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Heat Transfer Near the Trailing Edge of a Cooled Turbine Blade with Special Emphasis Given to Pressure Side Cut-Back Length

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ABSTRACT

Recent cooling system designs for the trailing edge region of a high pressure turbine blade frequently used narrow ejection passages on the concave (pressure) side of the blade. Traditionally, these designs incorporated a cut-back length on the pressure side of the blade. The coolant was ejected tangentially on the pressure side in the last 5-20% of the chord length. The cut-back length and the injection parameters are strongly influencing factors in terms of minimization of the hot spot temperature usually located at the extremity of the trailing edge. The current study is an effort to determine the optimum cut-back length for the reduction of the maximum blade temperature (i.e. the hot spot temperature) without changing the internal coolant mass flow rate already in use. Altering the cut-back length affects the conduction heat flow path and the coolant exit temperature at the exit of the trailing edge ejection passage. The effective length of the tangential cooling jet covering the pressure side of the blade near the trailing edge is also defined by the cut-back length. The present study uses realistic internal and external heat transfer information. The aero-thermal field is coupled by using a finite element methodology. The temperature distribution inside the blade is presented for five different cut-back lengths including the heat pick up levels imposed on the coolant flow originating from the internal passages of the blade. The study shows the direct relationship between the local wall temperatures of the trailing edge tip and the cut-back length. Maximum local temperature reduction in the order of 70 K is possible by altering the cut-back length.

NOMENCLATURE

A_c = Coolant-Side area, m^2
A_g = Gas-Side area, m^2
c_p = Specific heat at constant pressure, J/kg-K
h_f = $\frac{q''}{T_{aw} - T_w}$, Heat tr. Coeff. with film cooling,
h_g = $\frac{q''}{T_{0\infty} - T_w}$, Heat tr.co. without film cooling,
h_{max} = Maximum heat transfer coefficient on the trailing edge (without film cooling), W/m^2-K
k = Thermal conductivity, W/m-K
K = Degree Kelvin
L = Surface distance from the trailing edge, m ,also shell thickness
$\dot{m}$ = Mass flow rate of coolant, kg/s
n_x, n_y = Normals in x and y directions respectively
q'' = Heat flux, W/m^2
s = Distance (along the blade wall) from the beginning of the trailing edge exit slot, m
T_{aw} = Adiabatic wall temperature, K
T_f = Total temperature of the film, K
T_{max} = Maximum total mainstream gas temperature on the trailing edge, K
T_w = Temperature of the blade wall, K
$T_{0\infty}$ = Mainstream gas temperature, K
U = Overall heat transfer coefficient, W/m^2-K

Greek Letters

$\eta = \frac{T_{0\infty} - T_{aw}}{T_{0\infty} - T_w}$, Adiabatic wall effectiveness

θ = non-dimensional coolant temperature

Subscripts

aw = adiabatic wall
c = coolant side
g = gas side
f = film cooling
∞ = mainstream (freestream) conditions
max = maximum
w = wall

INTRODUCTION

Turbine blade cooling is a common practice in modern aircraft and ground based gas turbine engines operating at elevated cycle maximum temperatures. Internal blade cooling is accomplished, among other methods, by circulating the cooling air through an often serpentine passage in the core of the blade. Thermal energy transferred by forced convection and radiation to turbine blades must be removed by internal convection cooling and conduction. In general internal cooling flows are the primary mode of cooling action since conduction alone is insufficient to keep the blades below acceptable temperature and thermal stress limit. The external heat loads vary drastically in the radial direction due to total temperature profiles imposed by the combustor. The variation around the cross section of the blade at a given radius is also significant due to development of blade boundary layers in a highly accelerating turbulent and unsteady flow environment. Since thermal loads over the external surfaces of turbine blades vary non-uniformly, the internal cooling schemes must be arranged such that they can selectively remove higher heat loads at local hot spots.

The trailing edge of turbine blades is one of the most difficult regions for cooling purposes. Cooling air to be used in the trailing edge section of the airfoil is routed from appropriate compressor stages without passing through the combustor. The coolant is supplied to the interior of the blades where it passes through heat exchanger passages. Popular designs in the past have generally used short circular fins known as pedestals. The pedestals are staggered thereby promoting turbulent mixing in the narrow channels between the two side walls. Some important work concerning effectiveness on pin fin arrays have been carried out by Metzger et al. (1984, 1986), Van Fossen (1982) and Chyu et al. (1990, 1996).

The baseline cooling system for this work is a double impingement design that discharges coolant from a diffusing slot to the trailing edge of the gas turbine blade. The pressure side is cut back at the trailing edge to minimize the trailing edge thickness, and to provide coolant ejection on the pressure side of the trailing edge. The current configuration was provided by Downs and Soechting (1996). The primary goal of the present research is to determine the optimum cut-back length for the reduction of the blade temperatures on the trailing edge without changing the internal coolant mass flow rate already in use. Determining the accurate location of the hot spot for various trailing edge designs is also discussed. A finite element technique based on variational principle is used to achieve this goal.

FINITE ELEMENT METHODOLOGY

Thermal analysis of a film cooled turbine blade trailing edge is carried out by generating the local internal conduction heat flow pattern by using the external and internal surface convective heat transfer information provided by Downs and Soechting (1996). A computational method, Camci (1989) based on a variational principle was implemented in order to solve the heat conduction equation for a cooled turbine blade. The governing equation for a steady two dimensional temperature distribution in a film cooled gas turbine blade is given by,

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}=0 \tag{1}$$

Convective boundary conditions, free stream temperature and thermal conductivity of the blade material must be known to solve equation (1). The convective boundary condition is given as follows:

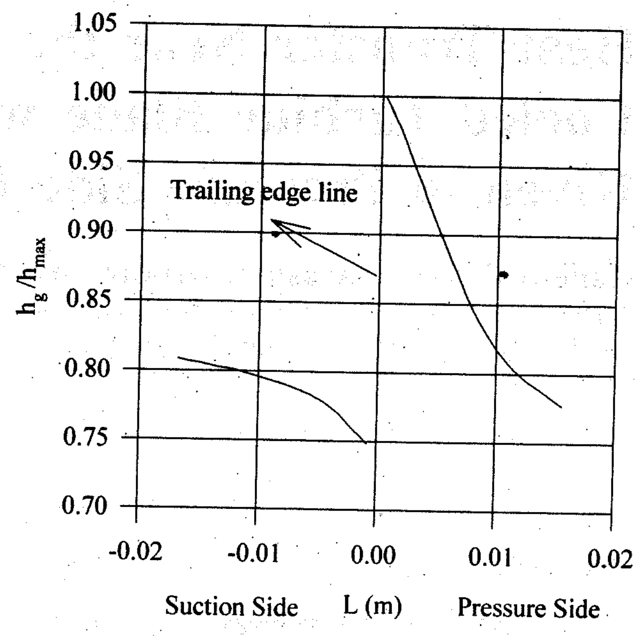

Figure 1. Heat transfer coefficient boundary condition (Downs and Soechting, 1996)

$$-k\left(\frac{\partial T}{\partial x}n_x+\frac{\partial T}{\partial y}\cdot n_y\right)=h\left[T_{0\infty}-T(x,y)\right] \tag{2}$$

The method of solution is based on a variational principle known as Euler's Theorem of variational calculus. The solution of equation (1), which satisfies the boundary conditions given in equation (2) is identical to minimizing an integral statement given in equation (3) over the whole domain.

$$I_F=\iint \frac{1}{2}k\left[\left(\frac{\partial T}{\partial x}\right)^2+\left(\frac{\partial T}{\partial y}\right)^2\right]dx\cdot dy+\int_L \frac{1}{2}h\left(T-T_{0\infty}\right)^2\cdot dL \tag{3}$$

dL is an infinitesimal length over the curved boundary of the domain. The 2-D domain is discretized using 8 noded (quadrilateral) second order iso-parametric finite elements on which I_F is minimized. This minimization procedure leads to a set of linear algebraic equations and the solution of this set of equations yields the temperature distribution in the domain of interest. This finite element technique does not require any iteration. Thus there is no such problem such as convergence of the scheme. The advantage of this technique is its exact nature except the numerical interpolations and integration performed in each element. Further details of this analysis can be found in Zienkiewicz (1971).

EXTERNAL HEAT TRANSFER COEFFICIENTS AND TOTAL GAS TEMPERATURE

The thermal performance of a gas turbine blade trailing edge cooling system depends on the magnitude of internal convection that can be generated and the amount of film

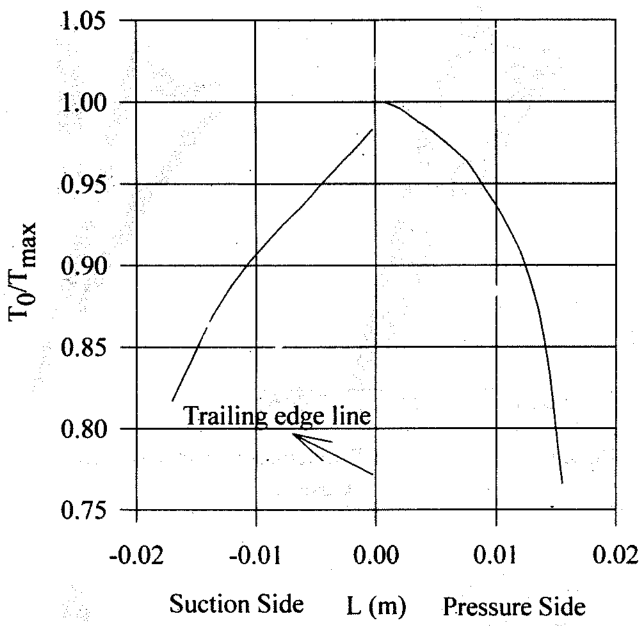

Figure 2. Total gas temperature boundary condition (Downs and Soechting, 1996)

cooling utilized in the region. The heat load imposed on the trailing edge system is a function of the convective heat transfer coefficient (h_g) generated on the external surface and the local gas temperature. The distribution of h_g (non dimensionalized by the maximum heat transfer coefficient on the trailing edge) used for this work is shown in Fig. 1, Downs and Soechting (1996). The effective wall temperature is influenced by the total gas temperature and any cooling effects from the film coolant injected upstream and the tangential wall jet used in the cut back region. The distribution of total gas temperature (non dimensionalized by the maximum gas temperature on the trailing edge) is shown in Fig. 2 Downs and Soechting (1996). The heat transfer coefficient (h_g) on the suction side, near the trailing edge of the turbine blade are lower than the pressure side values.

CALCULATIONS FOR OPTIMUM CUT-BACK LENGTH

As shown in Fig. 3, the thermal performance of the trailing edge cooling system was evaluated by dividing the geometry into 6 regions and examining each region independently. Region 1 is the trailing edge up-pass, regions 2 and 4 are crossover holes, regions 3 and 5 are impingement cavities and region 6 is the trailing edge exit slot. The coolant heat pickup in the trailing edge is obtained using 1-D analysis. Results of this analysis were found to be in agreement with the more complex 2-D and 3-D analyses. The following equation was used to determine the coolant exit temperature at each of the above sections:

$$\frac{T_f - Tc_{out}}{T_f - Tc_{in}} = e^{-\frac{UA}{\dot{m}c_p}} \qquad (4)$$

In Eq.(3), T_f is the total temperature of the coolant film, Tc_{in}

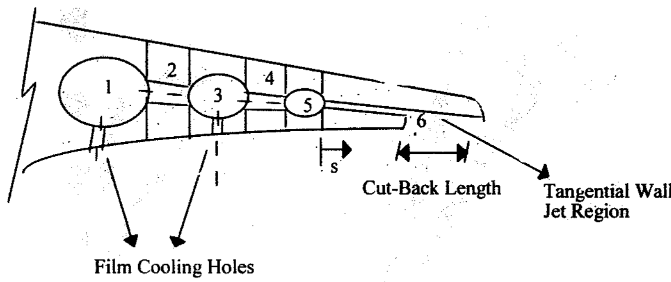

Figure 3. Schematic of trailing edge region showing detailed cooling flow distribution

and Tc_{out} are the inlet and exit temperatures for the 6 regions, A is the area and U is the overall heat transfer coefficient. The overall heat transfer coefficient is calculated as follows:

$$UA = \frac{1}{\frac{1}{h_g A_g} + \frac{L}{Ak} + \frac{1}{h_c A_c}} \qquad (5)$$

Equations 3 and 4 model each one of the 6 regions as an equivalent shell of thickness L. Eq.(3) incorporates the average cooling influence imposed by a film cooling layer on the gas side of an equivalent shell. Although these two equations do not represent the most accurate thermal model for local variations in each region, the model is adequate in calculating the coolant exit temperature from each region. Once coolant temperatures at the exit of each region is calculated from a 1D energy balance, the more accurate local blade metal temperatures are obtained from the solution of 2D heat conduction equation for the exact geometry using a variational method described in the previous paragraphs.

Using the above method, coolant exit temperatures were calculated for six trailing edge configurations with different cut-back lengths, including a fully extended blade with zero cut back length. With the calculated coolant exit temperatures, h (heat transfer coefficient boundary condition) was calculated for elements downstream of the cooling holes using the following equation:

$$\frac{h}{h_g} = \frac{h_f}{h_g}(1 - \eta\theta) \qquad (6)$$

The non-dimensional coolant temperature θ in Eq. (5) is defined as :

$$\theta = \frac{T_{0\infty} - T_{c_{out}}}{T_{0\infty} - T_w} \qquad (7)$$

Overall adiabatic wall effectiveness η of the coolant film covering the cut-back area can be predicted by superposing the film effectiveness data for the first film cooling row, second film cooling row, and the tangential wall jet in the trailing edge slot. This effectiveness was calculated using Seller's (1963) superposition hypothesis which is as follows:

$$\eta = 1 - (1 - \eta_1)(1 - \eta_2)(1 - \eta_3) \qquad (8)$$

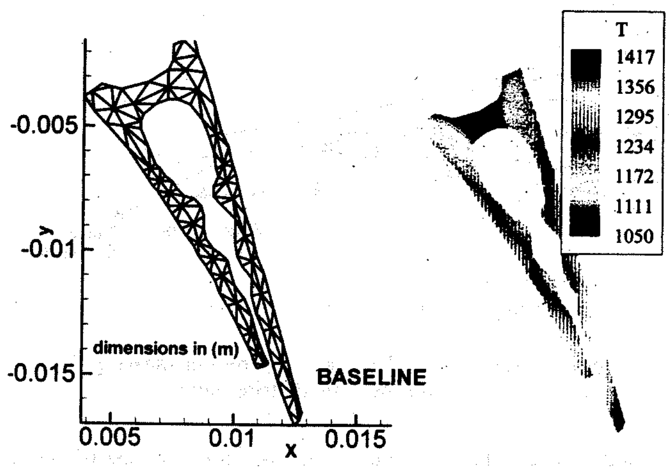

Figure 4. Grid And The Temperature Distribution In ^{o}C,
BASELINE Case

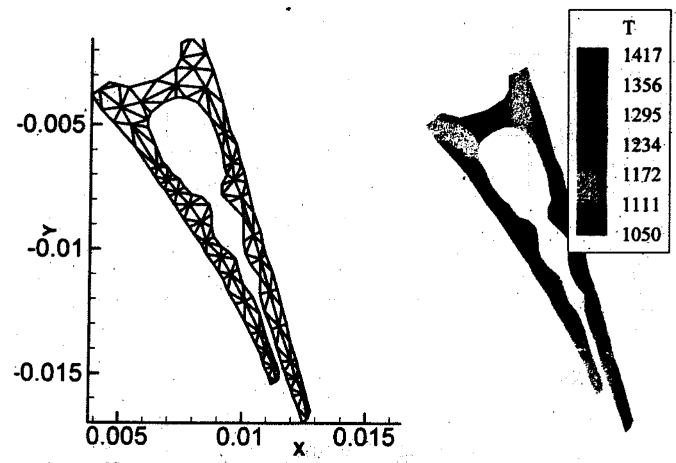

Figure 5. Grid And The Temperature Distribution In ^{o}C,
Case A

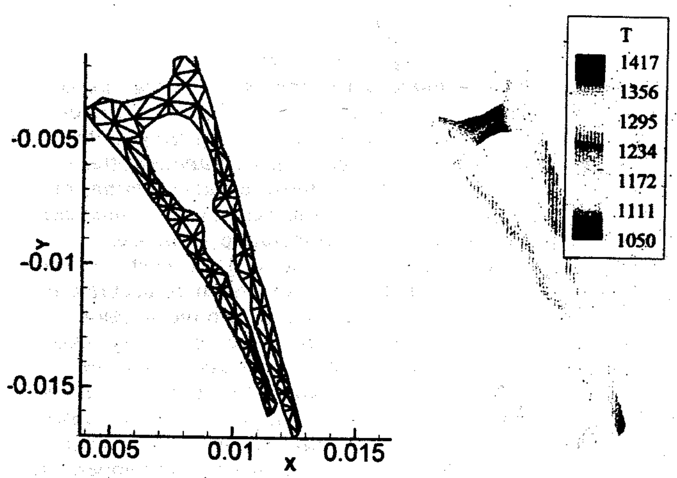

Figure 6. Grid And The Temperature Distribution In ^{o}C,
Case B

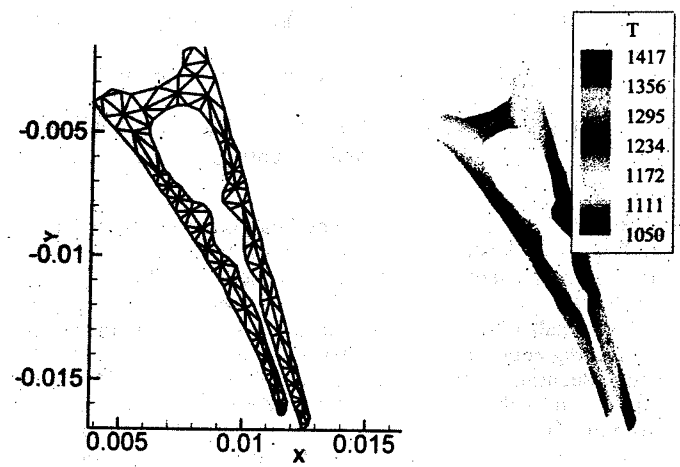

Figure 7. Grid And The Temperature Distribution In ^{o}C,
Case C

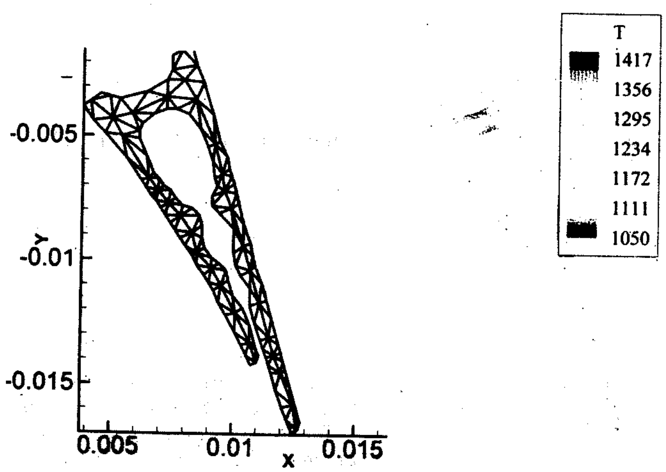

Figure 8. Grid And The Temperature Distribution In ^{o}C,
ase D

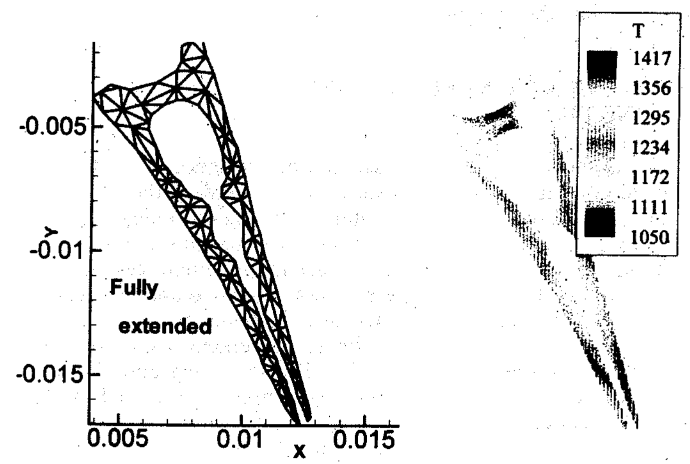

Figure 9. Grid And The Temperature Distribution In ^{o}C,
Fully Extended Case

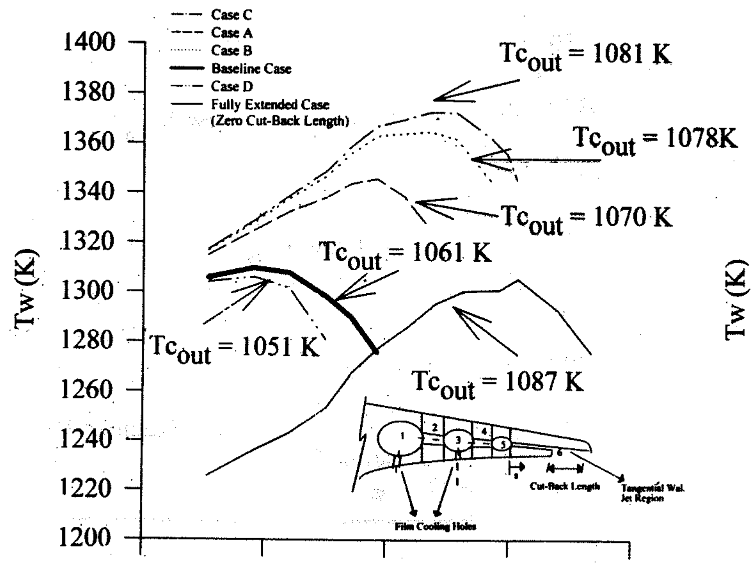

Figure 10. Blade Wall Temperatures on the Pressure Side (gas side, the region between 5 and 6 in Figure 3)

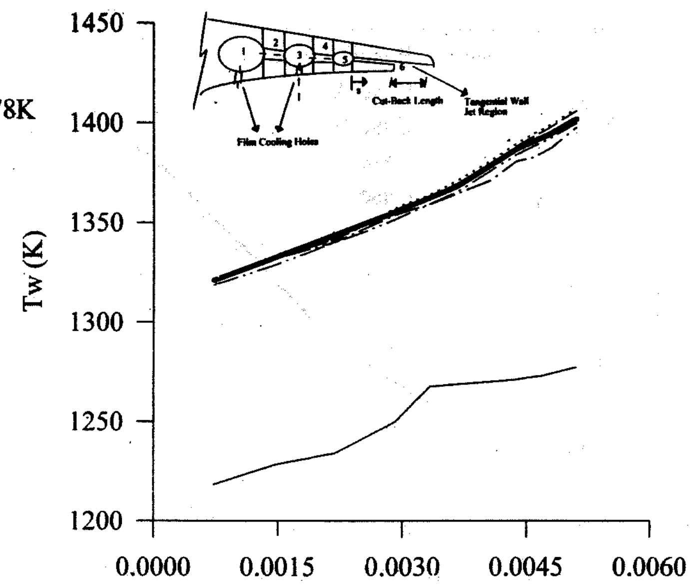

Figure 11. Blade Wall Temperatures on the Suction Side (gas side, the region between 5 and trailing edge in Figure 3)

η_1, η_2 and η_3 are the film effectiveness of the first cooling hole, second cooling hole and the trailing edge slot respectively, Downs and Soechting (1996). Using the values of η calculated from Seller's hypothesis (where h_f/h_g is approximately unity), and assuming an initial wall temperature T_w to be about 70% of $T_{0\infty}$, the temperature distribution in the blade metal for the trailing edge region was calculated. Using these new values of T_w, a new value for Θ and subsequently for h were calculated from Eq.(5), and a new wall temperature distribution was obtained. It took about 2-3 iterations for the convergence of wall temperatures.

RESULTS AND DISCUSSION

Figures 4 represents the computational grid and contour plot respectively for the baseline case provided by Downs and Soechting (1996). Coolant fluid issued from two discrete rows of film cooling holes on the pressure side mixes with the tangential wall jet near the trailing edge as shown in Figure 3. Figures 5-9 show the finite element meshes and contour plots of local temperatures respectively for the trailing edge region analyzed during the current study. Cases A, B and C have respectively one, two and three segments more on the pressure side than the baseline case. All three cases have a shorter cut-back length than the baseline case, Figure 3. Case D was obtained after removing one element from the baseline case on the pressure side, thereby having a longer cut-back length. The fully extended case has zero cut-back length. It is clear from all the contour plots that the tip of the trailing edge on the suction side is on an average hotter than its counterpart on the pressure side. This can be explained by the fact that the tangential wall jet provides some convective cooling in the cut-back region. The suction side is directly heated up by the hot mainstream gases and the cooling effect of the wall jet is not felt as much. Fast decaying nature of η imposed by the tangential wall jet also contributes to the excessive heating of this region.

Figure 10 shows the blade wall temperatures on the pressure side trailing edge exit slot. It is clear from the figure that the blade wall temperatures are reduced as the cut-back length is increased. A plausible explanation is that exit coolant temperatures at the trailing edge ejection passage are lower for longer cut-back lengths. Case D is clearly the best giving much lower temperatures than all the other cases with cut-back length. It is to be noted that the fully extended case gives the lowest wall temperatures on the pressure side. The maximum local temperature for the baseline case on the pressure side was found to be about 1310 K. This maximum temperature was located at s = 0.0014 m. The maximum local temperatures for the other cases A, B and C on the pressure side were found to be 1345 K (s = 0.0029 m), 1364 K (s = 0.0035 m), 1372 K (s = 0.0035 m) respectively. Thus, the maximum local temperatures on the pressure side also increased as cut-back length was decreased. Also, the location of the maximum temperature moved downstream towards the tip of the trailing edge exit slot. Case D (longest cut-back length) had the least local maximum temperature (1306 K at s = 0.0014 m) validating the statement that the local maximum temperature decreased with increasing cut-back length. The fully extended case had the same local maximum temperature as case D (1306 K) on the pressure side. Figure 11 shows the blade wall temperatures on the suction side trailing edge exit slot. It is evident from the figure that there is no significant difference in wall temperatures between all the cases (with cut-back length) studied. The fully extended though gives much lower wall temperatures. Also, the local maximum temperatures on the suction side for baseline case and cases A, B, C and D were found to be 1402 K, 1406 K, 1407 K, 1397 K, and 1400 K respectively. This local maximum temperature was always found at the very tip of the trailing edge exit slot on the suction side at s = 0.0051 m. So the local maximum temperature did not vary significantly in case of the suction side. If one considers both the pressure and the suction sides, the hot spot was always located on the suction side and that too at the very tip.

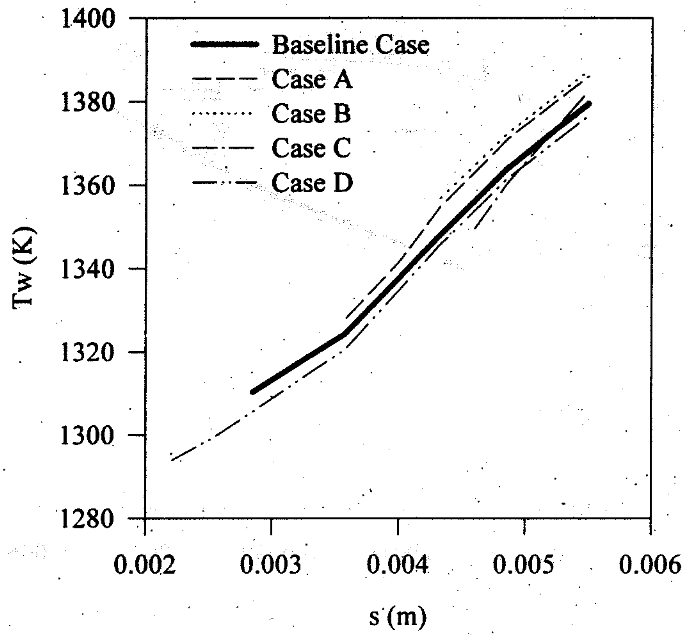

Figure 12. Blade wall temperatures in the tangential wall jet region

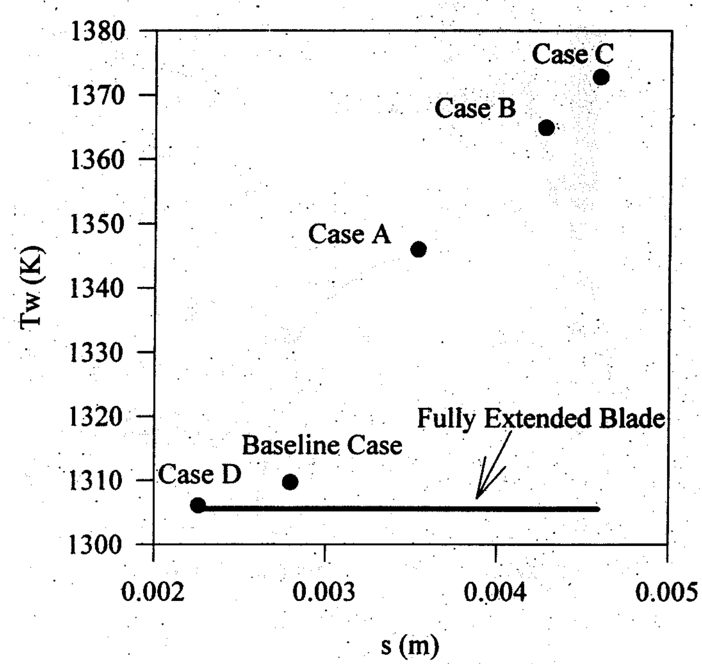

Figure 13. Variation of Maximum Temperature with Cut-Back Length (Pressure Side)

The aforementioned discussion was with regards to the external surface of both the pressure and the suction sides. If one carefully examines the contour plots, it is evident that the internal passages on both the pressure and the suction sides in exit slot region follow the same trend as for the external surfaces. Figure 12 shows the wall temperatures in the tangential wall jet region of the exit slot. There is no significant difference in blade metal temperatures between the various cases with different cut-back lengths.

Figure 13 shows the plot of local maximum wall temperature versus the total length of the blade wall from the exit slot on the pressure side. Longer the length of the blade wall from the exit slot, shorter is the cut-back length. From the figure, it can be deduced that maximum local temperature does reduce with shorter length of the blade wall, i.e. longer cut-back length. Again, a similar figure (Fig. 14) for the suction side shows hardly any difference in the maximum local wall temperatures between the cases with cut-back length.

From the above results, it can be concluded that, case D (with the longest cut-back length) gives the best results on the pressure side and hence is the optimum cut-back length. It would be worthwhile to analyze some more cases with cut-back lengths longer than case D. Although the thermal analysis favors longer cut back lengths for the reduction of metal temperatures near the trailing edge, caution should be exercised from aerodynamic penalty point of view. The wake of the viscous flow region downstream of the coolant slot on the pressure side is expected to be much wider than the wake corresponding to a design with relatively shorter cut-back length. The inviscid flow velocities near the trailing edge on both the suction side and the pressure side are near transonic values in high pressure turbines. The specific trailing edge area is extremely sensitive in terms of the generation of viscous losses. The overall aero-thermal design of the trailing edge region should incorporate an optimization process with dominant design issues being the minimized metal temperatures, minimized aerodynamic losses (including reduced coolant mass flow rate from the internal cooling system) and a geometrical design that is realistic from manufacturing point of view. Reducing the extreme gradients of metal temperatures in this region is also beneficial in reducing the thermal stresses.

CONCLUSIONS

Temperature distributions inside the trailing edge region of a gas turbine blade with five different cut-back lengths are presented. The results clearly indicate that longer the cut-back length, lower are the blade metal temperatures in the trailing edge exit slot on the pressure side. The trailing edge with the longest cut-back length gave the best results. The hot spot temperature on the pressure side has been significantly reduced for the case with the longest cut-back length (by more than 65 K) in comparison to the case with the shortest cut-back length. The suction side of the trailing edge exit slot did not show significant changes in blade metal temperatures for all the cases studied. Future work should focus on a few more cases with longer cut-back lengths. Aerodynamic losses in the wake region where there is strong interaction of the coolant jets and the free stream fluid, the thermal stresses and the manufacturability of new designs are significant parameters to consider in addition to local metal temperatures in the trailing edge region.

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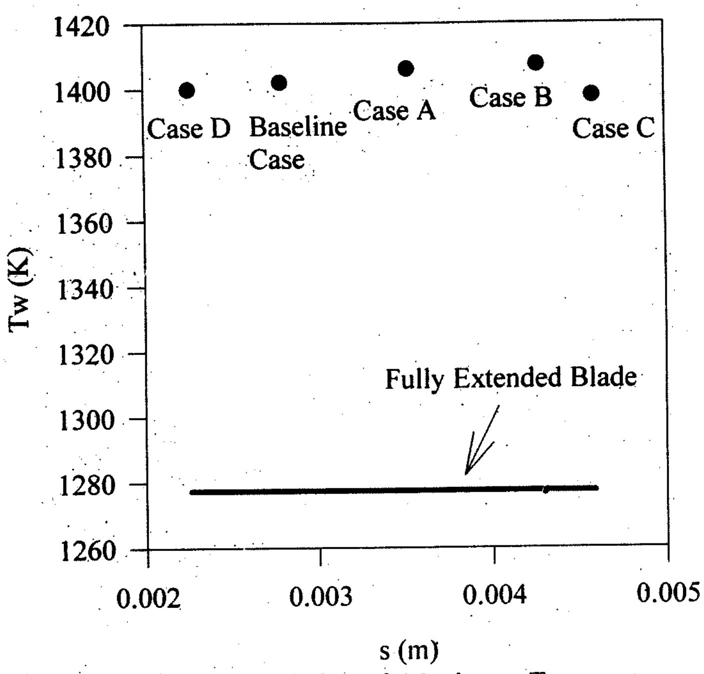

Figure 14. Variation of Maximum Temperature With Cut-Back Length (Suction Side)

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