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Determination of Stress Concentration Factors of a Steam Turbine Rotor by FEA

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***Abstract*—**Stress Concentration Factors are significant in machine design as it gives rise to localized stress when any change in the design of surface or abrupt change in the cross section occurs. Almost all machine components and structural members contain some form of geometrical or microstructural discontinuities. These discontinuities are very dangerous and lead to failure. So, it is very much essential to analyze the stress concentration factors for critical applications like Turbine Rotors. In this paper Finite Element Analysis (FEA) with extremely fine mesh in the vicinity of the blades of Steam Turbine Rotor is applied to determine stress concentration factors. A model of Steam Turbine Rotor is shown in Fig. 1.

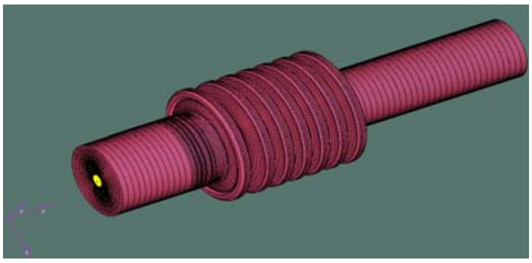

Fig. 1 Steam Turbine Rotor

***Keywords*—**Stress Concentration Factors, Finite Element Analysis, and ANSYS.

I. Introduction

GEOMETRIC discontinuities cause a large variation of stress locally, and often produce a significant increase in stress. The high stress due to the geometric discontinuity is called as 'Stress Concentration [1]. This can also appear when loads are applied over a small area or at a point. Geometric discontinuities are often called as 'Stress Risers'. Examples of stress risers include holes, notches, fillets and threads in a structural member. Often, Stress Risers are at the starting point of material damage. This ultimately leads to material failure by fracture. For this reason, it is important to realize the existence of stress concentrations and understand the overall behavior of some typical geometrical configurations at least for some critical applications. The ratio of the average or nominal stress to maximum stress is called Stress Concentration Factor and is denoted by K.

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Stress concentration factors are determined by the methods like 1. Theory of Elasticity 2. Photo Elasticity 3. The Grid Method 4. Brittle Coating Method 5. Strain Gauge Method 6. Flow Analogy Method and 7. Finite Element method [2].

The most comprehensive source of stress concentration factors for commonly encountered geometries has been compiled by Peterson (1953,1974). However in these references, the stress concentration factors for only filleted shafts are available and are only approximations based on photo elastic results for two-dimensional strips. The relation between two and three dimensional stress concentration factors is made by assuming an analogy exists between a circumferential fillet and a circumferential groove. This is the limitation of the Peterson Graphs for estimation of the stress concentration factors [3].

The numerical techniques are most effective due to advancement of high and large memory computers. These techniques can be applied for any minor change in the problem, which reduces the cost and time required for manufacturing and testing of several prototypes.

II. Modelling of Problems

The chosen problem is 3-D stress analysis and stress concentration factors are determined for the Steam Turbine Rotor with Blades. The material used for this application is Stainless Steel and its properties are given in Results Section.

A. Axi-Symmetry or Rotational Symmetry

If a shape can be defined by rotating a cross-section about a line, then it is said to be axi-symmetric. If the loads and boundary conditions are also axi-symmetric in nature, then axi-symmetric analysis may be carried out [4].

The problem is considered as axi-symmetric problem; hence only the resolving area is analyzed to reduce the considerable time of computations and tedious computer efforts [5]. FE Modeling of half of the steam turbine Rotor is shown in the Fig. 2 and boundary conditions in the Fig. 3.

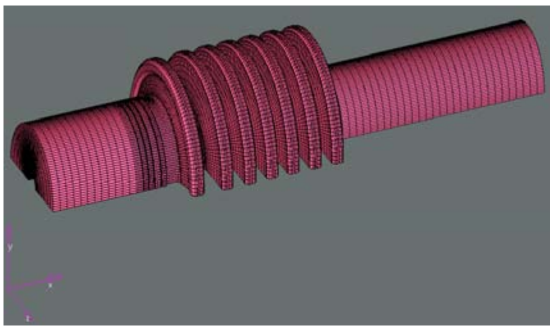


Fig. 2 FE Modeling of Half of the Steam Turbine Rotor

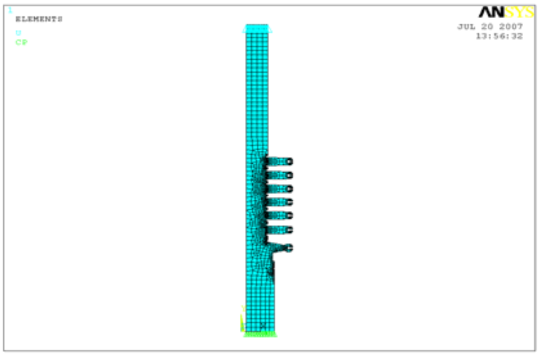


Fig. 3 Boundary Conditions

B. Element Used

Axi-symmetric elements are 2D planar in nature, and are used to modal a revolved 3D part in 2D space. Each element deforms as if it were a solid ring rotated about the axis of revolution. Axi-symmetric elements are available in most finite element packages and in a range of element shapes and types. No special boundary conditions have to be applied to these elements to achieve the symmetry condition. PLANE82 is a higher order version of the 2-D, four-node element is shown in the Fig. 4. It provides more accurate results for mixed (quadrilateral-triangular) automatic meshes and can tolerate irregular shapes without much loss of accuracy. The 8-node elements have compatible displacement shapes and are well suited to model curved boundaries. The 8-node element is defined by eight nodes having two degrees of freedom at each node: translations in the nodal x and y directions. The element may be used as a plane element or as an axisymmetric element. The element has plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities.

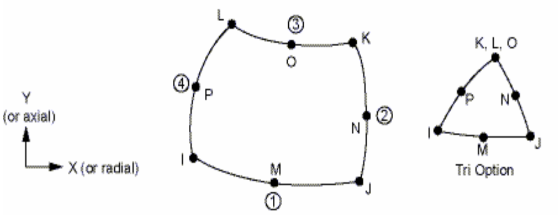


Fig. 4 Plane 82

C. The Software

The problem is analyzed by the software ANSYS 10.0. The flexibility, capability and options made the ANSYS program as user oriented and can be applied to wide variety of practical problems. The package contains many routines and all are interrelated to achieve a solution to the practical problems by finite element method.

III. Results

A. Material Properties

Material= Stainless Steel**,** Young's Modulus=190000 MPa, Poisson's Ratio=0.305,

Density= 8.17e-9 tons/mm^3 or 8.17e-6 kg /mm^3 or 8170 kg/m^3

B. Assumptions

Analysis is done on half of the model due to symmetry. The case studies analyzed for the estimation of stress concentration factors are 1. Neglecting the Bearings 2. Neglecting the Blades and Bearings 3. Providing Bearings to the Rotor

Case 1: Neglecting the Bearings

C. Pressure Calculation

Total no of blades provided on the rotor (n) = 34

Note: for axi-symmetric analysis total blades should be considered

Groove surface area = 66284.548 mm^2

Pressure exerted at rotor groove= Centrifugal force due to blades rotation / groove surface area.

Radius of groove = 494.957 mm, Centrifugal force = n(mrω^2)

= 34(3.05229 x 0.494957 x 314.1593^2) = 5069580.4898 N

Pressure = 5069580.4898/66284.548 = 76.4821 MPa = 76.4821e6 N/m^2

Fig. 8 shows the pressure applied at the Rotor Groove.

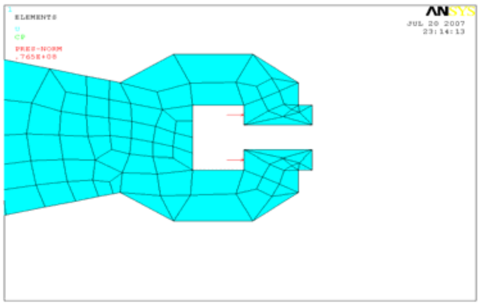


Fig. 8 Pressure applied at the Rotor Groove

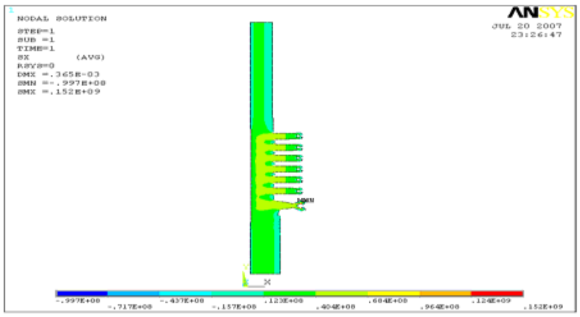


Fig. 9 Radial Stress

Fig. 9 shows the Max Radial Stress as 0.152e^9 N/m^2 = 152 MPa

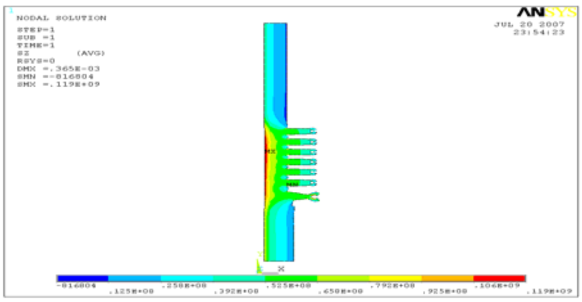


Fig. 10 Max Hoop Stress

Fig. 10 shows the Max Hoop Stress= 0.119e9 N/m^2 = 119 MPa

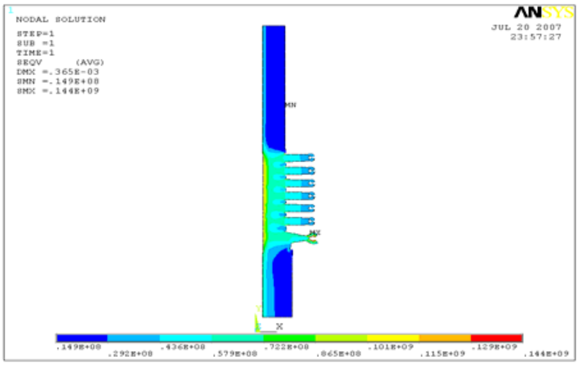


Fig. 11 Von-Mises Stress

Fig. 11 shows the Von-Mises Stress= 0.144e9 N/m^2 = 144 MPa

Case 2: Neglecting the Blades and Bearings

D. Load Calculations

Weight of the each blade (W) =30 N, Acceleration due to gravity (g) =9.81 m/sq.sec.

Mass of the blade (m) =W/g = 30/9.81=3.05229 Kg, Machine Rated Speed (N) =3000 rpm

Angular velocity (ω) = 2πN/60= (2x3.14159x3000)/60 = 314.1593 Rad/sec

No of blades provided on half of the model (n) = 17

Pressure exerted at rotor groove= Centrifugal force due to blades rotation / groove surface area.

Radius of groove = 494.957 mm, Centrifugal force = n (m$r\omega^2$)

= 17(3.05229 x 0.494957 x 314.1593^2) = 2534790.245 N

Groove surface area = 2x16571.137 = 33142.274 mm^2

Pressure = 2534790.245/33142.274 = 76.4821 MPa

= 76.4821e6 N/m^2

1. Max Radial Stress

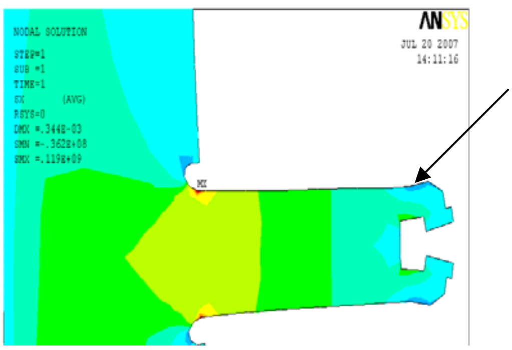


Fig. 5 Max Radial Stress

Fig. 5 shows the Max Radial Stress as 0.119e9 N/ m^2 = 119 MPa.

2. Max Hoop Stress

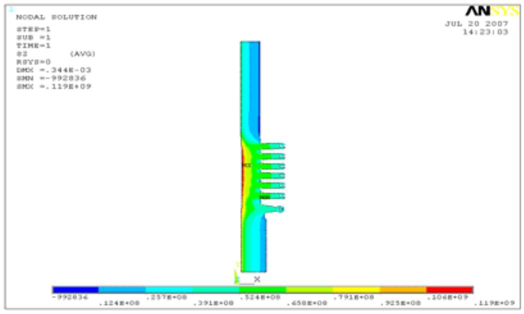


Fig. 6 Max Hoop Stress

Fig. 6 shows the Max Hoop Stress as 0.119e9 N/ m^2= 119 MPa.

3. *Von-Mises Stress*

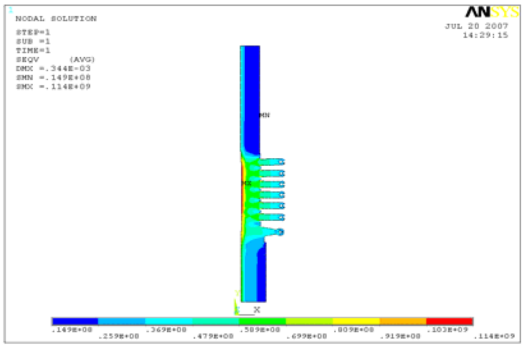


Fig. 7 Von-Mises Stress

Fig. 7 shows the Von-Mises Stress as 0.114e9 N/m^2 = 114 MPa.

Case 3: Providing Bearings to the Rotor

Description: FE modeling of the bearings is simplified by using rigid elements (CERIG). One end of the rotor is allowed to move in axial direction and FE Model is shown in Fig. 12.

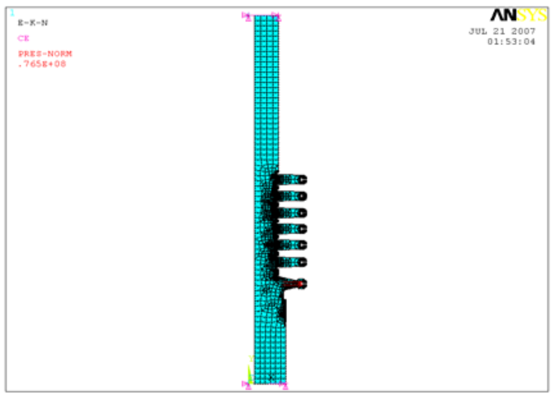


Fig. 12 FE Model

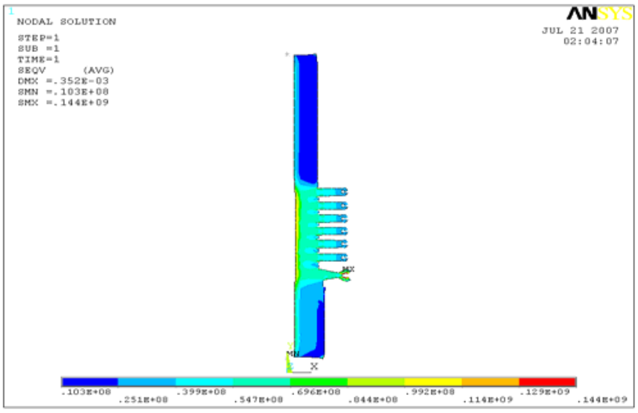


Fig. 13 Von-Mises Stress

Fig. 13 shows the Von-Mises Stress as 0.144e9 N/m^2 = 144 MPa. Fig. 14 shows the Symmetry Expansion Plot:

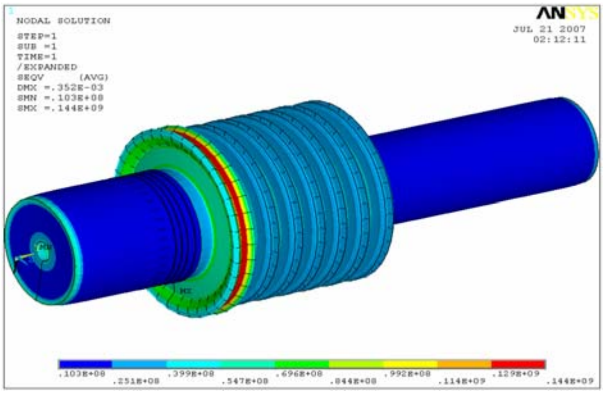
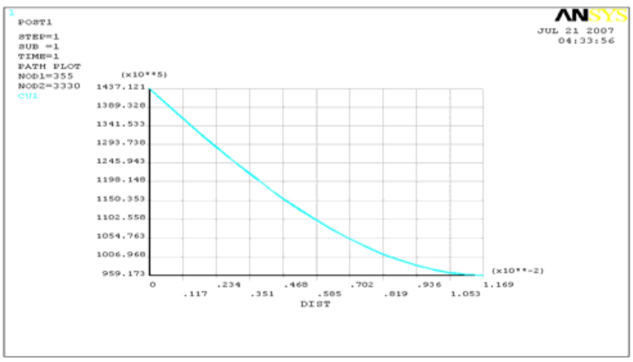


Fig. 14 Symmetry Expansion Plot

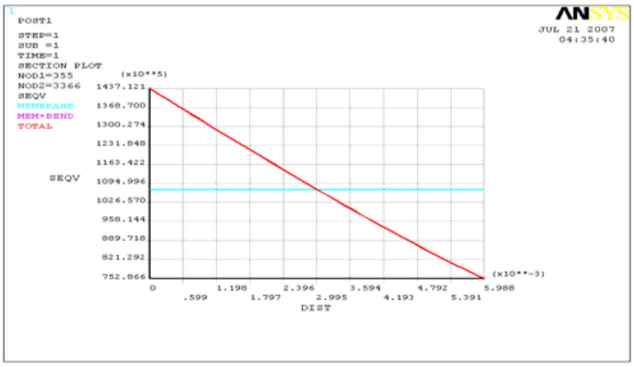
1. Theoretical Stress Concentration Factor* (Kt) *at Ambient Temperature (Nominal Stress)

Kt: A theoretical factor Kt expressing the ratio of the greatest stress in the region of Stress

Concentration to the corresponding nominal stress.



Graph. 1 Symmetry Expansion Plot



Graph. 2 Symmetry Expansion Plot

Graphs. 1&2 show the variation of theoretical stress concentration factor from the location of maximum stress.

Theoretical Stress Concentration Factor (Kt)= σmax/σnominal.

σmax= 144 MPa, σnominal= 107.2187 MPa

Kt = 144/107.2187

= 1.34

IV. Conclusion and Scope for Futurework

Peterson's theoretical stress concentration factor charts for filleted shafts have been a useful design tool for the past 40 years. In this work, the FEA is carried out for determination of stress concentration factors of Steam Turbine Rotor Shaft.

For the Case.1, neglecting Blades and Bearings, the Maximum Radial and Hoop Stresses are found as 119 MPa. In this case VonMises stress is found to be less than radial and hoop stress as 114 MPa.

For the Case.2, neglecting the Bearings, the Maximum Radial is increased up to 152 MPa and Hoop Stress is 119 MPa. In this case VonMises Stress is found to be 144MPa. for the case.3. Providing the Bearings to the Rotor, the VonMises stress is found to be 144MPa.

The Stress Concentration Factors for the case.3 are calculated and found as 1.34, which is within the safety limits. The above work can be extended for dynamic analysis and also be applied for thermal analysis.

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