A Computational Conjugate Thermal Analysis of HP Stage Turbine Blade Cooling with Innovative Cooling Passage Geometries

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***Abstract*: In most of the practical gas turbines, the turbine blades of HP stage are usually too small to employ the turbine blade cooling techniques effectively. The growing need for effective blade cooling techniques is a direct consequence of the continuous quest for greater fuel economy. It is very well known that the thermal efficiency and power output of gas turbines increase with increasing turbine entry temperature (TET). The current TET level in advanced gas turbines is far above the melting point of the blade material. Therefore, along with high temperature material development, a sophisticated cooling scheme must be developed for continuous safe operation of gas turbines for high performance. An attempt has been made in this paper to computationally analyze the coupled conjugate analysis of HP stage turbine blade for effective cooling using innovative cooling passages within the blade. An helicoidal shaped duct has been analyzed corresponding to different diameters and pitch length. It is found from the analysis that helicoidal cooling duct with larger diameter and with lower pitch length provides a vastly improved blade cooling in comparison to straight ducted cooling ducts for the HP stage turbine blade.**

***Index Terms*: Conjugate Thermal Analysis, HP Stage Turbine Blade Cooling, Innovative Cooling Geometries**

I. 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

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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 [15].

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 [14].

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 [14]. Several research works are being carried out to address the above problems.

II. Theoretical Formulations

A. Problem Statement and Assumptions

The objective of the analysis is to study the effect of helicoidal cooling ducts on 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. Material properties and other thermophysical properties are assumed to be constant with respect to temperature.
4. Flow is assumed to be turbulent with fully developed conditions.
5. Solution is marched in time and space and steady state is assumed to attain, when residuals are bought to set values.

B. Numerical Model

The commercial CFD software FLUENT (version 6.3.26) from Fluent, Inc. is employed for analysis. The simulation

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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 [2], [22]. 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^{-5}[22].

C. Governing Equations of Flow

All the governing partial differential equations are presented in cartesian tensorial notation.

Continuity:

$$\frac{\partial}{\partial x_i}(\rho U_i) = 0 \tag{1}$$

Momentum Transport:

$$\frac{\partial}{\partial x_j}(\rho U_i U_j) = \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) - \rho\overline{u_i u_j}\right] \tag{2}$$

Enthalpy:

$$\frac{\partial}{\partial x_j}(\rho U_i T) = \frac{\partial}{\partial x_j}\left[\frac{\mu}{\Pr}\frac{\partial T}{\partial x_j} - \rho\overline{u_i t}\right] \tag{3}$$

Turbulence Equations:

$$\frac{\partial}{\partial x_j}(\rho U_j k) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right) - \frac{\partial k}{\partial x_j}\right] + P_k - \rho\varepsilon \tag{4}$$

$$\textit{where,}\quad P_k = -\rho\overline{u_i u_j}\left(\frac{\partial U_i}{\partial x_j}\right) \tag{5}$$

$$\frac{\partial}{\partial x_j}(\rho U_j \varepsilon) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right) - \frac{\partial \varepsilon}{\partial x_j}\right] + c_{\varepsilon 1}\frac{\varepsilon}{k}P_k - \rho c_{\varepsilon 2}\frac{\varepsilon^2}{k} \tag{6}$$

III. METHOD OF SOLUTION

A. Blade configuration 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 Fig. 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 complete cooling process is internal convective cooling.

The cooling ducts having through circular ducts with hole radius of 1 mm, 1.5 mm, and 2 mm and helicoidal ducts of pitch length 4 mm and 6 mm with hole radius of 1 mm, 1.5 mm, and 2 mm are considered.

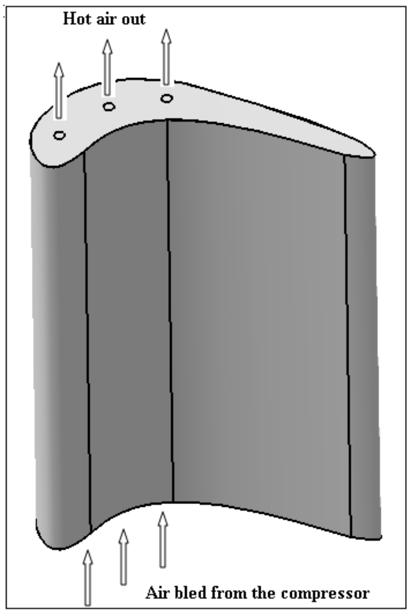

Fig. 1. Schematic of turbine blade with cooling ducts

B. Boundary Conditions

The main boundary conditions utilized both for gas turbine blade and cooling ducts are presented here. The parameters are associated with practical gas turbine operating conditions, corresponding to HP turbine stage featured with high temperature and velocity. The main flow has a convective boundary condition with free stream temperature of hot gas as 1561 K and convective heat transfer coefficient of hot gas as 2028 W/m^2K [22]. The coolant jet flow has an air temperature of 644 K and velocity of 106 m/s [22]. These settings are selected with a view to get a realistic representation of typical gas turbine operating conditions. The properties of air at 644 K and for the turbine blade material are as follows [25], [26].

TABLE I
PROPERTIES OF AIR AND BLADE

Properties of air at 644 K	Properties of blade material
Density: 0.54 kg/m^3 Specific heat: 1.06 kJ/kg K Thermal conductivity: 0.05 W/ m K Kinematic Viscosity: 59 × 10^{-6} m^2/s	Density: 8180 kg/m^3 Specific heat: 446 J/kg °C Thermal conductivity: 11.5 W/m °C

C. 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 (version V5 R14), which is an industry standard software for modeling.

To conduct numerical simulation, the computational domain as shown in Fig. 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.

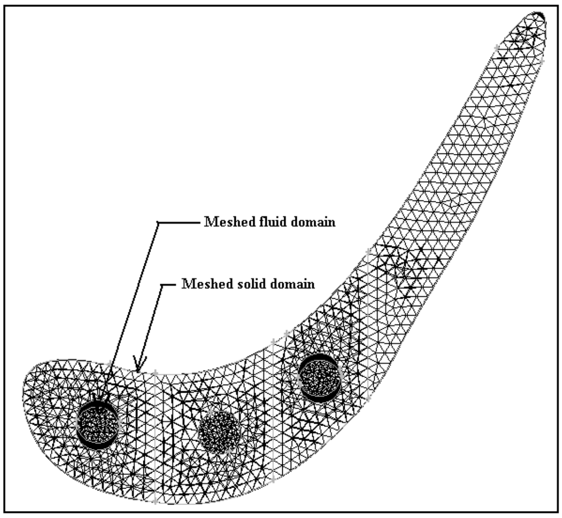

Fig. 2. Gas turbine blade modeled in CATIA and meshed in GAMBIT

IV. RESULTS AND DISCUSSION

The numerical heat transfer analysis carried over the domain blade is described in the following paragraphs, applying the parametric considerations for each case. The solution is assumed to be converged after suitable steady state condition is arrived at after the transient unsteady solution for converged solution.

In the results analysis that follows the cooling capabilities of circular duct is compared with helicoidal duct of the same diameter but with different pitch length.

<u>Case 1: Blade cooling with circular duct of 1 mm radius and helicoidal duct of same radius with pitch length of 6 mm and 4 mm.</u>

Fig. 3 shows the surface temperature along the span at 30% of chord length. This position for comparing the temperature distribution is made the same for all the cases. It is seen from the Fig.3 that with circular cooling duct along the span, the temperature has lower bound value of around 1529 K and upper bound of 1550 K only. The corresponding helicoidal duct of same radius but with pitch length 6 mm shows significant drop in blade surface temperature as shown in the Fig. 3. The blade surface temperature is from lower bound value of 1480 K to upper bound value of 1560 K 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 Fig. 3 with 4 mm pitch length the corresponding convective area for heat transfer increases even more than the previous case and is reflected in steep temperature change varying from 1422 K to 1560 K. Hence the utility of helicoidal cooling duct is established and it seems to be an efficient way for cooling a turbine blade with its space limitations.

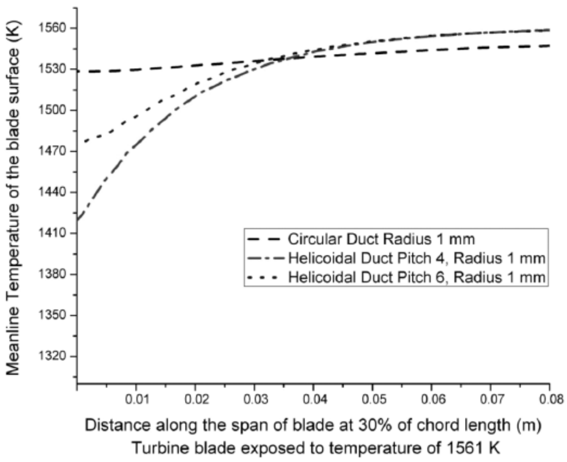

Fig. 3. Temperature variation along the span length of the blade

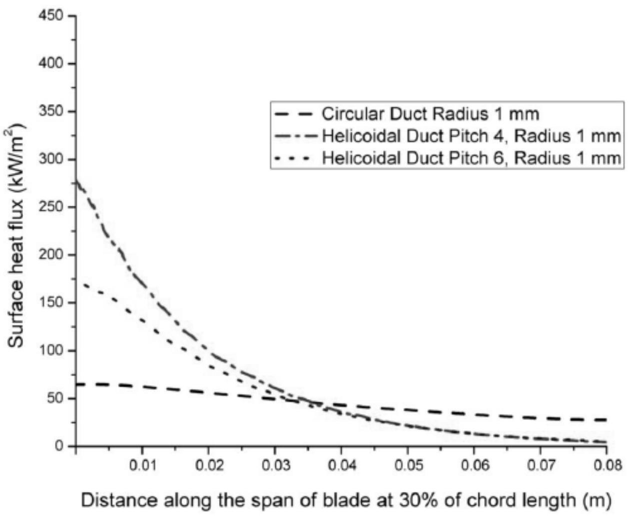

Fig. 4. Surface heat flux variation over the span of the blade

Fig. 4 shows heat flux dissipated from cooled blade surface. It is clear that for circular ducted blade that there is comparatively only negligible dissipation of thermal energy due to cooling of blade surface. Whereas for the helicoidal ducted blade as seen from the Fig.4., 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.

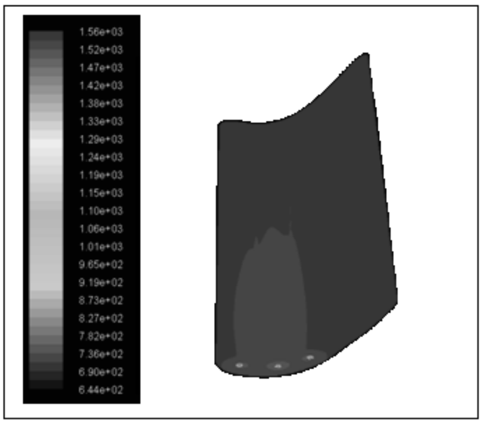

Fig. 5. Temperature distibution on blade surface having Circular cooling duct of radius 1 mm

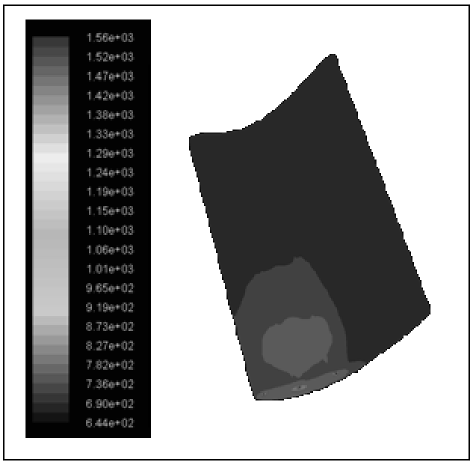

Fig. 6. Temperature distibution on blade surface having Helicoidal cooling duct of Pitch 6 mm and radius of 1 mm

Case 2: Blade cooling with circular duct of 1.5 mm radius and helicoidal duct of same radius with pitch length of 6 mm and 4 mm.

As seen in the Fig. 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 radius 1.5 mm instead of 1 mm as seen from the Fig. 8. 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 4 mm and 6 mm it could be easily seen that there is significant change with respect to minimum and maximum possible surface temperature drop along the span length of the blade.

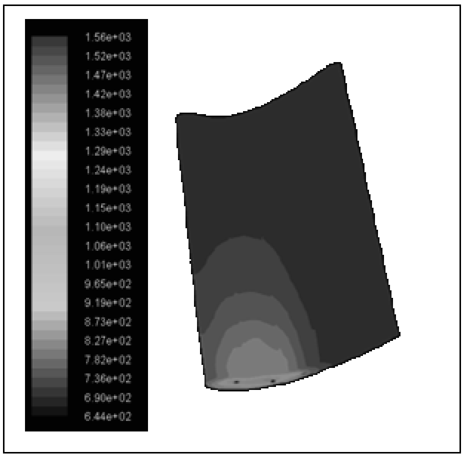

Fig. 7. Temperature distibution on blade surface having Helicoidal cooling duct of Pitch 4 mm and radius of 1 mm

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.

An interesting observation that can be made from Fig. 8 and 9 is that for helicoidal cooling ducts there is a minor fluctuation of temperature and heat flux over the span length of the blade. This is because of helicoidal nature of the cooling duct itself, in which the duct passage 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 sides. 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 and heat flux profile as shown in the Fig. 8 and 9.

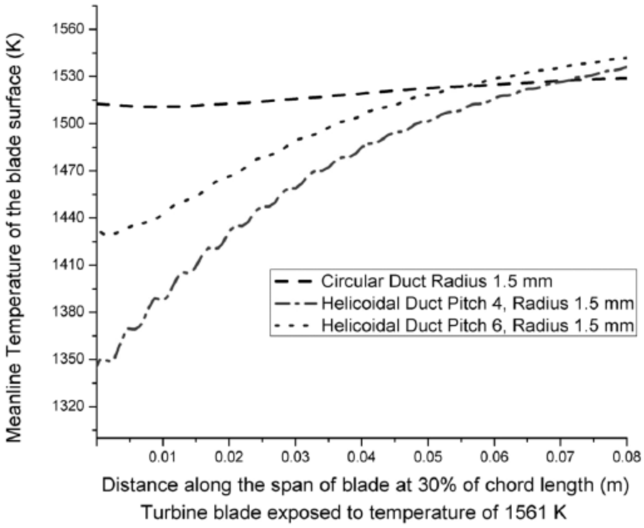

Fig. 8. Temperature variation along the span length of the blade

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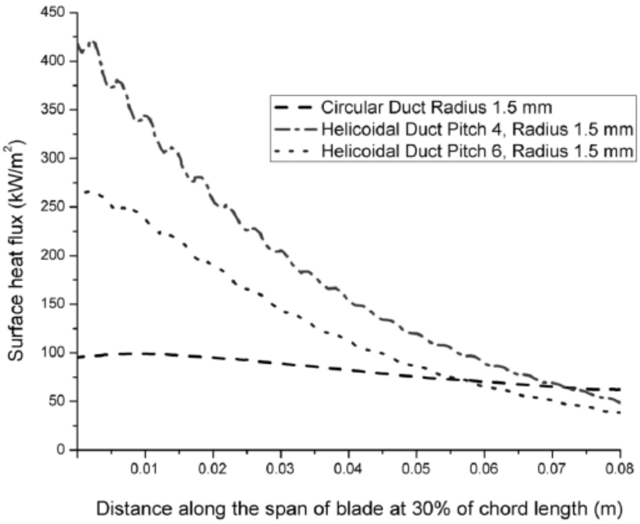

Fig. 9. Surface heat flux variation over the span of the blade

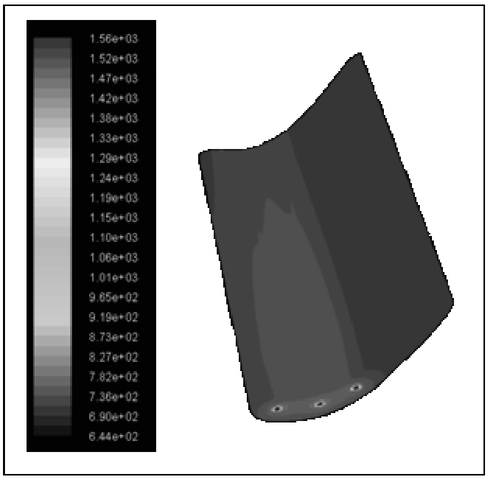

Fig. 10. Temperature distibution on blade surface having Circular cooling duct of radius 1.5 mm

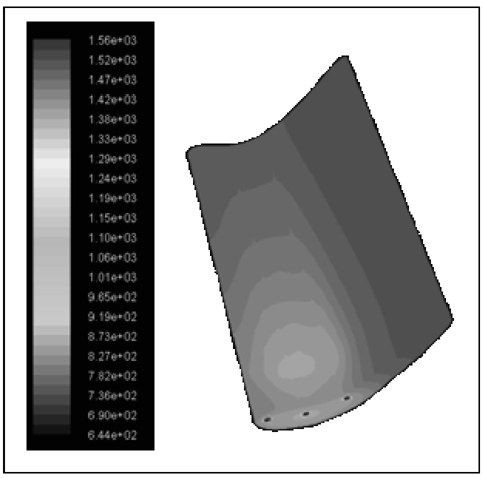

Fig. 11. Temperature distibution on blade surface having Helicoidal cooling duct of Pitch 6mm and radius of 1.5mm

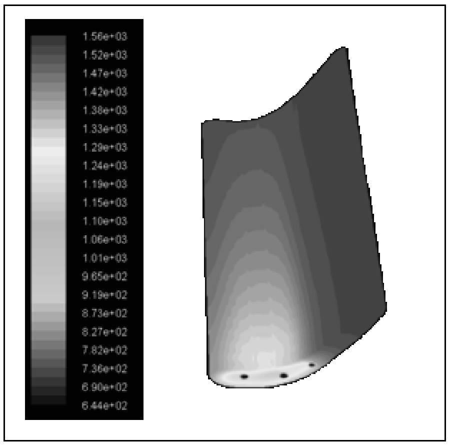

Fig. 12. Temperature distibution on blade surface having Helicoidal cooling duct of Pitch 4mm and radius of 1.5mm

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.

V. CONCLUSION

From present numerical analysis the following conclusions are derived.

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

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