EFFECT OF TAPER AND TWISTED BLADE IN STEAM TURBINES

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ABSTRACT

Turbo machinery is critical to various processes and their application range from steam turbines to aircraft turbines. Turbo machinery bladed disks are subjected to severe centrifugal loads. The centrifugal force is one of the problems that face the designers of blades especially the long ones. The severe loading leads to large peak stresses. One way to tackle this is by using twisted blades or using variable cross-section area blades. Rotating structures having the shape of pre-twisted blades commonly affect the modal characteristics. In order to design the rotating structures properly, their modal characteristics must be estimated accurately. The modal characteristics of rotating structures often vary significantly due to the effect of the centrifugal inertia force induced by the rotation motion. For this reason, the modal characteristics of rotating structures have to be investigated. In this paper effect of taper and twist in steam turbine blade is analyzed using ANSYS. Blades are idealized as tapered cantilever beams that are fixed to a rotating disc. The results obtained are then utilized in the design of the constant stress blade.

Keywords: Centrifugal Inertia Force, Peak Stresses, Pre-Twisted Blades, Turbo Machinery

1. INTRODUCTION

Structural integrity of all rotating components is the key for successful operation of any turbo machinery. This integrity depends on the successful resistance of the machine parts to the steady and alternating stresses imposed on them. The challenge with rotating equipment, such as turbo machinery, is often more severe due to the significance of the alternating loads that must be carried to satisfy their purpose.

In the most of the application the rotor blades are tapered from hub to tip. This reduces the weight of the peripheral portions of the blade and hence the hub stress, which must be carried under the high rotational speeds, will be reduced [1]. The centre line of the blade section will be normally located on the radial line to avoid the bending stress in the blade. But in some cases the blades are bent forward in the direction of motion to permit centrifugal bending moment to offset the air/gas thrust moment.

The rotor blades of high aspect ratio are twisted from hub to tip to compensate for the blade velocity variations and also to satisfy the radial pressure equilibrium conditions. Blade vibrations have also been the cause for failure in many cases [2, 3]. Hence it is important that the natural frequencies of the blade be such that it is outside the operating frequency range of the turbine not coincides with the harmonic speeds.

The blade of variable cross section with thin aerofoil sections at and near the blade tip are relatively heavy sections near the blade root. Therefore, where the peripheral speed of the blade is high, the mass per unit length of blade is minimum and where the resultant centrifugal force is high, the blade section is maximum. This tends to a more uniform blade stress than would be obtained in a blade of constant cross section. In order to obtain an even more uniform centrifugal stress, the blade is often tapered in width. The cross sectional area of a tapered blade may be varied in such a way that, centrifugal stress is uniform over greatest part of blade length. Centroid of the various cross sections should, as far as possible, lie on a radial line so as to eliminate bending due to inertia forces. By setting the line of centroids, a little forward of the radial line, the bending moment due to centrifugal force may be utilized partially to counteract the moment due to impulse. Taper influences both bending and torsional vibrations due to the variation in mass and stiffness along the length of the blade. Uniform stepped beams of rectangular cross section can approximate tapered blade. It is seen that, a decrease in the cross sectional dimensions of breadth with constant thickness increases the fundamental bending frequency as well as higher mode bending frequencies while decrease in thickness with constant breadth increases, the fundamental bending frequency and decrease other higher modes[4].

II OBJECTIVES OF THE PRESENT WORK

- To study the effect of taper and twist in steam turbine rotating blade as structural integrity.
- To study the effect of blade pull and average stress from gross yielding point of view and tip rub due to blade growth.
- To study the effect of blade stiffening on frequency shift to achieve the required separate margin.
- Role of stress stiffening and Spin softening on blade frequency.

III DESIGN CONSIDERATIONS

The Normal operating speed of the machine is 8650RPM and the blade material selected is Chrome Steel (X28CrMoNiV49) having properties like Density, Young's Modulus, Poisson's ratio, Yield strength and Coefficient of thermal Expansion are 7.85E-9 Ton/mm^3, 210E3 N/mm^2, 0.3, 550 MPa and 11.7E-6/°C (At operating Temperature 100°C). The thickness of the blade=2mm.

IV STRUCTURAL AND MODAL ANALYSIS

Steam turbine blades are subjected to centrifugal loads from high rotational speeds apart from steam bending forces. The blades are subjected to excitation frequencies ranging from the rotational speed and its harmonics. There are other excitations depending on the number of nozzles or vanes in each respective stage together with their harmonics. Therefore, it is inevitable that some blades or other in the flow path are subjected to resonance near the operational speed. In any case all the blades are subjected to some form of resonance during start up or shut down operations and in some cases at speeds very close to operational speed. In such blades that may operate in near-resonant conditions close to the operating speed, the designers have to provide sufficient damping to give the desired life These blades are susceptible for failures, unless very close grid frequency control is maintained.[5-7]

4.1. Spin softening

The vibration of a spinning body will cause relative circumferential motions, which will change the direction of the centrifugal load which, in turn, will tend to destabilize the structure. As a small deflection analysis cannot directly account for changes in geometry, the effect can be accounted for by an adjustment of the stiffness matrix, called spin softening [8].

4.2. Stress Stiffening

Stress stiffening (also called geometric stiffening, incremental stiffening, initial stress stiffening [9], or differential stiffening by other authors) is the stiffening (or weakening) of a structure due to its stress state. This stiffening effect normally needs to be considered for thin structures with bending stiffness very small compared to axial stiffness, such as cables, thin beams, and shells and couples the in-plane and transverse displacements.

4.3. Methodology of Prestress Modal Analysis

Pre stress modal analysis option is used to calculate the modes of a prestressed structure. By default, no prestress effects are included, i.e., the structure is assumed to be stress free.

4.4. Effect of Rotation

The turbine blade gets stiffened in bending because of centrifugal forces and thus, the bending natural frequencies increase the speed of rotation. There is no effect of centrifugal forces in torsional motion of the blade. The first consequence of centrifugal force is to influence the geometric form of blade. Those blades which are heavily twisted or which have non-symmetric cross sections are liable to change in their geometric form under running conditions.

4.5. Methodology of Stress stiffening-Increase stiffness

Stress stiffening, also called geometric stiffening or initial stress stiffening. Stress stiffening is the stiffening (or weakening) of a structure due to its stress state. This stiffening effect normally needs to be considered for thin structures with bending stiffness very small compared to axial stiffness.

4.6. Methodology of Spin softening: Linear effect-reduces stiffness

Adjustment of stiffness to account for the changes in geometry due to centrifugal effects. The vibration of a spinning body will cause relative circumferential motions, which will change the direction of the centrifugal load which, in turn, will tend to destabilize the structure. As a small deflection analysis cannot directly account for changes in geometry, the effect can be accounted for by an adjustment of the stiffness matrix, called spin softening. The meshed model and the Vonmises stress plot of straight, tapered and Taper & Twisted is as shown in Fig.1

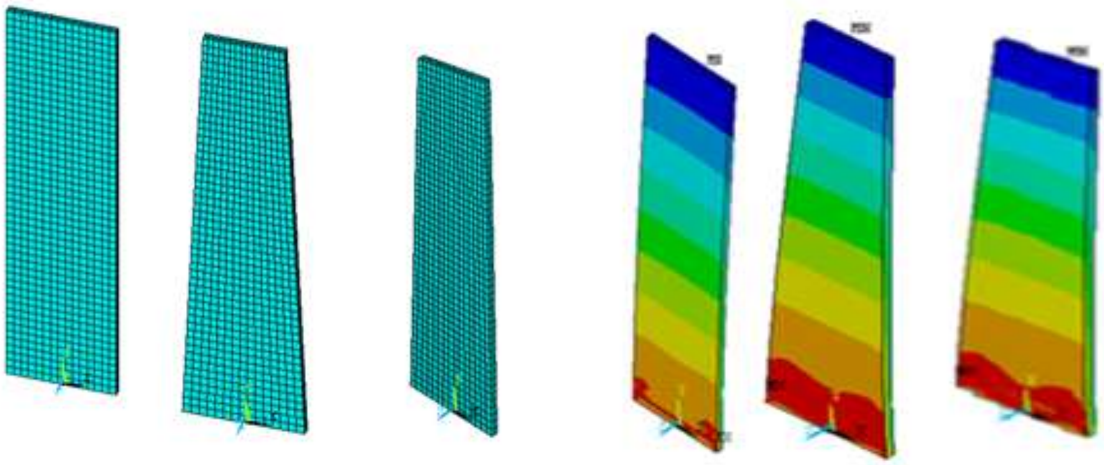


Fig 1. Meshed model and Vonmises stress plot of Straight, Tapered and Taper & Twisted

V RESULTS AND DISCUSSIONS

It is observed that at a rated speed of 8550rpm so much of 62.645N/mm^2 of stress is located at the hub of the aerofoil of straight blade with a certain mass of the blade. In order to reduce the peak stress and average stress in the aerofoil at the tip, the taper is introduced with knowledge base engineering sensitivity study was performed with respect to the taper in the blade and it was observed that the study was successful in achieving 0(zero) stress at the tip and 17% drop in the stress values with respect to the base line model. The mass of the blade also got reduced. The third parameter considered is the twist in the blade. The peak stresses in the blade along with the taper and twist of 5^0 increases the maximum stress by 2%(52.642N/mm^2).Similarly for 10^0 and 15^0 twist the percentage increase in the stress is moved upto 3%.As the twist increases the stress is also increased. Taking the overall picture with respect to the base line model in the 1st case with the introduction of taper by 17% reduction of stress was achieved whereas with the introduction of twist of 0^0 - 15^0, the margin comes down by 3%. So with the effect of Taper and Twist combined it is possible to achieve around 14% (17%-3%) gain in structural integrity with decrease in mass of the blade. The results and the plot are as shown in Fig 2-3 and Table 1-2.

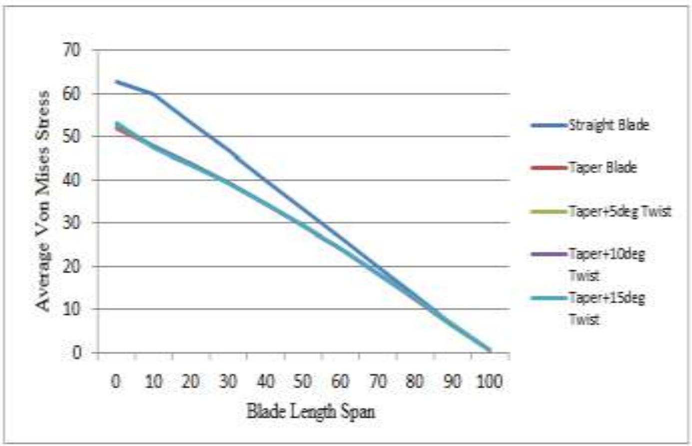
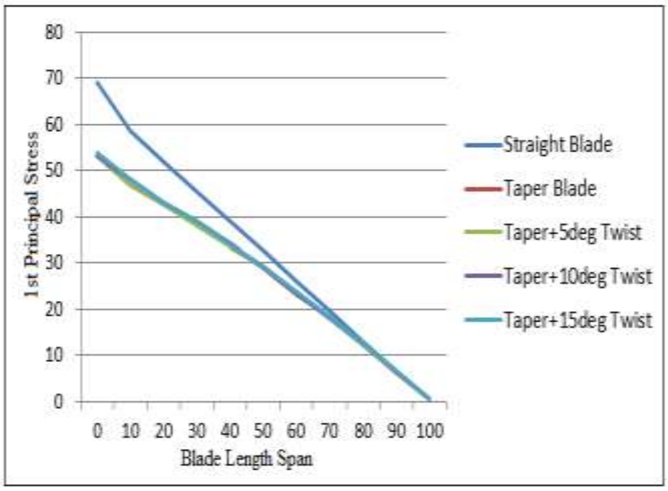


Fig 2. Average Von Mises Stress v/s % Blade Length Span

Blade Span Length	Straight Blade	Taper Blade	Taper +5deg Twist	Taper +10deg Twist	Taper +15deg Twist
0	62.6	52.1	52.6	53	53.1
10	59.9	47.7	47.76	47.7	47.5
20	53.4	43.8	39.3	39.2	39.1
30	46.7	39.3	39.3	39.2	39.2
40	40	34.5	34.5	34.4	34.4
50	33.3	29.4	29.4	29.3	29.3
60	26.5	24	24	24	24
70	19.9	18.3	18.3	18.4	18.4
80	13.3	12.5	12.6	12.6	12.6
90	6.7	6.4	6.5	6.6	6.6
100	0.75	0.6	0.6	0.6	0.6

Table 1. Results of Von Mises Stress

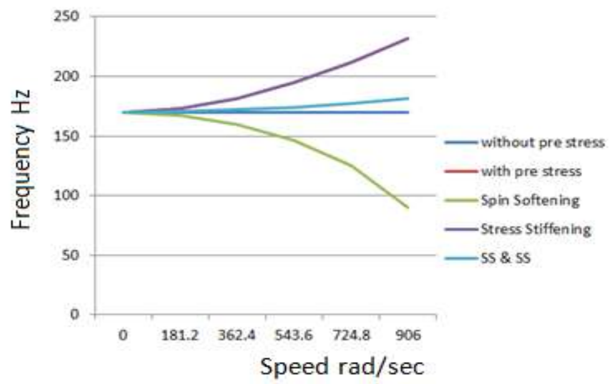


Blade Span Length	Straight Blade	Taper Blade	Taper +5deg Twist	Taper +10deg Twist	Taper +15deg Twist
0	69	53.4	53.3	53.2	54
10	58.6	47	47	48.4	48.2
20	52.1	42.7	42.7	43.3	43.1
30	45.5	38.5	38.2	39	38.9
40	39	34.1	33.5	34.1	34
50	32.5	29	29.1	29.2	29.1
60	26	23.2	23.5	23.5	23.8
70	19.5	18.2	18.1	18.1	18.3
80	13	12.3	12.4	12.5	12.5
90	6.5	6.1	6.4	6.5	6.5
100	0.59	0.53	0.6	0.61	0.71

Fig 3. Average 1st Principal Stress v/s % Blade Length Span **Table 2. Results of 1st Principal Stress**

From structural point of view, the natural frequencies of blade are equally important as that of the stress. The modal analysis is performed to evaluate the modal frequency and the blade dynamics. The aerofoil blade during centrifugal spin untwists itself due to centrifugal action and the increase in stiffness due to higher frequency is compared with straight blade frequency. From mass perspective the mass of the tapered blade is less than the straight blade due to twist resulting in higher frequency than the straight blade. This is very clearly evident as shown in the Table for straight, taper, taper and twist blades. The effect of stiffening is captured through prestress modal analysis and the results are tabulated in the tabular column 3-7. It is evident from the given blade length with and without prestress, the variation in the natural frequencies. With geometric softening the drop in the frequency is also indicated in Fig 4-8.The stress stiffening and spin softening effect is clearly captured for straight blade, taper blade and taper & twist blade. The twist option gives us a platform in tuning the blade by increasing or decreasing the twist. Hence the study is helpful in coming to conclusion to evaluate the sufficient separate margin required for structural integrity.

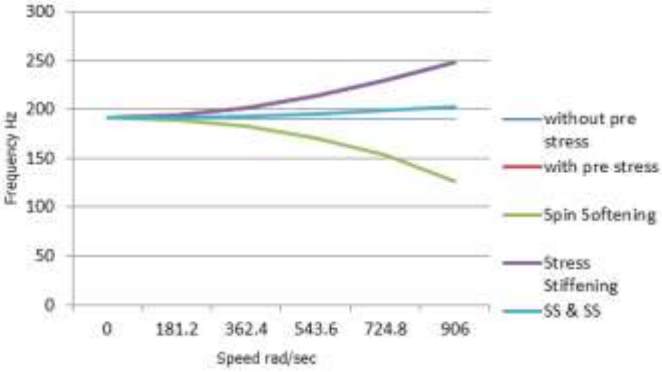
5.1. For Straight Blade



Angular speed 'rad/sec'	0	181.2	362.4	543.6	724.8	906
Without Pre stress	170.12	170.12	170.12	170.12	170.12	170.12
With Pre stress	170.12	173.03	181.46	194.67	211.71	231.7
Spin Softening	170.12	167.66	160.05	146.51	125.04	91.265
Stress Stiffening	170.12	172.95	181.39	194.57	211.65	231.64
Spin Softening and Stress Stiffening	170.12	170.53	171.97	174.3	177.45	181.28

Fig 4. Frequency v/s Speed Plot **Table 3. Frequency results for Straight Blade**

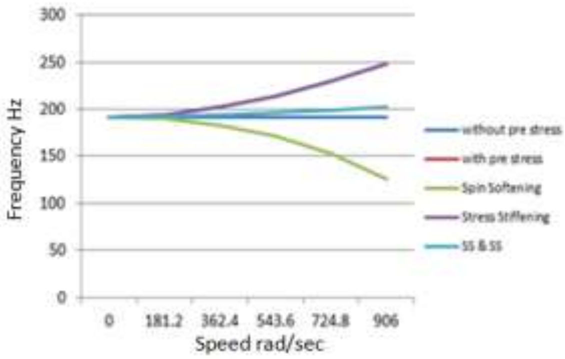
5.2. For Tapered Blade



Angular speed 'rad/sec'	0	181.2	362.4	543.6	724.8	906
Without Pre stress	191.69	191.69	191.69	191.69	191.69	191.69
With Pre stress	191.69	194.3	201.93	214.01	229.81	248.56
Spin Softening	191.69	189.51	182.82	171.05	153.09	126.29
Stress Stiffening	191.69	194.22	201.85	213.94	229.74	248.5
Spin Softening and Stress Stiffening	191.69	192.07	193.43	195.66	198.67	202.37

Fig 5. Frequency v/s Speed Plot **Table 4. Frequency results for Tapered Blade**

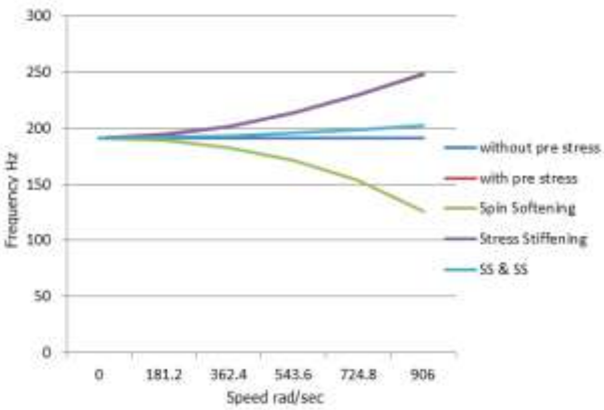
5.3. For Taper and 5^0 Twist Blades



Angular speed 'rad/sec'	0	181.2	362.4	543.6	724.8	906
Without Pre stress	191.71	191.71	191.71	191.71	191.71	191.71
With Pre stress	191.71	194.3	201.95	214.03	229.82	248.58
Spin Softening	191.71	189.52	182.82	171.07	153.11	126.32
Stress Stiffening	191.71	194.24	201.87	213.95	229.75	248.51
Spin Softening and Stress Stiffening	191.71	192.08	193.45	195.68	198.69	202.39

Fig 6. Frequency v/s Speed Plot **Table 5. Frequency results for Taper and 5^0 Twist Blade**

5.4 For Taper and 10^0 Twist Blades



Angular speed 'rad/sec'	0	181.2	362.4	543.6	724.8	906
Without Pre stress	191.76	191.76	191.76	191.76	191.76	191.76
With Pre stress	191.76	194.88	202	214.08	229.87	288.63
Spin Softening	191.76	189.58	182.88	171.13	153.18	126.41
Stress Stiffening	191.76	194.29	201.92	214.01	229.8	248.56
Spin Softening and Stress Stiffening	191.76	192.14	193.51	195.74	198.75	202.46

Fig 7. Frequency v/s Speed Plot **Table 6. Frequency results for Taper and 10^0 Twist Blade**

5.5 For Taper and 15^0 Twist Blades

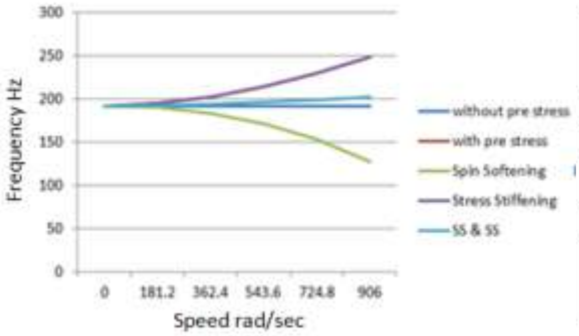


Fig 8. Frequency v/s Speed Plot

Angular Speed 'rad/sec'	0	181.2	362.4	543.6	724.8	906
Without Pre stress	191.86	191.86	191.86	191.86	191.86	191.86
With Pre stress	191.86	194.47	202.1	214.17	229.96	248.72
Spin Softening	191.86	189.68	182.98	171.24	153.31	126.56
Stress Stiffening	191.86	194.39	202.02	214.1	229.89	248.65
Spin Softening and Stress Stiffening	191.86	192.24	193.61	195.84	198.86	202.57

Table 7. Frequency results for Taper and 15^0 Twist Blade

VI CONCLUSIONS

The following are the conclusions made from the present work:

- In design of rotor blades, the centrifugal stress must be calculated because it is considered as the main source of stresses applied to the turbine rotor blade. The maximum stress is at the blade root.
- This paper reveals that the value of centrifugal stress can be controlled by just simply tapering the blades and the twist in the blade can incorporate the moments which are developed in the blade.
- Design of a constant stress blade gives a blade with less stress at the lower half and a similar stress at the upper half.
- The effect of taper and twist on the modal characteristics of cantilever beam are investigated. Variations of the two parameters affect the natural frequency of the blade which helps in blade tuning during design stage to avoid resonance at operating speed.

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