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STEADY STATE STRESS ANALYSIS AND HEAT TRANSFER ANALYSIS ON AN AXIAL FLOW GAS TURBINE BLADES AND DISK.

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ABSTRACT: To be able to design next generation of gas turbines it is necessary to improve the knowledge about the stress values produced in the turbine. A gas turbine rotating system is loaded with time by the changes in stress levels as a result of start-up and shutdown procedures, i.e. Low-cycle fatigue (lcf), as well as by steady forces caused by rotation (centrifugal stress) and thermal gradients, as also by high-cycle fatigue(hcf) in the course of normal operating conditions. Except during the start-up and shutdown procedures, the emphasis are given on the steady state analysis because of the turbine's maximum operating period is in almost steady state. The main entity of these versatile machines is made by the blades and vanes, which are subjected during operation to very high thermal and mechanical stresses (combined effects of centrifugal force and thermal gradient), in aggressive environment. The research on gas turbine cooling systems is coupled with the flow and heat transfer associated with rotating turbine.
Therefore this report has primarily focused on the heat transfer characteristics, centrifugal and the thermal stresses arising in the disk. The maximum stresses obtained from different analyses by using innovative high heat resistant material inconel 718 are found to be within the yield strength of the material. Interesting results obtained in terms of maximum operational radial stress, maximum operational hoop stress, maximum operational Vonmises stress, the temperature field etc. The values of the stresses indicate that the disk attains steady state after 450 seconds. And the disk is expected to perform well in spite of all the stringent operating conditions. The object is to provide understanding and information for designers to improve the life and efficiency of future generations of engines.

Key words: Steady state heat transfer, thermal stresses, Axial flow gas turbine

1. INTRODUCTION

The gas turbine is a primary energy deliverer not only for vehicular propulsion of such as air, land and water, but also for power generation. Several major factors affect thermal efficiency or specific fuel consumption of a gas turbine plant. These include:

- Increase in the turbine inlet temperature, namely firing temperature.
- Reduction of cooling air usage.
- Improving component efficiency.
- Cycle enhancement.

To consider the first factor as the factor to increase the efficiency the detail characteristics of the thermal load and heat transfer characteristics are to be known. Gas turbine blades operate under severe stress conditions induced by high gas temperatures and high rotating speeds.

The thermal performance and specific thrust of gas turbines can only be improved significantly by increasing the turbine inlet temperature. This approach is limited, however, by the availability of appropriate materials that withstand designed temperatures. The working temperatures that are encountered in other power plants are higher than those in gas turbines but there is an important difference. For instance, in the case of high output internal combustion engines, temperatures of the order 3000 deg C are encountered. These high temperatures are prevalent only for a milli-second. Moreover, the gas temperature oscillates from high to low values. So the metal surfaces do not attain such high temperatures and they are well cooled with either water or air. Therefore, the maximum metal temperature in the case of internal combustion engines rarely exceeds 200 -250 deg C. Over the last decade, the average temperature of combustion gas entering the first stage turbine in high performance gas turbines has increased from 1200 deg C to 1450deg C. Of this 250 deg C increase, improved alloys contributed to 65deg C while improved cooling contributed the rest. Highly sophisticated cooling techniques have to be employed in order to maintain acceptable life and safety requirements under such extreme operating conditions. Apart from these high temperatures, the blades and disks are subjected to tensile and compressive stresses. For satisfactory performance of gas turbines at elevated temperatures, cooling of gas turbine components is more promising than raising the strength of turbine materials.

The present study aims at carrying out steady state thermal analysis of an INCONEL turbine rotor which rotates at very high speeds of the order 50,000 rpm. This turbine is a single stage axial flow, partial admission impulse turbine which receives high temperature gases up to 1300 deg C. The turbine blades and disk faces are to be cooled against the hot gases. This is accomplished by impinging water on the turbine blades. The water is injected in the form of spray to cool the hot gases near the disk which in turn results in the cooling of disk faces. The hot gases after doing the work get accumu lated in the exhaust hood of the turbine for a little while before exiting through the output shaft. The churning action of the gases in the exhaust hood of the turbine causes the heating up of the downstream face of the disk. So, the downstream face of the disk should be cooled. This analysis is aimed at cross checking the adequacy of water cooling for the turbine disk. It should be emphasized that the present disk must be analyzed under two categories of stresses. The first kind is due to the centrifugal stresses that act on the disk due to its high rotational speed and the second kind, the thermal stresses that arise due to the thermal gradients. The design and analysis of turbine disk consists mainly the following parts: Structural analysis and thermal analysis with half- hot conditions are conducted using ANSYS. Steady state thermal and stress analysis, which consists of computation of heat transfer coefficients at various cross sections of the disk. Thermal analysis is conducted using ANSYS by making use of the heat transfer coefficients as boundary conditions.

2. NOMENCLATURE

Re Reynolds's number
Pr Prandtl number
St Stanton number
Nu Nusselt number
C_{fx} local friction coefficient
$Nu_r = 0.020\ Pr^{0.333}\ Re^{0.8}$

$$Nu_r = hr/k$$

$$Pr = \mu C_p / k$$

$$Re_r = \omega r^2 / \nu$$

$$\omega = 2\pi N / 60$$

Where
ω Angular velocity of the disk
N rotational speed of the disk
Re_r rotational Reynolds number.
ν kinematics viscosity
K thermal conductivity
E Young's Modulus
σ_u Ultimate strength
σ_y Yield strength
α Thermal coefficient
Cp Specific heat
ρ Mass Density

3. DETAILS AND DESIGN CRITERIA

3.1 Material

The turbine disk is simultaneously subjected to high temperature gradients and centrifugal forces that require a unique blend of material properties high strength, good fracture toughness, and low crack growth rate. The material chosen for the application is INCONEL 718 which is an alloy made from niobium, chromium and nickel. This material was chosen for its strength at elevated temperatures, corrosion resistance and the relative ease of manufacture. This material also retains its strength at elevated temperatures. The disk is forged as this method of manufacture has been very successful in producing uniform properties

3.2 Disk Details

The design life is 20 cycles, each lasting for 30 minutes.
The turbine-blade dimensions are as follows:
Mean diameter = 224.00 mm
Tip diameter = 238.60 mm
Hub diameter = 208.40 mm
Axial chord = 12.00mm
Blade height = 14.60 mm
As the blades are integrally machined from the disk forging, the above dimensions set the dimensions of the rim of the disk.
To prevent the excessive thrust load on the turbine shaft bearing, the turbine is provided with the pressure balance holes in the web region. The holes are defined by:
Hole PCD = 150.00mm
No. of holes = 6
Hole diameter = 6.00 mm.
The rotational speed of the disk is 50,000 rpm.

3.3 Mechanical properties

The physical properties of the material such as elastic modulus, Poisson's ratio, coefficient of thermal expansion etc. vary with the temperature. As the disk is subjected to severe thermal gradients, the nonlinear nature of the properties should also be taken into account. Generally, the density of the material is constant throughout the disk even under a wide range of temperatures. Poisson's ratio has only an insignificant effect on the stress distribution. Accurate stress calculations would therefore require accurate data of the elastic properties of the disk material.

Table 3.1

Property	Temp in ^oC	30	200	350	500	650
	Unit					
E	GPa	202.7	194.6	186.5	178.3	170.6

σ_u	MPa	1240	1161	1201	1117	1067
σ_y	MPa	1035	969	1002	932	890
K	W/m K	11.77	14.90	16.27	18.17	19.73
α	$*10^6/°C$	12.42	13.32	13.95	14.40	14.76
Cp	J/kg K	418.6	460.5	502.4	544.3	586.2
ν		0.29				
ρ	kg/m^3	8220				

3.4 Design Criteria

The turbine drive system utilizes single stage impulse turbines. The rotor disk and blades for this turbine are machined from a single integral forging. This turbine is clamped by four M6 bolts to the flange of main shaft circumferentially. The gas turbine operating conditions are very severe ranging from regions of low temperature and moderate stress to regions of very high temperature and high stress. Close dimensional control must be maintained which requires consideration of deformation and rigidity. The associated problems are service life, size, and weight, cost of materials and processing methods of materials. The life, size and weight are dictated by the functional requirements. It is necessary to know for a given temperature and a given expected life, the maximum amount of stress that can be carried within the assigned criterion for failure. The criterion for failure could be rupture or specified allowable deformation. The nature of the loads also should be considered as static loads lead to the consideration of static strength and fluctuating loads require consideration of endurance or fatigue strength. The three-dimensional flow around a guide vane should be studied numerically. The fluid flow and heat transfer in the region of the end plates should be the focus of the investigation. The purpose of this work should be to understand the influence of critical parameters on the platform heat transfer.

Fig 3.1 Axial flow gas turbine

3.5 Temperature gradient

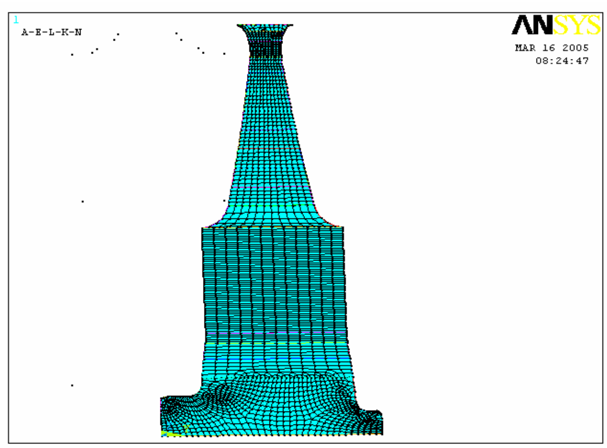
The disk is subjected to rapid heating by the combustion gases. The rim of the disk and the blades are heated by the primary flow of gases. The temperature of the combustion gas at nozzle entry is fixed at 1310º C. Since the turbine used here is the impulse type, most of the temperature drop takes place within the nozzle. The temperature of the gas coming out of the nozzle is around 887 ºC. The turbine disk faces are to be cooled against this temperature. This will be achieved by injecting water in the form of fine spray onto the blades and the disk faces. The quantity of the water injected is so decided that the temperature of the gas on the upstream and downstream faces is 260 ºC. The static temperature of the fluid in contact with the blade is 300 ºC. The analyses were conducted assuming these temperatures.

3.6 Fatigue

The turbines are known to suffer from fatigue on components that undergo small relative displacements under high forces. The blade disk attachments and roots of rotating air foils have to endure high centrifugal forces leading to elevated contact stresses which combine with relative small displacements that occur as the engine undergoes through its operational cycle. They make fatigue an important, if not the main design criterion. As the design life is 20 cycles, fatigue is not a major concern in the design

4. Finite element modeling

A two dimensional, axisymmetric finite element model was employed for the sake of simplicity. This model was also used to simulate the nonlinearity of the material properties. The physics environment approach is chosen. The thermal problem is defined. The thermal conductivity, specific heat and density values are given as material properties and are written in the thermal physics file. The thermal environment is cleared and then the structural problem is defined. The Young's modulus, expansion coefficients are given as material properties and are written in the structural physics file. The thermal physics file is then read and then the heat transfer coefficients are given as inputs along with the bulk temperature of the fluid at various sections of the disk. Solving the thermal problem gives the temperature distribution after attaining steady state conditions. The structural physics file is then read. The angular velocity and displacement constraints are applied. The temperature distribution from thermal results file is also read. This model is solved to obtain the stresses and displacements.



4.1 Meshing

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5. Structural Analysis:

The disk and the loads that are applied on it are axisymmetric in nature. Thus, the disk is modeled with axisymmetric, 8 noded, isoperimetric, quadrilateral elements whereas the blade section is modeled as plane stress elements with thickness as input. The stiffness effects of the blades were simulated by modeling the blades using plane stress elements. The centrifugal load due to the disk is applied by defining the angular velocity and the density of the material. ANSYS combines the angular velocity with the element mass matrices to form a body force load vector. The blade passage is divided into 12 equal sections and the area of each section is calculated by AUTOCAD. On dividing the area by the height of the blade, the blade thickness at each section is determined. The values are given in table 4.1. These calculated values are given as thickness input for the blade elements. The other input parameters being the speed of rotation 50,000 rpm and the density of the disk.
The values of radial, hoop and Vonmises stresses are depicted in the figures 5.1, 5.2, 5.3, 5.4 respectively. The maximum radial displacement is found to be 0.284 mm from figure 5.5.

Assumptions:

1. The disk material is completely elastic at the stress distribution induced by the centrifugal and thermal effects.
2. The stress is proportional to the strain.
3. All the variables of material properties are symmetric about the axes.
4. Temperatures are taken in the central plane perpendicular to the axis of the disk.

Table 4.1

Sl. No	Area from CAD mm^2	Wid-th mm	Thickness mm	No of Blades	Thick-ness input m
1	1.3312	1	1.3312	94	0.125
2	2.8717	1	2.8717	94	0.270
3	3.9405	1	3.9405	94	0.370
4	4.5103	1	4.5103	94	0.424
5	4.7367	1	4.7367	94	0.445
6	4.7618	1	4.7618	94	0.448
7	4.6580	1	4.6580	94	0.438
8	4.4313	1	4.4313	94	0.417
9	4.0072	1	4.0072	94	0.377
10	3.1025	1	3.1025	94	0.292
11	2.0420	1	2.0420	94	0.192
12	0.9745	1	0.9745	94	0.092

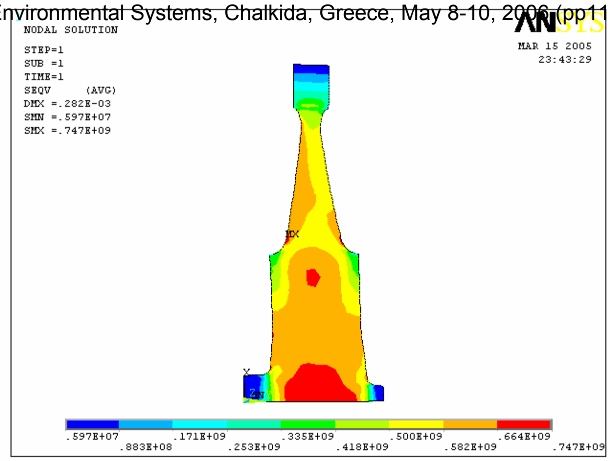

Fig 5.1 Structural Analysis Radial Stress Distribution (Pa).

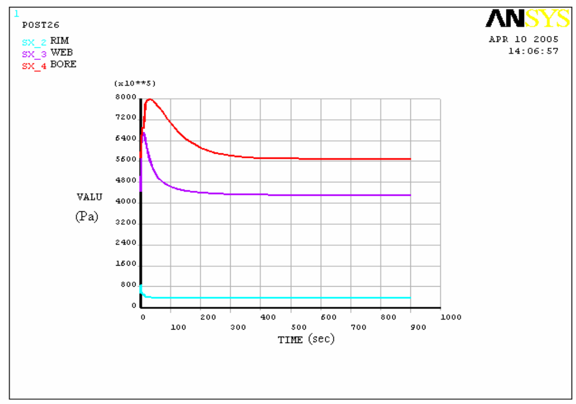

Fig 5.2. Radial stress variation in different parts of the disk with time.

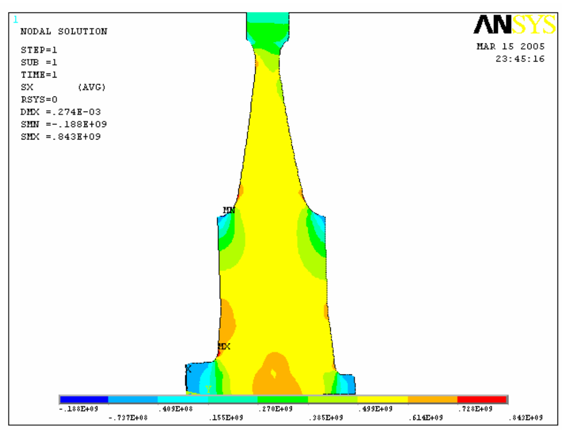

Fig. 5.3 Structural Analysis Hoop Stress Distribution (Pa).

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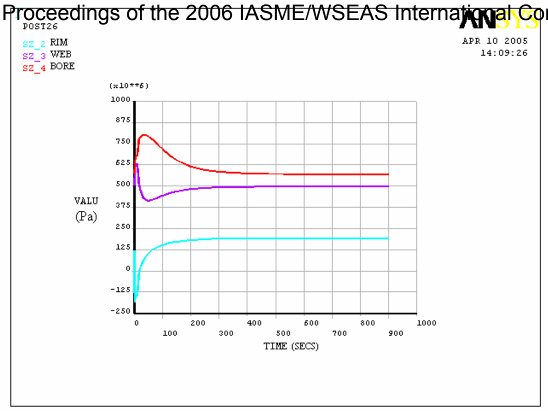

Fig.5.4 Hoop stress variation in different parts of the disk with time.

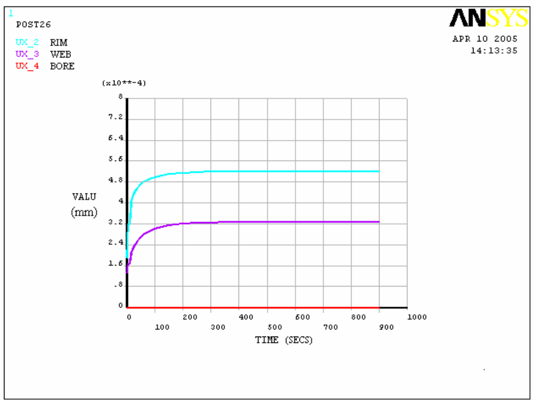

fig 5.5 Radial displacement variation in different Parts of the disk with time

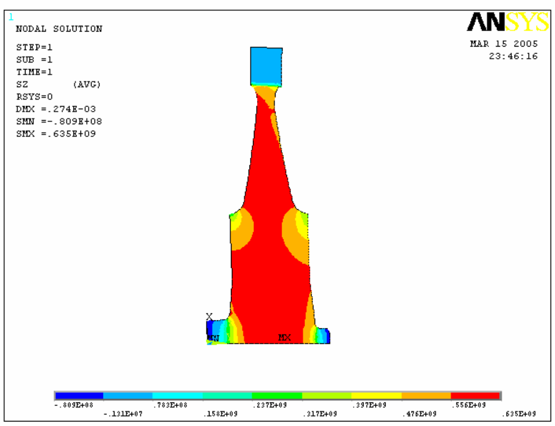

Fig. 5.6 Structural Analysis Vonmises Stress Distribution (Pa).

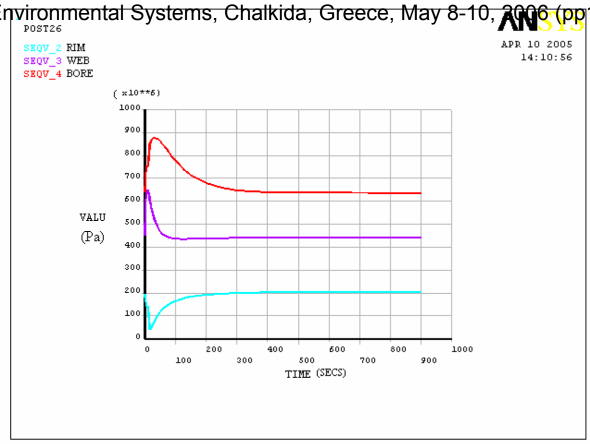

Fig. 5.7 Vonmises stress variation in different Parts of the disk with time.

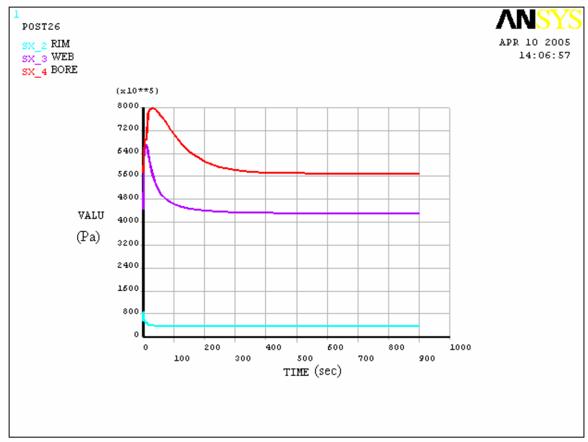

Fig 5.8 Variation of radial stress in different Parts of the disk with time

6. Heat transfer in blades and across disk faces:

The blade section is considered to be a flat plate and assuming that fully developed turbulent flow conditions exist, the heat transfer coefficient is calculated using the Chilton–Colburn analogy

$$St_x \, Pr^{0.666} = Cf_x / 2$$

C_{fx} local friction coefficient which is correlated by the expression of the form,

$$Cf_x = 0.0529 \, Re^{-0.2}$$

Which is applicable for: $Re < 10^8$, $0.1 < Pr < 60$.

The heat transfer coefficients across disk faces are calculated by making use of the turbulent local Nusselt number for an isothermal rotating disk.

$$Nu_r = 0.020 \, Pr^{0.333} \, Re^{0.8}$$

The convective heat transfer coefficients are calculated at various radial sections for various metal temperatures. Mean Heat transfer coefficient obtained at various sections are 1595.451, 2822.089, 3683.486, 4671.795, 5380.613, 5752.433, 6130.519, 6467.066 W/m^2 K

7 Thermal analyses:

The thermal form is similar to the structural form but has a single degree of freedom at each node, the temperature. Axisymmetric boundary conditions in the form of displacement constraints and thermal gradients are applied on the model. Preliminary thermal analysis is performed on the disk with half-hot condition i.e., the temperature of 500°C is applied on the nodes along the rim whereas the bore is assumed to be exposed to a temperature of 250°C. This forms the worst possible case that can be expected while the turbine is in operation. However, it was found from transient analysis that the stresses developed with the above boundary conditions were very high thus not meeting the design criteria. Therefore, it was decided to cool the blades down to 300°C by injecting water. Finally, the thermal analysis is conducted with 300 °C at the rim and 30 °C at the bore with linear variation of temperature in between. The blades are not modeled in this analysis. The thermal stresses are obtained from this analysis. Figures 7.1, 7.2, 7.3 represent the radial, hoop and Vonmises stress distributions across the disk.

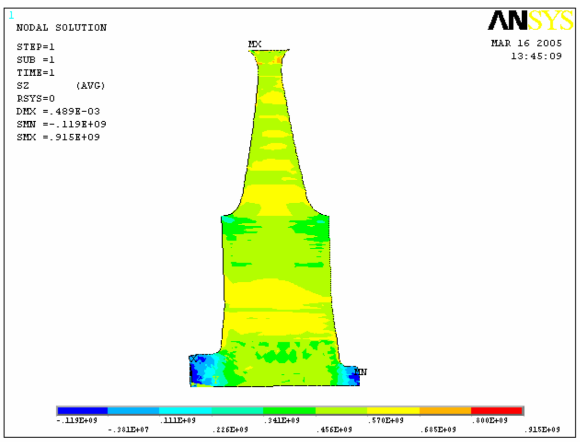

Fig 7.1 Preliminary Thermal Analysis Vonmises Stress Distribution. (Pa).

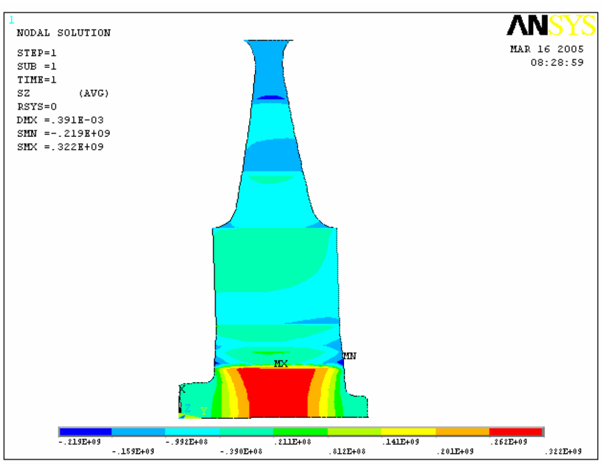

Fig 7.2 Preliminary Thermal Analysis Hoop Stress. (Pa).

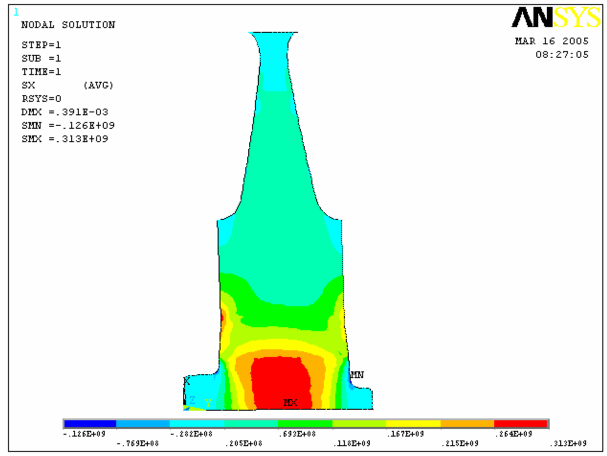

Fig 7.3 Preliminary Thermal Analysis Radial Stress. (Pa).

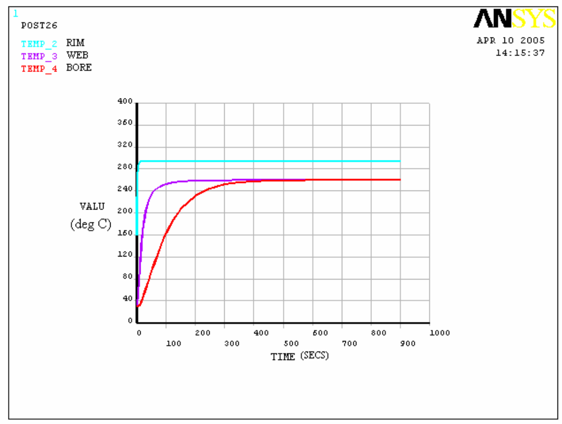

Fig 7.4 Temperature variation in deferent parts of the disk with time

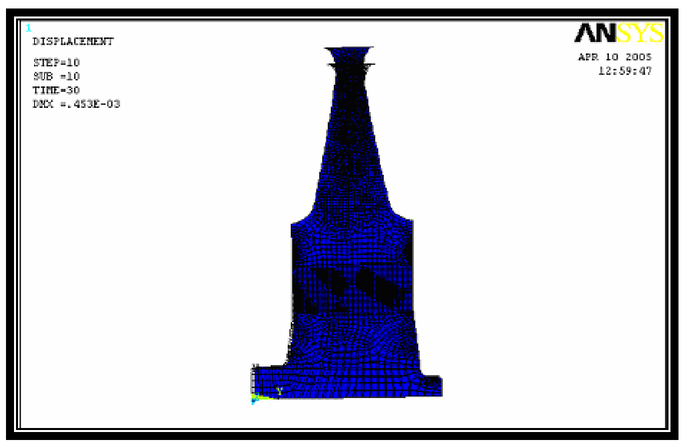

Fig 7.5 Steady state thermo structural analysis Displacement plot (pa)

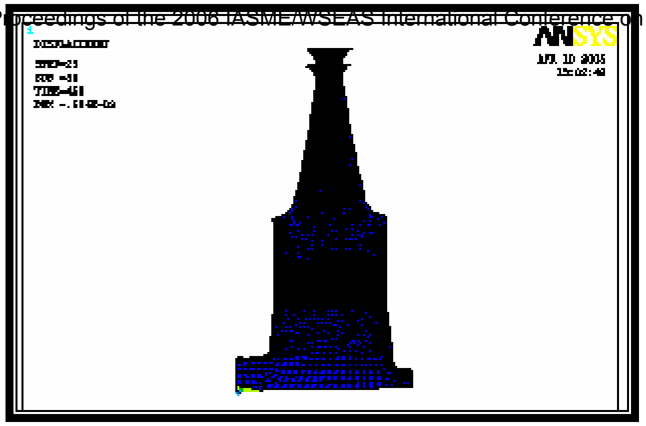

Fig 7.6 Steady state thermo structural analysis
Displacement plot (mm)

8. RESULTS & CONCLUSIONS:

The results and conclusions are presented of a study concerning the durability problems experienced with gas turbine engines. The investigation encompassed the design and failure history of hot gas path components in the engines. From the experimentation the Maximum operational radial stress which is a combination of thermal and structural analysis is obtained as 748 MPa. The maximum operational hoop stress is 571 MPa and finally the maximum operational Vonmises stress is 656 MPa. All the above values are within the yield strength of the material. From the graphs the stress values in the rim are greater than the stress values in the web and also the stress values in the bore. The stress values in the web are found to be intermediate because the rim of the disk is heated by contact with the hot gas. The bore being relatively far away from the gases is cooler.

The values of the stresses indicate that the disk attains steady state after 450 seconds.

It is concluded from fig. 6.5, 66 that the rim region undergoes the maximum displacement. The bore region doesn't show any expansion being far away from the hot gases.

The present study has conducted the detailed heat transfer analysis on the disk. This study has primarily focused on the centrifugal and the thermal stresses arising in the disk. From the results obtained it is noticed that the values are in tolerable limits. Hence the disk is expected to perform well in spite of all the stringent operating conditions.

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