Finite element analysis of metal matrix composite blade

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Finite element analysis of metal matrix composite blade

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Abstract. In this work, compressor rotor blade of a gas turbine engine has been analyzed for stress, maximum displacement and natural frequency using ANSYS software for determining its failure strength by simulating the actual service conditions. Static stress analysis and modal analysis have been carried out using Ti-6Al-4V alloy, which is currently used in compressor blade. The results are compared with those obtained using Ti matrix composites reinforced with SiC. The advantages of using metal matrix composites in the gas turbine compressor blades are investigated. From the analyses carried out, it seems that composite rotor blades have lesser mass, lesser tip displacement and lower maximum stress values.

1. Introduction

Composite materials can be defined as a material with two or more distinct macroscopical phases [1]. The constituents of composite material are easily distinguishable as individual components. Composites are used in various industries and in this paper, the application of composites in gas turbine blades has been studied. There are various structural scales and types of analysis in composite materials. The first level of analysis is known as micromechanics, which deals with the fiber and matrix, their arrangement and lamina properties. The next level of analysis is macromechanics, which deals with number of layers, thickness and other laminate properties. This is followed by the structural analysis that deals with the behavior of composite laminate structures under loading. The behavior of composite compressor rotor blades under loading has been studied using the finite element structural analysis approach. Finite element (FE) analysis has been performed on both composite and metallic compressor blades, and the advantages of one over the other have been established.

Composite materials are used in several industries because of its versatile application. The composites could serve as potential replacement of Ti alloy as they are lighter, stronger, tougher and have a higher corrosion resistant [2]. The deformation characteristics of the composite can be tailored to improve its performance to be better than that of Ti alloy. For example, stiffness and yield strength of the composites can be varied by altering the orientation of the fibers. There are different types of composites based on the matrix material: polymer matrix composite, ceramic matrix composite and metal matrix composite. Metal matrix composites (MMC) offers improved elevated temperature strength and modulus. It has an edge over alloys as the properties are tailorable for specific requirements [3]. Therefore, MMC is considered in the present study.

2. Literature survey

Jabbar *et al.* [4] have carried out a comparative study on structural analysis of first stage rotor blade made of titanium, steel and aluminum. Furthermore, Kim and Lee [5] discussed the modal characteristics and endurance strength of compressor rotor blade while Singerman and Jackson [6] described the state-of-the-art technology of Ti MMC aerospace fabrications, the potential payoffs and recent advances in processing that now lead to high quality, affordable Ti MMC components. The literatures on analysis of compressor blade made of Ti matrix composites reinforced with SiC are scarce. Therefore, the present work has been focused to investigate the performance of composites as compressor blade through static stress analysis using FE.

3. Material selection

This paper deals with the FE analysis of compressor rotor blades of gas turbine engine made of metal and metal matrix composites. The metal considered for this study is Ti 64, which constitutes 6% Al and 4% V. Meanwhile, the metal matrix composite considered for study is Ti 15-3 matrix with SiC reinforcements. Ti 15-3 constitutes 15% V, 3 % Cr, 3% Sn and 3 % Al and it has been chosen as the matrix material because of its two major advantages: it has better strength at elevated temperature and it is amenable to cold rolling for making foils that is not possible with Ti 64.The fibre volume fraction considered for analysis is 35%. Material properties considered for analysis are presented in Table 1 [8, 9].

Table 1. Material properties of Ti 64 and TMC

Material Properties	**Metal**	**TMC**				
	Ti 64	**Particulate**	**$[0]_8$**	**$[0/90]_{2S}$**	**$[0/\pm45/90]_S$**	**$[0_2/\pm45]_S$**
Density (g/cc)	4.54	3.86	3.86	3.86	3.86	3.86
Poisons ratio	0.3	0.28	0.33	0.25	0.33	0.38
Modulus of elasticity (GPa)	117	225	194	148	136	161
Thermal coefficient of expansion (x 10^{-6}/°C)	8.6	6.75	6.75	6.75	6.75	6.75
UTS (MPa)	930	1689	1500	1000	1150	1250

Static stress analysis and modal analysis have been carried out to obtain the stresses, deformation and natural frequency of the blade. These analyses were done on compressor rotor blade made of Ti alloy and TMC. In TMC, two cases were considered for analysis: particulate composites and laminate composites. Different types of laminates were considered for the analysis and they included uni-directional, cross-ply and quassi- isotropic laminates. Brief outline of various analyses carried out are given in Figure 1.

4. Finite element analysis

The finite element method is a numerical technique for analyzing structures with large number of degrees of freedom. It is extremely advantageous for analyzing structures with complex geometry and loading, which cannot be solved exactly using analytical methods. In this present work, ANSYS software was used for the analysis. Analysis of composite blades is different from the analysis of metallic blades because in case of composites, the concepts of layering, orientation of fibres, stacking sequence and also directional properties have to be taken into consideration. Static stress analysis and modal analysis had been carried out for metallic and composite blade.

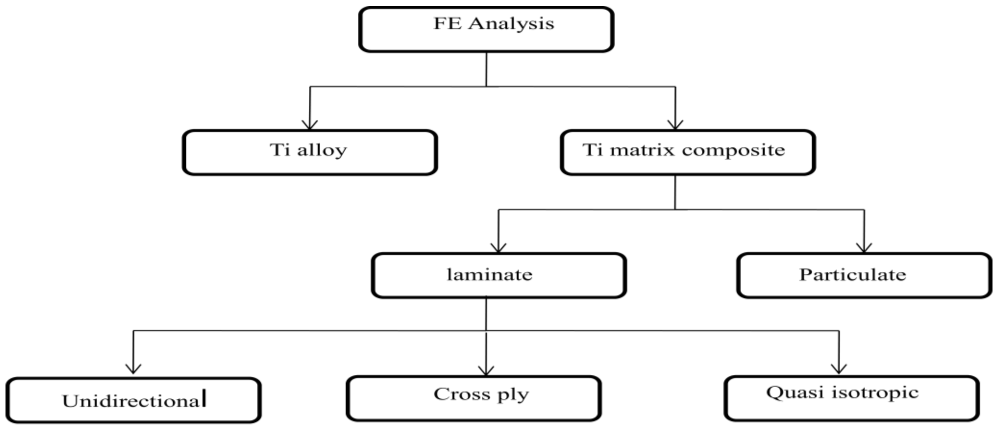

Figure 1. Outline of analysis for composite rotor blade

4.1. Mesh generation

The finite element model was developed using HYPERMESH software. The 3D model of the blade was divided into several zones and the 2D mesh was generated. 2D mesh was then converted into 3D mesh using appropriate commands. Solid 185 elements were used in the analysis of the metallic blades. The same elements cannot be used for the analysis of composites because non-availability of layering feature. Therefore, layered Solid 185 elements were used in the analysis of composite blades. Quality checks such as Jacobian, warpage and aspect ratio were ensured to be within permissible limits.

In FE analysis, a finer mesh results in a more accurate solution. However, as the mesh becomes finer, the computation time also increases. From convergence study, an optimized mesh that can give accurate results at considerably lesser time was obtained. H-Method was chosen to perform the convergence in ANSYS. This method involves running the analysis using a coarse mesh and the results obtained are compared with that obtained by analysing the same model but with finer mesh. In this method, the number of elements is increased and mesh refinement is done locally. This process is repeated until the difference between two successive solutions is less than 2 %. The final configuration of mesh that arrived from H-Method of convergence had 100526 nodes and 90784 elements.

The surfaces and 2D elements of the model were deleted. The 3D elements of the model were exported to ANSYS. User profile was selected in the preference and the element type was then set as Solid 185 in the utility menu. The components were deleted and the elements were renumbered. Finite element model of the blade was then exported to ANSYS. The 3D elements were now imported into ANSYS and the preference was set to be structural. The FE model of the blade is shown in Figure 2.

4.2. Loads and boundary conditions

The loads acting on the compressor rotor blades were centrifugal, gas and thermal loads. The major load acting on the blade is the centrifugal load as it is subjected to RPM of 16000. It acts along the radial direction of the blade. Gas loads act in the axial and tangential direction of the blade due to the difference in velocity and pressure between the leading edge and trailing edge of the blade. They act along X-axis and Y-axis, respectively. In this study, the blade was subjected to uniform temperature of 300°C. The boundary conditions that were provided to simulate the attachment of root on to the disc are shown in Figure 3. In the analysis of laminate composites, the sequencing of layups was carried out as shown in Figure 4 and the thickness of each layer was 0.5 mm.

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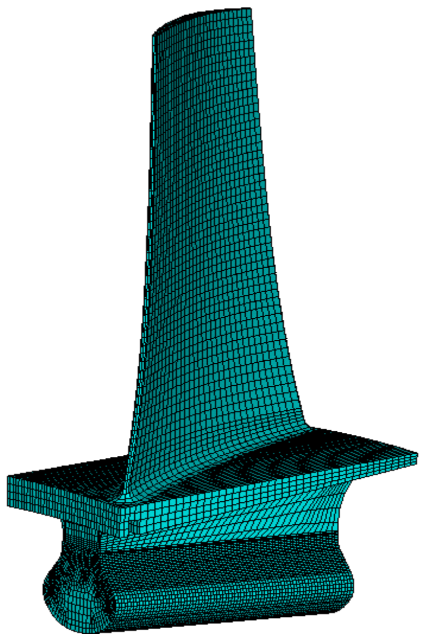

Figure 2. Finite element model of the compressor rotor blade

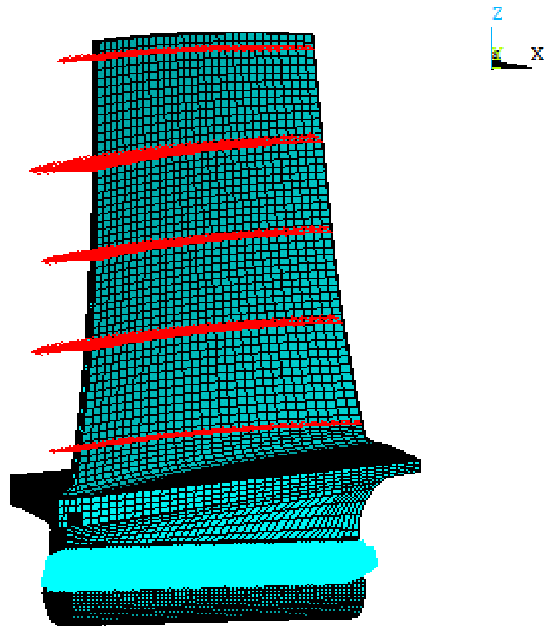

Figure 3. Loads and boundary conditions of compressor rotor blade

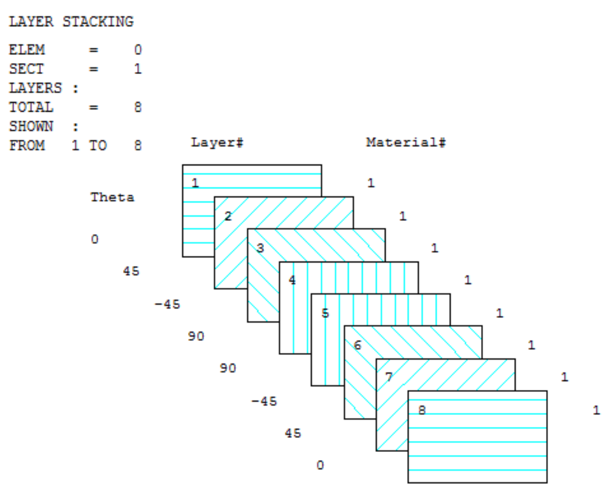

Figure 4. Sequence of lay up of composite laminate

4.3. Static stress analysis

Static stress analysis was carried out to determine the displacements and stresses on the compressor rotor blade. The stress plots of metallic and composite blade are given in Figure 5 and the analysis results are provided in Table 2. From Table 2, it is observed that the mass of the TMC blade was 27.6 g while the mass of the metallic blade was 32.5 g. From the analysis, it is evident that the radial tip displacement of Ti alloy was greater than that of the composites. This is because modulus of elasticity of Ti alloy is smaller when compared to that of TMC and the coefficient of thermal expansion of Ti alloy is greater than that of the TMC. The displacement is the least for unidirectional laminates and was 0.28 mm. The reason being, in unidirectional laminates, the fibres are oriented along the direction of loading. In the case of compressor rotor blade, the major load is the centrifugal load and it acts in the radial direction. As the fibres of the unidirectional laminates are along the radial direction, they have the highest stiffness in the same direction, resulting in minimal displacement. On the other hand, the particulate composites' displacement was 0.29 mm and was slightly higher than unidirectional laminates. In case of particulate composites, the stiffness was the same in all directions. The displacements of cross ply and quasi isotropic laminates were 0.37 mm and 0.38 mm, respectively. They were greater than unidirectional and particulate composites, but smaller than that of Ti alloy. This is because, in cross ply and quasi isotropic composites, fibres in few lamina are oriented in the direction of loading while few are not.

Considering Von mises stress, it is seen from Figure 5 that the stresses in composites are lesser than that in metal alloy. The maximum stress in case of Ti alloy is on the pressure side hub section whereas in the case of particulate composites, it is seen on the suction side hub section. Both Ti alloy and composites show gradual gradation in stress pattern. The red patch refers to the location of stresses greater than 180 MPa and it is comparatively smaller in case of particulate composites. The stress patterns of $[0]_8$, $[0/90]_{2S}$, $[0/\pm45]_S$, and $[0/\pm45/90]_S$ are more or less similar with a small variation in the maximum stress value. Considering the maximum principal stress, the maximum stress location is at the pressure side hub section for Ti alloy and the composites. The maximum principal stress of Ti alloy was greater than that of TMC. It is also seen that the stress patterns of all the composites were similar and the difference in maximum stress was not more than 2 MPa.

Table 2. Results of stress analysis of the compressor rotor blade

Properties	Metal	TMC				
	Ti 64	**Particulate**	**$[0]_8$**	**$[0/90]_{2S}$**	**$[0/\pm45/90]_S$**	**$[0_2/\pm45]_S$**
Mass of blade (g)	32.5	27.6	27.6	27.6	27.6	27.6
Von Mises stress (MPa)	225	207	219	202	201	212
Maximum principal stress (MPa)	233	200	201	197	197	199
Maximum radial displacement (mm)	0.70	0.29	0.28	0.37	0.38	0.32

4.4. Analytical estimation of stress

To validate the results of FE analysis, analytical estimation of the stresses was carried out. It is difficult to theoretically obtain the exact stresses acting on the rotor blade due to the complexity in blade structure and loading. However, the results can be obtained by considering the blade as a cantilever beam and the loads were uniformly distributed. This does not depict the exact structure and loading, but it has been used in an average sense to obtain the results. Centrifugal stresses acting on the blade depends on the size of the rotor blade and also on the rotational speed of the rotor. This is given by Equation 1.

$$\sigma_c = \frac{F_c}{A_h} = \int_{r_h}^{r_t} \rho\omega^2 \frac{A_b}{A_h} r dr \qquad (1)$$

where σ_c = centrifugal stress, A_b = area of interest at the required radius from the centre, ρ = density of the material, A_h = area at the hub, N = number of revolutions per minute, ω = angular speed, r_h = radius at hub and r_t = radius at tip.

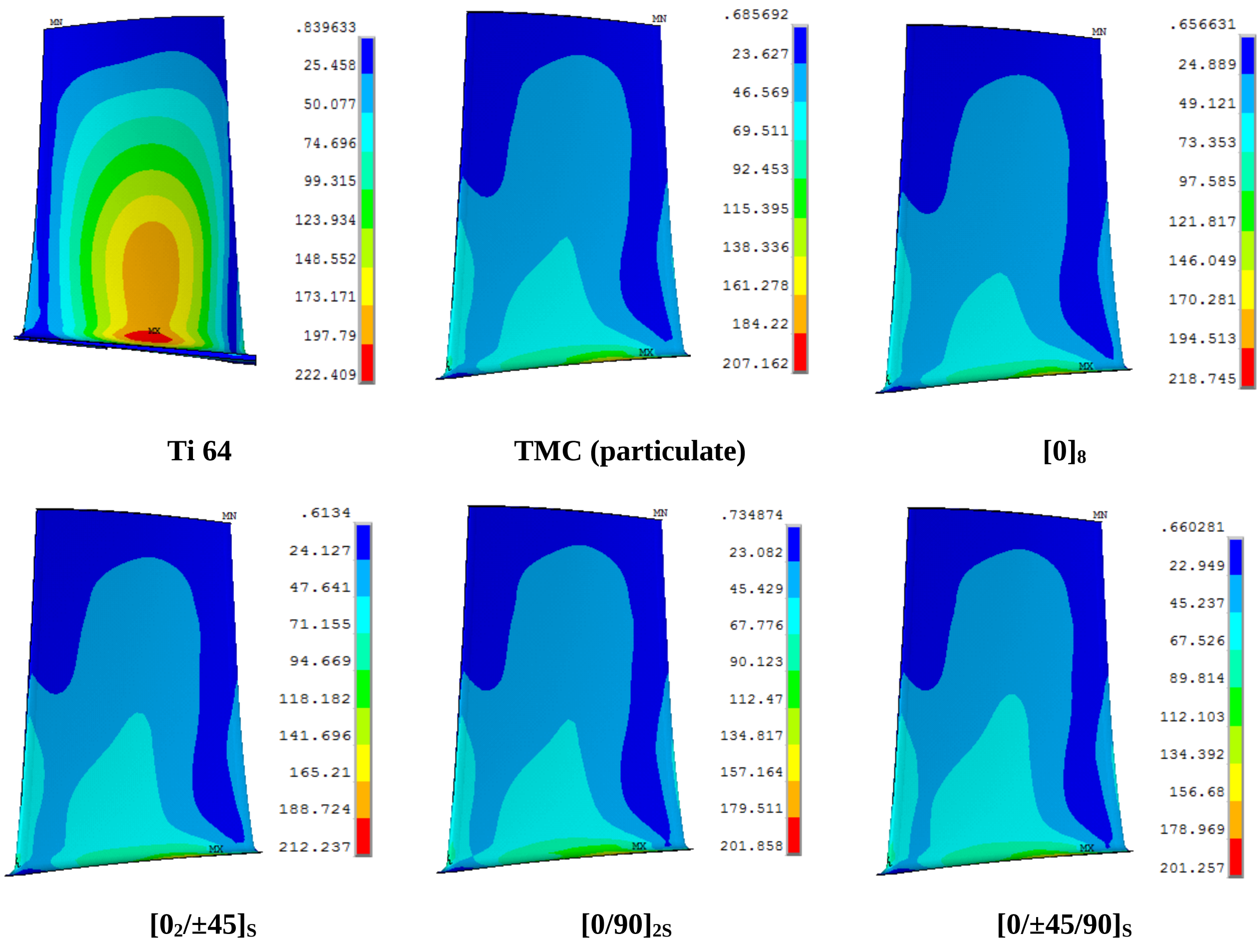

Figure 5. von Mises stress plot (MPa) of compressor rotor blade

Equation 2 shows a more simplified form of Equation 1, in which most parameters are pre-calculated. This shows that centrifugal stress is directly proportional to the $\rho A\omega^2$.

$$\sigma_c = \frac{\rho\omega^2 A}{4\pi}\left[2 - \frac{2}{3}\left(1 - \frac{A_t}{A_h}\right)\left(1 + \frac{1}{1+\frac{r_h}{r_t}}\right)\right] \tag{2}$$

$$\omega = \frac{2\pi N}{60} \tag{3}$$

$$N = 16028 \quad \text{and} \quad \omega = \frac{2\times\pi\times 16028}{60} = 1678.4$$

Centrifugal stress is the major stress acting on the blade. The centrifugal stress was estimated using Equation 2 for the compressor blade made of Ti alloy. The blade is assumed to be a cantilever beam with uniformly distributed load acting on it. The bending moment was estimated and it was used in calculating the bending stresses, which is given by Equation 4. The stresses were estimated at five different sections from hub to tip along the radius of the blade. The stresses acting at different sections of the blade are given in Table 3 and they clearly indicate that the magnitude of stresses decreases from hub to tip. This implies that the stress pattern matches with the results obtained from numerical analysis. The area of cross section and the radius were obtained from the solid model.

$$\sigma_b = \frac{My}{I} \tag{4}$$

where σ_b = bending stress, M = bending moment and I / y = section modulus.

Table 3. Variation of stress along the radius of the blade

Blade Section	Radius (mm)	Area (mm sq)	Stress (MPa)
Hub	215	260	406
Below Mean	225	74	103
Mean	235	56	71
Above Mean	245	47	58
Tip	255	33	38

The maximum stress obtained from FEA was 233 MPa and that obtained from analytical estimation was 406 MPa. The order of magnitude of stress matches with that obtained from FE analysis. The results did not exactly match because of the many assumptions that had been used in the analytical estimation of the stresses. The variation in the thickness and profile of the blade had not been taken into consideration. Thus these analytical results gave a gross estimation of stress and not as accurate as the results obtained from FE analysis.

4.5. Modal analysis

In gas turbines, when the operating speed of engine is equal to the natural frequency of the blade system, each of the individual blade begins to resonate. At resonance, a vibratory deflection of large amplitude is induced by relatively small amplitude stimuli. The result is severe mechanical stresses leading to failure. Modal analysis is used to determine the vibration characteristics such as natural frequencies and mode shapes of a structure, which are important parameters in the design of a structure for dynamic loading conditions. Modal analysis has been carried out with the pre-stressed compressor rotor blade and the first five natural frequencies have been obtained and are provided in Table 4. Natural frequency is directly proportional to stiffness and inversely proportional to the density of the body. As far as the metals are concerned, as the stiffness increases, the density also increases. The greatest advantage of composites here is that the increase in stiffness comes with decrease in density, resulting in increased natural frequency of the system.

Modal analysis has been carried out with the pre-stressed compressor rotor blade to capture the effects of change in stiffness due to centrifugal stresses. While estimating the natural frequency of the blade, the analysis is carried out using the compressor rotor blade without any loads acting on it. However, during actual operation of the engine, the blades will be subjected to centrifugal stress that results in a change of the blade's stiffness. Therefore, to capture this pre-stress effect, the blade was first subjected to the static analysis and then followed by modal analysis with pre-stress option on. The first five natural frequencies were obtained for Ti alloy, particulate and laminate composites. The results are provided in Table 4 and the mode shapes are provided in the Figure 6.

Table 4. Result of modal analysis of the compressor rotor blade

Frequency (Hz)	Metal	TMC				
	Ti 64	**Particulate**	**$[0]_8$**	**$[0/90]_{2S}$**	**$[0/\pm45/90]_S$**	**$[0_2/\pm45]_S$**
Mode 1	1393	1658	1677	1503	1463	1570
Mode 2	2628	3414	3431	3057	2886	3110
Mode 3	4903	5215	5286	4612	4442	4810
Mode 4	6420	9580	7731	6814	6489	7047
Mode 5	7016	10459	8776	7751	7337	7953

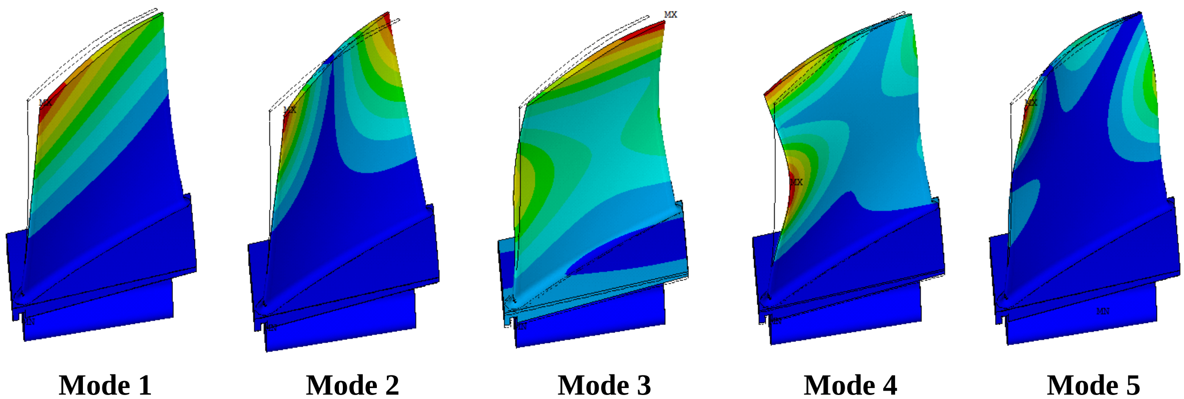

Figure 6. Mode shapes of Ti particulate composite rotor blade

It can be seen in Figure 6 that the mode shapes obtained for Ti alloy and Titanium matrix composites are similar. The first mode is a flexural mode, also known as 1F, in which there is only one nodal point. Animation in ANSYS shows the blade oscillating back and forth about its hub. The second mode is a torsional mode, also known as 1T, where the blade twists about its central line. The third mode is again a flexural mode, it is 2F because it has two nodal points wherein the animation shows a sine wave motion. The fourth and fifth modes could not be clearly distinguished as flexural or torsional mode because the animation showed a complex motion that was a combination of both flexure and torsion.

However, there is a considerable difference in natural frequencies of compressor rotor blade made of Ti alloy, particulate composites and laminates with various orientations in Table 4. Blades made of Titanium matrix composites have higher natural frequency compared to those made of Ti alloy. There is a difference of 300 Hz when comparing the first natural frequency of Ti alloy against that of TMC. For higher modes, the difference in frequencies increases. Unidirectional laminates have the highest natural frequency. The difference in natural frequencies of composites with various orientations is not more than 100 Hz.

5. Failure criteria

5.1. Maximum stress theory

Although there are several theories derived to identify the failure of a component, there is no single theory that is suitable for all kinds of material and loading. The selection of failure criteria is mainly based on the type of material and nature of loading. Hence it is difficult to conclude on only one of such theory. The maximum stress criterion had been employed in the case of TMC, which considers that the composite fails when the stress in the component exceeds the respective allowable limit. This is a simple and direct way to predict failure of composites as interaction between the stresses acting on the lamina is not considered [12]. The maximum allowable stresses of various laminates considered for analysis are in given in Table

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5. Due to limitations in availability of material properties, the maximum allowable stress in compression was assumed to be the same as in tension. The failure criteria option was checked in ANSYS and these allowable stresses were provided as input after which the analyses were carried out. The stresses in each lamina were checked against the allowable limits. It was found the stress experienced by the particulate, uniaxial, cross ply and quasi isotropic laminates were all within the allowable limits. Thus the safety of the compressor rotor blades was ensured under static loading condition. Maximum stress criteria is given by the following equations:

$$-\sigma_{11}^{C} < \sigma_{11} < \sigma_{11}^{T} \tag{5}$$

$$-\sigma_{22}^{C} < \sigma_{22} < \sigma_{22}^{T} \tag{6}$$

$$-\sigma_{33}^{C} < \sigma_{33} < \sigma_{33}^{T} \tag{7}$$

$$|\tau_{12}| < \tau_{12}{}^{y} \tag{8}$$

$$|\tau_{13}| < \tau_{13}{}^{y} \tag{9}$$

$$|\tau_{23}| < \tau_{23}{}^{y} \tag{10}$$

Table 5. Maximum allowable stress of composite laminates [13]

Maximum Allowable Stress (MPa)	0°	45°	90°
σ_L	1518	1275	380
σ_T	380	623	1518
τ_{LT}	326	569	326

5.2. Tsai – Wu criterion

The maximum stress failure criterion does not account for the interaction of stresses, thus Tsai Wu failure criterion that accounts for stress interaction was also employed. It is given by Equation 11. It was found that the resulting value of failure criteria check for composite laminates was less than 1. Thus the safety of the compressor rotor blades is ensured under static loading condition.

$$X_1\,\sigma_{11} + X_2\,\sigma_{22} + X_3\,\sigma_{33} + X_{11}\sigma_{11}^2 + X_{22}\sigma_{22}^2 + X_{33}\sigma_{33}^2 + X_{44}\tau_{23}^2 + X_{55}\tau_{13}^2 + X_{66}\tau_{12}^2$$
$$+ 2X_{12}\sigma_{11}\sigma_{22} + 2X_{13}\sigma_{11}\sigma_{33} + 2X_{23}\sigma_{22}\sigma_{33} < 1 \tag{11}$$

$$X_1 = \frac{1}{\sigma_1^T} - \frac{1}{\sigma_1^C} \tag{12}$$

$$X_{11} = \frac{1}{\sigma_1^T.\sigma_1^C} \tag{13}$$

$$X_{12} = -\frac{1}{2}\sqrt{X_{11}.X_{22}} \tag{14}$$

6. Results and discussion

It is mandatory to provide a sufficient tip clearance such that the tip of the blade does not rub against the casing during operation. This tip clearance is provided based on the radial growth of the blade. When the tip displacement of the blade is more, it implies that the tip clearance required is also more. As the tip clearance increases, the tip loses also increases and this results in decreased efficiency of the gas turbine.

Therefore, it is advantageous to have a lesser radial tip displacement that will require lesser tip clearance and eventually better efficiency. It is evident from the results of static stress analysis of the TMC (as in Table 2) that the maximum radial tip displacement has decreased by 0.4 mm compared to Ti alloy. This implies that for composite blades, the tip clearance can be smaller and hence during off design conditions, the losses would be minimized leading to increased efficiency.

The mass of the composite blades is less by 4 grams compared to the metallic blade, which is another major advantage. This is because of the lesser density of TMC against the Ti alloy. This reduction in mass is vital because it will result in weight reduction of nearly 15% per stage. If all stages of compressor are replaced with composites, the reduction in mass will be tremendous. The reduced mass of the blades could allow further weight saving in discs as well. This weight reduction will result in an overall engine weight reduction and reduced fuel consumption, thus providing operational cost savings.

As far as stresses are concerned, reduction in stress of around 30 MPa is seen in TMC compared to that of metallic blade. It is observed from the stress plots (as in Figure 5) that the major part of the composite airfoil reveals stresses less than 70 MPa against the metallic blade. In case of laminate composite blades, the stresses of individual laminas were also obtained. Tsai Wu criterion and maximum stress criterion have been employed as failure criteria for Ti alloy and TMC, respectively, and the stresses were found to be within the allowable limit. The increase in stiffness was made possible through fibre reinforcement and it provides blade's designers with an opportunity to tune the performance of the blade under load. The increasing material stiffness also changes the resonant frequency of the blade, thereby allowing removal of any damaging vibration modes from the engine running range without any excessive section thickening or added weight.

7. Conclusion

The static stress and modal analysis of Ti alloy and TMC for the compressor blade application have been carried out using FE analysis. The stresses of composite blades obtained in static analysis are within the allowable stress limit. First five natural frequency of the rotor blades made of Ti alloy and TMC were obtained. It is concluded that TMC blades have lower weight, lesser maximum radial tip displacement, better strength, better stiffness and higher natural frequency. The strength of composite has been compared with Ti alloy and its advantages are substantiated quantitatively in the present work. Failure criteria checks show that the metallic and composite rotor blades are both safe under static loading condition. However, a detailed study has to be carried out considering HCF and LCF. Further testing is essential to conclude on the replacement of metallic blades with composites.

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