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HEAT TRANSFER IN COOLED AERO-DERIVATIVE TURBINE BLADE: A NUMERICAL ANALYSIS

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Abstract
Aero-derivative gas turbines have found extensive applications as mechanical drives and in medium-sized utility power plants. It has a higher efficiency due to its high pressure and temperature operations; hence, the need for proper cooling techniques to achieve the required creep life and attain reliability. In this paper, the heat transfer in a cooled aero-derivative gas turbine blade is determined numerically using the Alternating Direction Implicit (ADI) scheme of Computational Fluid Dynamics. The convective heat transfer coefficient of the governing Newton's law of cooling equation is the basis. A solver was developed for the heat transfer problem based on the selected boundary conditions and designed cooling parameters of the GE PGT25+ aero-derivative gas turbine to obtain the temperature distribution within a cooled blade for 30 minutes in-service operation. There is no significant change in the temperature profiles across the nodal points, varying between 90°C – 600°C. The temperatures within the blade are significantly constant throughout the operating time of the turbine blade, inferring that there was effective heat transfer from the blades to the cooling air since the temperature variation did not exceed the melting point of the blade material. The ADI strategy is, therefore, suitable for heat transfer design computations for complex systems like the gas turbine engine.

Keywords: Aero-derivative, Blades, Cooling, Heat transfer, Temperature

1.0 INTRODUCTION
Aero-derivative gas turbines are engines derived from aircraft gas turbines. They have found extensive applications on offshore platforms and in petrochemical industries as mechanical drives, and in medium-sized utility power plants of up to 50MW capacity; this is mainly due to its lower maintenance, its high heat-recovery potentials, as well as its higher efficiency. Their compactness, lightweight and multiple fuel application make gas turbine a natural power plant for offshore platforms (Boyce, 2002). The most widely used alloys for gas turbine blades are the Inconel-738 (IN-738) and the GTD-111, though other alloys of varying composition exist; and the safe blade temperature is about 650°C which would minimize oxidation and thermal stresses to the blade (Boyce, 2002). During gas turbines operations, hot gases from the combustion system flow past the rotating turbine blades, expanding in the process; consequently, the components in the hot section of the gas turbine subjected to these high temperatures often require

cooling (Arnalet al, 2007). The destructive processes affecting gas turbine vanes and blades begins with the destruction of the aluminium coating, resulting in the parent material of vanes and blades exposed to the direct thermal and chemical effects of the exhaust gases; which will lead to the overheating of that material as shown in Plate I. Hence the condition of the blades is of crucial importance to the reliability and lifetime of the entire turbine (Józef & Pawlak, 2011).

(a) (b)

Plate I: Gas turbine blades: (a) – a new turbine blade, (b) – the in-service damaged one

Source:(Józef & Pawlak, 2011).

Modern gas turbines operate at very high temperatures to increase their efficiency and performance; these temperatures may exceed the material melting temperature of the turbine blades (Balajiet al, 2016) making the study of the heat transfer within the gas turbine blade essential.

Gas Turbine Blade Cooling

Gas turbine blades are cooled with air removed from the compressor stage. There are three major internal cooling regions in a turbine blade. The film cooling region in the leading edge cooled by jet impingement, the pressure and suction surfaces in the middle section cooled by serpentine rib roughened passages with local film cooling, and the blade tip region cooled by pin fins with trailing edge injection (Han, 2004). Several types of research have been done to improve blade cooling and heat transfer dynamics .Guntur et al, (2010) simulated the heat/mass transfer characteristics in 45º and 90° angled ribbed channel cases with bleed flow. They found that the Reynolds Stress Model (RSM) was better in predicting heat transfer in the channels than the k-ω and k-ε models, for the designed conditions and geometries of channels. The study conducted on the internal cooling performance of dendritic cooling systems revealed that there were superior internal heat transfer coefficients and reduced exit blowing ratio for a given pressure margin through the cooling passage compared to typical designs. CFD models were employed to analyse the flow field in the cooling passages (Batstone et al, 2012). Scherhag et al, (2016), investigated numerically and experimentally an efficient cooling method for turbine liner segments based on pulsating impinging jets; they discovered the formation of distinct toroidal vortices separating from the nozzle indicating a potential heat transfer improvement. The development of an analytical model for the prediction of gas turbine blade cooling, calculated the distribution of coolant Mass Flow Rate (MFR) and metal temperatures of a turbine blade using the mass and energy balance equations at given external and internal boundary conditions (Chowdhury et al, 2017). Another development of a combined thermodynamic model for blade cooling based on mass/energy balances and heat transfer correlations for predicting a one-dimensional temperature profile on the blade external surface, as well as the required cooling air flow rates to prevent turbine blade material from creep (Masci & Sciubba, 2018). Ba et al, (2018), also developed an aero-thermal coupled blade cooling model capable of predicting blade surface temperature distribution and internal coolant flow conditions for initial blade cooling design with a minimal amount of input information by considering the interaction effect between the main flow and the coolant. This paper studies the heat transfer of a cooled aero-derived gas turbine blade using the Alternating Direction Implicit (ADI) strategy, employing the impingement cooling method.

2.0 METHODOLOGY

The Alternating-Direction Implicit (ADI) scheme was used to investigate the heat transfer in the cooled aero-derivative gas turbine blade considering the convective heat transfer rate governed by the Newton's law of cooling in equation (1):

$$q = Ah_g(T_g - T_w) \tag{1}$$

Where: q represents the heat transfer rate, h_g is the hot gas heat transfer coefficient (W/m^{2}K), A is the heat transfer surface area (m^2), $T_g$is the temperature of the hot gas and $T_w$is the wall metal temperature. The ADI scheme is based on the approximated and modified Peaceman-Rachford strategy by (Chapra & Canale, 2015) into two steps to give equations (2) and (3):

$$\frac{T_{i,j}^{n+\frac{1}{2}} - T_{i,j}^{n}}{\frac{\Delta t}{2}} = h_g\left[\frac{T_{i+1,j}^{n} - 2T_{i,j}^{n} + T_{i-1,j}^{n}}{(\Delta x)^2} + \frac{T_{i,j+1}^{n+\frac{1}{2}} - 2T_{i,j}^{n+\frac{1}{2}} + T_{i,j-1}^{n+\frac{1}{2}}}{(\Delta y)^2}\right] \tag{2}$$

and

$$\frac{T_{i,j}^{n+1} - T_{i,j}^{n+\frac{1}{2}}}{\frac{\Delta t}{2}} = h_g\left[\frac{T_{i+1,j}^{n+1} - 2T_{i,j}^{n+1} + T_{i-1,j}^{n+1}}{(\Delta x)^2} + \frac{T_{i,j+1}^{n+\frac{1}{2}} - 2T_{i,j}^{n+\frac{1}{2}} + T_{i,j-1}^{n+\frac{1}{2}}}{(\Delta y)^2}\right] \tag{3}$$

The main idea of implementing the ADI scheme is to split the heat transfer computations in two steps, with the first step being an implicit method in the x-direction and an explicit method in the y-direction, producing an intermediate solution for time; while the second step involves an implicit method in the y-direction and an explicit method in the x-direction (Araújo et al, 2014). A MATLAB solver was developed considering a square grid mesh of 25 x 25 mm dimension with 5 mm spacing resulting into a mesh with six nodes at $\Delta t = 0$ as illustrated in figure 2. Computations for the heat distribution on the nodes along the j direction constitutes the first half step at t + 1/2 of the ADI scheme while the computations along the *i* direction constitute the second time step at t + 1. The dirichlet boundary conditions at $\Delta t = 0$ would assume that the temperature T will also be Zero. Other boundary conditions as the time increases are the turbine inlet temperature of 1,400^{o}C, the temperature of the cooling air of 350^{o}C and the temperature of the cooling air exiting the blade of 500^{o}C. Other parameters considered for the development of the MATLAB solver were adapted from the GE PGT25+ engine and are summarized as followed: Turbine Inlet Temperature (TIT) = 1400^{o}C (1673K); Turbine Exit Temperature (TET) = 500^{o}C (773K); Compressed air Cooling Temperature = 350^{o}C (623K); Kinematic viscosity of the compressed air = 54.85 x 10^{-6} m^2/s; Pressure ratio for the hot gas P_g = 21.5 bar; Heat Rate for the engine q = 8, 751 kJ/kWhr (2.43 kJ/kWs); Diameter of cooling Passages D = 0.00125m; Mass flow rate of cooling air = 435 kg/s and Reynolds number = 60000 (Siemens, 2019).

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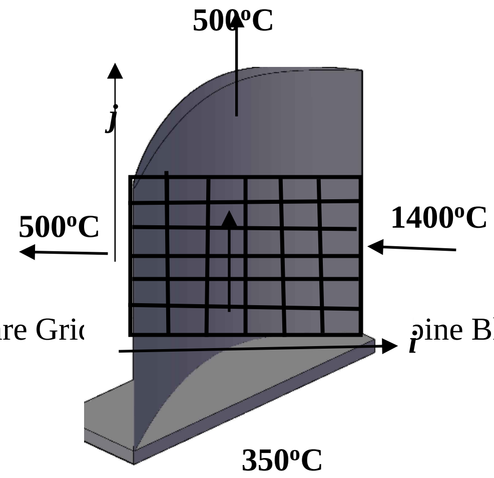

Figure 1: The Square Gri ine Blade Model

3.0 RESULTS AND DISCUSSIONS

The results for the MATLAB solver yielded the temperature distributions at the several nodal points in the turbine blade as shown in table 1 to table 6 for corresponding times of 5, 10, 15, 20, 25 and 30 minutes respectively of the turbine blade in service.

Table 1: Temperature distributions within the nodes after 5 minutes of Turbine blade in service

At t = 300s						
	N1	**N2**	**N3**	**N4**	**N5**	**N6**
T1	285.965	202.304	175.81	180.037	223.439	346.127
T2	229.734	123.763	90.2047	95.5588	150.534	305.939
T3	218.845	107.297	71.9723	77.6082	135.477	299.061
T4	231.521	119.973	84.6484	90.2843	148.153	311.737
T5	293.114	187.144	153.585	158.939	213.914	369.319
T6	526.81	443.149	416.655	420.882	464.284	586.972

Table 2: Temperature distributions within the nodes after 10 minutes of Turbine blade in service

At t = 600s						
	N1	**N2**	**N3**	**N4**	**N5**	**N6**
T1	285.965	202.304	175.811	180.038	223.439	346.127
T2	229.734	123.764	90.2052	95.5593	150.534	305.939
T3	218.846	107.298	71.9729	77.6088	135.477	299.061
T4	231.522	119.974	84.6489	90.2848	148.153	311.737

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T5	293.115	187.144	153.586	158.94	213.915	369.319
T6	526.81	443.149	416.656	420.883	464.284	586.972

Table 3: Temperature distributions within the nodes after 15 minutes of Turbine blade in service

At t = 900s						
	N1	**N2**	**N3**	**N4**	**N5**	**N6**
T1	285.965	202.304	175.811	180.038	223.439	346.127
T2	229.734	123.764	90.2054	95.5595	150.534	305.939
T3	218.846	107.298	71.973	77.6089	135.477	299.061
T4	231.522	119.974	84.6491	90.285	148.153	311.737
T5	293.115	187.144	153.586	158.94	213.915	369.32
T6	526.81	443.149	416.656	420.883	464.284	586.972

Table 4: Temperature distributions within the nodes after 20 minutes of Turbine blade in service

At t = 1200s						
	N1	**N2**	**N3**	**N4**	**N5**	**N6**
T1	285.965	202.304	175.811	180.038	223.439	6.127
T2	229.734	123.764	90.2055	95.5596	150.534	305.939
T3	218.846	107.298	71.9731	77.609	135.477	299.061
T4	231.522	119.974	84.6492	90.2851	148.153	311.737
T5	293.115	187.144	153.586	158.94	213.915	369.32
T6	526.81	443.149	416.656	420.883	464.284	586.972

Table 5: Temperature distributions within the nodes after 25 minutes of Turbine blade in service

At t = 1500s						
	N1	**N2**	**N3**	**N4**	**N5**	**N6**
T1	285.965	202.304	175.811	180.038	223.439	346.127
T2	229.735	123.764	90.2055	95.5596	150.535	305.939
T3	218.846	107.298	71.9732	77.6091	135.477	299.061
T4	231.522	119.974	84.6492	90.2851	148.153	311.737
T5	293.115	187.144	153.586	158.94	213.915	369.32
T6	526.81	443.15	416.656	420.883	464.284	586.972

Table 6: Temperature distributions within the nodes after 30 minutes of Turbine blade in service

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At t = 1800s						
	N1	**N2**	**N3**	**N4**	**N5**	**N6**
T1	285.965	202.304	175.811	180.038	223.439	346.127
T2	229.735	123.764	90.2056	95.5597	150.535	305.939
T3	218.846	107.298	71.9732	77.6091	135.477	299.061
T4	231.522	119.974	84.6493	90.2852	148.153	311.737
T5	293.115	187.144	153.586	158.94	213.915	369.32
T6	526.81	443.15	416.656	420.883	464.284	586.972

Graphical representations of the temperature versus the nodal points obtained for 5, 15 and 30 minutes in service are shown in figures 2, 3 and 4 respectively. From the graphs, it can be observed that there is no significant change in the temperature profiles across the nodal points from 0s to 1800s (30 minutes) in service of the turbine blade and the temperatures within the blade is significantly constant throughout the operating time of the turbine blade, signifying the cooling of the blades when the bled air is passed through it; the temperatures within the nodes varied between 90^{o}C – 600^{o}C inferring that heat transfer from the cooling air is effective as it maintains the temperatures within blade to not more than 650^{o}C.

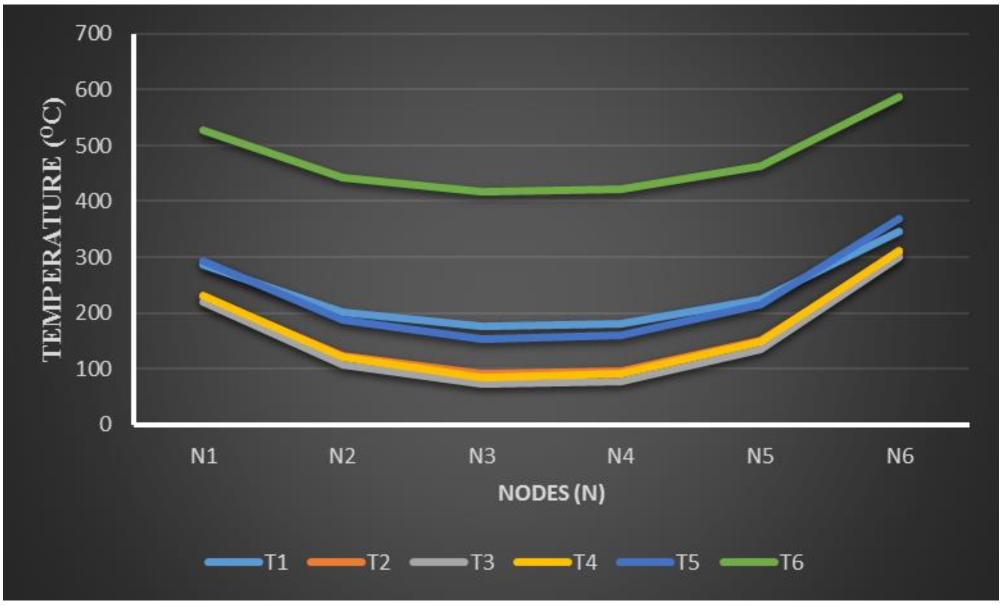

Figure 2: Temperature distributions within the nodes at 300s in service

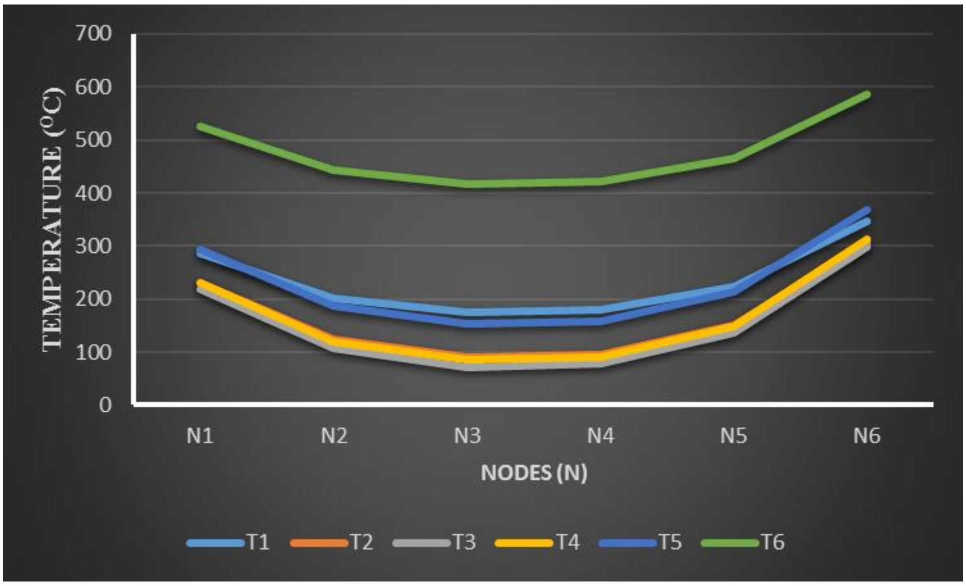

Figure 3: Temperature distributions within the nodes at 900s in service

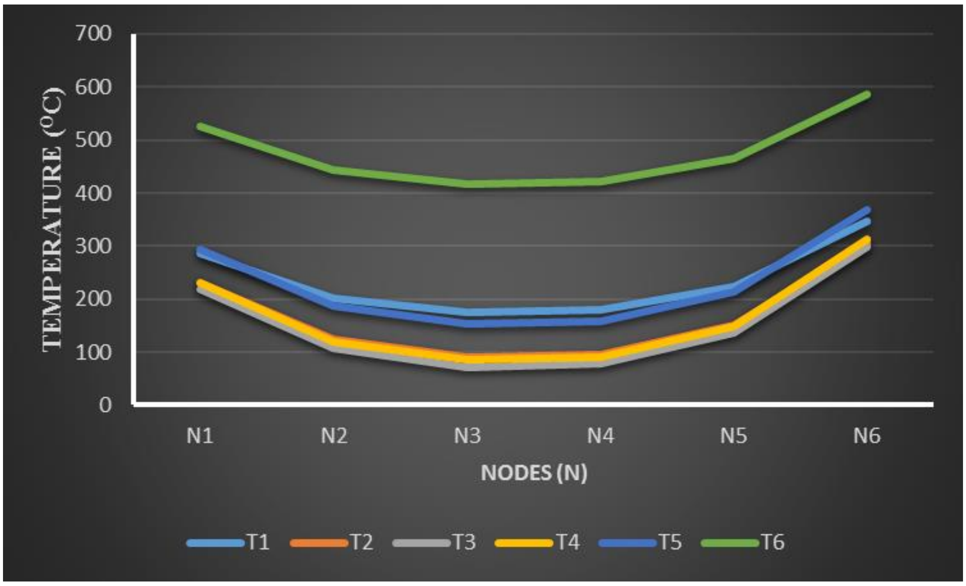

Figure 4: Temperature distributions within the nodes at 1800s in service

4.0 CONCLUSION

The heat transfer in a cooled aero-derivative gas turbine blade is determined numerically, based on the convective heat transfer process employing the Alternating Direction Implicit (ADI) scheme from computational fluid dynamics (CFD), considering design and operating parameters for a High Pressure (HP) turbine blade such as Turbine Inlet Temperature (TIT), Turbine Exit Temperature (TET), Compressed air cooling temperature, Kinematic viscosity, Pressure ratio, Diameter of cooling holes, Mass flow rate of cooling air and Reynolds number. The design and operating conditions for GE PGT25+ gas turbine engine were applied. A

MATLAB solver was developed to compute the temperature distribution within the meshed section of the blade model considering existing boundary conditions around the blade. There was effective heat transfer from the blades to the cooling air since it maintained the temperatures within the blade to acceptable limits not exceeding the melting point of the blade material. The ADI strategy is, therefore, suitable for heat transfer design computations for complex systems like the gas turbine engine. The CFD technique could, therefore, be used to study and solve heat transfer design problems

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