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STRESS ANALYSIS OF A GAS TURBINE ROTOR USING FINITE ELEMENT MODELING

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ABSTRACT

In this paper, stresses in a gas turbine rotor are investigated using finite element (FE) analysis. Due to the complexity of the geometry, a three-dimensional solid modeling of a group of blades and a sector of the disc was first created on a commercial CAD software and subsequently exported to a general-purpose FE package for analysis. The FE model consists of approximately 25,000 solid tetrahedral elements to mesh both the blades and the disc. Stresses developed due to the turbine operating conditions at high rotational speeds and thermal gradients were analyzed, taking into consideration the material behavior at elevated temperatures as well as loads on the rotor blades due to gas pressure. Results of the present model are validated with earlier simplified analytical predictions reported in the literature. For the present design, stresses at the disc center are shown to be higher than those on the blades; a result that may serve as a guideline in the selection of the materials for both the disc and blades.

1 INTRODUCTION

In recent years, there has been an increased interest in the design and analysis of critical components in gas and steam turbines. In aircraft applications, a gas turbine is an indispensable equipment for power generation for both the main and auxiliary systems. Blade failures in gas turbines have a severe impact on the power generation system. A failure condition may initiate due to a number of phenomena, each having the ability to exclude a unit from service [1]. The most common failure mechanisms which occur within a mature unit are normally associated with the material properties. Failures occur from time to time in aircraft applications, as they do in other engineering structures. However, they are not always examined closely to identify the causes. Similarly, when the failures are observed from time to time, repairs are often made without any careful analysis being undertaken. Most blade failure investigations end with a metallurgical report [5]. A metallurgical examination of the blade establishes whether the failure is related to material defects, heat treatments, etc.

An excessive research has been expended on the analysis of the stresses developed in the blades of gas turbine rotors, with the majority of investigations relying on the use of FE procedures. Attempts have been made by Hou *et al.* [2], who employed nonlinear FE analysis to determine the steady-state stresses and dynamic characteristics of a gas turbine blade. Meguid *et al.* [3] conducted FE analysis to study the fir-tree region in turbine disc assemblies.

In this paper, the FE technique is used to predict rotational, bending and thermal stresses in turbine discs and blades. These are compared with a simplified analytical routine.

2 APPROXIMATE ANALYTICAL SOLUTION

Figure 1 shows an illustration of the turbine rotor under investigation. The rotor consists of a number of blades fitted on a disc of non-uniform thickness. All the blades are constrained by an outer ring called a belt. The outer diameter of the turbine rotor is equal to 0.3 meters.

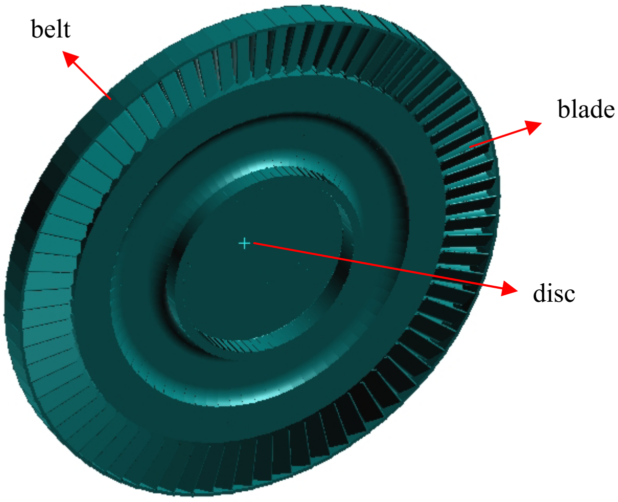

Fig. 1 Gas turbine rotor

2.1 Stress Distribution in a Thin Rotating Disc

Hot pressurized gases are forced into the cavity formed by the blades causing the rotor to operate at a rotational speed exceeding 20,000 rpm. Regardless of the complexity of geometry, a simplified analytical solution for the stresses will first be considered. The gas turbine rotor may approximately be treated as a uniform rotating disc problem, while adjusting the boundary conditions at the outer (free) surface to account for the presence of a certain number of blades. Beside the stresses due to rotation, gas pressure and thermal stresses do exist.

Due to the non-uniform temperature distribution within the disc, thermal stress components would develop along the radial and circumferential directions. Theses stresses can be evaluated using analytical closed form formulas available in text books [4,7].

2.2 Stress Distribution in the Blade

The blades of the rotor are fixed to the outer area of the disc at their roots. The blade is subjected to a flow of hot pressurized gases which force the rotor to rotate at its required speed. Consequently, the blade experiences centrifugal forces which impose a tensile stress, as well as a bending stress due to the gas pressure on the profile of the blade.

The blade may be considered as cantilever as shown in Fig. 2 with fixed end at the blade root and free end at the tip of the blade. Thus we get the maximum stresses on the front and back sides of the blade using elementary strength of materials [4].

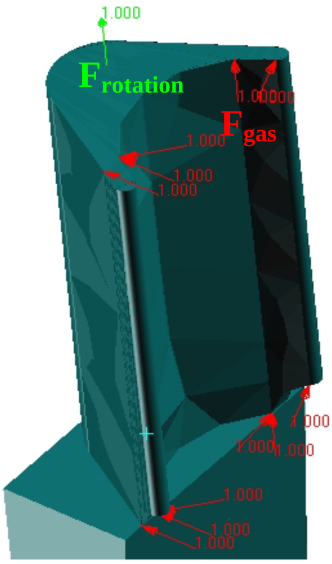

Fig. 2 Distribution of forces on the blade

3 FINITE ELEMENT MODELING

3.1 Geometry

Due to complexity of the blade profile geometry, the pro-engineer CAD software was used to draw a three dimensional quarter of the disc with a group of three blades. In order to compare the results with the simplified analytical solution two models were created where the first model is established with uniform disc thickness and the other is drawn with non-uniform disc thickness. Later, both models were exported to the FE package PATRAN in the form of a parasolid file type.

3.2 Meshing

After importing the parasolid files of both models, four-noded solid tetrahedral element was used to mesh the three blades and the disc. The FE model consists of more than 25,000 elements as shown in Fig. 3 and 4.

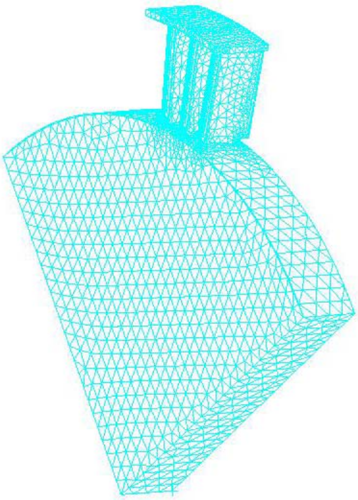

Fig. 3 FE mesh, uniform disc thickness

Fig. 4 FE mesh, non-uniform disc thickness

3.3 Loads and boundary conditions

Three different types of loading were applied to the model. The inertial load was simulated by imposing 400 revolutions per second rotational velocity in the z-direction. The gas pressure load was assigned by a value of $4.5x10^5$ N/m^2 normally on the front side of each blade. Finally, considering the relatively short operating period and high speed of the turbine rotor, all blades are assumed to be subjected to an abrupt high temperature of 800°C from the hot gases upon startup. Due to thermal conduction, heat is transferred from the blades to the disc for the subsequent 100 seconds representing one operational life cycle of the turbine.

For symmetrical boundary conditions (B.C.) of the displacement of the disc, all the nodes at the center were fixed in the three directions x, y and z. The displacement of nodes lying in the y-z plane was fixed in the x-direction. On the other hand, the displacement of nodes lying in the x-z plane was fixed in the y-direction.

3.4 Material used

A chrome-nickel steel alloy was selected, which is a valid representative of the material used for gas turbines [6]. Table 1 shows the alloy chemical composition.

Table 1 Chemical composition

C	Mn	Si	Cr	Ni
0.11	0.5	0.61	23.6	20.65

The material properties were taken to vary with temperatures from 0°C up to 800°C [8]. This is elaborated in Table 2.

Table 2 Engineering constants

Temperature (°C)	0	100	200	300	400	600	800
K	22	22	22	22	24	24	26

Temperature (°C)	25	95	205	315	425	595	815
E (GPa)	200	195	185	175	166	154	136

where E: Young's modulus (GPa), υ: Poisson's ratio, K: Thermal conductivity (W/m°C),
ρ : Density = 7835 (kg/m^3)
α : Coefficient of thermal expansion = 10x10^{-6} (/°C)
C_P: Specific heat = 460 (J/kg°C)

4 RESULTS AND DISCUSSION

The PATRAN FE package was used to run a linear static analysis to solve five different loading cases. These load cases are rotational load (loading case 1), gas pressure load (loading case 2) and temperature gradient as a thermal load (loading case 3), the mechanical loads (rotation and gas pressure – loading case 4), and combining mechanical and thermal loads (loading case 5). In order to obtain the thermal load, a temperature distribution was carried out from a separate solution for transient thermal problem as shown in Fig. 5. Mesh refinement was carried out by systematically increasing the number of elements until converged solutions of the resulting stresses were achieved.

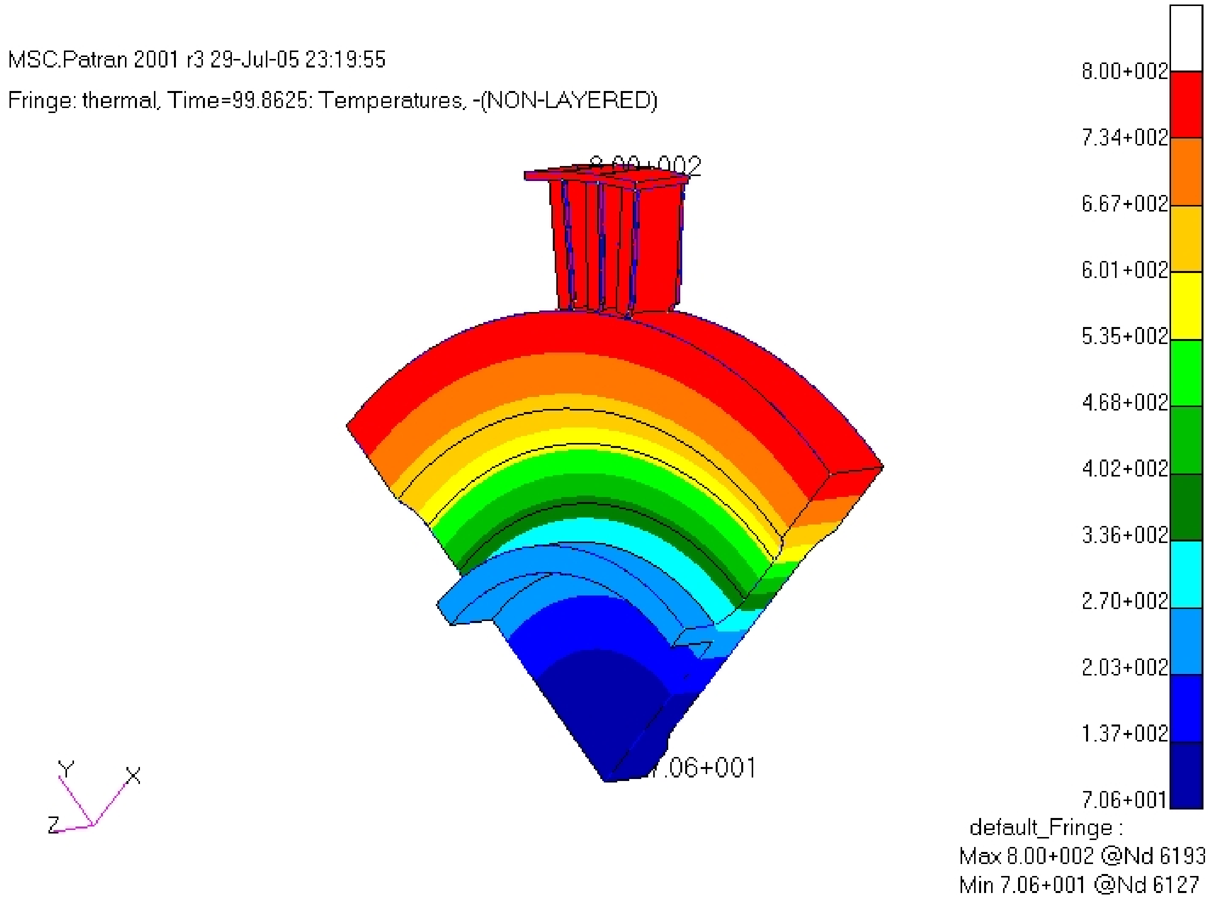

Fig. 5 Temperature distribution in the rotor

A comparison of the results of the present FE analysis of the rotational load of the uniform disc is made with the analytical simplified solution for the rotating disc. The comparison between the two methods is shown in Table 3.

Table 3 Comparison between Analytical and FE solutions (uniform disc)

	Approx. Analytical solution	FE solution
At center of disc	$\sigma_r =$ 197 MPa	$\sigma_r =$ 220 MPa
	$\sigma_\theta =$ 197 MPa	$\sigma_\theta =$ 220 MPa
At outer surface of disc	$\sigma_r =$ 52 MPa	$\sigma_r =$ 62 MPa
	$\sigma_\theta =$ 114 MPa	$\sigma_\theta =$ 130 MPa

As indicated, the stresses computed from the analytical method are in reasonable agreement with the FE predictions, which supports the validity of the three dimensional FE procedure employed.

The first four previously mentioned load cases were then applied to the model with a non-uniform disc thickness, and the results in terms of the Von Mises stress distributions are shown in Figs. 6 – 9. The fifth loading case is shown later.

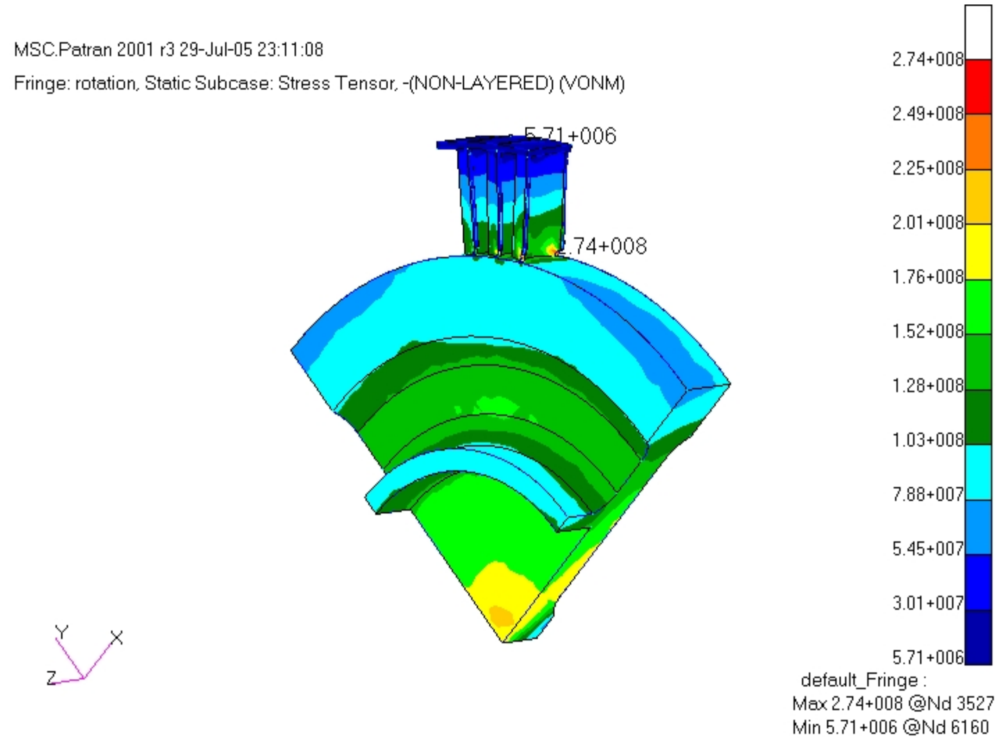

Fig. 6 Case 1: rotational load

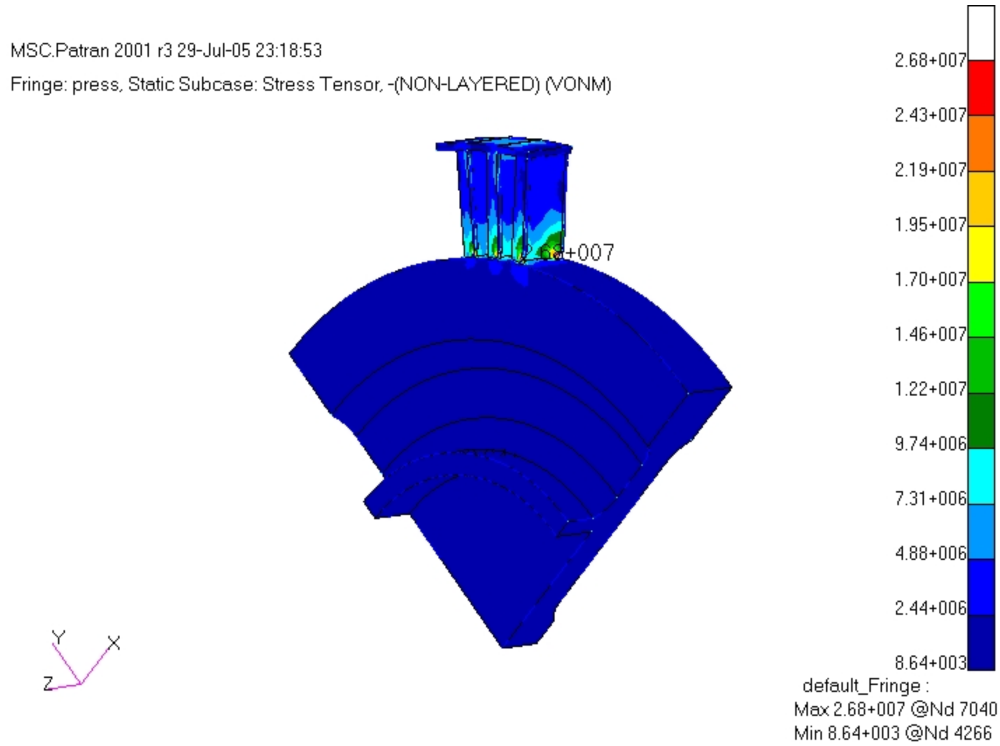

Fig. 7 Case 2: gas pressure load

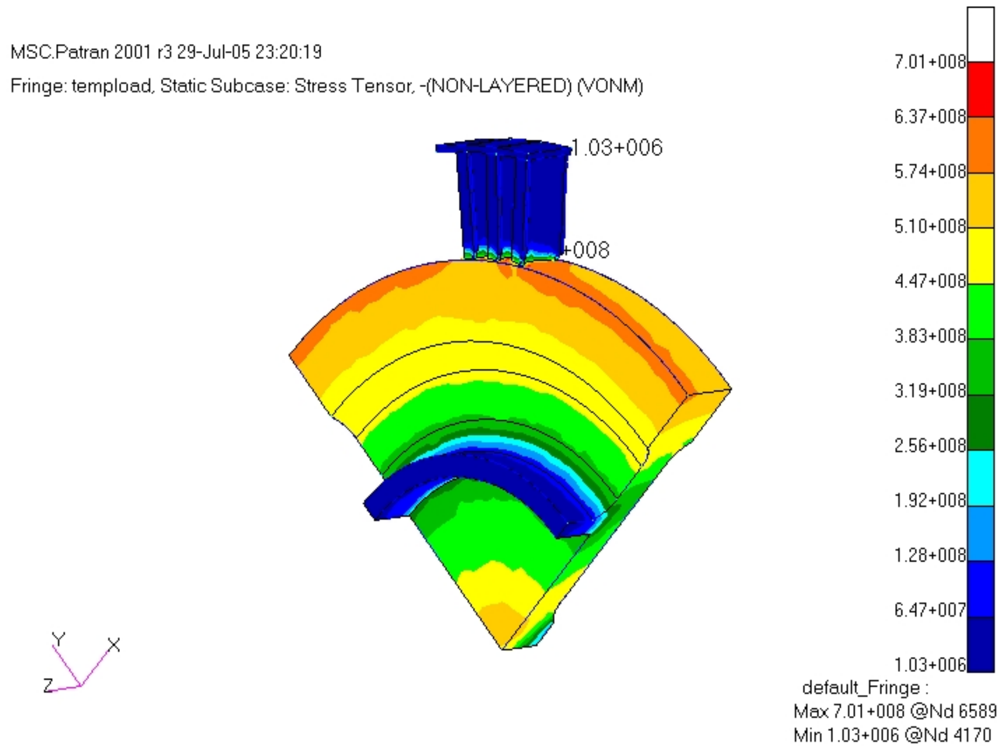

Fig. 8 Case 3: thermal load

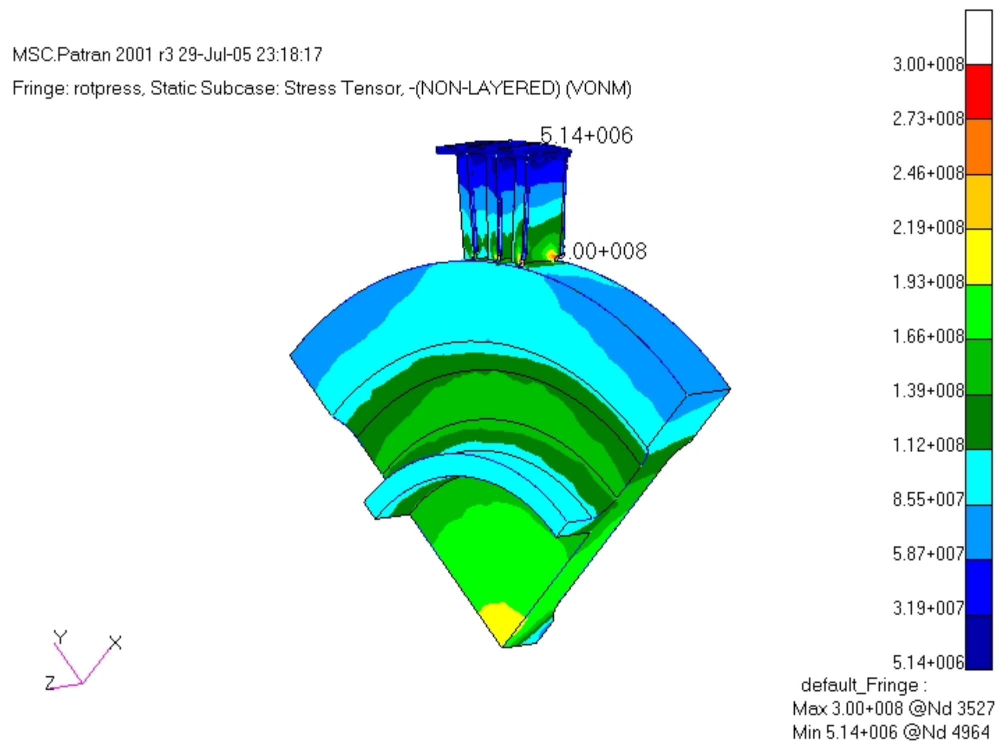

Fig. 9 Case 4: rotation and gas pressure load

In addition, a comparison between the FE analysis and the analytical method is also done for the front side and back side of the blade as indicated in Table 4.

Table 4 Comparison of stresses on front and backside of the blade

	Approx. analytical solution	FE solution
Max. Von Mises stress at front side of blade	σ_{max}= 97 MPa	σ_{max}= 112 MPa
Max. Von Mises stress at back side of blade	σ_{max}= 175 MPa	σ_{max}= 139 MPa

Once again, results obtained from the analytical solution compare fairly well with the FE predictions.

Finally, the FE analysis of non-uniform disc for the rotation, gas pressure and thermal load (loading case 5) is elaborated in Fig. 10 and 11 to show the maximum von Mises stresses as well as the resultant deformation.

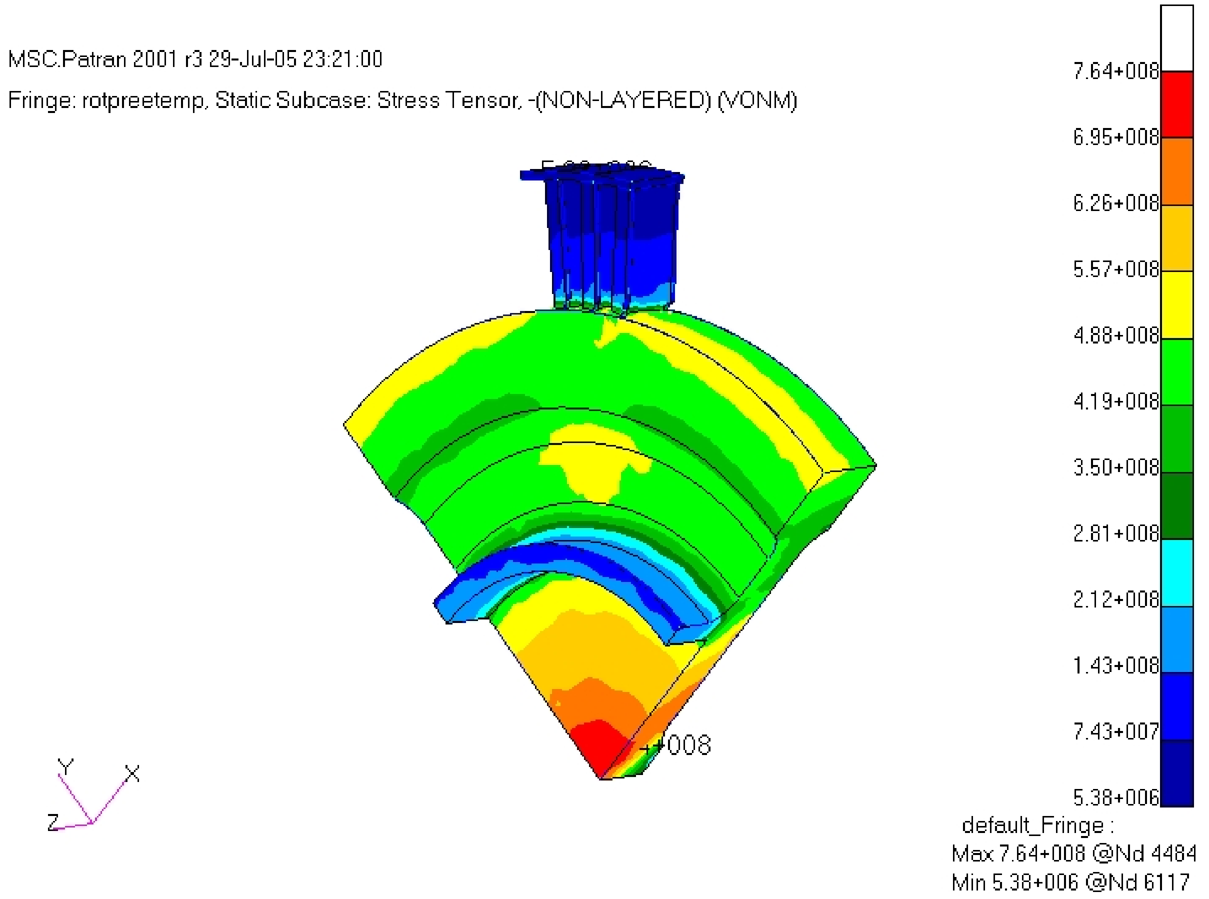

Fig. 10 Case 5: Stress distribution for rotation, gas pressure and thermal load

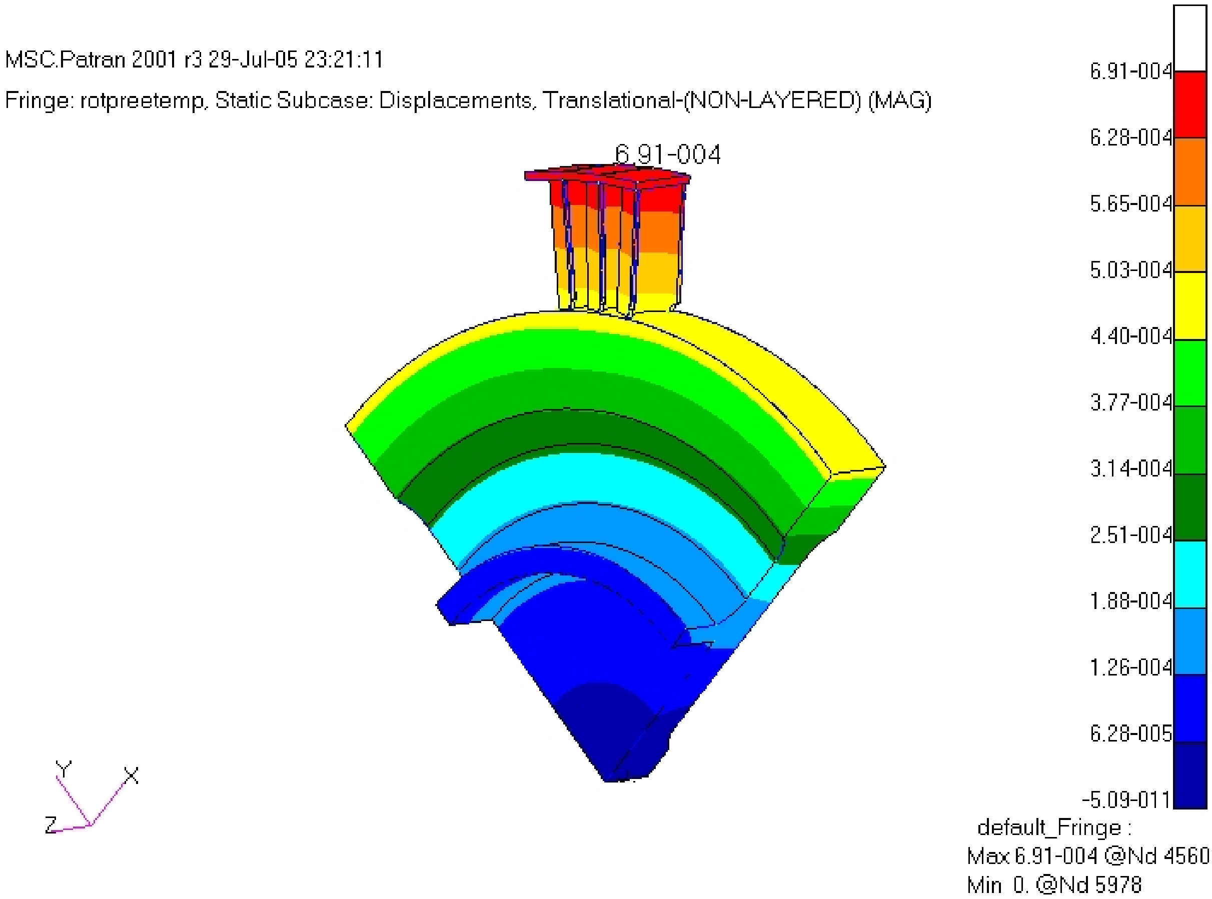

Fig. 11 Displacement distribution for rotation, gas pressure and thermal load

5 CONCLUSION

This paper presents a FE analysis of the stresses developed in a gas turbine rotor. Due to the complexity of the geometry, a 3-D solid modeling of a group of blades and a sector of the disc was first created on a commercial CAD software and subsequently exported to a general-purpose FE software for analysis. The FE model consists of approximately 25,000 solid tetrahedral elements to mesh both the blades and the disc. Several loading cases were investigated in order to assess the relative contribution of the various loads imposed on the turbine during operation, namely stresses due to rotation of the turbine, gas pressure on the blades, and stresses resulting from thermal gradients. Linear static analysis, as well as transient thermal analysis, were conducted, taking into consideration the material behavior at elevated temperatures. The results were compared with a simplified analytical solution that gives approximate values of the stresses at the disc and blades for a disc with uniform thickness. The FE model was extended to handle the non-uniform disc problem.

Inspection of the results reveals that the greatest stresses in the disc and blades result from rotational loads, owing to the relatively high operating speed of the turbine. Bending stresses on the blades due to gas pressure are significantly lower than those due to rotation. Thermal stresses on the blades are also quite low as the blades are free to expand. However, thermal stresses on the disc are quite high owing to the sharp temperature gradient that exists in the turbine disc.

For design purposes, material selection is quite crucial since the material must maintain its mechanical properties at elevated temperatures. For the present design, stresses at the disc center are shown to be higher than those on the blades; a result that may serve as a guideline in the selection of the materials for both the disc and blades.

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