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Design and Thermo-Structural Analysis of Gas Turbine Rotor Blade

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ABSTRACT: *As airfoil is fundamental building block of blade, the paper has selected GOE 798 profile and it has been provided acceptable cambering. Various gas angles and velocities first calculated for mean radius and subsequently other radial sections of blade. The aspect ratio calculated for the blade is 1.35 and throat opening calculated is 0.02136m. The rotor blade angles calculated vary from root to tip section of blade which also gives varying velocity triangles. To have uniform velocity triangles from root to tip the designed blade possesses small amount of twist. Nickel based alloy RENE 41 has been used as blade material. The tensile centrifugal force and gas bending force calculated analytically and used for structural stress simulation. For the assumed turbine inlet temperature* **1400k***, the rotor inlet temperature analytically determined as* **1267k** *and steady state temperature distribution over the blade has been obtained through simulation. The paper also talks about cylindrical shape cooling passage running from blade root to tip and calculates coolant temperature along height theoretically for coolant of* **600k** *at root. The paper ends with the comparison of temperature across cross section of blade at different height. Four common points across cross section of blade have been taken namely a, b, c and d and temperature distribution along height at respective points has been estimated and compared.*

Keywords: *Turbine inlet temperature, rotor inlet temperature, structural analysis, thermal analysis, blade cooling*

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1. INTRODUCTION

The paper talks about geometrical design of first stage gas turbine rotor blade in axial flow case for open cycle gas turbine engine. The design is followed by static state structural and thermal analysis. The paper has also included construction of simple cylindrical cooling passages design running from blade root to tip and steady state thermal analysis of model.

It is stated that special series that are used for turbine airfoils, NACA 4412, NACA 4415, T6, C7 and RAF 27 [1]. The paper has used GOE 798 [2] profiles as blade profile with suitable circular arc bending. The profile has maximum thickness to chord ratio of 0.2. Dr.Ch S Naga Prasad [3], Nickel based turbine blade was designed with four holes and six holes and he concluded blade with six holes is better as thermal flux had increased with increasing number of holes. RDV Prasad, G Narasa Raju, MSS Srinivasa Rao, N Vasudeva Rao [4], they had examined optimum temperature over gas turbine blade made up of Inconel 1155 and Inconel 718 with number of holes (5, 9 and 13).With selection of profile the paper will include three dimensional design for first stage gas turbine rotor blade with detail theoretical calculation. RENE 41 Nickel based supper has been used as blade material and the blade is analyzed for structural effects. Further the blade is also analyzed for thermal effects. In addition the paper includes design of 22 cylindrical cooling passages from root to tip and temperature distribution along height has been estimated by passing coolant of 600k from root of the blade.

2. ROTOR BLADE DESIGN

The ideal assumptions that were made to design blades were no other components carry pressure losses, mass flow rate is constant and fluid is same through cycle. The various blade absolute and relative values of angles and velocities will have triangle which is called velocity triangle. The triangle was first designed at mean of blade section and subsequently at other radial sections. In Figure 2.1 Alpha is notation of absolute angle and beta is notation of relative angle in velocity triangle of angles. All flow angles are measured from axial direction. The alphabet c represents absolute velocity and w represents relative velocity. Alphabet p represent pressure. It shows the axial flow velocity triangle for first stage turbine. Initially the gas enters in stator blade rows with static pressure P_1, static temperature T_1, and flow absolute velocity C_1 at an absolute angle α_1. C_{x_1}, axial velocity, has been assumed constant.

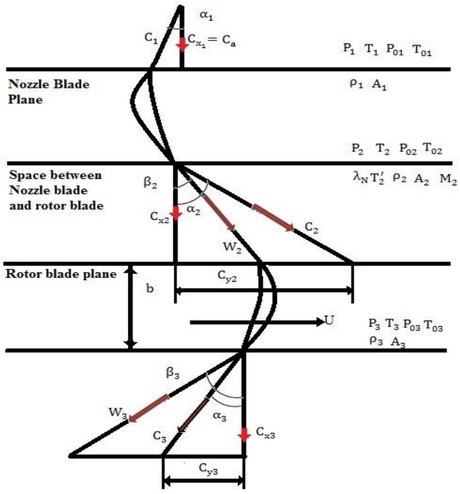

Figure 2.1: Velocity Triangle for axial flow turbine

'x', subscript for axial direction and 'y', subscript for tangential direction of blade. Station one represents nozzle blade plane, station two represents space between nozzle blade and rotor blade and station three represents rotor blade plane.

The flow then expanded in stator blade and leaves it by P_2, , T_2, and C_2 with and increased C_2 at an angle α_2. P_{01}, T_{01}, P_{02}, T_{02} Corresponding stagnation absolute components. Velocity relative to blade inlet at an angle β_2 is W_2. On further expansion in rotor blade or being deflected further the gas leaves P_3, , T_3 and C_3. Corresponding stagnation component for absolute components are P_{03}, T_{03}. The relative velocity further expanded to W_3, more than W_2. Absolute velocity C_2 decreases to C_3. C_3 , leaves at an angle α_3 and W_3 leaves at an angle β_3. α_3 , is swirl angle and C_3 will be next absolute velocity for next stator. In multi stage turbine C_1 and α_1 probably equal to C_3 and α_3 so that same successive blade shape can be used in successive stage. $C_{y2} + C_{y3}$ Represents the change in whirl component or tangential component of momentum per unit mass flow which produces useful torque. The change in axial component $C_{x2} + C_{x3}$ produce axial thrust on rotor [5].

Table i: Input parameters for blade design [5]

S.N	Parameter	Value	Unit	Description
1	Isentropic compression (η_{isen})	0.9		
2	Flow coefficient (ϕ)	0.85		It is in range (0.8-1.0) satisfactory
3	Shaft rotational speed (N)	9600	RPM	It is compressor matching data which is fixed (Assumed).
4	Mean blade speed (U)	350		m/s Assumed within limit to reduce stressing in blade
5	Total head inlet Temperature (T_{01})	1400	k	
6	Total head inlet Pressure (P_{01})	10*10^5	N/m^2	Assumed
7	Pressure ratio(P_{01}/P_{03})	1.6		Assumed
8	Mass flow rate ($\dot{m}_{gas}$)	75	kg/s	Assumed
9	Nozzle loss Coefficient (λ_N)	0.1		Assumed
10	Temperature drop (ΔT)	150	k	

11	c_p	1.2005	$kJ/kg-k$	At 1400k
12	c_υ	0.9134		At 1400k
13	R	0.287	$kJ/kg-k$	Gas constant
14	γ	1.314		Constant
	$\frac{\gamma}{\gamma-1}$	$4.18\approx 4$		
15	Ambient temperature (T_a)	293	k	
16	Ambient pressure (P_a)	101325	N/m^2	

The rotational speed is matched speed with compressor and it is fixed. One could not modify or change once compressor is designed. The pressure loss in turbine blades are balanced by pressure increase by compressor stages design to make turbine blade design easy. The blade height varies with respect to successive stage but the profile is repeated as it is assumed $C_{x3} = C_{x2}, C_1 = C_3, \alpha_3$, is rotor blade exit angle. It is also called swirl or whirl component [5].

2.1. Blade loading, gas angles and swirl angle at mean [5]

$$\psi = \frac{2 * c_p * \Delta T_{0s}}{(U)^2} \tag{1}$$

The swirl angle increases the losses in jet pipe. The swirl angle less than 20 degree is desirable and beyond this the losses would be high. As jet is efficient for straight our design effort would be to minimize whirl component. There be no loss for $\alpha_3 = 0°$

$$\tan\alpha_3 = \tan\beta_3 - \frac{1}{\phi}, \tan\beta_3 = 1.18 \tag{2}$$

To maintain calculated temperature drop coefficient value of 2.94 and less value of α_3, it might be necessary to have degree of reaction less than 0.5 at mean diameter.

$$\tan\beta_3 = \frac{(0.5\psi + 2\Lambda)}{2 * \phi}, \Lambda = 0.265, \text{i.e. } 26.5\% \text{ at mean diameter} \tag{3}$$

The reaction value is low at mean radius and it probably would have negative reaction at root of blade, this imply high losses due to recompression of gas in rotor blade, so another swirl angle is chosen less than 20° as $\alpha_3 = 10°$

$$\tan\alpha_3 = \tan\beta_3 - \frac{1}{\phi}, \tan\beta_3 = 1.18 + 0.17 = 1.35$$

$$\tan\beta_3 = \frac{(0.5\psi + 2\Lambda)}{2 * \phi}, \Lambda = 0.4125, \text{moderte degree of reaction}$$

For $\alpha_3 = 12°$

$$\tan\alpha_3 = \tan\beta_3 - \frac{1}{\phi}, \tan\beta_3 = 1.389$$

$$\tan\beta_3 = \frac{(0.5\psi + 2\Lambda)}{2 * \phi}, \Lambda = 0.45, \text{satisfactory degree of reaction}$$

$$\tan\beta_3 = 1.389, \beta_3 = 54.25°$$

$$\tan\beta_2 = \frac{(0.5\psi - 2\Lambda)}{2 * \phi}, \beta_2 = 18.54°$$

$$\tan\alpha_2 = \tan\beta_2 + \frac{1}{\phi}, \alpha_2 = 56.51°$$

So far the gases angles at mean are

$$\alpha_2 = 56.51°, \beta_2 = 18.54°, \beta_3 = 54.25°, \alpha_3 = 12°$$

2.2. Velocities

Axial velocity ($C_{x2} = C_{x3}$)

$$U * \emptyset, 297.5 \text{ m/s}$$

Absolute velocity ($C_{2,} C_3 = C_1 = C_{x1}$)

$$C_{2,} = \frac{C_{x2}}{\cos\alpha_2}, 539.15 \text{ m/s } \textit{and } C_3 = \frac{C_{x3}}{\cos\alpha_3}, 304.14 \text{ m/s}$$

Relative velocity (W_2, W_3)

$$W_2 = \frac{C_{x2}}{\cos\beta_2}, 313.78\,\mathrm{m/s}\ \ and\ W_3 = \frac{C_{x3}}{\cos\beta_3}, 509.2\,\mathrm{m/s}$$

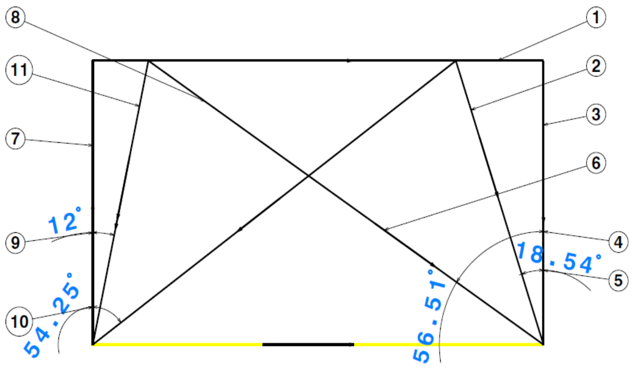

Figure 2.2: Velocity Triangle at mean radius

Table ii: Parameters Nomenclature

S.N	Symbol	Description
1	U	Blade speed
2	W_2	Relative velocity at station2
3	C_{x2}	Axial velocity at station 2
4	β_2	Rotor blade inlet angle
5	α_2	Stator blade exit angle
6	C_2	Absolute velocity at station 2
7	C_{x3}	Axial velocity at station 3
8	W_3	Relative velocity at station 3
9	α_3	Swirl angle
10	β_3	Rotor exit velocity
11	C_3	Absolute velocity at station 3

2.3. Theoretical calculation for Space between rotor and stator blade [5]

There is no work done in nozzle so $T_{01} = T_{02}$

The temperature equivalent of outlet velocity for nozzle is

$$T_{02} - T_2 = \frac{{C_2}^2}{2C_p}$$

The ideal temperature would be less due to friction loss in nozzle.

$$T_2' = T_2 - \lambda_N \frac{{C_2}^2}{2C_p}$$

As expansion is isentropic static pressure P_2 can be found from relation

$$\frac{P_{01}}{P_2} = \left(\frac{T_{01}}{T'_2}\right)^{\frac{\gamma}{\gamma-1}}$$

The critical pressure value for [5] $\gamma = 1.314$

$$\left(\frac{\gamma+1}{2}\right)^{\frac{\gamma}{\gamma-1}}, = 1.79$$

$$\frac{P_{01}}{P_c} = \left(\frac{T_{01}}{T'_2}\right)^{\frac{\gamma}{\gamma-1}}, = 1.49$$

The actual pressure ratio 1.49 is less then critical pressure ratio 1.79 so the nozzle are not chocking and pressure in the plane of throat is equal to 6.7 bar.

Density of gas at station two

$$\rho_2 = \frac{P_2}{RT_2}$$

Area of plane

$$A_2 = \frac{\dot{m}}{\rho_2 C_{x2}}$$

Checking for Mach number at inlet of rotor blade weather it exceed 0.75 value or not, if it exceed 0.75 Mach value at inlet of rotor blade at mean diameter, there could be more losses as shock wave is formed in blade passage [5].

$$M_2 = \frac{W_2}{\sqrt{\gamma R T_2}}$$
$$= 0.45 \text{ less than } 0.75$$

2.4. Calculation for nozzle blade plane [5]

Temperature equivalent of inlet and outlet energy is

$$\frac{(C_1)^2}{2C_p}$$

Static temperature

$$T_1 = T_{01} - \frac{(C_1)^2}{2C_p}$$

$$\frac{P_{01}}{P_1} = \left(\frac{T_{01}}{T_1}\right)^{\frac{\gamma}{\gamma-1}}, P_1 = 8.95 \text{ bar}$$

$$\rho_1 = \frac{P_1}{RT_1}$$

$$A_1 = \frac{\dot{m}}{\rho_1 C_{x1}}$$

2.4.1. Calculation for Rotor blade plane [5]

Temperature of outlet of blade or first stage

$$T_{03} = T_{01} - \Delta T_{0s}$$

$$T_3 = T_{03} - \frac{(C_3)^2}{2C_p}$$

$$\frac{P_{03}}{P_3} = \left(\frac{T_{03}}{T_3}\right)^{\frac{\gamma}{\gamma-1}}, P_3 = 3.5 \text{ bar}$$

$$\rho_3 = \frac{P_3}{RT_3}$$

$$A_3 = \frac{\dot{m}}{\rho_3 C_{x3}}$$

2.4.2. Calculation for blade height, root radius and tip radius at rotor blade plane

$$U = 2\pi N r_m, r_m = 0.35 \text{ m}$$

Height of blade [5]

$$h = \frac{AN}{U}, \approx 0.12 \text{ m}$$

Root radius

$$r_r = 0.35 - \frac{0.12}{2}$$

Tip radius

$$r_t = 0.35 + \frac{0.12}{2}$$

$$\frac{r_t}{r_r} = \frac{0.41}{0.29}, = 1.41, \text{Accepted as it slightly exceeds range } (1.2 - 1.4)$$

2.4.3. Calculation of blade angles at root radius and tip radius at rotor blade plane [5]

At root radius

$$\frac{r_m}{r_r} = 1.20, \frac{r_r}{r_m} = 0.83$$

Nozzle outlet gas angle at root radius

$$\tan \alpha_{2r_r} = \frac{r_m}{r_r} \tan \alpha_{2m}$$

Rotor inlet angle at root is

$$\tan\beta_{2r_r} = \frac{r_m}{r_r}\tan\alpha_{2m} - \frac{1}{\phi}$$

Rotor outlet at root is

$$\tan\beta_{3r_r} = \frac{r_m}{r_r}\tan\alpha_{3m} + \frac{r_r}{r_m}\frac{1}{\phi}$$

Swirl component at root radius

$$\tan\alpha_{3r_r} = \frac{r_m}{r_r}\tan\alpha_{3m}$$

Absolute velocity ($C_{2,}C_3 = C_1 = C_{x1}$)

$$C_{2,} = \frac{C_{x2}}{\cos 61.12}, C_3 = \frac{C_{x3}}{\cos\alpha_3}, C_{x2} = C_{x3}$$

Relative velocity ($W_{2,}$ W_3)

$$W_2 = \frac{C_{x2}}{\cos 32.36}, W_3 = \frac{C_{x3}}{\cos 50.99}$$

Temperature equivalent at root

$$T_{2r} = T_{02} - \frac{(C_{2r})^2}{2 * c_p}, T'_{2r} = T_{2r} - \lambda_N \frac{(C_{2r})^2}{2 * c_p}$$

Mach number at root [5]

$$M = \frac{W_2}{\sqrt{\gamma R T_{2r}}}, = 0.51, \text{less than } 0.75$$

At tip radius

$$\frac{r_m}{r_t} = 0.85, \frac{r_t}{r_m} = 1.17$$

Nozzle outlet at tip radius

$$\tan\alpha_{2r_t} = \frac{r_m}{r_t}\tan\alpha_{2m}$$

Rotor inlet at tip radius

$$\tan\beta_{2r_t} = \frac{r_m}{r_t}\tan\alpha_{2m} - \frac{1}{\phi}$$

Rotor outlet at tip radius

$$\tan\beta_{3r_t} = \frac{r_m}{r_t}\tan\alpha_{3m} + \frac{r_t}{r_m}\frac{1}{\phi}$$

Swirl component at tip radius

$$\tan\ \alpha_{3r_t} = \frac{r_m}{r_t}\tan\alpha_{3m}$$

Absolute velocity ($C_{2,}C_3 = C_1 = C_{x1}$)

$$C_{2,} = \frac{C_{x2}}{\cos 52.10}, C_3 = \frac{C_{x3}}{\cos 10.24}, C_{x2} = C_{x3}$$

Relative velocity ($W_{2,}$ W_3)

$$W_2 = \frac{C_{x2}}{\cos 5.97}, W_3 = \frac{C_{x3}}{\cos 57.36}$$

Temperature equivalent at tip

$$T_{2t} = T_{02} - \frac{(C_{2t})^2}{2 * c_p}, T'_{2t} = T_{2r} - \lambda_N \frac{(C_{2t})^2}{2 * c_p}$$

Mach number at tip

$$M = \frac{W_2}{\sqrt{\gamma R T_{2t}}}, = 0.32, \text{less than } 0.75$$

Table iii: Calculated blade angles

Stator blade			
	Root radius	Mean radius	Tip radius
Outlet (α_2)	61.12°	56.51°	52.10°
Rotor blade			
Inlet (β_2)	32.36°	18.54°	5.97°

Outlet (β_3)	50.99°	54.25°	57.36°
Swirl (angle) (α_3)	14.30°	12°	10.24°

Table iv: Calculated thermodynamic parameters for blade design

Parameters	Values			Unit
	Station one	Station two	Station three	
T_1	1361.5			k
P_1	8.95			bar
ρ_1	2.29			kg/m^3
A_1	0.11			m^2
T_2		1279		k
T'_2		1267		k
P_2		6.7		bar
ρ_2		1.82		kg/m^3
A_2		0.14		m^2
T_3			1211.5	k
T_{03}			1250	k
P_3			3.5	bar
ρ_3			1	kg/m^3
A_3			0.25	m^2

As blade angles are calculated in different radial section, the profile design of blade is followed by providing necessary bents to the selected GOE 798 airfoil. The profile is bent in such a way that it provides $\cos^{-1}\left(\frac{o}{s}\right) \approx \beta_3$ at mean radius, 'o' is opening of throat and's' is pitch or blade spacing [6]. Nickel based isotropic material alloy that is RENE 41 is applied to the blade which possesses following properties. Nominal composition of RENE 41 alloys consists of 56Ni19Cr10.5Co9.5Mo3.2Ti1.7Al0.01Zr0.008CO0.015B [7].

Table v: RENE 41 alloy properties [7] [8] [9]

Material properties	Value	Unit
Density	8248.6	kg/m^3
Modulus of elasticity	$2.1994*10^{11}$	N/m^2
Poisson ratio	0.31	
Thermal expansion coefficient	$1.1934*10^{-5}$	/k
Heat capacity	460.55	J/kg. k
Thermal conductivity	107.31	w/m. k
Yield strength	$1.0618*10^{9}$	N/m^2
Melting point	1645	k
Shear modulus	$8.3948*10^{10}$	N/m^2
Bulk modulus	$1.9293*10^{11}$	N/m^2

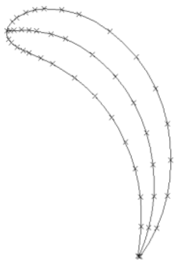

Figure 2.3: Profile with circular cambering

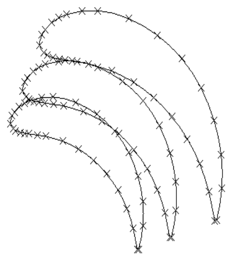

Figure 2.4:Blade profile at root, mean and tip section in ascending order

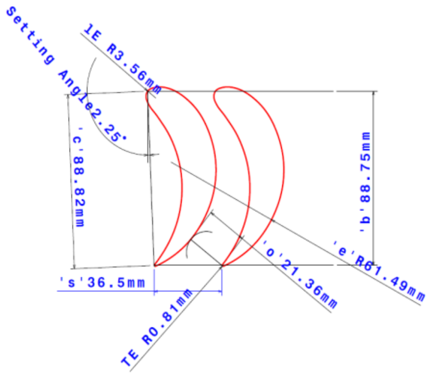

Figure 2.5: Throat opening geometrical estimation

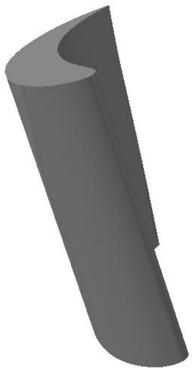

Figure 2.6: Three dimensional Nickel based blade

Table vi: Geometrical values obtained from blade profile

S.N	Parameter	Value	Unit	
1.	Profile chord (c)	0.08882	m	From profile (Figure 2.5)
2.	Axial chord (b)	0.08875	m	From profile (Figure 2.5)
3.	Thickness (t), max	0.0178	m	As$\left(\frac{t}{c}\right) = 0.20$
4.	Leading edge radius (R_{le})	0.00356	m	From profile i.e. 0.0975s [6]
5.	Trailing edge radius (R_{te})	0.81	mm	
6.	Suction surface curvature (e)	0.06149	m	From profile (Figure 2.5)
7.	Pitch (s)	0.0365	m	As permissible $0.25 < \left(\frac{s}{e}\right) < 0.625$, $\left(\frac{s}{e}\right) = 0.593$ permissible [6]
8.	Throat opening (o)	0.02136	m	Geometrical interpretation (Figure ***2.5***)
9.	Setting angle (λ)	2.75°		(calculated as $\cos^{-1}\left(\frac{b}{c}\right) = 2.27°$) from profile [6]
10.	Flow coefficient (ϕ)	0.85		Adopted value of range 0.8 to 1.0 [5]
11.	Height of blade (h)	0.12	m	
12.	Radius at root (r_r)	0.29	m	
13.	Radius at tip (r_t)	0.41	m	
14.	Radius at mean	0.35	m	

15.	Radius ratio $\left(\frac{r_t}{r_r}\right)$	1.41	It slightly beyond range (1.2-1.4) [5]
16.	Aspect ratio $\left(\frac{h}{c}\right)$	1.35	
17.	$\cos^{-1}\frac{o}{s}$	54.25°	54.18° ≈54.25° (From calculation)
18.	N_R	60	Number of rotor blade [5] $N_R = \frac{2\pi r_m}{s}$

3. STRUCTURAL AND THERMAL ANALYSIS

As blade is subjected to very high thermal and mechanical stresses, they provide centrifugal forces and thermal gradient. The centrifugal force mainly provided by shaft rotational speed and thermal gradient is provided by hot gas flow. This paper is mainly based on steady state analysis. The finite element model was generated by meshing of blade.

3.1. Structural analysis [5] [9] [10]

3.1.1. Centrifugal tensile force

This kind of stress mainly due to shaft rotational speed, the centrifugal forces due to rotational shaft, mathematically it is calculated as [11]

$$F = \rho_b * A * \omega^2 \left(\frac{r_t^{\,2} - r_r^{\,2}}{2}\right), 388714\ N \quad (4)$$

3.1.2. Gas bending force

The force arising from change in angular momentum in tangential direction, which produces useful torques, it also produces gas bending moment about axial direction. Theoretically it is calculated as [10]
Tangential force $\dot{m} * (C_{y2} + C_{y3})$ equivalent with

$$\dot{m} * C_a(\tan\alpha_2 + \tan\alpha_3),\ 33997\ \mathrm{N} \quad (5)$$

3.1.3. Pressure force in axial direction

The pressure force in axial direction is calculated as [5]

$$F_p = (P_2 - P_3) * \frac{2\pi(r_t - r_r)}{n}, 4021\mathrm{N\ per\ unit\ length} \quad (6)$$

3.1.4. Meshing and simulation of model

For the model before simulation, mesh convergence test is done to obtain continuous results of deformation and stresses. The meshing parameter is varied to create smaller size of element. When mesh density increases it gives more accurate results that value obtained is more close to actual solution. For mesh element size 0.0014mm and above, the peak deformation value obtained is constant. 0.0014mm mesh element size was taken for which average element quality is about 0.78.

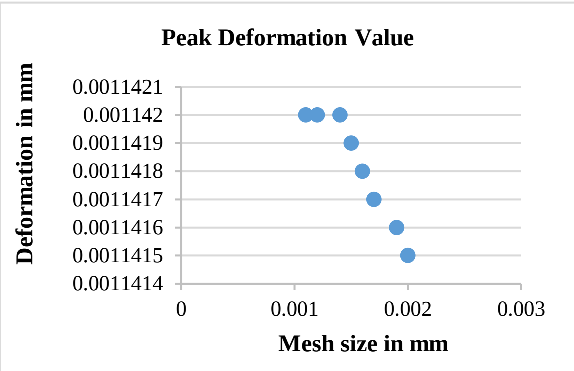

Figure 3.1: Mesh convergence test

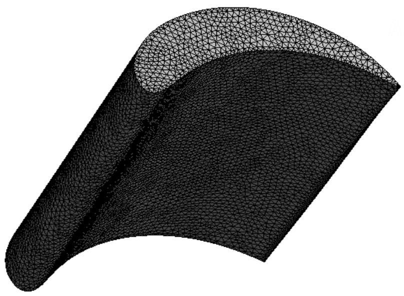

Figure 3.2: Meshing of blade (Elements:75408, nodes:115408)

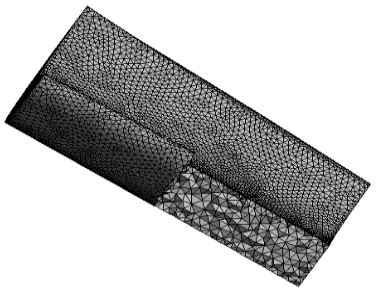

Figure 3.3: Cut section views with growth rate 1.0

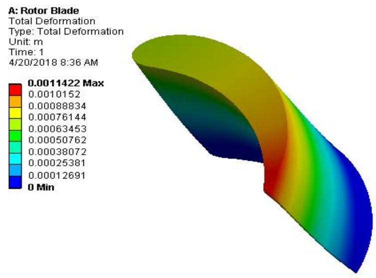

Figure 3.4: Blade total deformation

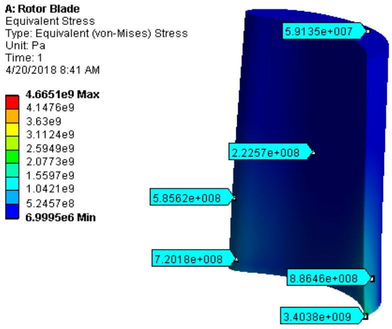

Figure 3.5: Stress distribution

3.2. Thermal analysis

As we have assumed gas is flowing at rate of 75kg/s in turbine annulus, the flow will enter third station that is rotor station and occupy area around $0.25\mathrm{m}^2$. The length it covers in third plane is about 0.08882m that is chord length. The entering temperature to rotor blade is calculated as $T_2' = 1267\mathrm{k}$ and exit static temperature calculated as $T_3 = 1211.5$. The specific heat of the gas is 1.2005KJ/kg°k. Thermal conductivity of Ni based alloy is 107.31w/m.k. Convection coefficient or heat transfer coefficient is measure how fluid effectively transfer heat from convection. In case of gas it varies with speed of gas [12]. The ambient temperature as assumed to be 293k and inlet temperature to the rotor calculated 1267k. The total heat for a single blade is summation of heat from conduction and convection.

Reference temperature is $\frac{1267+293}{2}, 780K$ [13]

Table vii: Gas properties at temperature 780k [13] [14]

Parameter	Value	Unit
Specific heat (c_p)	1.093	kJ/kg.K
c_υ	0.8065	kJ/kg.K
Kinematic viscosity (υ_g)	$171.38*10^{-6}$	m^2/s
Dynamic Viscosity (μ_g)	$47.75*10^{-6}$	kg/m.s
Thermal conductivity (k_g)	80.78	mW/m.K
Universal gas constant (R)	0.2870	kJ/kg.k
γ	1.355	
Chord (c)	0.08882	m
Ambient temperature	293	k

Prandtl number

$$\mathrm{Pr.} = \frac{\mu_g c_p}{k_g}, 0.646 \tag{7}$$

Reynold number

$$R_e = \frac{V * c}{\upsilon_g}, 213265.43 < 5 * 10^5 \quad (8)$$

It shows flow is laminar.
Nusselt number

$$N_u = 0.664 R_e^{\frac{1}{2}} Pr.^{\frac{1}{3}}.265.1 \quad (9)$$

Total heat transfer coefficient (h_t)

$$\frac{N_u k_g}{c}, 241.1 \ w/m^2.k \quad (10)$$

3.3. Blade cooling

The universal coolant for gas turbine rotor blade is air. Air is favorites as it could be discharged to the main flow. It is necessary to be able to estimate the cooling gas flow required to achieve a specified blade relative temperature for any aerodynamic design. The cooling air will be tapped from compressor usually about 3 percent of main flow. Inside cooling passage as coolant passes from root to tip, it possesses continuous heat exchange between blade internal surface and coolant. Coolant temperature along height of blade is calculated as:

$$\frac{T_{gas} - T_c}{T_{gas} - T_{cr}} = \frac{k_2 k_3}{k_1 + k_1 k_2 + k_2} * e^{-k_4 h}, T_c \text{ is coolant t from } h = 0 \text{ to } h = h \ [15] \quad (11)$$

Table viii: Input parameters for cooling design

Parameter	Value	Unit	Description
Mass of coolant ($\dot{m}_c$)	0.0375	kg/s	$\frac{75 * 0.03}{60}$
Gas temperature (T_g)	1267	k	Entry temperature to rotor blade
Temperature of coolant at root (T_{cr})	600	k	Assumed
Reference temperature $\frac{1267+600}{2}$, $\mathbf{933.5}\boldsymbol{k}$ [14]			
Specific heat (c_{pc})	1.176	kJ/kg. K	
c_υ	0.8881	kJ/kg. K	
Dynamic Viscosity (μ_c)	39.84*10^{-6}	kg/m. s	
Kinematic viscosity (υ_c)	105.35*10^{-6}	m^2/s	
Thermal conductivity (k_c)	64.29	mw/m. K	
Universal gas constant (R)	0.2870	kJ/kg − k	
γ_c	1.324		
Ambient pressure (P_a)	101325	pa	
Coolant pressure (P_c)	1781768	pa	
Ambient temperature (T_a)	293	k	
Density (ρ_{ca})	10	kg/m^3	
Hole diameter (D)	0.005	m	
Cross section Area of cooling hole (A_c)	1.9625*10^{-5}	m^2	
Velocity of coolant (V_c)	9	m/s	$\frac{\dot{m}_c}{\rho_c A_c}$, A_c cross sectional area of cooling hole

Total number of hole is 22
Mass flow of coolant per hole is 0.0017 kg/s per hole
Cross sectional area of cooling hole is $\pi * (0.0025)^2, 1.9625 * 10^{-5} m^2$

$$V_c = \frac{\dot{m}_c}{\rho_c A_c}, 9\ m/s$$

Prandtl number

$$Pr. = \frac{\mu_c c_{pc}}{k_c}, 0.728$$

Reynold number (Total)
At $h = 0.12m$

$$R_e = \frac{V * h(\text{Height of blade})}{\upsilon_c}, \frac{9 * 0.12}{\upsilon_c}, 10251.5 < 5 * 10^5$$

At $h = 0.117m$

$$R_e = \frac{V * h(\text{Height of blade})}{\upsilon_c}, \frac{9 * 0.12}{\upsilon_c}, 9995.25 < 5 * 10^5$$

At $h = 0.06m$

$$h = 0.06m$$
$$R_e = 5125.77 < 5 * 10^5$$

At $h = 0.01m$

$$R_e = 854.3 < 5 * 10^5$$

All value of Reynold number shows flow is laminar.
Nusselt number (Total)

$$N_u = 0.664 R_e^{\frac{1}{2}} Pr.^{\frac{1}{3}}, 60.45, 42.77, 17.45$$

Total internal heat transfer coefficient (h_c)

$$= \frac{N_u k_c}{D(\text{diameter of hole})}, 777.64\ w/m^2.k, 767.86\ w/m^2.k, 549.94\ w/m^2.k, 224.38\ w/m^2.k$$

At tip gas wetted perimeter (s_g) is 0.22m (Approximately)
Coolant wetted perimeter (s_c) is

$$22 * 2\pi * 0.0025(\text{radius of hole}) = 0.3397m$$

$$k_1 = \frac{h_c s_c}{h_g s_g}, = 4.98, k_2 = \frac{k_b}{h_g s_g}, = 2.02, k_3 = \frac{h_c s_c}{\dot{m}_c c_{pc}}, 132.13$$

$$k_4 = \frac{k_2 k_3}{k_1 + k_1 k_2 + k_2}, = 15.65$$

To calculate coolant temperature along height, the heat transfer coefficient $h_c = 777.64\ w/m^2.k$, *at* $h = 0.12m$ is used as it is final value of heat transfer coefficient when coolant leave from tip.

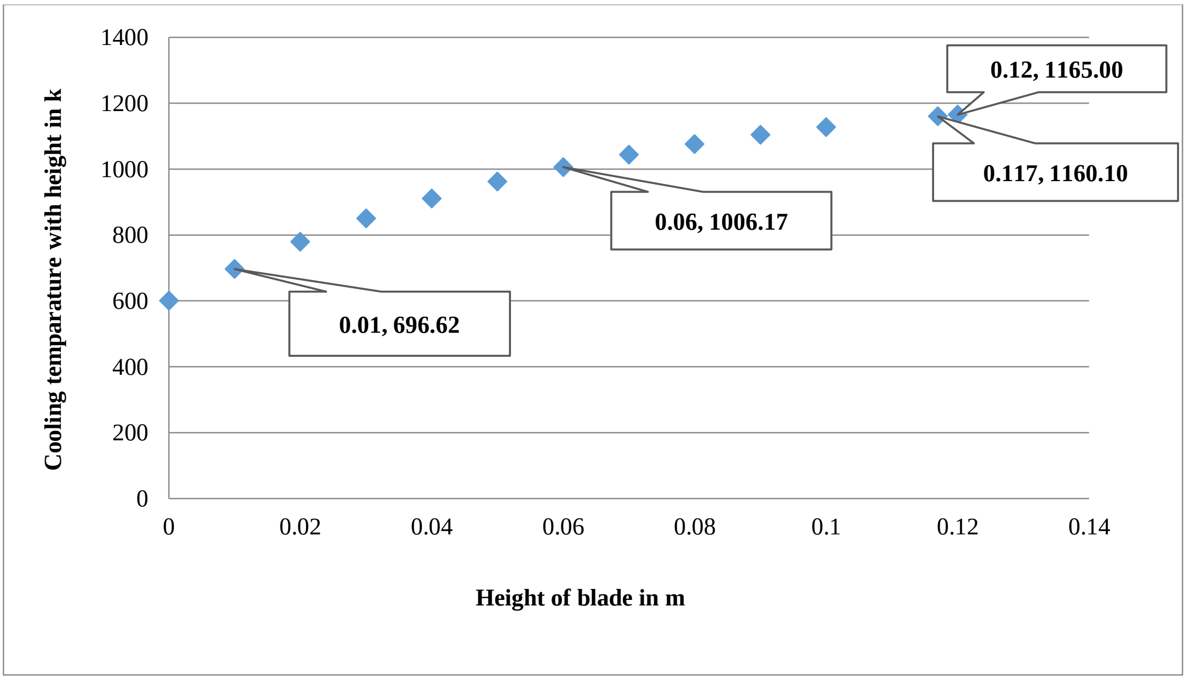

Figure 3.6 Blade coolant temperature vs. Blade height

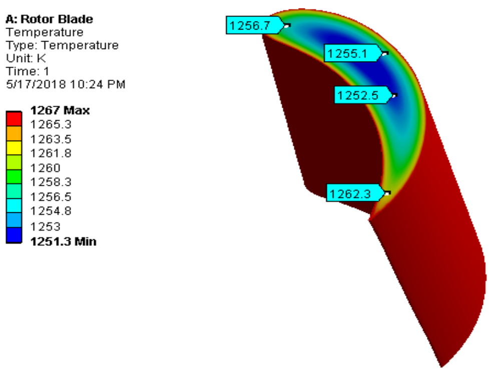

Figure 3.7 Temperature distribution before cooling

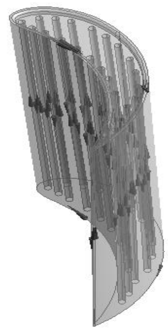

Figure 3.8 Cooling Design from root to tip

Coolant at root ($\boldsymbol{T_{cr}}$) :600k

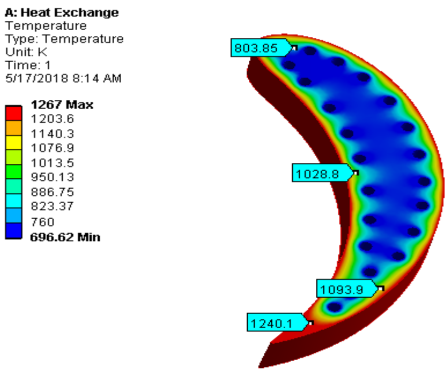

Figure 3.9 Temperature distribution at $\boldsymbol{h = 0.01m}$

$\boldsymbol{h_c = 224.38\, w/m^2.k\, , T_{c\, at\, h=0.01} = 696.62k}$

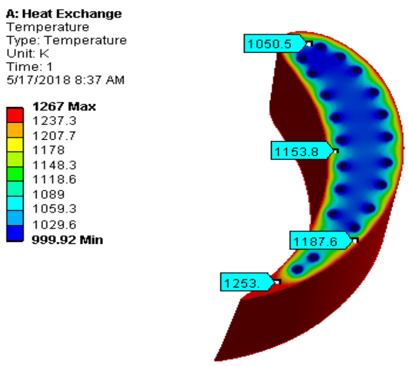

Figure 3.10 Temperature distribution at $\boldsymbol{h = 0.06m}$

$\boldsymbol{h_c = 549.94\, w/m^2.k\, , T_{c\, at\, h=0.06} = 1006.17k}$

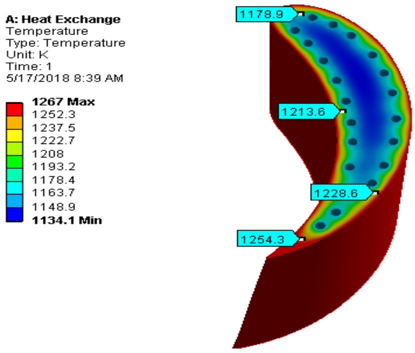

Figure 3.11 Temperature distribution at $\boldsymbol{h = 0.117m}$

$\boldsymbol{h_c = 767.68\, w/m^2.k\, , T_{c\, at\, h=0.01} = 1160.1K}$

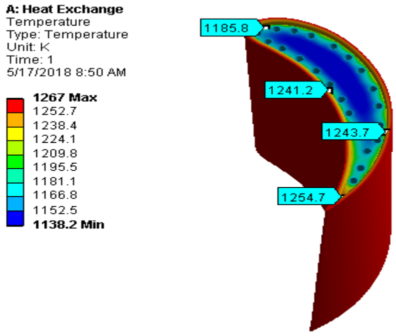

Figure 3.12 Temperature distribution at $\boldsymbol{h = 0.12m}$

$\boldsymbol{h_c = 777.64\, w/m^2.k\, , T_{c\, at\, h=0.12} = 1165k}$

Table ix Temperature distribution comparison along height

Blade height (h) in m	Temperature in k		
	Common point	Without cooling	With cooling
0.01	a	1262.3	1240.1
	b	1252.5	1093.9
	c	1255.1	1028.8
	d	1256.7	803.85
0.06	a	1262.3	1253

	b	1252.5	1153.8
	c	1256.7	1187.6
	d	1255.1	1050.5
0.117	a	1262.3	1254.3
	b	1252.5	1228.6
	c	1256.7	1213.6
	d	1255.1	1178.9
0.12	a	1262.3	1254.7
	b	1252.5	1243.7
	c	1256.7	1241.2
	d	1255.1	1185.8

Figure 3.6 depicts the different value coolant as it passes from root to tip and it is achieved using equation (11). Blade heights 0.01m, 0.06m, 0.117m and 0.12m is taken as reference and respective coolant temperatures estimated analytically. Points a, b, c and d are considered as common points across cross section of blade at different height. At Table ix temperatures along across cross section of blade at different height and at assumed points has been listed. (Figure 3.7) Shows the temperature distribution across cross section of blade before cooling design.

4. RESULT AND DISCUSSION

Table x: Verification of parameters within design limit

Parameter	Designed range	Calculated value	Description
ψ	3-5	2.94	Moderate
ϕ	0.8-1.0	0.85	Satisfactory (Assumed)
Λ	0.4-0.5	0.45	Satisfactory
Swirl angle (α_3)	$0 \leq \alpha_3 \leq 20°$	12°	Satisfactory
$\frac{r_t}{r_r}$	1.41	1.2-1.4	Moderate
$\frac{h}{c}$		1.35	Moderate
$\frac{s}{e}$	$0.25 < \left(\frac{s}{e}\right) < 0.625$	0.593	Satisfactory
R_{le}	$0.05s \leq R_{le} \leq 0.1s$	0.0975s	Satisfactory
t	$0.1 \leq t \leq 0.22$	0.2	Satisfactory

As thesis proposed of designing first stage gas turbine rotor blade, the aspect ratio obtained is moderate to reduce vibration effect. The satisfactory blade loading coefficient means blade could withstand on given temperature drop. As gas angles and gas velocities are vary from root to tip of the blade the velocity triangles also vary. So blade has small amount of twist. The setting angle that is angle between chord alignment and axial axis is 2.75°. It means the angle of incidence of axial velocity is very less could be compare to zero. The profile has leading edge radius is 0.0975s and trailing edge has radius of 0.81mm. As it possess trailing edge with radius it helped for cooling technology implementation. The critical pressure ratio compared with calculated pressure ratio to find weather nozzle is chocking it was found value of calculated pressure ratio at station two is less than critical pressure ratio so nozzle is not chocked and throat pressure is equivalent with static inlet pressure to rotor blade.

As steady state structural analysis is concern, it was observed blade leading and trialing region across root section are stressed excessively. The blade tip section has more propensities to have deformation. As steady state thermal analysis is concern, the blade is simulated with analytically calculated static gas temperature 1267k all over the blade surface. The temperature distribution across cross section of blade is observed and minimum temperature explored as 1251.3k. The model was further simulated with 22 cylindrical cooling passages extending from blade root to tip. It could be observed (Figure 3.6), the coolant temperature at tip is about 1165k when 600k coolant passed through root of blade. Along the height it could also be observed (Table ix) increase of blade temperature across cross section with increase of blade height.

5. CONCLUSION

The forces acting on blade were determined for structural stresses and deformation analysis. The simulation demonstrated, axial force affects trailing edge most, bending force affects pressure side of blade most and tangential force affects root of the blade most. The blade also analyzed thermally. The rotor inlet temperature was calculated theoretically as 1267k for given assumed turbine inlet temperature 1400k. The temperature distribution for given material blade was observed and the vicinity where maximum temperature built up was identified. The place nearby wall of the blade was seen thermally disturbed more so the cooling passages was constructed and coolant of 600k at root was used for steady state thermal analysis and it was perceived the reduction of temperature along height of blade cross section after application of cooling design.

The bending of the profile only based on geometrical interpretation, the cascade test could give precise throat opening so the result could be more accurate. The thesis only based on study state thermal and structural analysis. The CFD analysis could give more genuine results. The paper only has discussed about internal convection cooling, it could include film and jet impingement cooling design for further optimization of temperature distribution.

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