Stress Analysis of Axial Turbine Rotor For Gas Turbine Engines

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Abstract-*An analytical method is presented for the calculation of elastic stresses induced in the symmetrical disks of the gas turbines. These disks cannot be simply analysed with conventional analytical method. So Finite Difference Method was adopted to evaluate the elastic stresses in disk. An excel program was written to find out distribution of hoop stress, radial stress. The program doesn't evaluate axial stress because of axis symmetrical nature of loading circumference and the shearing stresses will be zero everywhere. Hoop and Radial stresses are assumed to be constant along the thickness and the error introduced by the assumptions is small when the disc (diameter/thickness) ratio is large. In excel program the stress concentration feature such as bolt whole are not modelled so a 3-D analysis with all geometrical features in ANSYS 14.0 FEM program was done. A thermal analysis was done to get temperature distribution over turbine rotor.*

Keywords-Stress Analysis, Axial turbine, Rotor, Gas turbines

I. INTRODUCTION

Gas turbine in many respects is the most satisfactory one of the various means of producing mechanical power. The absence of reciprocating and rubbing members means fewer balancing problems that the lubricating oil consumption is exceptionally low, and reliability can be high. The gas turbine in its most common form is a heat engine operating by means of a series of process consisting of compression of air taken from the atmosphere, increase of gas temperature by the constant pressure combustion of fuel in the air, the whole being a continuous flow process. Gas turbine engine are most widely used for two reasons their primary use is in the aircraft engine. Gas turbine engine are also used in heavy industry for the purpose of power generation and in marine propulsion. The turbine used in gas turbine engines is intended for expansion of hot gases and thereby extracting power from hot gases. This power is used to drive compressors. The turbine disks have to sustain various loads like centrifugal loads by blades and its own mass, thermal loads etc. These turbine disks are generally symmetrical. The objective of this project is to find out stresses induced in a typical turbine disk and to calculate factor of safety. This calculation is done for alternate materials and the objective is to find out most suitable material for the turbine disk.

II. OBJECTIVE& METHODOLOGY

The 2-D Disc was theoretically solved using discretized ring technique with sum and difference method. The result is compared with ANSYS output. Both the results are similar for 2-D Disc therefore it is assumed as ANSYS output and theoretical values are similar. The turbine used in gas turbine engines is intended for expansion of hot gases and thereby extracting power from hot gases. This power is used to drive compressors. The turbine disks have to sustain various loads like centrifugal loads by blades and its own mass, thermal loads etc. These turbine disks are generally symmetrical. The objective of this project is to find out stresses induced in a typical turbine disk and to calculate factor of safety. This calculation is done for alternate materials and the objective is to find out most suitable material for the turbine disk.

III. TURBOMACHINE DISC DESIGN

The primary function of a turbo machine disc is to locate and rotate the rotor blades in the gas flow path. This could also be achieved by locating the blades on the surface of a drum; however, there are instances when a single discrete component such as a disc is the most efficient means of meeting the requirement. The secondary functions of a rotor disc are to act as a pressure barrier and to form part of the torque load path. The way in which the cob-diaphragm rim configuration carries the inertia load generated by the blades, the fixings, the shrouds, the lock plates etc. has been previously discussed in loads „Loads Acting on a Turbo machine‟. It has been shown, through the concept of the „free hoop radius‟ that the cob and material inside the free hoop radius helps to carry the material outside the free hoop radius.

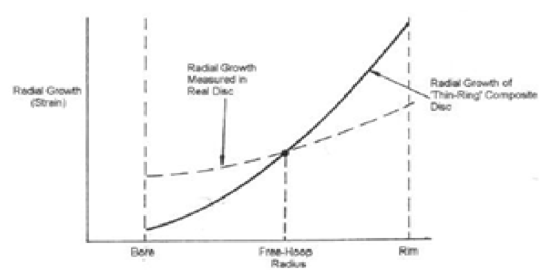

Fig 1. Free Hoop radius

III. DISCRETISED RING TECHNIQUE

First The general turbo machine has an irregular shape. A typical disc profile is shown in fig 10 and has a thickened portion close to the bore and another thickened portion at the rim. For such a shape, there is no closed- form analytical solution and some form of approximation must be used. The popular finite element and boundary element methods are themselves approximate techniques and accuracy is enhanced by the selection of a sufficiently large number of elements. Naturally, for a disc model that involves a large number of elements, computational methods are necessary. However, it is possible to represent a typical turbo machine disc with relatively few elements. Since the turbo machine disc is rotationally symmetric, it can be represented, with a high degree of accuracy, by a number of constant thickness rings. The disc shown in fig 02 has been represented in this way

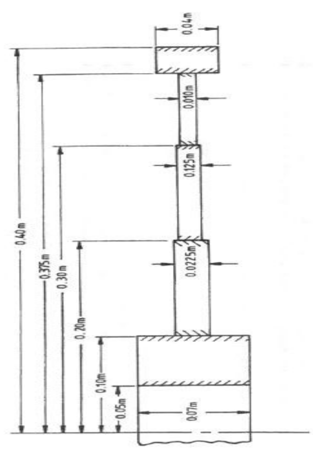

Figure 2 Turbo machine Disc

If two rings can be represented as shown in fig 03, then to progress from one ring to the next the increments in radial stress and hoop stress across the interface between the two rings must be evaluated. Naturally in real rings radial stress and hoop stress will be continuous and it would be unusual for there to be a step change.

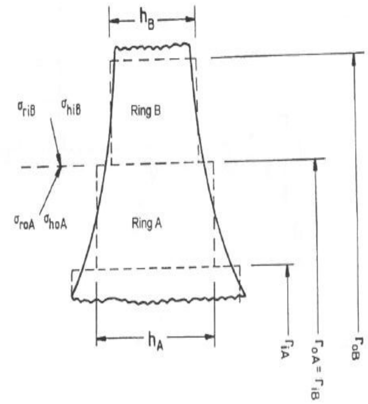

Figure 3 Adjacent Ring Elements

The definition of quantities in the progression from one element to the next is achieved by the use of appropriate subscripts. The manipulation of the stress equation is simplified by the use of „sum" and „difference" terms. The aim of the exercise is to calculate the increments of radial and tangential stress as progress is made from one ring to the next.

IV. DETERMINATION OF STRESS INCREMENTS

By progressing radially outward through the ring elements of the disc, if the sum and differences of the radial and hoop stresses at the inside of element A are known, then the sum and difference terms at the outside of element A can be calculated from a knowledge of the element radii, the density of the disc material and the disc rotational speed by means of Sum and Difference Method for the Calculation of Stress Increments.The elements A and B now need to be linked together so that the increments of stress can be obtained so that progress can be made from one ring to the adjacent ring. The classic way of relating conditions at the outer radius of element A.The equilibrium condition requires that the total outward radial force acting on the outer surface of element A is just balanced by the total inward force on the inner surface of element B. Similarly the compatibility condition requires that the strain at the outer face of element A is equal to the strain at the inner surface of element B. Compatibility considerations also allow the inclusion of the effect of temperature gradient from bore to rim.However, properties such as the elastic modulus, E, temperature, T, and the coefficient of thermal expansion, alpha will need to be averaged over the depth of each element.This method presents a difficulty in that there are not enough boundary conditions at the bore or the rim to allow the evaluation of the two constants of integration. The radial stress at the bore is usually zero for most modern gas turbine design or some known compressive stress if the disc has been shrunk on to the shaft as is often the case with steam turbine design. The radial stress at the rim can also be readily determined since it represents the inertial load produced by the blades, spacers, lock plates etc. in the form of a rim stress. Unfortunately the hoop stresses at bore and the rim are unknown. The way to proceed is to assume a hoop stress at the bore and to calculate the resulting stress distribution through the disc. The radial stress at the rim, calculated from an assumed hoop stress at the bore, can then be compared with the real radial stress at the rim. The hoop stress at the bore can be adjusted in an iterative process until the calculated rim stress is equal to the real rim stress.The calculation is facilitated if the numbers are subjected to an Excel spreadsheet in which the correct stress distribution can be obtained after very few iterations.

V. PROBLEM DESCRIPTION

A Suitable material is to be selected for a turbine disk which runs at 7823 rpm. The temperature varies at a range of 408 to 700 ^{o}C from bore to the rim. The turbine disk rim is attached with 32 blades. A suitable material is to be selected for the given disk and the results from the analytical (Sum and Difference Method) are compared with the ANSYS results. The Materials selected for the analysis are,Nimonic-90, Inconnel-718, Hr Steel(Type 17-22a(S)). The material selected for the analysis are calculated the stress values with temperature gradient, blade load and the given dimensions. The Analysis is carried out in two stages. First The disk profile is analysed with the help of Sum and Difference Method. In this method the disk is analysed at four different stations. By this the Radial, Tangential and Equivalent stresses are calculated. Then The 3D Blade profile is created in ANSYS and with the input parameters the stress values are obtained from the result.

A. Meshing of the model

Meshing is the process of reducing the given models into elements and nodes so that the given problem could be solved with high accuracy. Four iterations are carried out with four different mesh parameters and corresponding boundary conditions

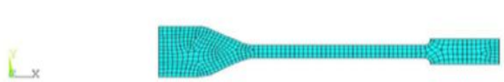

Fig 4 Meshed single rotor

Component	Dimensions
Inner radius	0.2767 m
Outer radius	0.3215 m
thickness	0.08136 m
Hub to tip ratio	0.855

Table 1 2D Rotor Dimensions

VI. RESULTS AND DISCUSSION

A. 2 D rotor blade analysis

The 2D disk stress distribution and comparison of theoretical and actual disk and hoop stress is illustrated below

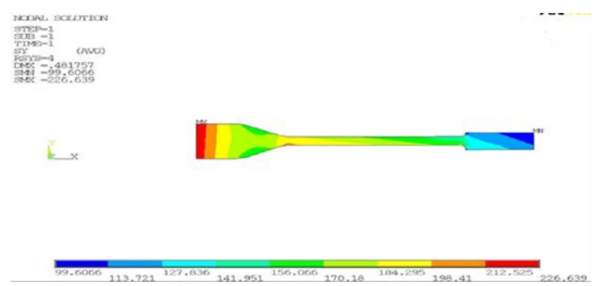

Figure 5 Stress Contour for 2D rotor

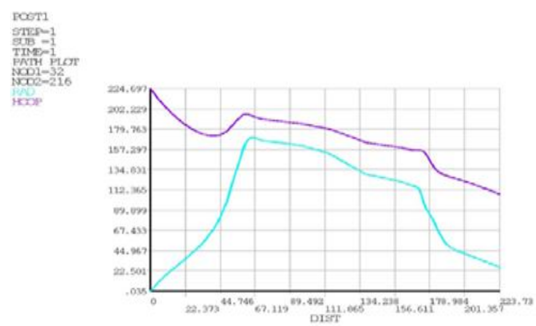

Figure 6 Rotor Disc stress Distribution

S. no	Stress	Theoretical	Numerical
1	Hoop	231	226
2	Radial	170	205

Figure 7 Theoretical and Numerical values of hoop and radial stress

B. D Disc stress analysis

To model stress concentration features like bolt holes and non-axisymmetric features like turbine blade profile needs to be modelled in much complex detail. The following figure is the 3D D-Disc modelled in computational plane.

Figure 8 D-Disk Mesh

The following are von-mises stress contours of the D-Disk for 6 NIMONIC-**90,** INCONNEL-718, HR STEEL-Type 17-22A(S)respectively

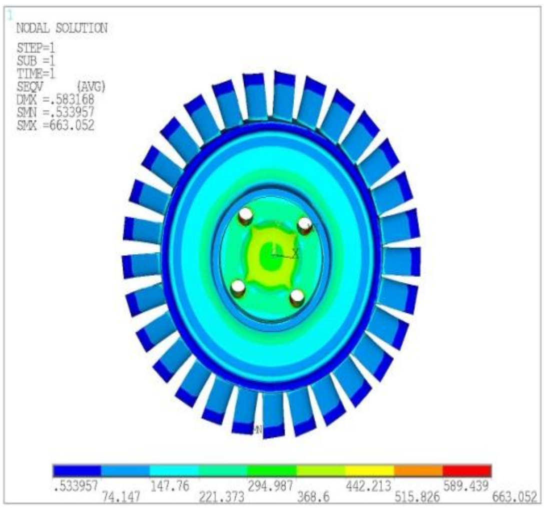

Figure 9 Von-mises stress contour for Blade material

C. FOS at Critical Point bolt level

The factor of safety values at maximum stress location of the D-Disk rotor is enlisted in the following table.

S. No	Material	Factor Of Safety At Maximum Stress Location(Bolt Hole)
1	NIMONIC-90	0.93
2	INCONEL-718	1.49
3	HR STEEL 17-22A(S)	1.72

Table 2 FOS Values at Maximum stress Points

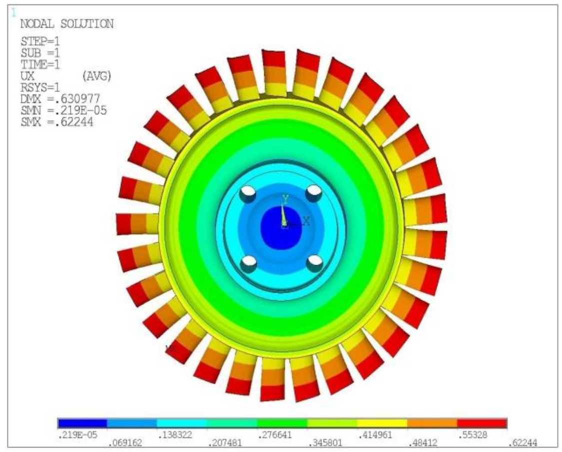

Figure 10 Radial Growth for Blade material

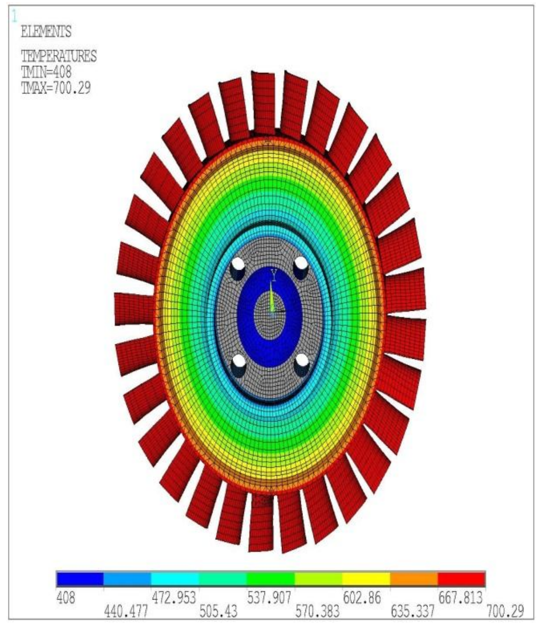

Figure 11 Temperature contour for blade material

VII. CONCLUSION

The finite difference method used for theoretical calculation gives fairly accurate results for variable cross sectional area disc which are axisymmetric in nature. This theoretical result is verified by repeating the same analysis in ANSYS FEM Software and comparing both results. To model stress concentration features like bolt holes and non-axisymmetric features like turbine blade we need it do 3-D analysis. This analysis was carried out in ANSYS FEM Software. The Factor of Safety value in the critical points such as Disc centre, bolt holes and in the blade are calculated using ANSYS results. It can be concluded from the above table that HR Steel Type17-22A(S) has highest Factor of Safety in all points. So, it can be chosen as the most suitable material for the turbine rotor.

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