High Pressure Impulse Steam Turbine Blade

Mr. Leela Krishna Vegi

Department of Mechanical Engineering

Jawaharlal Nehru Technological University, Kakinada, Andhra Pradesh, INDIA

E-mail ID: krishna.leeven@gmail.com

Abstract— In high pressure turbine the temperature of the steam will be in a range of 5400c – 5600c and the flow of the fluid flow will be subsonic and supersonic. The pressure and temperature on the blade effects the surface of the blade, life cycle, deformation of the blade. In order to withstand blade at high pressure, temperatures, stresses and thermal loads the model of the blade is designed by introducing the cooling channels to the blade by supplying the cooling air through the rotor shaft which is extracted from the atmosphere and condenser to supply the coolant to the turbine blade. Using Grey cast iron material the blade profile can be able to withstand maximum pressure and temperatures and reduces the deformation levels. The aerodynamic profile, root notch and the shroud are designed using the CATIA v5 3D modeling software. The part is analysis with a material in ANSYS analysis software to observe the static structural, thermal and other required stresses at critical conditions in order to increase the life cycles and give better performance of the blade by using the Grey cast iron main high pressure steam turbine blade.

Keywords Impulse blade, steam turbine blade, cooling channels, high pressure turbine blades, fire-tree root notch, Design and Modeling, Analysis.

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1. INTRODUCTION

Steam turbine impulse blade is one of the most important part in the steam turbine rotor equipment system. Impulse blade is one of the high pressure withstanding blade which is placed in the starting stage of the turbine rotor arrangement in between the nozzle and the reaction blades. The steam produced from the boilers at high temperatures and pressures in a super saturated state and is supplied through the convergent and divergent nozzle to the starting stage of the steam turbine blade are called the Impulse blades. Its working principle is to convert the high pressure of the super saturated steam into higher velocity and rotates the shaft about 3000 r.p.m to 10,000 r.p.m which depends upon the capacity of thee turbine and the boiler.[Here the working fluid is the steam which strikes the turbine blade at a speed of supersonic flow].In order to withstand high pressure and thermal stresses the model of the blade should be in such a way that the blade profile is fixed with the shroud, cooling circuits and two different types of roots which are capable enough to withstand high pressures and loads.

2. DESCRIPTION OF THE PROJECT

In this project we are showing root notches fixed to the blade and a shroud added to top of the blade to reduce the stress with a specific material. Applying different axial and centrifugal forces and loads on the blade to the maximum conditions.
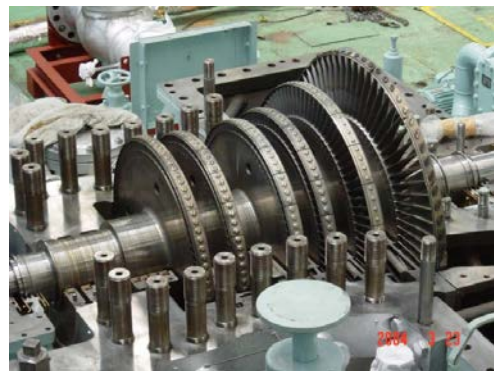

Fig. 1 High pressure stage of steam turbine

We can find the value of withstanding forces and temperatures' using the ANSYS software shows the result in simulation view so that with the help of the diagram we can identify the deformation of the profile at any particular part and observe the physical, stresses, place of deformation. Using

Gary cast iron for the analysis on the modeled steam turbine impulse blade on various like testing stress analysis, thermal analysis, static analysis, heat flux etc. with the better result of the best economical metal is chosen for the manufacturing for better performance of the blade. With the help of the theoretical calculations some of the unknown values are known for the better analysis and comparison calculated for the three different metals.
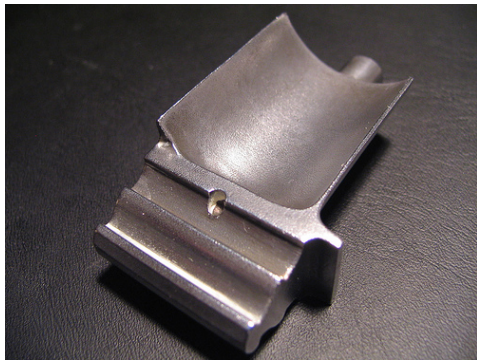
2.1 Impulse blade

Fig. 2 Impulse Blade

In impulse there is no change in the pressure of the steam as it passes through the moving blades. There is change only in the velocity of the steam flow when the steam strikes the impulse blade through the nozzle inlet. An impulse turbine has fixed nozzles that orient the steam flow into high speed jets. These jets contain significant kinetic energy, which the rotor blades, shaped like buckets, convert into shaft rotation as the steam jet changes direction. A pressure drop occurs across only the stationary blades, with a net increase in steam velocity across the stage.

As the steam flows through the nozzle its pressure falls from inlet pressure to the exit pressure (atmospheric pressure, or more usually, the condenser vacuum). Due to this higher ratio of expansion of steam in the nozzle the steam leaves the nozzle with a very high velocity. The steam leaving the moving blades has a large portion of the maximum velocity of the steam when leaving the nozzle. The loss of energy due to this higher exit velocity is commonly called the "*carry over velocity*" or "*leaving loss*".

In impulse turbines, high-velocity steam from fixed nozzles impacts the blades, and this impulse drives the blades forward, causing the rotor to turn. The main feature of these turbines is that the heat drop per stage can be quite large, allowing for large blades and a smaller number of stages.

2.2 Blade stresses

The determination of blade stresses is a critical factor in the design of blades. The severest stresses are imposed by the centrifugal forces due to high rotative speeds. Bending stresses are also imposed by centrifugal forces, fluid-pressure differences and vibration. Centrifugal stresses are a function of the mass of material in the blade, blade length and speed. The component of centrifugal force acting radially outward exerts a tensile stress at the root. Sufficient cross sectional area must be provided in the blade at the root and a material capable of withstanding stress without fatigue must also be provided.

The centrifugal force on an element **dr** at a radius **r** (fig) is given by

$$dF = (\gamma a \, dr)\omega^2 r$$

Where, γ = specific weight of blade material, **kg/m³**

a = blade cross-sectional area, **m²**; and

ω = angular velocity, **rad/sec**

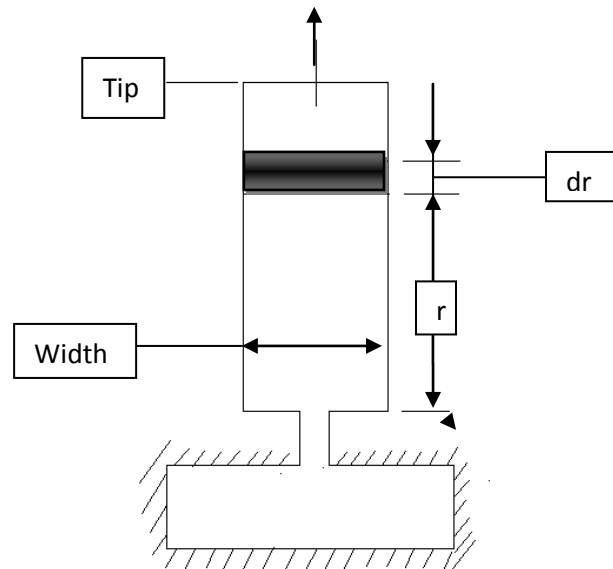Total centrifugal force exerted at the blade root is

Fig. 3 Centrifugal and bending forces on the turbine blade

$$F_c = \int_{rr}^{r1} \gamma a dr \, \omega^2 \, r = \frac{\gamma a \omega 2}{2} (r_t^2 - r_r^2) \rightarrow [1]$$

Where, r_t = tip radius, and r_r = root radius

$$Fc = \frac{\gamma a \omega^2}{2\Pi} A = \frac{\gamma a A}{2\Pi} [\frac{2\Pi N}{60}]^2 \rightarrow [2]$$

Where A = annular area = $\Pi (r_t^2 - r_r^2)$

The centrifugal or the tensile stress at the blade root is thus

$$S_c = \frac{Fc}{a} = \gamma^2 A (\frac{N}{23.94})^2 \rightarrow [3]$$

If the blade is trapped, the mass of material is reduced, thereby reducing the centrifugal stress.

Since the stress exerted at any section of the blading decreases radially, reaching a minimum near the tip, a constant cross-sectional area is not required for strength. Hence, where the centrifugal stresses are severe the blade is tapered by decreasing both its thickness and width. Impulse blades are subject to bending from centrifugal stress and the tangential force exerted by the fluid. Reaction blades have an additional bending stress due to large axial thrust because of the pressure drop which occurs in the blades. All turbine blades may be subjected to bending because of vibration. The total stress at a given point on a turbine blade is found by adding the centrifugal stress at that point to the bending stress.
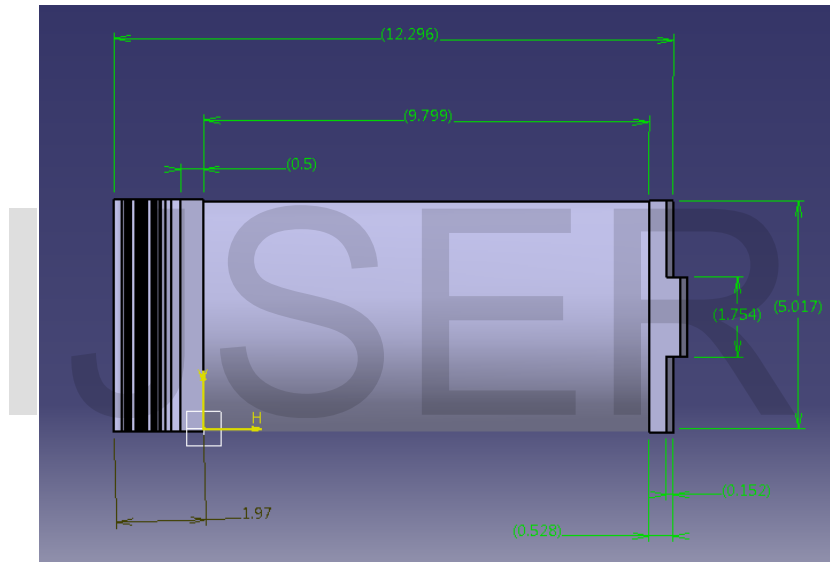
3. MODELING OF TURBINE BLADE

Fig. 4 Overall dimensions of the steam turbine impulse blade with a solid and Skelton view for cooling circuit blade profile

Blade profile design these are the following coordinates used for the generating the profile of the blade
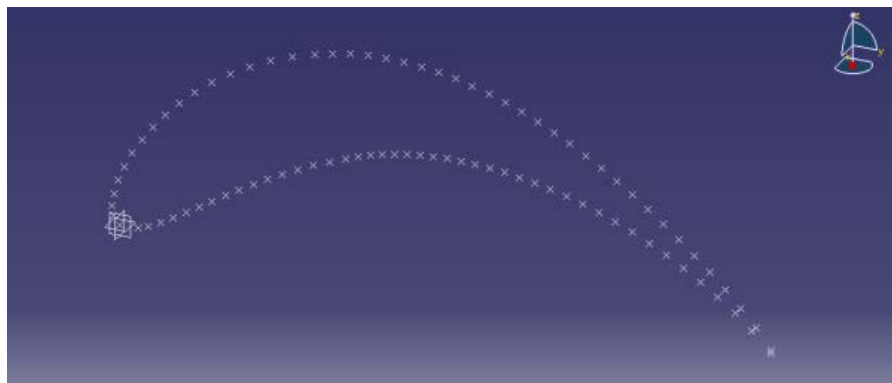

Fig. 5 blade profile generated with the coordinates shown in the below

These are the following coordinates used to generate the aerodynamic profile of the high pressure impulse blade on considering the inlet and outlet angles of the blade.

X-coordinates	Y- coordinates	X-coordinates	Y- coordinates	X-coordinates	Y- coordinates
1. 1	-0.002692	32. 0.273144	0.139633	64. 0.326314	0.330709
2. 0.970943	0.026901	33. 0.249786	0.127814	65. 0.353526	0.335437
3. 0.945122	0.049222	34. 0.226982	0.115549	66. 0.38105	0.338513
4. 0.918544	0.069353	35. 0.204776	0.102977	67. 0.408612	0.33976
5. 0.892009	0.087425	36. 0.183171	0.0902	68. 0.435764	0.339257
6. 0.865509	0.103765	37. 0.16216	0.077353	69. 0.462504	0.337196
7. 0.839077	0.118555	38. 0.141683	0.064516	70. 0.488897	0.333701
8. 0.8128	0.1319	39. 0.121579	0.051742	71. 0.514991	0.32888
9. 0.786731	0.143892	40. 0.101678	0.039133	72. 0.540825	0.322787
10. 0.760918	0.154602	41. 0.081897	0.026911	73. 0.56638	0.315487
11. 0.735401	0.164096	42. 0.062441	0.015613	74. 0.59168	0.30704
12. 0.710216	0.172426	43. 0.044032	0.00621	75. 0.616742	0.297481
13. 0.685391	0.179644	44. 0.027925	-0.000029	76. 0.641577	0.286854
14. 0.660955	0.185793	45. 0.01504	-0.002453	77. 0.666206	0.27519
15. 0.63693	0.190913	46. 0	0	78. 0.690647	0.262516
16. 0.613332	0.195045	47. -0.010036	0.012399	79. 0.714918	0.248853
17. 0.590177	0.198224	48. -0.011655	0.027498	80. 0.739039	0.234221
18. 0.56748	0.200486	49. -0.009067	0.046818	81. 0.763035	0.218631
19. 0.545245	0.20186	50. -0.002601	0.068828	82. 0.786926	0.202087
20. 0.523478	0.202382	51. 0.006964	0.09199	83. 0.810735	0.184594
21. 0.502194	0.202073	52. 0.019094	0.115468	84. 0.834486	0.166144
22. 0.481363	0.200959	53. 0.033584	0.138889	85. 0.858201	0.146725
23. 0.461007	0.199074	54. 0.050395	0.162083	86. 0.881906	0.126308
24. 0.441159	0.196415	55. 0.069548	0.184897	87. 0.905627	0.104856
25. 0.421867	0.192996	56. 0.091031	0.20712	88. 0.92938	0.082198
26. 0.403108	0.188781	57. 0.114818	0.228533	89. 0.95322	0.058097
27. 0.384421	0.183688	58. 0.140829	0.248841	90. 0.977409	0.033022
28. 0.365441	0.177771	59. 0.168929	0.267763	91. 1	0.002638
29. 0.347214	0.171407	60. 0.198994	0.285033		
30. 0.321465	0.161425	61. 0.230857	0.30038		
31. 0.297041	0.150877	62. 0.264357	0.313567		
		63. 0.298905	0.324225		

This is the design of the blade which is designed with the cooling channels provided to the blade profile with shroud on the top and the external fir tree type of root notch which is modeled in the Catia v5 modeling software.
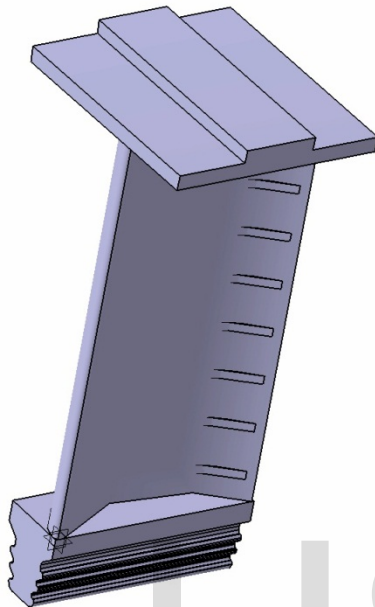

Fig. 7 Designed model imported in to Ansys

Fig. 6 Design of the impulse blade

There are eight rectangular sections of the cooling pass way is created from the vertical cooling circuit which are projected from the bottom of the blade profile to the top of the shroud. A rectangular pocket is created at the bottom of the profile. A channel or the passage is created to the root notch in order to allow the cooling air through the cooling circuit to draw the heat or the hot gases passing through the blade.

4. ANALYSIS OF THE TURBINE BLADE

Using the Ansys analysis software of the v12 version different pressure and thermal loads are applied on the blade profile to find the Equivalent stresses and normal stresses at a fixed position at different directional axis [X, Y, Z directions] so that we can observe the Maximum and minimum withstanding conditions of the blade profile on different materials and better performance and positive value of the material is considered like grey cast iron, ceramics and other alloy steels. ASTM Grade 40 [ISO grade 300, EN-JL1050] grey cast iron steel is used for the analysis of the blade.
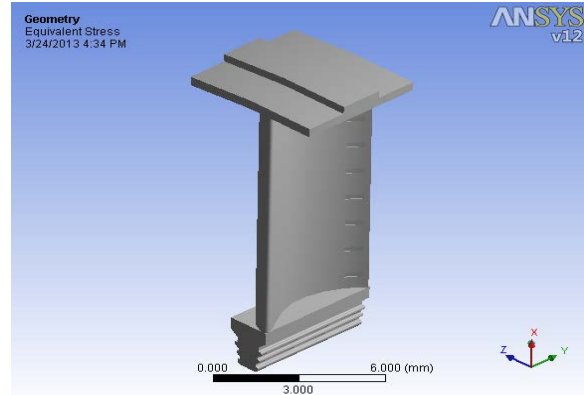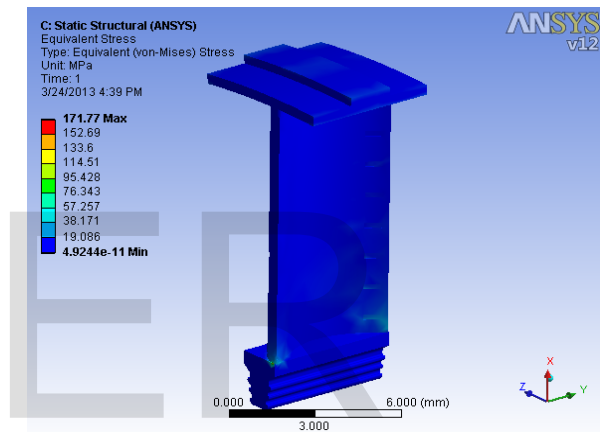

Fig. 8 Equivalent stress for Grey cast iron

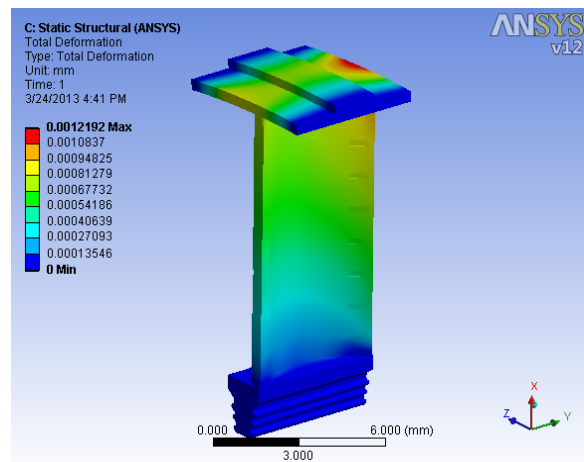**Fig. 9 Total deformation for Grey cast iron**

4.1 Static Analysis of the Grey cast iron

Sno.	Types of stress	Max (MPa)	Min (MPa)
1.	Equivalent stress	171.77	$4.9244e^{-11}$
2.	Normal stress(x-axis)	115.9	-12.642
3.	Normal stress(y-axis)	29.901	-11.338
4.	Normal stress(z-axis)	39.964	-29.902
5.	Shear stress(xy plane)	86.712	-8.7274
6.	Shear stress(yz plane)	12.605	-6.1494
7.	Shear stress(zx plane)	17.037	-12.494
8.	Total deformation	0.0012192	0

These are the inlet and outlet temperatures that are introduced into the turbine blade in the Ansys software for the thermal Analysis.

Table 2
Temperature parameters

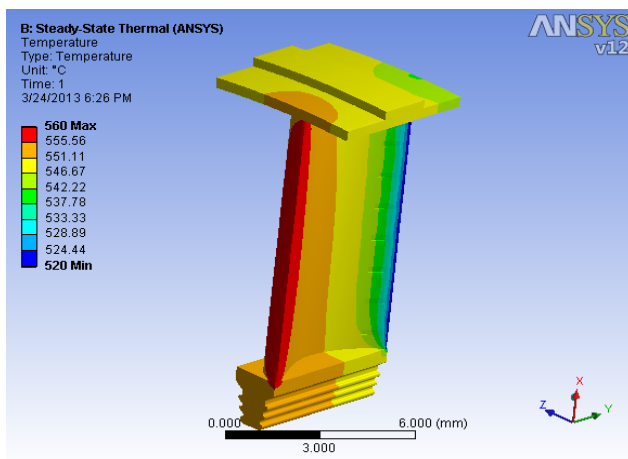| Temperature at inlet	$T = 560^{0}c$
Temperature at outlet	$T = 520^{0}c$

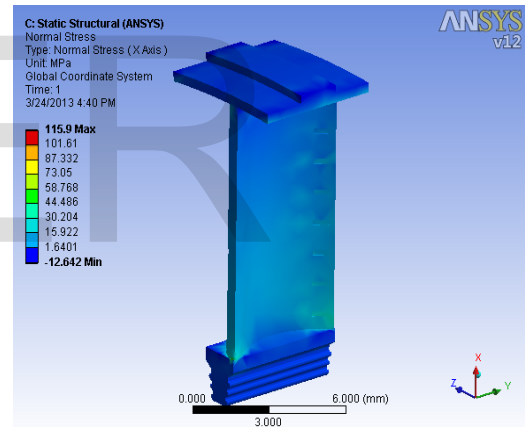**Fig. 10 Boundary conditions for the blade**

Fig. 11 Normal stresses on x – direction
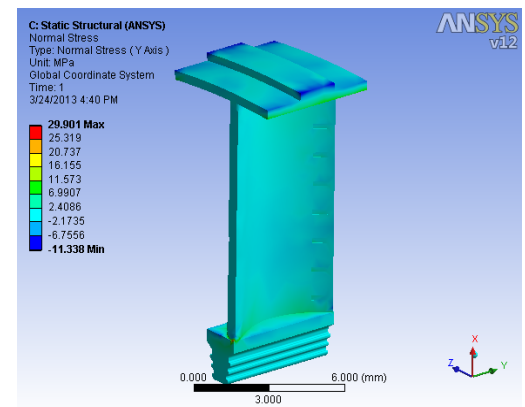
Fig. 12 Normal stress on y – direction

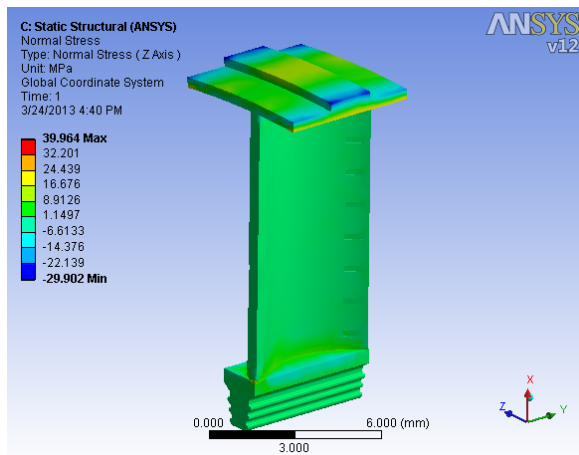**Fig. 13 Normal stress on z - direction**

Fig. 14 Total heat flux

5. CALCULATIONS

TABLE NO. 3
Mechanical properties of Grey cast Iron

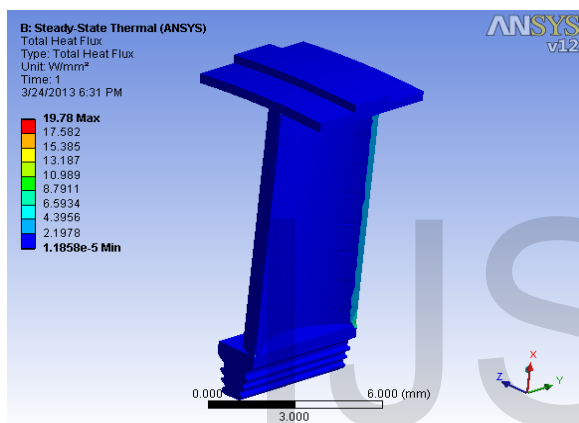| Mechanical Properties	Grey cast iron
Density(g/cc)	7.4
Modulus of elasticity, (Gpa)	124
Yield strength, (Mpa)	195
Ultimate strength ,S_u,(Mpa)	298
Poison ratio	0.26
Hardness Brinell	235
Shear Modulus, (Gpa)	55 – 69
Compressive Strength, (Psi)	139×10³,965 Mpa
Bulk Modulus, (Gpa)	76 – 95
Thermal Conductivity	46w/m-k [ambient]

5.1 THEORETICAL CALCULATIONS

The centripetal force is the external force required to make a body follow a curved path. Any force (gravitational, electromagnetic, etc.) or combination of forces can act to provide a centripetal force. This force is directed inwards, towards the center of curvature of the path.

A simplified 2-D figure of the blades under discussion is shown in Figure.
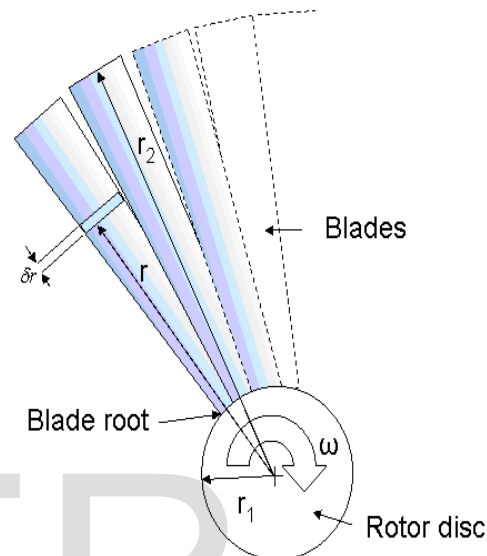

Fig. 15 Simplified blade dimensions

The general equation for centripetal force is:

$$F= mr\omega^2 \ldots (1)$$

Where, is the mass of the moving object is the distance of the object from the center of rotation (the radius of curvature) and $mr\omega$ is the angular velocity of the object.

In the case under consideration, we need to account for the fact that the mass of the blade is distributed over its length and the radius of curvature also changes along the length of the blade.

Consider a small segment of mass δm, of length having width δr at a distance r from the centre. Then the equation for the centripetal force δF on this small segment is given by:

$$\delta F = \delta m\omega^2 r \ldots (2)$$

In practice, a blade tapers in thickness towards its tip; but, for simplicity, assuming the blade to have a constant cross sectional area A (m²) and material density ρ (kg/m3), we can write:

$$\delta m = \rho A \delta r$$

and equation (2) becomes

$$\delta F = \rho A \omega 2 r \delta r$$

or formally,

$$dF = \rho A \omega^2 r dr \ldots (3)$$

Let r_1 be the radius of the rotor disc and be the distance between the centre of the rotor disc and tip of the blade. Then, integrating equation (3) along the total length of the blade, the total centripetal force acting on the blade is given by:

$$F = \rho A \omega^2 \int_{r1}^{r2} r dr$$

So,

$$F = \rho A \omega^2 \left(\frac{r_2^2 - r_1^2}{2} \right) \ldots (4)$$

We can convert the angular velocity from revolutions per minute (rpm) to radians per second using the following relationship:

$$\omega = \frac{rpm \times 2\pi}{60} \ldots (5)$$

Knowing the values for the cross-sectional area, density, angular velocity and radii, we can then calculate the force on one blade. Once this force has been calculated, we can estimate the nominal stress σ on the blade root using the following relation:

$$\sigma = \frac{F}{A_{root}} \ldots (6)$$

Where, A_{root} is the cross-sectional area of the blade root.

5.2 Input Data

Absolute velocity of steam entering the blades, $C_1 = 600$ m/s

Absolute flow angle (nozzle angle) $\alpha_1 = 20^0$

Blade velocity co-efficient, $K = 0.8$

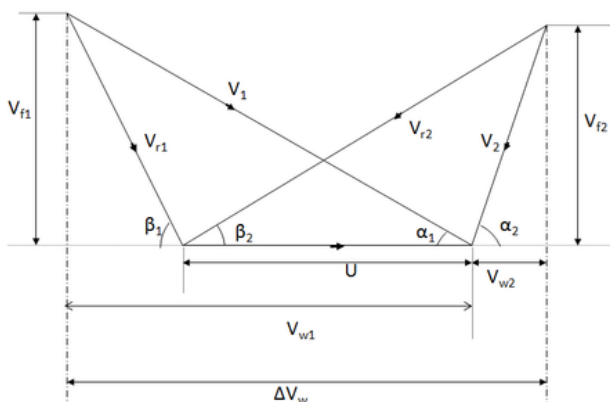Mass of steam flowing through the turbine, $m_s = 1000$ kg/h

Fig. 16 the velocity triangles at inlet of first stage rotor blades were constructed as shown

Diameter of blade mid-span D = 1.0687 m,

Design speed of turbine N = 3000 r.p.m,

Peripheral speed of rotor blade at its mid-span (blade velocity) $C_{bl} = \frac{\pi D N}{60}$

From the velocity triangles in figure

We get,

Whirl velocity $C_w = C_{w1} + C_{w0}$

$\quad\quad\quad\quad = 790$ m/sec

Velocity of flow at entrance, $C_{f1} = 225$ m/sec

Velocity of flow at exit, $C_{f2} = 180$ m/sec

Relative velocity at entrance, $C_{r1} = 450$ m/sec

Now, $C_{r0} = K\, C_{r1}$

$\quad\quad\quad\quad = 0.8 \times 450$

$\quad\quad\quad\quad = 360$

Relative velocity at exit $C_{r0} = 360$ m/sec

Entrance angle of blade $\beta_2 = 29^0$

Exit angle of blade $\beta_1 = 29^0$

As the blades are symmetrical,

$\quad\quad\quad \beta_2 = \beta_1 = 29^0$

Absolute velocity of steam leaving the blades, $C_0 = 320$ m/s

5.3 Finding tangential Force (P_t) and Axial force (P_a)

Tangential force, $P_t = m_s (C_{w1} + C_{w0})$

$\quad\quad\quad\quad = 1000 / (60 \times 60) \times (790)$

$P_t = 219.44$ N

Axial force, $P_a = m_s (C_{f1} - C_{f0})$

$\quad\quad\quad = 1000 / (60 \times 60) \times (45)$

P_a **$= 12.5$ N**

5.4 Evaluation of centrifugal force

Centrifugal force,

$$F_c = \int_{r2}^{r1} \gamma a \, dr \, \omega 2 \, r = \gamma \, a \, \omega^2/2 \times (r_t^2 - r_r^2)$$

Where, r_t = Tip radius, and r_r = Root radius.

$$F_c = \gamma \, a \, \omega^2/2\pi \quad (A)$$

$$= \gamma \, a \, A/2\pi \, (2\pi N/60)^2$$

Where,

A = annular area = **$\pi (r_t^2 - r_r^2)$**,

γ = density of material,

ω = angular velocity, rad/s

Now,

Density of material	γ	= 7850 kg/m³,
Tip radius	r_t	= 93.10 mm
Root radius	r_r	= 58 mm
Blade cross-sectional area	a	= 2.791 mm²

F_c = 7850×10⁻⁹ × 2.71 × π [(93.10)² – (58)²]/2π × [(2π×3000)/60]²

F_c = 5734 N

Now centre of gravity of blade in X, Y, Z directions are

G_x = 4.594 mm

G_y = 2.134 mm

G_z = 0.848 mm

5.5 Calculation of Factor of safety

$$F.S = \frac{\text{yield point stress}}{\text{Design stress or working stress}} \rightarrow (1)$$

(Or)

$$F.S = \frac{\text{ultimate stress}}{\text{Design stress or working stress}} \rightarrow (2)$$

The above two formulas depends on the hardness of materials:

1. Equation (1) for hard materials

2. Equation (2) for soft materials

5.6 Structural Analysis for Grey cast iron with cooling channels

$$\text{Factor of Safety [F.S]} = \frac{\text{ultimate stress}}{\text{Design stress or working stress}}$$

$$= \frac{240}{171.77}$$

F.S = 1.39

5.7 Result of the weight for Grey cast iron

Density of Grey cast iron	= 7.4× 10⁻⁶ kg/mm³
Volume	= 1788686.732 mm³
Weight	= volume × density
	= 1767552.586 ×7.4×10⁻⁶
	= 13.08 kg

Conclusion

- From the Literature Survey the various Input Parameters for the Analysis of impulse steam turbine blade has been collected.
- By using Ansys software analysis (Structural, Thermal) of the impulse steam turbine blade has been carried out successfully. We had done the analysis on considered different type materials from this grey cast iron is having the better performance and very much economical.
- Static structure analysis is analyzed by applying different loads [tangential force, axial force and centrifugal force] at the boundary conditions. We get equivalent stress, normal stress in x, y, z directions and shear stress in xy, yz and zx planes. By comparing the stresses for different materials it is observed Grey cast iron is giving better performance to impulse steam turbine blade by introducing cooling circuit to the blade.
- Thermal analysis is analyzed by applying inlet and outlet temperatures at boundary conditions. We get total heat flux and directional heat flux x, y, z directions.
- By using the Stresses obtained from the static structural Analysis the Factor of Safety is obtained.

- On calculated weight for three materials we found that the Grey cast iron is having the less weight so, that it can withstand stresses.
- So, here we conclude that the impulse steam turbine blade is observed to give a better performance by using the Grey cast iron material.
- The life of the blade can be increased to 40 – 50 cycles. By using the density of 7000 kg/cm^2.The weight of the blade of can be reduced and the work done by the blade will be increased.

Nomenclature

a	=	Area of cross section
A	=	Blade cross-sectional area, m2; and
A_{root}	=	Cross-sectional area of the blade root
F_c	=	Total centrifugal force
$F.S$	=	Factor of Safety
G_x	=	Centre of gravity on x- direction
G_y	=	Centre of gravity on y- direction
G_z	=	Centre of gravity on z- direction
N	=	Number of stages
P_i	=	Pressure of steam at inlet
P_o	=	Pressure of steam at outlet
V_i	=	Velocity of steam at inlet
V_{os}	=	Velocity of steam at outlet
V_a	=	Absolute velocity of steam
V_r	=	Relative velocity of steam
V_b	=	Blade velocity
θ	=	Nozzle angle
ϕ	=	Blade entrance angle
γ	=	Blade exit angle
δ	=	fluid exit angle
γ	=	specific weight of blade material, kg/m3
ω	=	angular velocity, rad/sec
r_t	=	tip radius
r_r	=	root radius
S_c	=	centrifugal or the tensile stress at the blade root
δ_m	=	small segment of mass
δ_r	=	distance r from the centre
ρ	=	material density, (kg/m3)
γ	=	density of material

Acknowledgement

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