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008326

NASA Contractor Report 198542
ICOMP-96-10; 95-GT-142

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October 1996

Prepared for
Lewis Research Center
Under Cooperative Agreement NCC3-370

National Aeronautics and
Space Administration

INSTITUTE FOR COMPUTATIONAL MECHANICS IN PROPULSION
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ABSTRACT

The rate of heat transfer on the tip of a turbine rotor blade and on the blade surface in the vicinity of the tip, was successfully predicted. The computations were performed with a multiblock computer code which solves the Reynolds Averaged Navier-Stokes equations using an efficient multigrid method. The case considered for the present calculations was the SSME (Space Shuttle Main Engine) high pressure fuel side turbine. The predictions of the blade tip heat transfer agreed reasonably well with the experimental measurements using the present level of grid refinement. On the tip surface, regions with high rate of heat transfer was found to exist close to the pressure side and suction side edges. Enhancement of the heat transfer was also observed on the blade surface near the tip. Further comparison of the predictions was performed with results obtained from correlations based on fully developed channel flow.

NOMENCLATURE

A	Flow area
b	Hydraulic diameter=2C
C	Gap spacing
C_p	Constant pressure specific heat
k	Thermal conductivity
$\dot{m}$	Mass rate of flow
Nu	Nusselt number
Pr	Prandtl number
r	Local Radius
Re	Reynolds number
S	Distance along the blade surface measured from the stagnation line
St	Stanton number
T	Temperature
y+	Dimensionless distance from the wall
Z	$r\text{-}r_{hub}/r_{tip}\text{-}r_{hub}$

Subscripts

0	Fully developed value
1	Inlet condition
n	Coordinate normal to the wall
t	Total value at the stage inlet
w	Wall condition

INTRODUCTION

Modern gas turbine engines rely on elevated turbine inlet temperatures to attain high cycle efficiencies. The objective of turbine heat transfer research is to provide tools and information which enable designers to use high inlet temperatures to increase performance and concurrently minimize component temperature levels and temperature gradients to extend operating life (Metzger et al., 1993).

The tip of turbine rotor blades experience very large thermal loads, comparable to those at the stagnation line. In recent years many research activities have been aimed at acquiring better understanding of the tip clearance flow physics as well as development of tools to simulate the flow patterns and compute the heat transfer.

On the numerical simulation side, much work has been done in the area of simulation of the flow in the tip clearance region. The approach taken has been mostly to solve the three-dimensional flow field around the entire blade, including the tip gap region. The heat transfer predictions however, have used simpler schemes such as empirical correlations or at most two-dimensional simulations on simplified geometries and boundary conditions.

The flow in the gap between the blade tip and the shroud, i.e., the tip clearance, is mainly driven by the pressure differential between the pressure and suction surfaces of the blade. This flow has a detrimental effect on the turbine stage efficiency and is responsible for the high rate of heat transfer to the blade tip and the blade suction surface in the vicinity of the tip. To date several attempts have been made to obtain better understanding of the physics of flow in the tip gap region as well as to assess the dominant mechanisms of heat transfer in order to minimize the negative impact of this flow. These attempts have been analytical and numerical as well as experimental.

Mayle and Metzger (1982) demonstrated that the tip clearance flow is composed mainly of mainstream blade passage fluid. They also concluded that the flow in the tip gap region, viewed from a coordinate system fixed to the blade, is analogous to a duct entrance flow with the opposite wall moving. They hypothesized that, since the major resistance to turbulent heat transfer is within the viscous sublayer and the buffer layer adjacent to the wall, for a small enough dimensionless gap length the rate of heat transfer on the tip surface can be directly taken from experiments without a moving wall. They showed this to be true experimentally for a range of dimensionless rotation parameter as well as a range of dimensionless gap length and Reynolds numbers. This latter conclusion has been the basis of many experimental measurements of the blade tip heat transfer.

Metzger et al. (1989) used the conclusions from the aforementioned study to investigate the heat transfer on a grooved turbine blade tip model. They studied the tip heat transfer using a stationary shroud and showed that for a flat tip case, the rate of heat transfer displays a maximum value a short distance downstream of the entrance contraction, as is the case for the abrupt pipe entrance flow, and that this maximum was as much as approximately 100% above the fully developed value.

Moore et al. (1989) made two-dimensional velocity calculations for a simplified incompressible turbulent tip clearance flow for two cases. One of these cases had a pressure side/ blade tip corner at a 70 degree chamfer and the other case had a 90 degree corner. They showed that the effect of chamfering the corner was to reduce the size of the separation bubble formed on the tip surface. They solved the energy equation and showed good agreement with experimental results of Metzger and Bunker (1985), who measured a larger peak augmentation of heat transfer coefficient for the right angle corner.

Another notable work in the area of the prediction of tip heat transfer was performed by Metzger, Dunn and Hah (1991). In that work the authors used the fully developed duct correlation developed by Boelter et al. (1945). This correlation requires the calculation of the channel Reynolds number based on the averaged flow velocity. To obtain the average velocity of the flow through the gap, they used the results of a three-dimensional Navier-Stokes calculation to obtain the leakage flow rate. They assumed an augmentation factor of 2.3, based on the data of Metzger et al. (1989), to estimate a heat transfer coefficient representing the case they considered. A comparison with the experimental data taken on the tip from the leading edge to about 30 percent chord showed good agreement.

The rate of heat transfer on the blade surface near the tip is greatly modified due to the tip clearance flow. Near the tip, on the suction side of the blade, the blade thermal loading is increased due to the roll up of clearance flow and formation of a vortex (Rued and Metzger, 1989). On the pressure side the increase in thermal loading is due to the sink like behavior of the leakage flow (Metzger and Rued, 1989).

In this paper we calculate the rate of heat transfer to the blade surface and the tip surface by employing a Navier-Stokes solver. The entire blade surface is included in the computations and the effect of the tip clearance flow on the blade surface heat transfer is isolated and discussed. The computations are performed using the code TRAF3DMB (TRAnsonic Flow Multi-Block of Steinthorsson et al., 1993), which is a multiblock version of TRAF3D previously shown to be successful in predicting the blade surface and endwall heat transfer (Ameri and Arnone, 1994a,1994b). This modified computer code solves the Reynolds Averaged Navier-Stokes equations using an efficient multigrid method. The case considered for the present calculations is the SSME (Space Shuttle Main Engine) high pressure fuel side turbine. Blade surface and tip heat transfer data were presented by Dunn et al., (1994) and Dunn and Kim (1992). In addition to the measured data, the computed tip heat transfer results are compared with predictions obtained from correlations developed for fully developed channel flow.

COMPUTATIONAL METHOD

The simulations performed in this study were done using a recently developed computer code called TRAF3D.MB (Steinthorsson et al. 1993). This code is a general purpose flow solver, designed for simulations of flows in complicated geometries. The code is based on the TRAF3D, an efficient computer code designed for simulations of flows in turbine cascades (Arnone et al. 1991). The TRAF3D.MB code employs the full compressible Navier-Stokes equations. To handle complex geometries, the code uses multiblock grid systems (i.e., non-overlapping "zoned" grids), but has the added capability of handling grids with non-contiguous grid lines across branch cuts. The TRAF3D.MB code was described in detail by Steinthorsson et al. (1993). Some aspects of the formulation used in the code are the same as those described by Arnone et al. (1991). Only major aspects of the code will be described below.

Discretization of the Governing Equations:

The governing equations used in TRAF3D.MB are the compressible Navier-Stokes equations. The equations are transformed to a rotating reference frame using the formulation introduced by Chima and Yokota (1990). The governing equations are discretized using a second order accurate (in space), cell-centered finite volume discretization. In the present calculations, central differencing was used for the convective terms, with mixed fourth- and second-order artificial dissipation to damp out oscillations due to odd-even decoupling. The artificial dissipation employed is essentially the same as described in Arnone et al.(1991), i.e., eigenvalue scaling based on the work of Martinelli and Jameson (1988) and Swanson and Turkel (1987) is used to minimize the amount of artificial dissipation inside shear layers. The viscous terms of the Navier-Stokes equations are discretized in a finite volume fashion using central differencing.

Time-Stepping Scheme and Convergence Acceleration:

Solutions to the discretized governing equations are obtained by marching to steady state with a four-stage Runge-Kutta time integration scheme. To reduce the cost of the computations, viscous fluxes are evaluated only in the first Runge-Kutta stage, and artificial dissipation terms are evaluated only in the first and second stage. Convergence to steady state is accelerated by using (a) local time stepping, (b) implicit residual smoothing (Martinelli and Jameson, 1988) and (c) multigrid scheme (Jameson (1983), Brandt (1977), Hackbusch and Trottenburg (1982) and Briggs (1987)). The local time step sizes are computed considering both inviscid and viscous contributions (Arnone et al. (1991). The implicit residual smoothing is applied independently within each block of the multiblock grids and is used on every level of the multigrid scheme. The multigrid scheme is implemented using full approximation storage (FAS). The full multigrid procedure (FMG) is also implemented to generate good initial conditions for the finest grid. The convergence acceleration techniques employed in TRAF3D.MB are described in some detail by Steinthorsson et al.(1993).

Multiblock Scheme:

The multiblock capability implemented in TRAF3D.MB is completely general in the sense that there is no limitation on the number of blocks or how they can be connected. For storage, a single one-dimensional array is used for all the individual blocks, with pointers to the first element in each block. The code automatically orders the blocks into memory. If enough memory is available, all blocks are stored in memory at all times. Otherwise, the code separates the blocks into groups or clusters that fit into memory, one at a time. Block clusters that are not in memory are saved in a file (Solid State Device or "scratch").

In TRAF3D.MB, ghost cells are used to accomplish the necessary communication among blocks in the multiblock grid system. Two layers of ghost cells are used around each block in the multiblock grid system, effectively creating an overlap of four cells between neighboring blocks. Transfer of data among blocks involves loading into the appropriate ghost cells of each block the data from neighboring blocks that it needs access to. The code automatically decides from connectivity information and grid point coordinates whether, upon transfer, rotation of vector quantities should be done (e.g., as is needed in azimuthally-periodic geometries).

In the current implementation, blocks communicate on each level in the multigrid scheme but only before the first stage in the Runge-Kutta scheme. This reduces overhead due to communication among blocks without significantly impacting the rate of convergence to steady state.

Boundary Conditions

The types of boundary conditions encountered in solving the problem at hand are as follows:

1) Inlet: The inlet boundary condition for axially subsonic flows is treated by specifying the total inlet temperature and total inlet pressure as well as the inlet angle profiles. The outgoing Riemann invariant is extrapolated to the inlet from within. For a supersonic inlet all the conditions are specified. The total temperature and pressure profiles are determined to match the law of the wall for the specified hydrodynamic and thermal boundary layer thicknesses on the hub and/or shroud.

2) Exit: At the exit boundary, for the subsonic axial flow, the pressure is specified and all the other conditions are extrapolated from within. The pressure at the exit plane is computed by integration of the radial equilibrium equation with specified hub endwall pressure at the exit. In case of supersonic flow, all variables are extrapolated.

3) Walls: At the walls, the normal pressure gradient is set to zero, the temperature or the gradient of the temperature is specified, and the no-slip condition is enforced. The density and total energy are computed from the pressure and the temperature.

4) Periodic boundaries are computed as interior points.

5) Non-matching grid: This type of boundary condition, which occurs on the wake line and on the inner boundary of the O-grid in the tip region, is set by linear interpolation of the primary variables of the phantom cells.

Turbulence and Transition Models

In the present study, the Baldwin and Lomax (1978) algebraic turbulence model is employed. This model is patterned after the Cebeci model modified to obviate the need for finding the edge of the boundary layer. It is a two-layer algebraic eddy viscosity model. The inner part which is very important in the calculation of heat transfer uses the Prandtl-Van Driest formulation. The thermal diffusivity is calculated by assuming constant turbulent Prandtl number.

The transition, as affected by the normal and unsteady wake passing processes on the blade is modeled using the method described by Mayle (1991). The application of the turbulence and transition model to the three-dimensional turbine blade heat transfer calculations is described in Ameri and Arnone (1994a). In the tip region, the turbulence model is implemented separately for the tip surface and the shroud surface assuming fully turbulent flow.

COMPUTATIONAL GRID

The calculations are performed using a grid system with three blocks. Two of the blocks, shown in Figure 1, are C grids that wrap around the turbine blade. The third block shown in Figure 2, is of the O type and covers the tip clearance. The C-grid is of the so-called 'non-periodic' type (Arnone et al. 1992). This means that the number of transverse C-grid lines on the pressure side of the wake line do not correspond with the number of grid lines on the suction side. This allows better control over the grid point distribution and grid skewness. The C-grid is generated by using an elliptic scheme to construct an "inviscid" grid and the viscous grid is made by embedding grid lines in the vicinity of the no-slip boundaries with the desired wall spacing. The hub to tip grid is constructed by radial stacking of the C grid. This is done using geometric stretching. The first grid lines parallel to the hub or the shroud endwall are constructed to lie within the viscous sublayer.

The O-grid used for the tip clearance is generated algebraically. The transverse grid lines span the distance between the mean camber line and the corresponding grid node on the C-grid. Figure 3 shows a picture of the grid located on the blade tip. The grid is refined close to the edge of the blade to allow for the large variations in the flow in that region. The O-grid is also stacked in the spanwise direction using geometric stretching.

RESULTS AND DISCUSSIONS

The case chosen for this study is the first stage rotor of the two stage Rocketdyne Space Shuttle Main Engine (SSME) high pressure fuel turbine. For this rotor, the tip clearance is 2 percent of the span.

Dunn and Kim (1992) have performed extensive vane and blade surface pressure and heat transfer measurements on this turbine. They also measured the rate of heat transfer on the tip along the mean camber

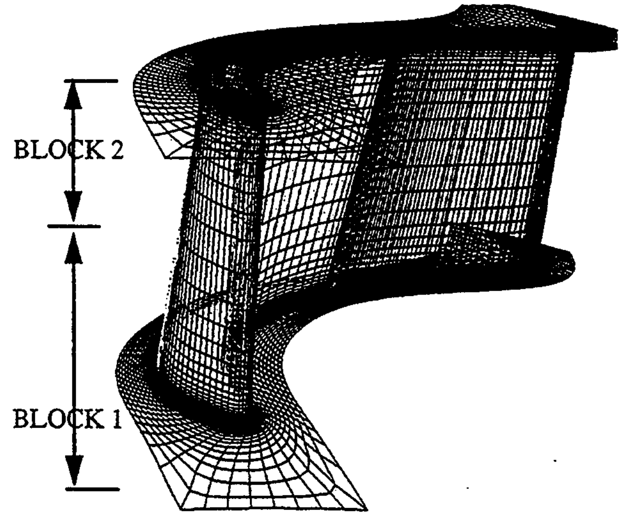

Fig. 1 The computational grid.

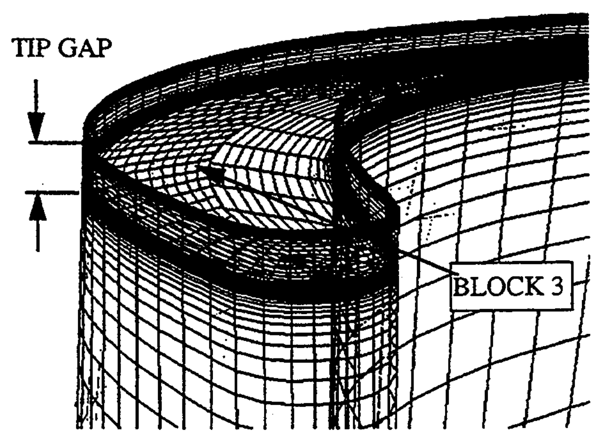

Fig. 2 Close up view of the grid in the vicinity of the blade tip.

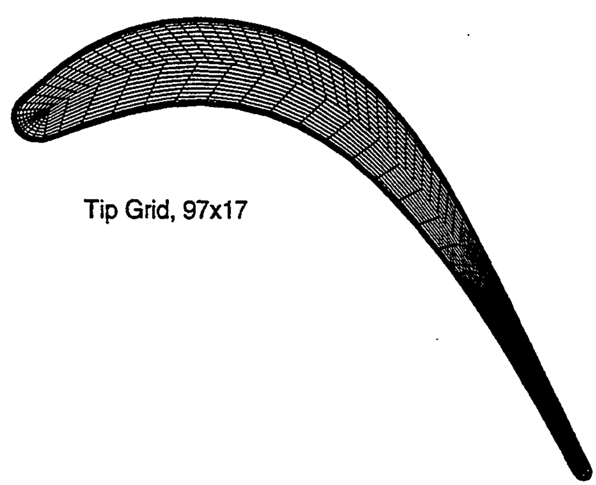

Fig. 3 O-grid, placed in the tip clearance.

Exit Reynolds Number	$2.0x10^5$
Rotation rate (RPM)	9600.0
Relative Inlet Angle(nominal) at Mid-Span	42.5°

TABLE 1. Run Conditions

line of the first stage blade. This measured data will be used to gauge the accuracy of the current predictions.

The flow conditions used in the experiment and current predictions are as given in Table 1. Additional information required, such as the inlet angle profile to the blade and the static to absolute total pressure ratio across the blade row, was obtained by the analysis of the turbine using the MTSB code (Katsanis and McNally, 1977).

Simulations of the flow and heat transfer on the blade and tip gap surfaces were performed using three multi-block grid schemes of varying grid densities to be described later.

The three grid schemes will be described below. Subsequently, results of the simulations of the tip clearance flow will be discussed briefly. Lastly, the heat transfer predictions for the tip and the blade surface will be presented and compared with experimental data of Dunn and Kim (1992) as well as with relevant empirical correlations.

<u>Grid System</u>

Multi-block simulations of the flow and heat transfer on the blade and tip gap surfaces were performed using three grid schemes of varying grid densities. The grid schemes are designed to allow investigation of the required resolution for attaining accurate prediction of the rate of heat transfer on the blade tip as well as on the blade surface. The three grid schemes used in the multi-block calculations are referred to as grids A, B and C. The dimensions of the hub to tip and tip to shroud C-grids and the tip clearance O-grid are given in Tables 2 and 3. Note that the regions specified in Table 2 do not correspond to the blocks. The spanwise grid stretching ratios are also given. In table 4 the gridding schemes are classified as fine or coarse in the two regions.

All the grids are constructed such that the distance of the cell centers adjacent to solid walls, in wall units (y^+) is close to unity. Grid B violates this rule by not resolving the hub endwall boundary layer. For that grid, the value of y^+~5.

<u>Tip Flow and Secondary Flows</u>

Figures 4 and 5 illustrate the flow patterns on the mid gap height of the tip clearance and the trace of the flow close to the tip surface. The flow as shown is relative to the blade tip. Figure 4 shows the direction of the flow in the mid-gap height of the tip clearance. In Figure 5, the tip leakage vortex on the suction side and tip separation vortex on the blade tip can be identified. The presence of these vortices has a large effect on the rate of heat transfer to the blade tip and to the blade surface. This effect will be described subsequently.

	Grid A	Grid B	Grid C
Circumferential	193	193	193
Transverse	49	49	49
Spanwise, hub to Tip Stretching Ratio	65 1.27	33 1.3	65 1.27
Spanwise, tip to shroud Stretching Ratio	25 1.5	33 1.3	37 1.24

TABLE 2. C-grid dimensions.

	Grid A	Grid B	Grid C
Circumferential	97	97	97
Transverse	17	17	17
Spanwise, tip to shroud Stretching Ratio	25 1.5	33 1.3	37 1.24

TABLE 3. O-grid dimensions.

	Grid A	Grid B	Grid C
Blade	Fine	Coarse	Fine
Tip Clearance	Coarse	Fine	Fine

TABLE 4. Grid classifications in the two computational regions.

<u>Tip Heat Transfer</u>

In the figures to follow, the definition of the Stanton number is:

$$St \equiv \frac{-k\partial T/\partial n|_w}{(\dot{m}/A)_1 C_p (T_w - T_t)} \tag{1}$$

The predictions obtained using the three grid schemes are shown in Figure 6 and compared with the experimental data of Dunn and Kim (1992). The results of the runs using grids B and C in Figure 6 show the predictions to be more sensitive to

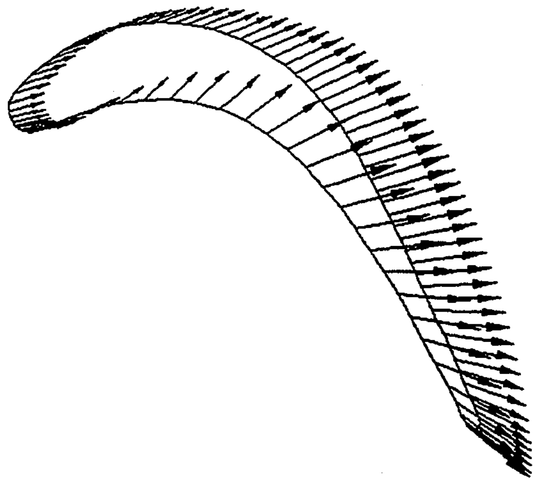

Fig. 4 Flow at mid-gap height of the tip clearance

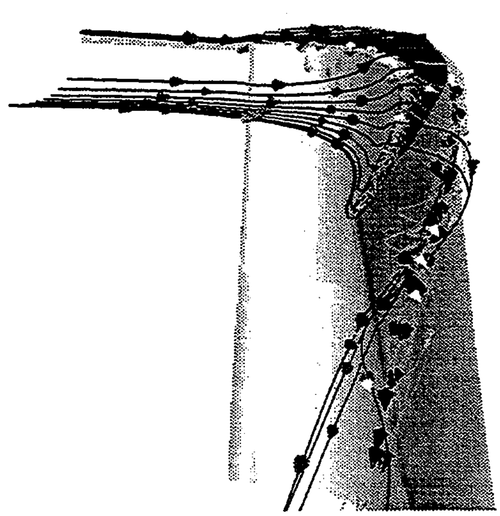

Fig. 5 Trace of the flow in the tip gap region.

the spanwise grid stretching ratio in the tip region rather than to the general resolution of the aggregate grid. This agrees with the general understanding that the flow inside the tip gap is essentially independent of the surrounding flow in the blade passage. Definite proof of grid convergence however still requires further refinement of the grid in the tip.

Mean camber line. The flow in the tip region is a pressure driven flow. It is thought that the flow runs from the pressure side to the suction side due to the differences in the pressure on the two sides. A correlation suggested by Boelter et al. (1945) is often used to estimate the lower bound on the heat transfer to the tip surface. The correlation is as follows:

$$Nu_0 \equiv 0.023 Re_b^{0.8} Pr^{0.4} \qquad (2)$$

where Nu_0 is the Nusselt number for the fully developed flow and Re_b is the Reynolds number based

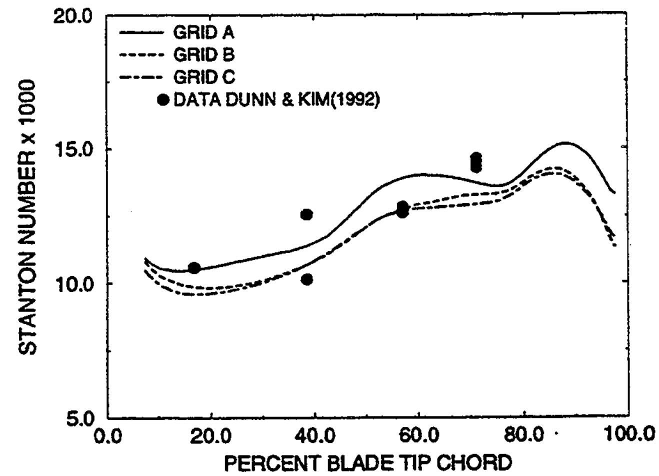

Fig. 6 Grid resolution study of the mean camber line heat transfer on the tip surface.

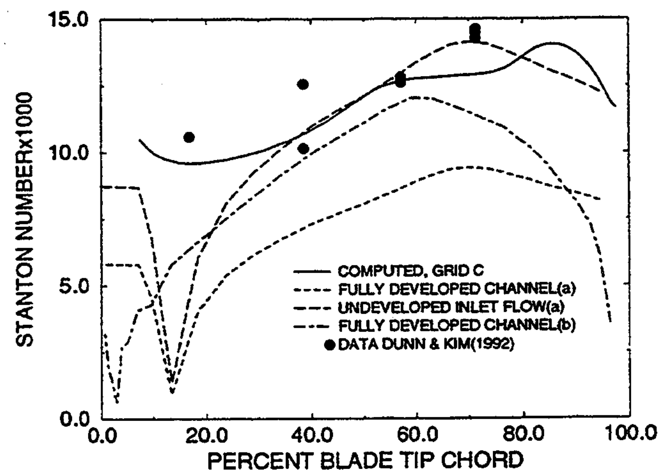

Fig. 7 Comparison of the predicted mean camber line tip heat transfer with the fully developed channel flow correlation and the assumed developing flow heat transfer behavior.

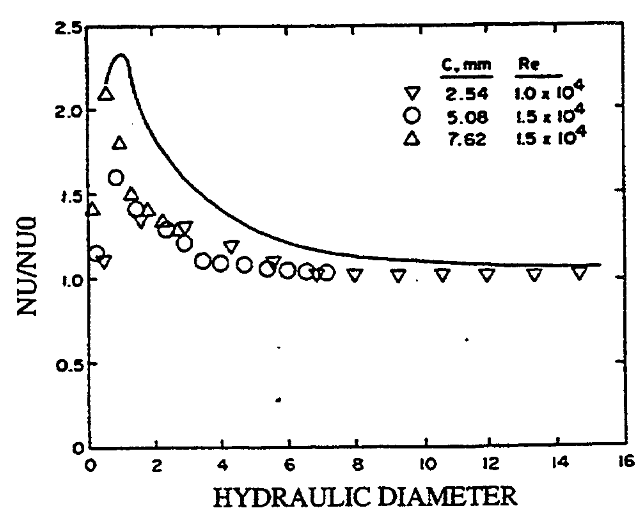

Fig. 8 Tip surface heat transfer development, Metzger et al.(1989)

on the average velocity and the hydraulic diameter "b". In this case the hydraulic diameter is twice the gap height. The above correlation was applied along the mean camber line of the blade using a) the average velocity normal to the mean camber plane at each axial location and b) a velocity computed at each axial location based on the pressure differences across the gap and assuming purely tangential flow.

Method "a" can be used as a product of post-processing of tip clearance velocity information obtained using coarse grids. In Figure 7 the numerical prediction obtained using the correlation and the averaged velocities is shown as the short dashed line. The long dashed line in that figure assumes a 50% increase in heat transfer compared to the fully developed prediction. The numerical predictions agree with the 50% enhancement line in the region where the flow runs along the perpendicular direction to the mean camber line. In the upstream region where the flow direction is more axial this assumption fails. The justification for the choice of 50% enhancement factor is as follows: Measurements of Metzger et al. (1989), shown in Figure 8, performed using heated air on a channel floor downstream of an inlet, show that the fully developed heat transfer is achieved at a length of at least 4 times the hydraulic diameter (8 gap heights) along the floor surface. Those measurements show, that the peak rate of heat transfer occurs close to the inlet and varies with Reynolds number and the gap height, from about 40-120% above the fully developed rate. This peak value of heat transfer is quite localized. For the case at hand, the blade thickness at the tip is approximately 6 times the clearance gap (3 hydraulic diameters) at ~32% chord. This reduces to approximately 1.5 times the clearance gap (.75 hydraulic diameter) in the downstream portion of the blade. Based on the above information, therefore, an estimated enhancement factor of 1.5 is quite justifiable.

Figure 7 also contains the prediction obtained using the procedure described as item "b" above. This latter method is more likely to be used because of the usual availability of pressure distribution information. This method provides a quick means of estimating the magnitude of the heat transfer rate on the tip surface.

Tip surface. Figure 9 exhibits the variation of the heat transfer rate in terms of Stanton number on the tip of the blade. As expected, large heat transfer rates occur close to the edge of tip surface due to the entrance effect. This effect is quite pronounced in the leading edge region as well as the edge of the blade on the pressure side where the flow crosses the blade(cf. figs. 4 and 5). The lower values occur in the regions where the flow is more developed due to the longer travel distance of the flow over blade. The largest values of Stanton number on the blade tip, surprisingly, appear to occur close to the edge of the suction surface. A probable cause of this effect can be deduced from figure 10, where large favorable pressure gradients can be seen to exist at those locations, causing local acceleration of the flow and a commensurate increase in the rate of heat transfer. Figure 9 shows the complex heat transfer pattern that exist on the tip surface of the blade. Obviously, aside from the regions experiencing entrance type of flow effect, the rest of the tip surface is not amenable to simplified analysis.

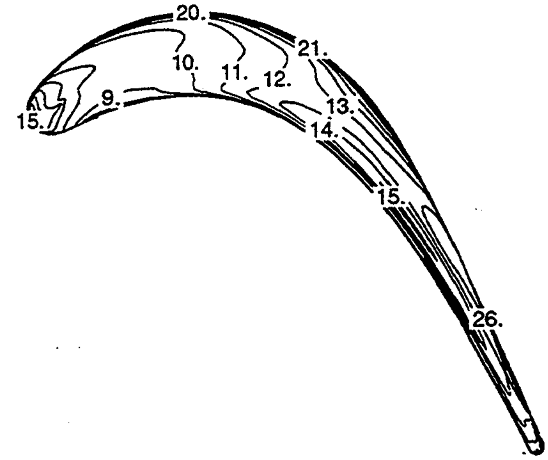

Fig. 9 Tip heat transfer, Stanton number x 1000

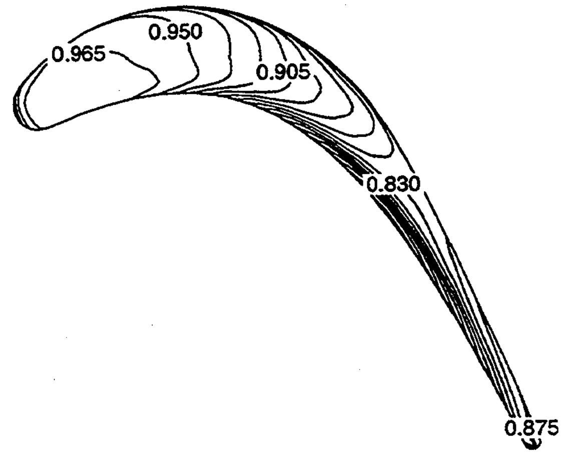

Fig. 10 Tip surface pressure distribution normalized by the relative total pressure.

Blade Surface

Figure 11 shows the predictions of the rate of heat transfer at 90% span using the three different grid schemes and comparison with the experimental data.

The rate of heat transfer on the surface of the blade is greatly affected by the tip clearance flow. It was noted by Rued and Metzger (1989) and Metzger and Rued (1989) that near the tip, the rate of heat transfer is increased on the suction side of the blade due to the roll up of clearance flow and formation of a vortex. On the pressure side, the increase is due to the sink like behavior of the leakage flow.

Figures 12(a) and 12(b) illustrate the variation of the rate of heat transfer on the blade surface as a whole and

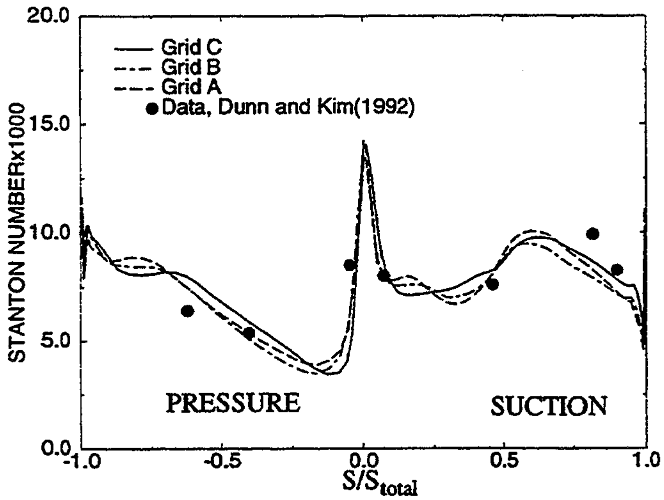

Fig. 11 Blade surface heat transfer at 90% span.

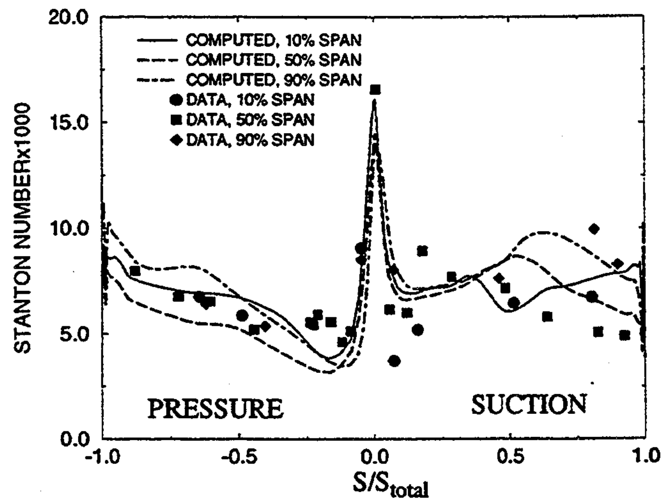

Fig. 13 Blade surface heat transfer at all measured locations with grid C.

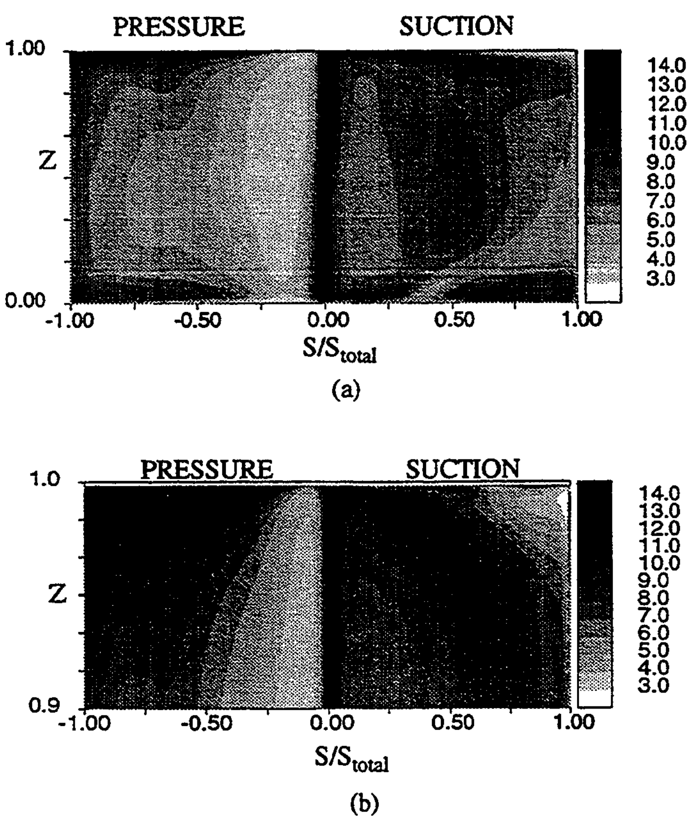

Fig. 12 Blade surface heat transfer, Stanton numberx1000, (a) hub to tip, and (b) upper 10% of radial distance to the shroud.

near the blade tip. The evidence of the effect of the tip vortex on the suction side can be clearly seen from those figures. Looking to the pressure side, the increase in the rate of heat transfer in the vicinity of the blade tip is also evident. This is in agreement with the experimental observations and theoretical considerations noted by Metzger and Rued (1989).

Finally, Figure 13 shows the comparison of the predictions using the finest grid employed, namely grid C, with the experimental measurements for the three spanwise locations of 10, 50 and 90%. The agreement with the data is seen to be quite satisfactory.

SUMMARY AND CONCLUSIONS

In this work the effect of the tip clearance flow on the rate of heat transfer to the blade surface and the tip surface was studied. The case considered for the calculations was the SSME (Space Shuttle Main Engine) high pressure fuel side turbine.The prediction of the mid-camber line heat transfer on the tip surface matched the available measured data reasonably well. Large rates of heat transfer were observed near both the pressure and suction side edges of the tip surface. It was observed that on the blade tip surface heat transfer enhancement is due to the entrance effect as well as flow acceleration. The usefulness of commonly used correlations for the rate of heat transfer on the tip surface was found to be limited.

The blade surface heat transfer close to the tip was found to be greatly modified due to the presence of the tip vortex on the suction side and the sink flow on the pressure side as expected. The predictions in that region agreed well with the measurements.

Further studies of the effect of grid resolution and distribution in the tip gap is ongoing.

ACKNOWLEDGEMENT

The authors wish to express their gratitude to Dr. Raymond Gaugler Chief of the Turbomachinery Flow Physics Branch, as well as to Dr. Louis Povinelli Chief of the Internal Fluid Mechanics Division of NASA Lewis

Research Center for their support and encouragement of this work. Thanks are also due to Mr. Robert Boyle of NASA Lewis Research Center and Prof. A. Arnone of University of Florence, Italy for the enlightening discussions and their guidance. The first author wishes to thank Professor S. Farokhi of University of Kansas for sharing his knowledge of tip clearance flows.

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REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	October 1996	Contractor Report

4. TITLE AND SUBTITLE

Prediction of Unshrouded Rotor Blade Tip Heat Transfer

5. FUNDING NUMBERS

WU–505–90–5K
NCC3–370

6. AUTHOR(S)

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8. PERFORMING ORGANIZATION REPORT NUMBER

E–10492

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135–3191

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

NASA CR–198542
ICOMP–96–10
95–GT–142

11. SUPPLEMENTARY NOTES

Prepared for the 40th Gas Turbine and Aeroengine Congress and Exposition sponsored by the American Society of Mechanical Engineers, Houston, Texas, June 5–8, 1995. A.A. Ameri, AYT Corporation, Brook Park, Ohio; and E. Steinthorsson, Institute for Computational Mechanics in Propulsion, NASA Lewis Research Center (work funded under NASA Cooperative Agreement NCC3–370). ICOMP Program Director, Louis A. Povinelli, organization code 2600, (216) 433–5818.

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Unclassified - Unlimited
Subject Category 34

This publication is available from the NASA Center for AeroSpace Information, (301) 621–0390.

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

The rate of heat transfer on the tip of a turbine rotor blade and on the blade surface in the vicinity of the tip, was successfully predicted. The computations were performed with a multiblock computer code which solves the Reynolds Averaged Navier-Stokes equations using an efficient multigrid method. The case considered for the present calculations was the SSME (Space Shuttle Main Engine) high pressure fuel side turbine. The predictions of the blade tip heat transfer agreed reasonably well with the experimental measurements using the present level of grid refinement. On the tip surface, regions with high rate of heat transfer was found to exist close to the pressure side and suction side edges. Enhancement of the heat transfer was also observed on the blade surface near the tip. Further comparison of the predictions was performed with results obtained from correlations based on fully developed channel flow.

14. SUBJECT TERMS

Turbomachinery; Heat transfer; CFD

15. NUMBER OF PAGES 11

16. PRICE CODE A03

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