Numerical Analysis of Gas Turbine HP Stage Blade Cooling with New Cooling Duct Geometries

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Abstract: The turbine blades of HP stage are rather small to incorporate the blade cooling techniques efficiently. The current turbine entry temperature levels in advanced gas turbines exceed the melting point of the blade. So, along with high temperature material development, a novel cooling scheme must be developed for safe operation of HP stage. An effort has been made in this paper to computationally analyze the HP stage turbine blade for effective cooling using new duct within the blade. An helicoidal duct with circular cross-section has been analyzed corresponding to different duct diameters and pitch length of the helix. It is found from the analysis that helicoidal cooling duct with larger diameter and with lower pitch length provides a vastly improved cooling compared to straight ducted cooling ducts for the HP stage.

Keywords: HP Stage Turbine Blade Cooling, Innovative Cooling Geometries

Introduction:

Gas turbines are extensively used for aircraft propulsion, land-based power generation, and industrial applications. The gas turbine is a power plant, which produces a great amount of energy for its size and weight. Its compactness, low weight and multiple fuel application make it a natural power plant for many applications. Thermal efficiency and power output of gas turbines increase with increasing turbine entry temperature (TET). It is clear from Brayton cycle that the increase in pressure ratio increases the gas turbine thermal efficiency accompanied with increase in turbine firing temperature. The increase in pressure ratio increases the overall efficiency at a given temperature. However increasing the pressure ratio beyond a certain value at any given firing temperature can actually result in lowering the overall cycle efficiency [Horlock J. H. (2001)].

As the TET increases, the heat transferred to the blades in the turbine also increases. The level and variation in the temperature within the blade material (which causes thermal stresses) must be limited to achieve reasonable durability goals. Also, the temperatures are far above the permissible metal temperature due to which there is a need to cool the blades to operate without failure [Han J. et al. (2011)].

Therefore, along with high temperature material development, a sophisticated cooling scheme must be developed for HP stage turbine blade for continuous safe operation with high performance. Several research works are being carried out to address the above problems.

Problem Statement and Assumptions:

The objective of the analysis is to study the effect of helicoidal cooling ducts on gas turbine blade cooling by varying the geometric parameters and comparing the same with circular cooling ducts. The following assumptions are made.

1. The computational domain is assumed to be made of solid-liquid interface with conjugate heat transfer.
2. Steady incompressible flow for the fluid.
3. Flow is assumed to be turbulent with fully developed conditions.
4. Solution is marched in time and steady state is assumed to be obtained after all residuals are bought to prescribed constant values.

Numerical Model:

The commercial CFD software FLUENT (version 6.3.26) from Fluent, Inc. is employed for analysis. The simulation uses the segregated solver, which employs an implicit pressure-correction scheme. The SIMPLE algorithm is used to couple pressure and velocity. First order upwind scheme is selected for spatial discretization of the Reynolds Averaged Navier Stokes (RANS) equations as well as energy and turbulence equations. Converged results are obtained after the residuals were found to be less than the specified values. A converged result renders an energy residual of $10^{-6}$, and momentum and turbulence kinetic energy residuals
being $10^8$ [Li X. et al. (2008)].

**Flow Equations:**

All the equations are presented in Cartesian tensor notation.

**Mean Flow Equations:**

**Continuity:**

$$\frac{\partial}{\partial \mathbf{x}} (\rho \mathbf{U}) = 0 \; (1)$$

**Momentum Transport:**

$$\frac{\partial}{\partial \mathbf{x}} (\rho \mathbf{U} \mathbf{U}) = \frac{\partial P}{\partial \mathbf{x}} + \frac{\partial}{\partial \mathbf{x}} \left[ \mu \frac{\partial \mathbf{U}}{\partial \mathbf{x}} + \rho \mathbf{U} \frac{\partial \mathbf{U}}{\partial \mathbf{x}} \right] - \rho \mathbf{U} \frac{\partial \mathbf{U}}{\partial \mathbf{x}} \; (2)$$

**Enthalpy:**

$$\frac{\partial}{\partial \mathbf{x}} (\rho \mathbf{U} T) = \frac{\partial}{\partial \mathbf{x}} \left[ \frac{\mu + \mu_t}{\text{Pr}} \frac{\partial T}{\partial \mathbf{x}} - \rho \mathbf{U} \frac{\partial T}{\partial \mathbf{x}} \right] \; (3)$$

**Turbulence Modeling Equations:**

Zonal k-ε model: This consists of the high-Re k-ε in the fully turbulent core:

$$\frac{\partial}{\partial \mathbf{x}} (\rho \mathbf{U} k) = \frac{\partial}{\partial \mathbf{x}} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial \mathbf{x}} \right] + P_k - \rho \mathbf{U} \mathbf{U} \; (4)$$

$$P_k = -\rho \mathbf{U} \frac{\partial \mathbf{U}}{\partial \mathbf{x}} \; (5)$$

$$\frac{\partial}{\partial \mathbf{x}} (\rho \mathbf{U} \varepsilon) = \frac{\partial}{\partial \mathbf{x}} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial \mathbf{x}} \right] + c_{\varepsilon \varepsilon} \frac{\varepsilon}{k} P_k - \rho \mathbf{U} \mathbf{U} \mathbf{U} \mathbf{U} \mathbf{U} \; (6)$$

**Computational Domain for the Analysis:**

While the configuration of the internal cooling ducts in real gas turbine airfoil can be highly complicated, in the present analysis, a simplified geometric model with different ducts of varying geometry is being studied for the effective cooling consideration as shown in figure 1. In general, the coolant air enters the cooling ducts from the blade root, flows through entire length of ducts and finally leaves from blade tip. The cooling process is entirely by internal convective cooling only.

The cooling ducts having through ducts of circular cross section with hole radius of 1.5 mm and 2 mm, helicoidal ducts of pitch length of 4 mm and 6 mm with circular cross section with hole radius of 1.5 mm and 2 mm are considered.

**Boundary Conditions:**

The required boundary conditions implemented in the study of blade profile and for inner cooling ducts are presented here. The boundary conditions are derived from the practical gas turbine operating conditions, corresponding to HP stage blade that are generally exposed to high temperature and velocity. The through flow of hot gases over the turbine blade has a free steam temperature of hot gas as 1561 K and convective heat transfer coefficient of hot gas as 2028 W/m²K. On the other hand the coolant air admitted at the duct of the blade root has an entry temperature of 644 K [Li X. et al. (2008)] and mass flow rate of 90 kg/hr. These settings are selected with a view to get a realistic representation of typical gas turbine operating conditions.

**Table 1: Thermo Physical Properties of air at 644 K**

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>0.54 kg/m³</td>
</tr>
<tr>
<td>Specific heat</td>
<td>1.06 kJ/kg K</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>0.05 W/m K</td>
</tr>
<tr>
<td>Kinematic Viscosity</td>
<td>$59 \times 10^{-6}$ m²/s</td>
</tr>
</tbody>
</table>

**Table 2: Thermal Properties of Blade Material**

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>8180 kg/m³</td>
</tr>
<tr>
<td>Specific heat</td>
<td>446 J/kg °C</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>11.5 W/m °C</td>
</tr>
</tbody>
</table>

**Numerical grid, Meshing and Simulation Procedure:**

The information for geometric model for HP stage gas turbine was derived directly by measuring the coordinates of the blade profile using Coordinate Measuring Machine available in the Metrology Lab of Mechanical Engineering Department of MIT, Manipal. Gas turbine blades along with different cooling ducts configuration were modeled using the software CATIA.
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(version V5 R14), which is an industry standard software for modeling.

Figure 2: Gas Turbine Blade Modeled in Catia and Meshed in Gambit

To conduct numerical simulation, the computational domain as shown in figure 2, is meshed with control volumes built around each grid using GAMBIT (version 2.4.6), which is the preprocessor for FLUENT (version 6.3.26). The mesh for blade simulation is an unstructured type consisting of 2, 19,774 tetrahedral cells. But refined tetrahedral mesh was employed for cooling ducts as dimensions were very small, so as to extract good accuracy.

The grid independence test was performed to the quality of mesh for solution accuracy. The influence of further refinement did not change the result by more than 1.25% which is taken here as appropriate mesh quality for computation.

Results and Discussions:

The results of the thermal analysis in each case corresponding to parametric considerations of the geometry of the cooling duct are detailed below. The results analysis follows the cooling capabilities of helicoidal duct of various pitch lengths viz-a-viz the circular duct of same diameter.

The Non-dimensional temperature $\theta$ is defined as

$$\theta = \frac{T_{\infty} - T_a}{T_{\infty} - T_0}$$

where $T_{\infty}$ is hot gas temperature surrounding the blade profile in a convective ambience, $T_a$ is the temperature along the span length of the blade and $T_0$ is temperature of the cooling air admitted to the cooling duct at the blade root. Similarly the Non-dimensional span length $\delta$ is defined as

$$\delta = \frac{L}{S}$$

is the distance along the span length which the blade temperature is measured and $S$ is overall span length

Case 1: Blade Cooling with Through Duct of Circular Cross Section of 1.5 Mm Radius and Helicoidal Duct of Circular Cross Section and Same Radius with Pitch Length of 6 Mm and 4 Mm.

Figure 3 shows the non-dimensional temperature distribution of the exposed blade surface along the span at 30% of chord length. This non-dimensional temperature distribution is taken at this position for all the different cases for the purpose of comparison. It is seen from the figure that with circular cooling duct along the span, the non-dimensional temperature has lower bound value of around 0.31 and upper bound of 0.45 only. The corresponding helicoidal duct of same radius but with pitch length 6 mm shows significant drop in blade surface temperature change as shown in the figure 3. The blade non-dimensional surface temperature change is from lower bound value of 0.36 to upper bound value of 0.54 along the span length.

This drastic cooling of turbine blade can easily be ascribed to large convective area available within the helicoidal duct corresponding to a pitch length of 6 mm. The advantage of the helicoidal cooling duct is accentuated more if we consider the same helicoidal duct with different pitch length of 4 mm. As seen in figure 9, with 4 mm pitch length, the corresponding convective area for heat transfer increases even more than the previous case and is reflected as steep non-dimensional temperature change varying from 0.40 to 0.585.

Figure 4 shows heat flux dissipated from cooled blade surface. It is clear that for circular ducted blade there is comparatively only negligible dissipation of thermal energy due to cooling of blade surface.
Figure 4: Surface Heat Flux Variation over the Span of the Blade

Figure 5: Temperature of Blade Surface Having Helicoidal Cooling Duct of Pitch 6 mm and Radius Of 1.5 mm

Figure 7: Temperature of Blade Surface Having Circular Cooling Duct of Radius 1.5 mm

Whereas for the helicoidal ducted blade as seen from the figure, a large flux dissipation occurs over the blade surface due to better cooling corresponding to the enhanced area available for helicoidal ducted blade. It is also possible that due to helicoidal nature of cooling duct, the heat dissipation rate is augmented because of turbulence that is brought about due to the path of the cooling duct. The combined effect is beneficial for maximum possible blade surface cooling.

Case 2: Blade Cooling With Through Duct of Circular Cross Section of 2 Mm Radius and Helicoidal Duct of Circular Cross Section and Same Radius with Pitch Length of 6 mm.

Since it is physically not possible to place a helicoidal duct of 2mm radius with pitch length of 4 mm inside the blade, this model is not taken for comparison purposes in this case. As seen in the figure 8 and 9 the phenomena explained for case 1 are manifested to almost the same extent for the level of cooling that is achieved when using larger cooling duct of 2 mm instead of 1.5 mm. In the case of circular duct there is negligible drop in temperature. But the gradient across the cooling surface is almost unchanged whereas for helicoidal cooling ducts with pitch length 6 mm it could be easily seen that there is reasonable change with respect to minimum and maximum possible surface temperature drop along the span length of the blade. This is due to the fact that by providing larger radius for the duct the effective convective area for the duct to exchange heat with the hot surface of the blade increases. And also possibly due to higher turbulence of the flow, as explained earlier.
nature of the cooling duct itself in which the duct passage is alternately will be in the vicinity of the suction and pressure side of the blade, whereas at other positions the cooling duct is away from these two surfaces. Hence when the cooling duct is near to the surfaces there is enhanced cooling of the blade surfaces than when the cooling duct is positioned away from the two surfaces. This results in the fluctuations in the temperature profile as shown in the figure 8 and 9.

Figure 11: Temperature of Blade Surface Having Helicoidal Cooling Duct of Pitch 6 mm and Radius of 2 mm

The efficacy of using helicoidal cooling ducts is therefore justified but associated difficulties with respect to practical manufacturing of a turbine blade with helicoidal cooling ducts remains to be explored.

Conclusion:

From present numerical analysis the following conclusions are derived.

- It is seen that an innovative helicoidal cooling passage, provides an augmented convective area for better heat dissipation.
- The helicoidal path also acts as a turbulators providing extended dissipation rates due to turbulence of flow.
- The diameter and pitch length plays a major role in optimizing the geometry of the helicoidal cooling passage.
- The temperature distribution shows wiggles in the solution due to proximity of the suction and pressure side alternately for the helicoidal ducted passage

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References:


