^oF$, $(\Delta\rho/\rho)_i=0.13$.

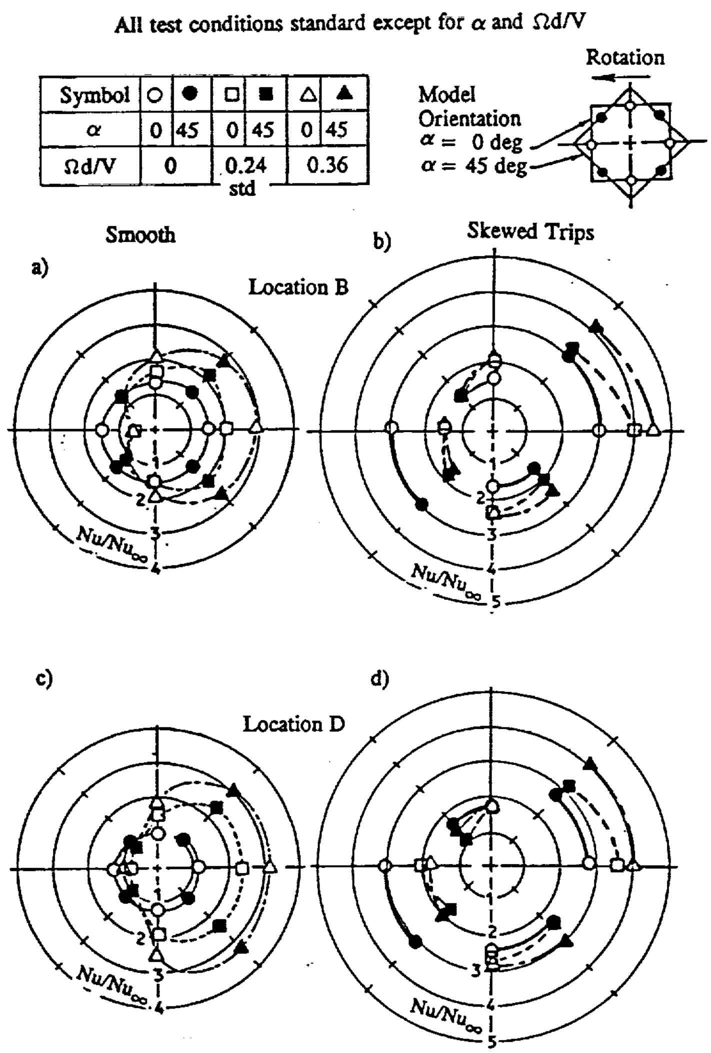

Figure 7. Comparison of Heat Transfer Results From First Leg of Smooth and Rough Wall Models; $\Delta T=80^oF$, $(\Delta\rho/\rho)_i=0.13$, Re=25,000.

- Variations of 20 to 50 percent in the heat transfer ratio were noted due to α = 45 or 0 or to $+\Omega$ or $-\Omega$ orientations downstream of the turns before the flow became developed
- The effect of model orientation has little effect in the second leg where flow is inward.

ACKNOWLEDGEMENTS

The work published in this paper was supported by the NASA/Lewis Research Center under the HOST Program, Contract No. NAS3-23691 to the Pratt and Whitney Commercial Engine Business/Engineering Division and by the United Technology Corporation's independent research program. The authors gratefully acknowledge the assistance of Ms. S. Orr (UTRC) in the performance of this program. The authors are appreciative of the support and guidance by the HOST management team at NASA/Lewis Research Center and by their colleagues at P&W and UTRC.

Symbol Flag	○ (flag up)	○ (flag right)	○ (flag down)	○ (flag left)
$(\Delta\rho/\rho)_i$	0.07	0.13	0.18	0.22

Symbol	○	□	◇	▲	▼	★
α	0	0	0	45	45	45
Ω	Positive	Positive	Negative	Positive	Positive	Negative
Trips	Smooth	Skewed	Smooth	Smooth	Skewed	Smooth

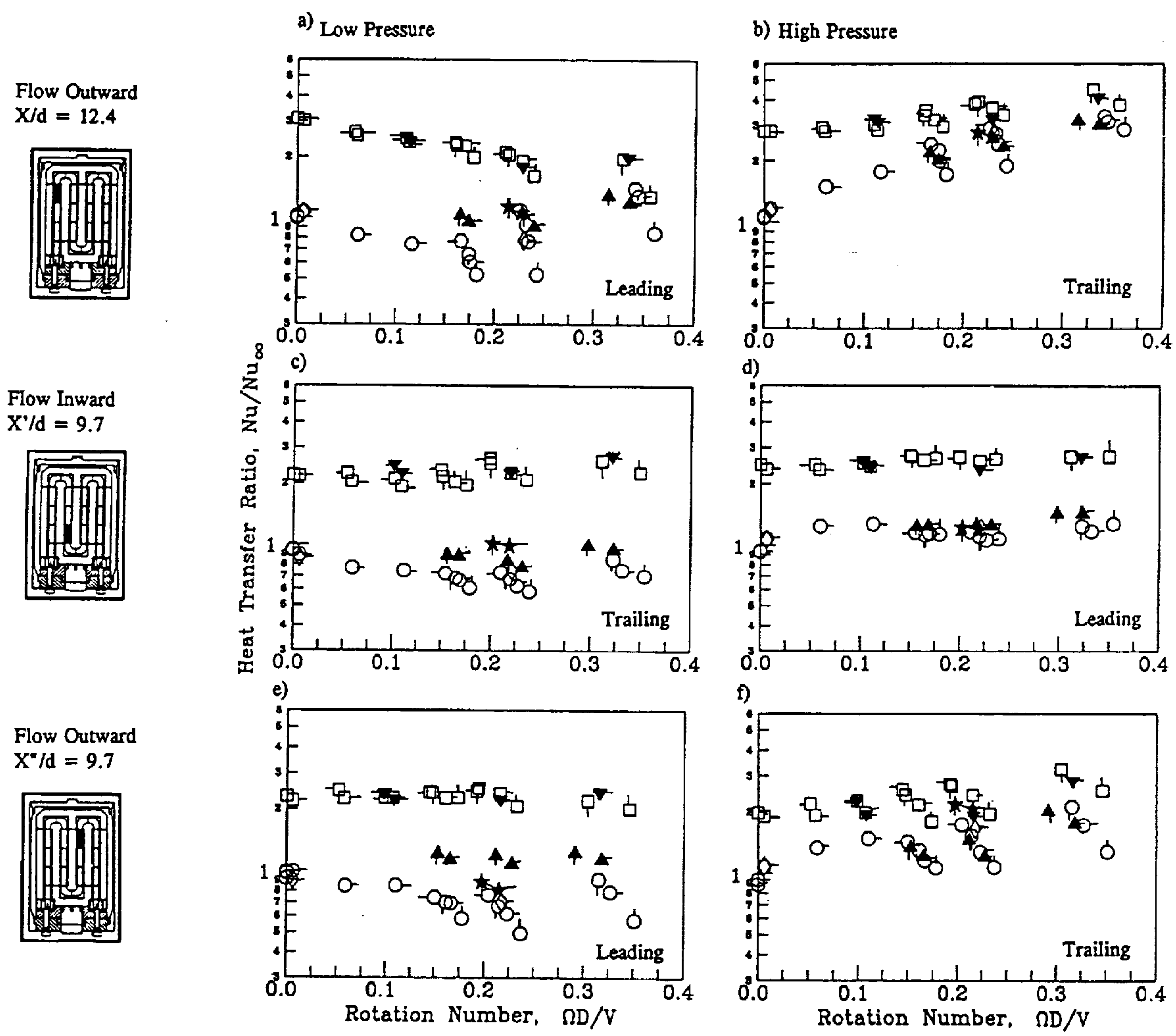

Figure 8. Effects of Rotation Number and Inlet Density Ratio on Heat Transfer Ratio; Re=25000, R/d=49.

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REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)

2. REPORT DATE
April 1993

3. REPORT TYPE AND DATES COVERED
Technical Memorandum

4. TITLE AND SUBTITLE
Heat Transfer in Rotating Serpentine Passages With Selected Model Orientation for Smooth or Skewed Trip Walls

5. FUNDING NUMBERS
WU–505–62–52

6. AUTHOR(S)
B.V. Johnson, J.H. Wagner, G.D. Steuber, and F.C. Yeh

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
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Cleveland, Ohio 44135–3191

8. PERFORMING ORGANIZATION REPORT NUMBER
E–7793

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
Washington, D.C. 20546–0001

10. SPONSORING/MONITORING AGENCY REPORT NUMBER
NASA TM–106126

11. SUPPLEMENTARY NOTES
Prepared for the 38th International Gas Turbine and Aeorengine Congress and Exposition, Cincinnati, Ohio, May 24–27, 1993. B.V. Johnson and J.H. Wagner, United Technologies Research Center, 411 Silver Lane, East Hartford, Connecticut 06108; G.D. Steuber, Pratt & Whitney, 400 Main Street, East Hartford, Connecticut 06108; and F.C. Yeh, NASA Lewis Research Center, Cleveland, Ohio. Responsible person, F.C. Yeh, (216) 433–5872.

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Unclassified - Unlimited
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12b. DISTRIBUTION CODE

13. ABSTRACT ***(Maximum 200 words)***

Experiments were conducted to determine the effects of model orientation as well as buoyancy and Coriolis forces on heat transfer in turbine blade internal coolant passages. Turbine blades have internal coolant passage surfaces at the leading and trailing edges of the airfoil with surfaces at angles which are as large as +/−50 to 60 degrees to the axis of rotation. Most of the previously-presented, multiple-passage, rotating heat transfer experiments have focused on radial passages aligned with the axis of rotation. The present work compares results from serpentine passages with orientations 0 and 45 degrees to the axis of rotation which simulate the coolant passages for the midchord and trailing edge regions of the rotating airfoil. The experiments were conducted with rotation in both directions to simulate serpentine coolant passages with the rearward flow of coolant or with the forward flow of coolant. The experiments were conducted for passages with smooth surfaces and with 45 degree trips adjacent to airfoil surfaces for the radial portion of the serpentine passages. At a typical flow condition, the heat transfer on the leading surfaces for flow outward in the first passage with smooth walls was twice as much for the model at 45 degrees compared to the model at 0 degrees. However, the differences for the other passages and with trips were less. In addition, the effects of buoyancy and Coriolis forces on heat transfer in the rotating passage were decreased with the model at 45 degrees, compared to the results at 0 degrees. The heat transfer in the turn regions and immediately downstream of the turns in the second passage with flow inward and in the third passage with flow outward was also a function of model orientation with differences as large as 40 to 50 percent occurring between the model orientations with forward flow and rearward flow of coolant.

14. SUBJECT TERMS
Heat transfer; Rotating blade passages; Turbulators

15. NUMBER OF PAGES
10

16. PRICE CODE
A02

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. Z39-18
298-102