A Numerical Analysis of Heat Transfer and Effectiveness on Film Cooled Turbine Blade Tip Models

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A NUMERICAL ANALYSIS OF HEAT TRANSFER AND EFFECTIVENESS ON FILM COOLED TURBINE BLADE TIP MODELS

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ABSTRACT

A computational study has been performed to predict the distribution of convective heat transfer coefficient on a simulated blade tip with cooling holes. The purpose of the examination was to assess the ability of a three-dimensional Reynolds-averaged Navier-Stokes solver to predict the rate of tip heat transfer and the distribution of cooling effectiveness. To this end, the simulation of tip clearance flow with blowing of Kim and Metzger was used. The agreement of the computed effectiveness with the data was quite good. The agreement with the heat transfer coefficient was not as good but improved away from the cooling holes. Numerical flow visualization showed that the uniformity of wetting of the surface by the film cooling jet is helped by the reverse flow due to edge separation of the main flow.

NOMENCLATURE

b	width of the cooling hole
C_p	constant pressure specific heat
D	hydraulic diameter of the experimental tunnel ($4.63b$)
G	mass velocity (ρV)
H	height of the channel
h	heat transfer coefficient, eq.(1)
M	blowing parameter G_f/G_m
Nu_D	Nusselt number based on hydraulic diameter
Pr	Prandtl number
q	wall heat flux
R	gas constant
Re	Reynolds number $G_m D_h/\mu$
T	temperature
Tu	turbulence intensity
V	channel bulk velocity
X	distance downstream of cooling holes
y^+	dimensionless distance from a wall, $y^+ = y\dfrac{v^*}{v}$
η	film cooling effectiveness, eq. (2)
μ	viscosity
v	kinematic viscosity

Subscripts

f	film value
m	main flow
R	reference value
w	wall value

INTRODUCTION

The tips of unshrouded rotor blades in axial turbine stages are exposed to high free stream temperatures and large convective heat transfer rates. As a result, tips of blades are prone to early failure. Film cooling is often employed to provide protection against tip burn-out and corrosion. In a typical engine the tip leakage flow is somewhere near 3% of the primary flow and the tip injection mass flow is nearly half that amount (Chen et al. 1993). Therefore efficient use of this cooling air is quite important.

The ability to accurately compute the rate of heat transfer and cooling effectiveness on the tip surface can form the basis of a system enabling the design of efficient tip cooling schemes. Numerical models would enable the designer to design cooling schemes that use the cooling air efficiently thus producing the desired effect of protecting the blade tip without the side effect of contributing to the losses. We have in the past shown that it is possible to predict the rate of blade tip heat transfer (Ameri and Steinthorsson, 1995, 1996 and Ameri et al. 1997, 1998). More recently comparisons were made with a complete map of experimentally measured tip surface heat transfer rates with favorable agreement (Bunker et al. 1999 and Ameri et al. 1999).
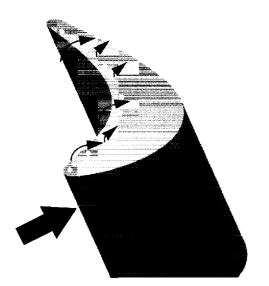
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* Member AIAA

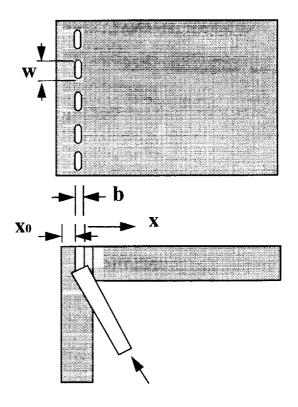**Fig. 1** Typical turbine blade with tip cooling holes

In a previous numerical work Chen et al. (1993) showed, by employing two-dimensional simulations and comparison with experimental measurements the variation of the discharge coefficients with parameters such as the tip flow Reynolds number, isentropic Mach number, length of the flow path along the tip gap and the positive effect of the utilization of squealer tips in conjunction with secondary jets issuing into the tip gap thus reducing the discharge coefficient of the tip leakage flow. Obviously two-dimensional simulations pose severe limitations on the range of geometries that can be considered. In addition, due to the lack of three-dimensional structures such as streamwise vortices, the penetration of the free stream into the film cannot be simulated.

Experimentally, tip cooling has been investigated by Kim et al.(1995) and Kim and Metzger (1995) for flat and recessed tips and for cooling holes of different shapes and issuing from different locations. We have focused our attention on the flat tip case as will be discussed in the ensuing sections.

In this work we seek to determine the suitability of typical CFD methods to tip heat transfer predictions in the presence of the cooling holes. We will also attempt to determine the suitability of the scheme for prediction of cooling effectiveness. This would be the first work addressing tip cooling using numerical simulations.

EXPERIMENTAL SETUP OF KIM AND METZGER

Kim and Metzger set out to study the heat transfer problem as shown in Fig. 1. The tip of the blade is exposed to high velocity flow of hot air. The flow runs from the pressure to suction side of the blade following the pressure gradient. Cooling air is injected through the holes (shown as white dots) over the tip to provide a film of cooler air to protect the blade tip against burnout. A physical model of the blade tip was constructed by Kim and Metzger as

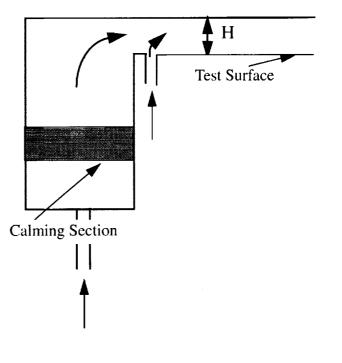**Fig. 2** Details of the test surface of Kim and Metzger(1995)

Fig. 3 Schematic of the testing tunnel

shown in Figs 2 and 3. Figure 2 shows the details of the test surface of the experiment. In this experimental model the primary flow is supplied to a narrow channel simulating the clearance gap above a plain blade tip and the secondary film flow is supplied to the tip surface through a line array of discrete normal injection holes near the upstream or pressure side. Figure 3 shows the schematic of the tunnel and manner in which the main stream and secondary air stream are introduced. The film is injected into the main flow near the channel entrance and is discharged downstream thus simulating the pressure side to suction flow in the tip region of a blade with injection near the edge on the pressure side. The upper wall is adjustable such that the ratio of H/b can be adjusted.

It should be noted that the type of flow considered here is quite sensitive to the sharpness of the corner as has been stressed by many authors including Chen et al. (7). It is believed that modeling the corners as sharp is appropriate for the experiment of Kim and Metzger (9)

The available runs are for values of H/b of 1.5 and 2.5 and for main stream Reynolds numbers of 15000, 30000 and 45000 and blowing ratios (M) ranging from zero to 0.9. The no blowing case was performed without covering the hole.

In this work numerical calculations for the Reynolds number of Re=45,000, H/b=2.5 and M of zero and 0.3 were made.

NUMERICAL SIMULATIONS

COMPUTATIONAL METHOD

The simulations in this study were performed using a multi-block computer code called Glenn-HT, previously known as TRAF3D.MB (Steinthorsson et al. 1993) which is based on a single block code designed by Arnone et al. (1991). This code is a general purpose flow solver designed for simulations of flows in complicated geometries. The code solves the full compressible Reynolds-averaged, Navier-Stokes equations using a multi-stage Runge-Kutta based multigrid method. It uses the finite volume method to discretize the equations. The code uses central differencing together with artificial dissipation to discretize the convective terms. The overall accuracy of the code is second order. The present version of the code (Rigby et al. 1996, 1997 and Ameri et al. 1998a) employs the k-ω turbulence model developed by Wilcox (1994a,1994b) with modifications by Menter (1993). The model integrates to the walls and no wall functions are used. For heat transfer a constant value of 0.9 for turbulent Prandtl number, Pr_t, and 0.72 for Pr were used. Viscosity is a function of temperature through a 0.7 power law (Schlichting, 1979) and Cp is taken to be a constant.
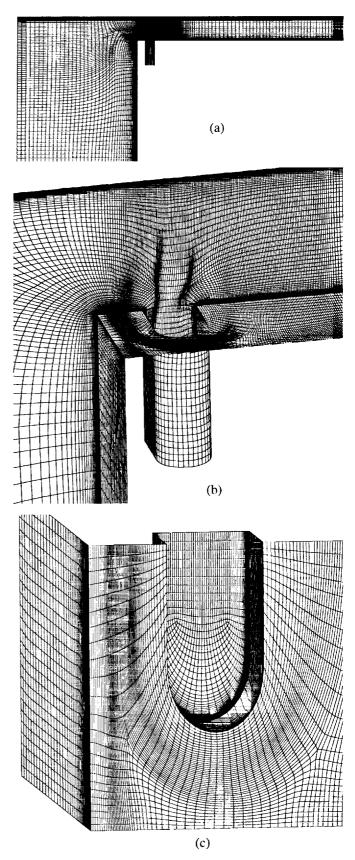

(a)

(b)

(c)

Fig. 4 Various views of the computational grid

The problem considered is a three temperature problem, involving the main flow fluid temperature, the film temperature and the wall temperature. Kim and Metzger define:

$$q_w = h(T_R - T_w) \qquad (1)$$

Where T_R is a reference temperature that renders h independent of the temperature and T_w is the wall temperature as long as temperature differences are not too high and fluid property variations can be reasonably small. Both h and T_R are unknowns. The cooling effectiveness is defined as:

$$\eta = \frac{(T_R - T_m)}{(T_f - T_m)} \qquad (2)$$

If the wall temperature is the only parameter that is varied (density ratio held constant), then T_R is constant and simultaneous solution of eq. (1) for two runs provides h and T_R and the effectiveness is obtained from eq.(2).

THE GRID AND THE BOUNDARY CONDITIONS

Figure 4 shows the grid built for the model problem. Figure 4(a) shows the total view of the problem. Figure 4(b) shows the details near the edge and the injection hole while 4(c) shows more clearly the details of the grid around and into the hole. The spanwise symmetry between the holes and within the holes is used to minimize the size of the computational domain and symmetric boundary conditions are used along those boundaries. The grid is refined to resolve gradient near the walls. The resolution is such that the average value of y^+ is around 0.25, The total number of cells for this grid was 655488 cells. No slip boundary condition is applied to all the walls except to the left wall of the reservoir in Fig. 4(a). This was done to relieve some of the grid density requirement. At the two inlets, total temperature and the average normal momentum were specified.
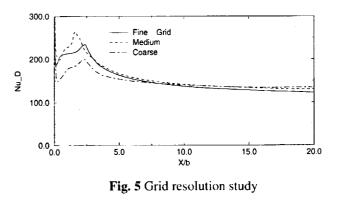
RESULTS AND DISCUSSION

GENERAL REMARKS

First for validation purposes the case of fully developed pipe flow is presented. This is followed by clearance flow simulations. The case of Re=45,000, H/b=2.5, with the blowing parameter M of zero and 0.3 were chosen for study.

FULLY DEVELOPED PIPE FLOW

As a check on the accuracy of the results we performed two simulations to predict the rate of heat transfer for a fully developed pipe flow. For this we chose two cases of Reynolds numbers of 17,000 and 33,000. The pipes were 65 diameters long. The calculations were done on a longitudinal slice with periodic boundary conditions in the tangential direction.
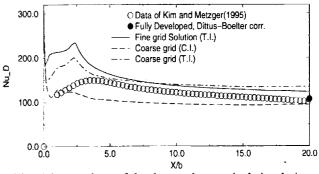

Fig. 5 Grid resolution study

Fig. 6 Comparison of the data and numerical simulations for the no blowing case.

A 5x49x49 grid was used. The results were compared with the commonly used Dittus-Boelter correlation (Boelter et al.(1945))

$$Nu = 0.023 Pr^{0.4} Re^{0.8} \qquad (3)$$

The following table contains the results of that comparison.

Re	Eq. 3	CFD
33000	85.08	88.20
17000	48.86	53.47

Both of the above cases were run using a constant wall temperature 1.1 times the total inlet temperature.

NO BLOWING CASE

For this case in the experiment the supply of injection air into the tip gap was shut off but the hole was not covered. Similarly, for the simulations it was still necessary to model and grid the injection pipe but the inlet to the injection pipe was closed by assigning a slip boundary condition to that surface.

The question of the minimum grid resolution required to obtain grid independence was first addressed by running our problem at three different grid levels, namely, coarse, medium and fine levels. The medium grid was obtained by discarding alternate grid points from the generated fine grid

and the coarse was subsequently generated from the medium grid in the same manner.
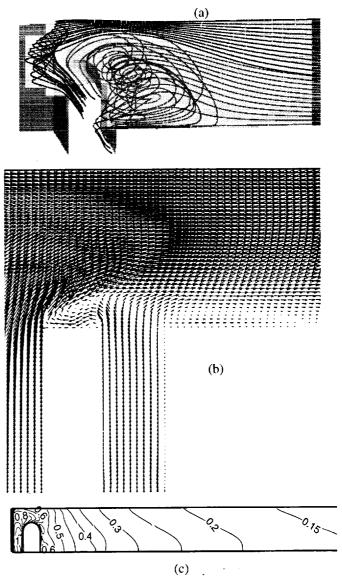
Figure 5 shows the distribution of the span averaged Nusselt number downstream of the hole and the experimental data. The Nusselt number is based on the hydraulic diameter of the experimental channel which in the present case is 4.63 times the hole diameter. The medium and the fine grid results differ by less than 3% away from the hole and less than 10% within two diameters of the hole. Given that the fine grid has 8 times as many cells as the medium grid we decided that the fine level grid was resolved enough for our purposes. The current grid resolution will be used for the blowing cases as well to be discussed later.
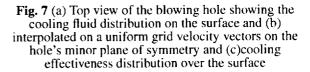
Figure 6 compares the simulation results with the experimental measurements of Kim and Metzger(1995) where the data is somewhat over-predicted. It was speculated by the experimenters that the flow might have relaminarized. Numerical experimentation using the coarse grid showed that the flow inside the channel could be laminar, semi-laminar (upper wall turbulent and the lower wall laminar) or turbulent depending on the state of the incoming flow. The calm inlet (C.I.) and the turbulent inlet (T.I.) solutions are presented on the same figure to bolster this point. Since the solutions using clam inlet condition are unsteady and difficult to converge and more importantly the true state of the flow in turbine environments is seldom laminar, it was decided to run all the cases using the (T.I.) conditions. Considering the very complex nature of the flow near the edge and the presence of the hole and the reported 8% error band on the data we deem the comparison satisfactory.

BLOWING CASE

To test the calculation technique discussed earlier, three cases (instead of two) with differing wall temperatures or total inlet film temperatures were run keeping the film to main temperature ratio close together to avoid differences due to density ratio. This was done using a coarse grid. The results of the computations were post-processed pairwise and the resulting heat transfer coefficient and cooling effectiveness distributions were calculated and found to be identical. Having convinced ourselves of the accuracy of the technique, using a fine grid two cases of differing wall temperatures for the case of M=0.3 and Re=45,000 and H/b=2.5 were run. The film to main flow temperature ratio for both cases was 0.85 and the wall to main flow temperature ratios were 0.85 and 0.9.
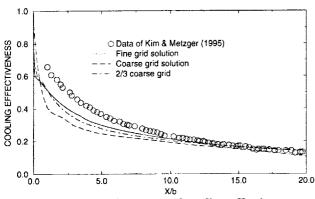
Figure 7 show the flow near the hole. In Fig. 7 (a) it can be seen that the coolant air has covered the tip surface immediately downstream of the hole. Due to the flow separation and backward flow of the main stream air the fluid emerging out of the hole flows upstream (fig. 7(b)) and spills out the sides of the hole thus covering the surface between holes. This action appears to help the spanwise uniformity of h and η. Thus the proximity of the hole to the pressure side edge of the

Fig. 7 (a) Top view of the blowing hole showing the cooling fluid distribution on the surface and (b) interpolated on a uniform grid velocity vectors on the hole's minor plane of symmetry and (c)cooling effectiveness distribution over the surface
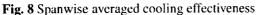
blade in addition to the ones already considered here appears to be an important factor in tip film cooling. In Fig. 7 (c) the distribution of cooling effectiveness over the entire surface is shown where the relative uniformity is evident. In the experiment of Kim and Metzger the resulting effectiveness was completely uniform.

Figure 8 shows the comparison between the computed spanwise averaged cooling effectiveness and the experimentally measured values. The overall agreement is quite good and improves with distance from the hole. In Fig. 8 the distribution obtained from

Fig. 8 Spanwise averaged cooling effectiveness

Fig. 9 Spanwise averaged Nusselt number downstream of the cooling hole

the coarse grid is also shown. The solution shown as 2/3 coarse grid was obtained on a grid similar to the coarse grid except in the normal direction from the heat transfer surfaces where the grid was as refined as the fine grid. The agreement with the experimental measurements is quite similar to the fine grid solution whereas the true coarse grid shows agreement with the data away from the holes. The comparison of the calculated spanwise averaged Nu_D with the measured values is shown in Fig. 9. The agreement with the experimental data near the hole is not very good but improves away from the hole. The agreement away from the holes is encouraging since the objective of film cooling is to provide a blanket of coverage persisting far downstream of the holes. The near hole area is generally overcooled and not normally a failure region. Nevertheless, the near hole results if used in a design would lead to a conservative design. Figures 8 and 9 show that the effectiveness can be computed with better reliability than the heat transfer coefficient.

SUMMARY AND CONCLUSIONS

The use of CFD code to predict the rate of heat transfer and the cooling effectiveness for tip cooling of blades was investigated. To this end the simplified physical model of Kim and Metzger and the corresponding data were used. The three temperature problem was solved by setting up two cases with differing wall temperatures and solving for h and η. The process was repeated using the two cases above with a third case pairwise to ensure the consistency of the method.

The heat transfer coefficient required higher grid resolution than the effectiveness. There was a large variation of the heat transfer coefficient with the grid density. This is logical since the flow near the hole, still under the influence of the tip clearance entrance effect, is quite complex and nonuniform. The variation of effectiveness with the grid density was much reduced. The agreement of the computed effectiveness with the
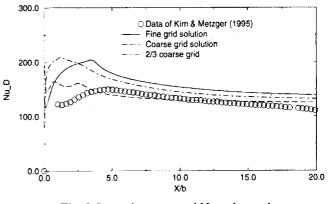
data was quite good. The agreement with the heat transfer coefficient very near the holes (up to 3 to 4 diameters) was not very good but improved away from the holes. The agreement away from the holes where the effectiveness of cooling is diminished is encouraging. The near hole area is generally overcooled and not normally a failure region.

Numerical flow visualization showed that the uniformity of wetting of the surface by the film cooling jet is helped by the reverse flow due to edge separation of the main flow and that the distance from the pressure side edge to the edge of the film cooling hole may be an important variable.

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