Failure Analysis of Steam Turbine Rotor Due to Low Flow Conditions

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Abstract – *Normally a steam turbine used for generating electric power is treated as complex system with numerous rotary and stationary parts working precisely under controlled conditions. Due to precise design conditions the steam turbine units and the rotary speeds at which they operate, the malfunction of one component can lead to severe damage of entire unit. The rotary and stationary parts of steam turbine sections are prone to failure due to corrosive environment and high temperature along with high pressure gradient in the turbine. During the shed down period if the separation margins and the frequency evaluation is not taken care during preliminary and detailed design conditions the turbine experiences severe failure accompanied by a loud noise near its natural frequency. This results in fractured stationary and rotary blades. The degree of damage may not be economical under repair. The unexpected excessive forces or impact load have led to brittle fracture of the discs. In the light of the above one such effort is made to design the blades with shrouds optimizing the geometry interfering with stationary blades well-tuned to avoid resonances during low flow conditions or shutdown conditions. The turbine experiences resonance with low amplitude of vibratory stress causing the fatigue damage. In order to prevent the damage control classical theories blended with simulative results work as an effective design tool indicating the root cause of the failure in blades. The vibratory tools such as Camble diagram, SAFE diagram, interference diagram and Goodman diagram are effectively utilized in shroud design with an optimum geometry and a locking mechanism to prevent catastrophe.*

Key Words: Steam turbine, Static and modal analysis, Equivalent stresses.

1. INTRODUCTION

The vibration of rotors and rotor frameworks has been a worry of specialists and researchers for an over a century. In 1869, Rankine distributed an article, "On the Centrifugal Force of Rotating Shafts", which is the soonest reference to vibrations of a pivoting framework. From this timeframe, dynamical examination of shaft was started. Present day outlines of rotor-bearing frameworks for the most part go for expanded power yield and change in productivity. The requesting prerequisites set on current pivoting machines, for example, turbines, electric engines, electrical generators, compressors, have presented a requirement for higher speeds and lower vibration levels. Since the innovation of the wheel, rotors have been the most ordinarily utilized parts of machines and systems. Rotational movement is utilized to accomplish interpretation, as from the wheel to the pivot; to store vitality, as in the antiquated sling or present day flywheels; to exchange power starting with one point then onto the next by utilizing belts, cogwheels, or gear trains; to get dynamic vitality from different sorts of vitality, for example, thermal, substance, atomic, or wind vitality. Rotors utilized as a part of machines and instruments give various preferences as respects effectiveness, wear, and simple alterations.

While satisfying essential parts in hardware, the rotors are, in the meantime, the primary wellspring of irritation of typical operation of the machines. Rotational movement around a fitting pivot, at appraised, outline forced, rotational speed, speaks to the critically required dynamical state for rotors. In all pragmatic cases in turning apparatus, the collected rotational vitality can't, be that as it may, be completely utilized for the plan reason. This vitality has a potential for genuine breaks and can without much of a stretch be changed into different types of vitality. Normally, as in all other mechanical components, some vitality misfortune because of dissipative instruments dependably happens, irreversibly changing the rotor rotational vitality into thermal vitality, which in the long run gets irreversibly scattered. With the exception of this sort of reaction, in rotors there exist extra wellsprings of vitality holes, changing the rotor rotational vitality into different types of mechanical vitality.

1.1 Rotor Supported on Rigid Supports

Figure 1 demonstrates an established Jeffcott rotor – a rotor with an aggregated mass at the inside and bolstered by orientation at each end.

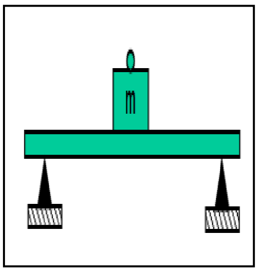

Figure 1: Jeffcott Rotor with inflexible backings

Give us a chance to accept that the mass is accumulated at the mid traverse. The heading are thought to be inflexible backings. Subsequently the rotor can be thought to be basically bolstered beam, utilizing the hypothesis of beams, the stiffness of the just upheld beam can be composed as, $K = \frac{48EI}{L^3} = \frac{48E*\pi d^4}{64L^3}$ 2.1

Utilizing the condition for normal recurrence, we get,

$$\omega n = \sqrt{\frac{k}{m}} = \sqrt{\frac{48E\pi d^4}{64mL^3}} \quad 2.2$$

If we assume distributed mass of the shaft of diameter "d" and length "L", the above equation can be reduced to,

$$\omega n = \sqrt{\frac{48E\pi d^4 * 4}{64L^3 * \rho\pi d^2 L}} = \sqrt{\frac{3Ed^2}{\rho L^4}} = f\left(\frac{d}{L^2}\right) \quad 2.3$$

As it were, the common recurrence is straightforwardly corresponding to the distance across and contrarily relative to the second energy of the length. This is a vital relationship, which we will be utilizing regularly to physically comprehend the impacts of the plan changes to the machine.

The "length" refers to the bearing span (axial distance between the bearing centrelines). Just as an example, if you increase the diameter of the shaft, without changing the length, the natural frequency will increase. On the other hand, if you increase the bearing span, then the natural frequency drops at a greater rate (since "L" is raised to the second power). If the bearing span is reduced, then the natural frequency increases.

1.2 Discussion of Journal bearings

The liquid (oil) film bearings are the most well-known methods for supporting turbo apparatus. Obviously, moving component bearings are likewise utilized as a part of many gas turbine applications. The term diary applies to the bit of the shaft that is upheld by a bearing. A comprehension of the pretended by the bearings on the dynamic conduct of the rotor is a basic prerequisite for architects who manage the plan and operation of bearings for turbo apparatus. Figure 2.5 demonstrates a straightforward kind of diary bearing (otherwise called sleeve bearing). The shaft pivots in the leeway space (overstated) of the bearing as appeared. The freedom is more often than not around 1.5 mils/inch of the diary breadth.

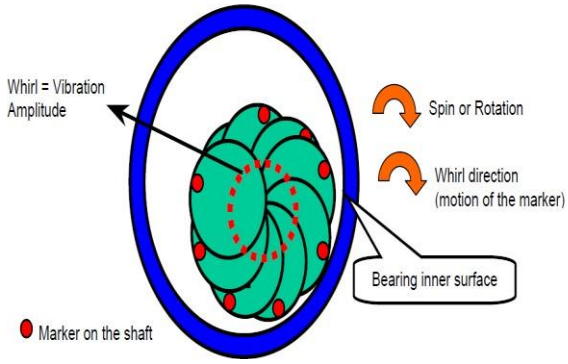

Figure 1.2 Example of forward whirl in the bearing clearance space

1.3 Motion of the shaft in the bearing

At the point when the shaft pivots in the bearing freedom space, it has two sorts of movements.
The main movement is the pivot or the turn of the shaft. This is same as the running rate of the shaft.

The second sort of movement is the precession - which is the revolution of the Center of the shaft as for the geometric Center of the bearing. (A relationship would be the movement of the earth – it turns about its pivot and furthermore spins round the sun) This precession, in rotor dynamic phrasing, is all the more generally known as whirl. This whirl movement is additionally named forward whirl and in reverse whirl. Forward whirl is the movement in which the focal point of the shaft moves in an indistinguishable course from the turn of the shaft. The regressive whirl is the movement in which the focal point of the shaft moves the other way as the revolution of the shaft. All in all, the whirl orbit is elliptical.

1.4 Bearing stiffness and damping coefficients

At the point when the shaft is not turning, the shaft is resting at the base of the freedom space in the bearing. With the leeway space loaded with oil, as the shaft begins to turn, the shaft goes about as a "pump", "pushing" the oil underneath itself! This creates the lift of the shaft. At any consistent turning speed, the Center of the shaft is found far from the geometric focal point of the bearing as appeared in Figure 2.6. This is known as the capriciousness of the diary. The oil "wedge" underpins the shaft.

The properties related with the oil film are firmness and damping. These are natural properties of the oil and are a component of oil sort, thickness, temperature, and so on. For analysis purposes the solidness and damping are arranged towards the even and vertical tomahawks – henceforth, the bearing is said to have a flat firmness and vertical firmness (same for the damping). The flat firmness is demonstrated by Kxx and the vertical solidness by Kyy. So also the damping is shown as Cxx and Cyy. To muddle (genuine living!) things, on the grounds that the oil film is consistent around the shaft, there exist parts of the solidness and damping in the x-y course too! The pretended by this cross-coupled firmness (Kxy) is critical in understanding the strength of rotor-bearing frameworks.

The meaning of Kxx is "Drive of the rotor in x course because of a dislodging in the x bearing". Also, Kxy is "Constrain of the rotor in x heading because of an uprooting in the y course". Presently, Kxy and Kyx are complimentary. Due to the "connect" in the x and y strengths and removals, these are known as the Cross-coupling coefficients.

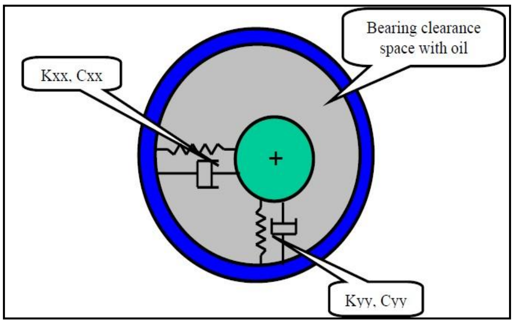

Figure 1.3 A simple journal bearing geometry

2. LITERATURE REVIEW

A large portion of the turbo machines working today keep running on or exceptionally close to the second basic speed. Some keep running with no evident vibration issues. Others are marked issue machines, requiring consistent regard for keep the vibration beneath the trek level. A few machines are intentionally intended to work close to the second basic speed on account of the requirement for higher execution necessities and hence higher paces. Others are intended to keep running beneath the second basic speed yet wind up running specifically on the second. It was by and large expected that the second basic was no less than three times the first and, in this manner, of little concern. Real advances have been made over the most recent twenty years in investigative rotor and bearing dynamics that have prompt enhanced basic speed expectations. In the fifties, before the general accessibility of liquid film bearing dynamic examination codes, the rotor basic were anticipated in view of unbending bearing investigations. With the improvement of the fast PC, dynamic bearing projects wound up plainly accessible in the late seventies.

Rotor Dynamics is unique in relation to Structural Dynamics, as we manage a pivoting structure. Fundamentally, all the vibration wonders will be substantial, in any case, there are a few contrasts and we need to set up methodology on dealing with the rotors and their vibratory marvels. Rankine [2] is credited to have said the presence of a basic speed of a rotor in 1869. He characterized this as a farthest point of speed for centrifugal spinning. There were many questions whether a rotor can cross such a basic speed? It was assumed that it will be temperamental in the wake of intersection the basic speed. This is fairly like Speed of sound and whether one can cross this hindrance in flying. We need to sit tight for almost 50 years to have an unmistakable comprehension on this point.

The primary fruitful rotor model was proposed by Foppl in 1895. It comprised of a solitary plate halfway situated on a roundabout shaft, without damping. It exhibited that supercritical operation was steady. Lamentably, Foppl distributed his work in a German structural building diary, which was little perused, if by any means, by the rotor dynamics group of his day.

Jeffcott in 1919[3] regarded the issue as constrained vibration and recognized the essential standards of rotor dynamics.A large portion of the turbo machines working today keep running on or exceptionally close to the second basic speed. Some keep running with no evident vibration issues. Others are marked issue machines, requiring consistent regard for keep the vibration beneath the trek level. A few machines are intentionally intended to work close to the second basic speed on account of the requirement for higher execution necessities and hence higher paces. Others are intended to keep running beneath the second basic speed yet wind up running specifically on the second. It was by and large expected that the second basic was no less than three times the first and, in this manner, of little concern. Real advances have been made over the most recent twenty years in investigative rotor and bearing dynamics that have prompt enhanced basic speed expectations. In the fifties, before the general accessibility of liquid film bearing dynamic examination codes, the rotor basic were anticipated in view of unbending bearing investigations. With the improvement of the fast PC, dynamic bearing projects wound up plainly accessible in the late seventies.
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Dr. A.V.Ruddy [4] in their paper title "An Introduction to the influence of the bearings on the dynamics of rotating machinery",in this approach he concerned with the lateral vibration of rotor influence that other machine components have on that vibration. When investigating the dynamic behaviour of turbo machine. When investigating the dynamic behaviour of a turbo machine rotor and bearing is most important. The plant engineer is concerned with the level of vibration in the running speed range and, in order to ensure an acceptable dynamic behaviour of turbo machine. The critical of rotor speed map shows the interaction of rotor and support stiffness. It is important note that, generally increasing the support stiffness increasing the natural frequencies. So the bearing play important role in rotor dynamics.

3. OBJECTIVES AND METHODOLOGY

3.1 Objectives

There are several objectives to be fulfilled within a standard rotor dynamic and structural analysis of steam turbine rotors. The following objectives are considered for the present work

- To conduct the static stress analysis of single and multi stage rotor to check the static strength and structural integrity
- Fatigue life estimation of multistage rotor

Although the aim of the present work is to develop single and multi rotor model and evaluate the rotor dynamic capabilities of Ansys software. Thus, verification of simple model and rotor dynamic analysis method had become one of the dispositions of this work.

3.2 Methodology

Develop an easy-to-use and easy-to-understand analysis procedure for free and forced vibration problems in rotor dynamics.

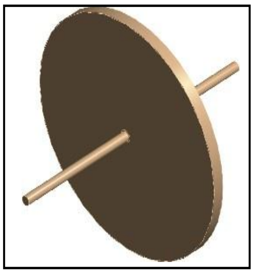
To determine the free and forced frequency of a single rotor systems and to understand the concepts of mode shapes

Develop harmonic frequency response function estimation

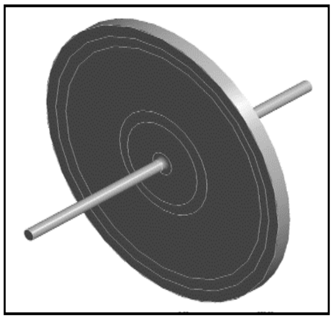
4. STATIC ANALYSIS OF SINGLE ROTOR SYSTEM

4.1 Problem Definition

Present work involved to configuration of single stage rotor, the base line modal consists of solid type rotor whereas second configuration was with varying cross section disc. The difference in geometry is as shown in sketch



Solid section



I solid section

A single rotor system is considered which is to be analyzed statically and dynamically to visualize the behavior of stresses and directional deformation of the rotor system. Two types of rotor systems are taken into consideration i.e. (i) solid rotor (ii) I section rotor and comparative analysis is carried out between the two cases of rotor systems.

Preliminary design considerations

The properties of the model are summarized as follows

Shaft properties
Length of the shaft, L= 1200mm
Diameter of the shaft, D= 40mm
Young's Modulus, E= 210E3 N/mm^2
Poisson ratio = 0.3
Density = 7800 kg/m^3
Speed N = 5000 rpm

Disk properties
Case 1: Solid rotor system
Mass, m = 307 Kg (Solid Rotor)

Diameter of the solid rotor = 1000 mm
Thickness of the rotor = 50 mm

Case 2: I Section rotor

m = 282 Kg
Thickness of the rotor = 50 mm
Total distance from axis of shaft to rotor end is 500 mm
Distance between axes to center of gravity is 510 mm

For the specified material property at rated rpm of 4000 rpm is considered to be operating speed and over speed 120 percentage is considered for present analysis however the present work involved linear static analysis at100 percent speed with 3 principal stresses resulted at 100 percent speed and von misses stresses, deformation is tabulated in table and corresponding stresses as shown below,

4.2 Static Analysis of single solid rotor system

Figure 4.1 shows the 3D model of solid rotor system.

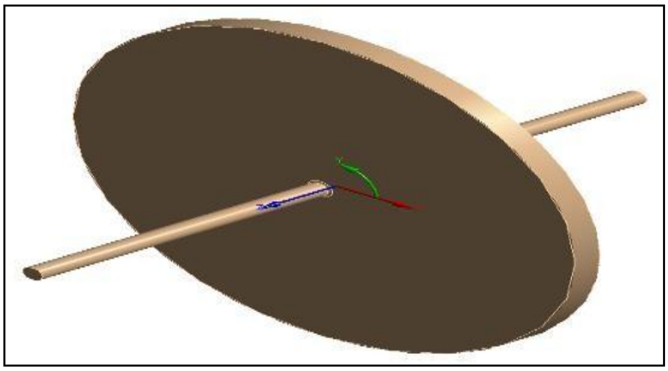
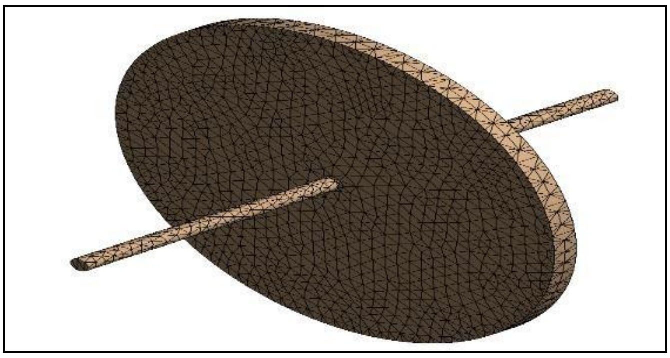


Figure 4.1: 3D model of solid rotor system.

Finite element model

The mesh is made up of 3D tetra elements and is shown



Boundary Conditions

The boundary conditions for the single rotor system is as follows and shown in figure 5.5 All degrees of freedom is arrested at one end of the shaft and on the other end, shaft is free to move in one direction i.e. axial direction and pressure is applied to the end of the rotor.

Centrifugal Pressure due to blade pull:

Speed = 5000 rpm.

Angular Velocity = (2 X 3.147 X 5000 / 60) = 523.39 rad/sec

Circumferential area of Disk = $\pi D t = \pi * 1000 * 50 = 0.157$ m^2

Centrifugal force = (m (Angular Velocity)^2 X r) = (307*523.39^2*0.51)= 42.89MN

Centrifugal stress for solid rotor = pressure =(F/A) = (42.89E6/0.157) = 273.18 MPa

Centrifugal stress for I section rotor = 251 MPa

Fig 4.3 shows the pressure is applied to the end of the rotor

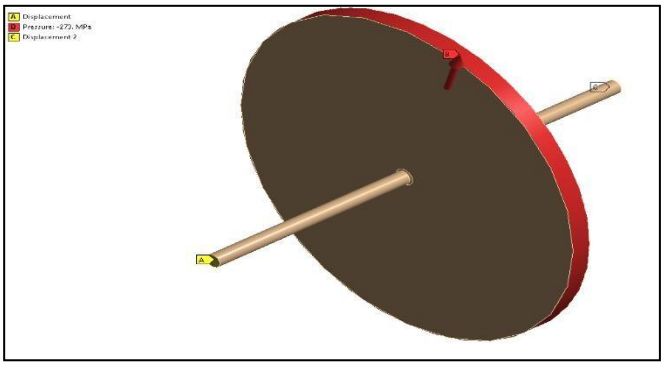


Figure 4.3: FE Model

Static stress results

Case 1: Solid rotor

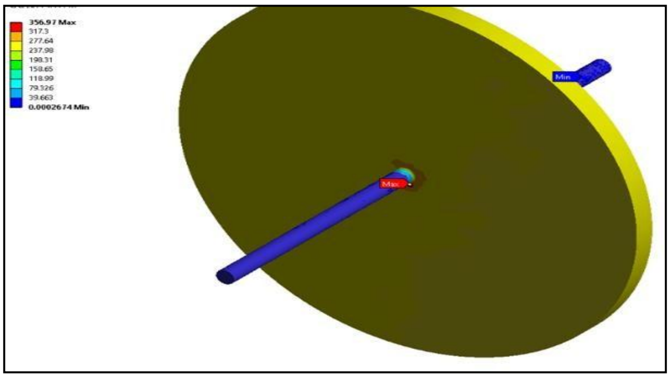


Figure 4.4: Von-mises stress

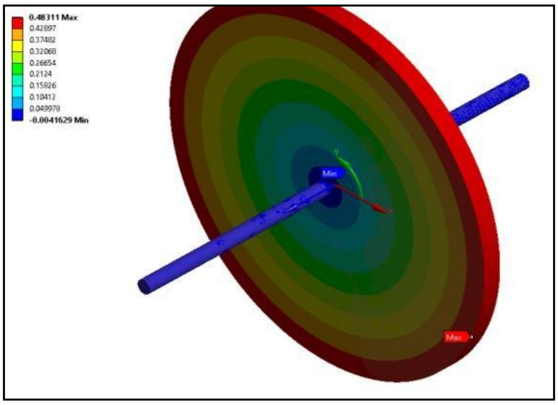
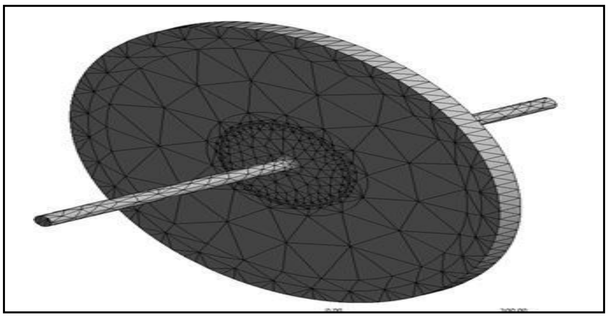


Figure 4.5: Directional deformation

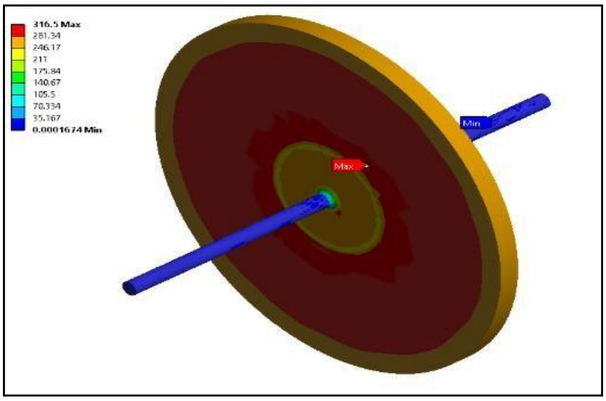
Table : For Solid rotor

Von misses stress	Valve (MPa)
σ 1	356.97
σ 2	317.3
σ 3	277.64
σ_{von}	356.11

Case 2: I Section rotor



I section rotor meshing



Von misses stress

Table: For I section rotor

Von misses stress	Valve (MPa)
σ 1	344.83
σ 2	305.83
σ 3	266.83
σ_{von}	316.5

Over speed margin;

Yield stress (σ_y) =585 Speed (n) =4000rpm

a)For 100% speed ;

σ_{avg} = 269 Mpa

$=\sqrt{\sigma_y/\sigma_{work}}$

$=\sqrt{585/269}$

=1.47>1.21

Hence over speed margin is achieved

b) For 121% speed;

Average stress = 269×(1.21×1.21)

=393.8 Mpa

Ultimate stress (σ_u)= 780 Mpa

Burst speed $=\sqrt{\sigma ult / \sigma work}$

$=\sqrt{(780/393.8)}$

=1.40>1.21

5. MODAL ANALYSIS of ROTOR SYSTEM

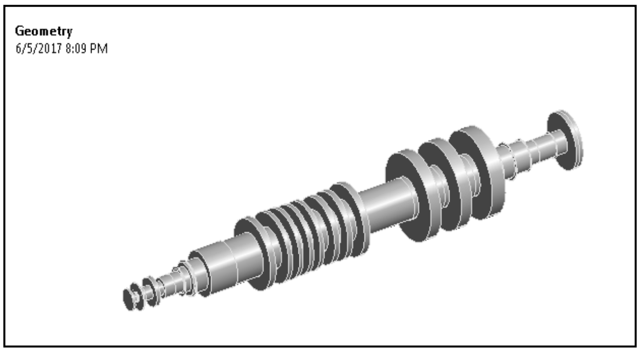
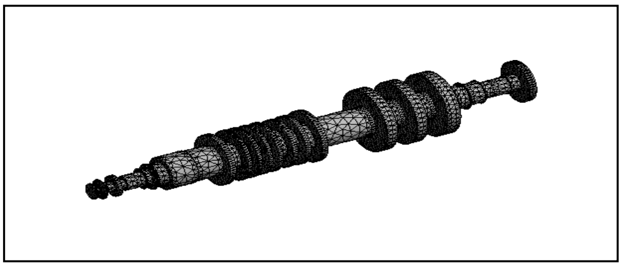


Fig 5.1, 3d Modal of multi stage rotor



Statistics	
Nodes	66095
Elements	37637
Mesh Metric	None

Fig 5.2, Meshing of multi stage rotor total no element 37637,

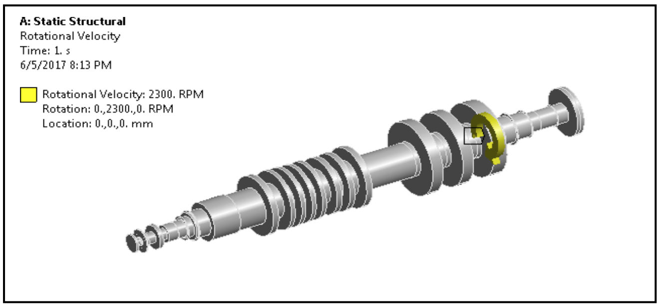
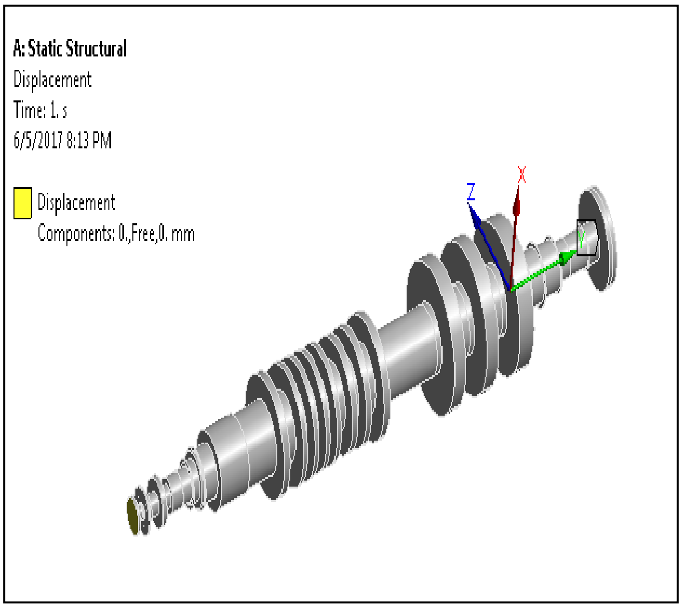


Fig 5.3, rotational velocity in multi stage rotor

Fig5.4 , displacement in multi stage rotor



Static analysis

Equivalent stress

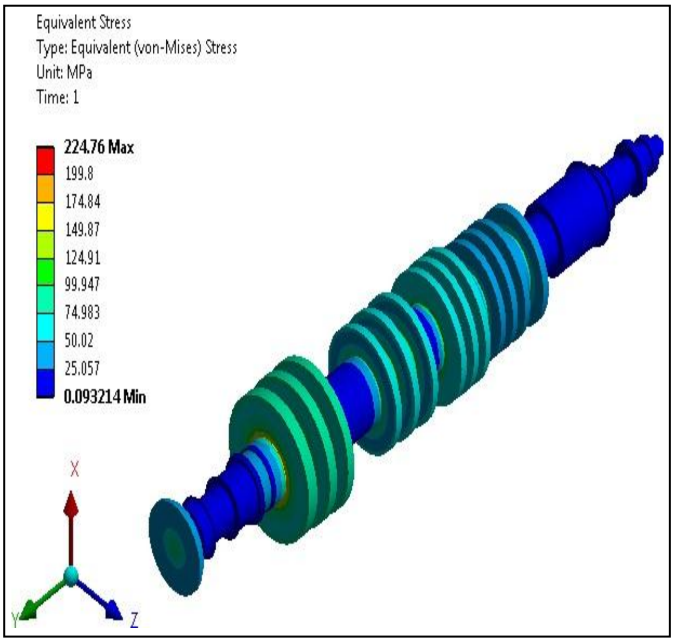
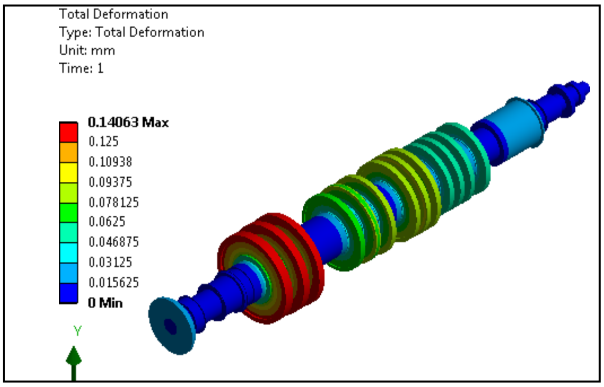


Fig 5.5, max equivalent stress is 224.76 in multi stage rotor

Total deformation



Fig,5.6, max total deformation is 0.14063 mm in multi stage

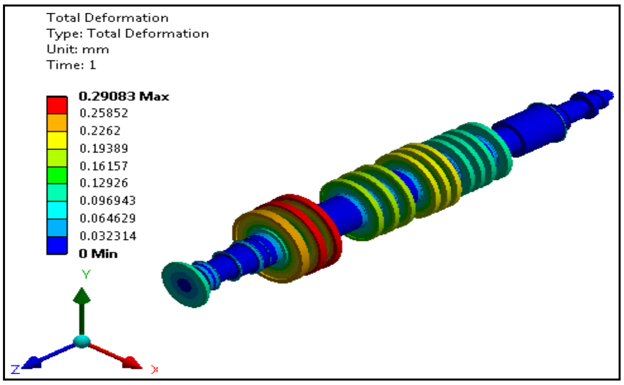
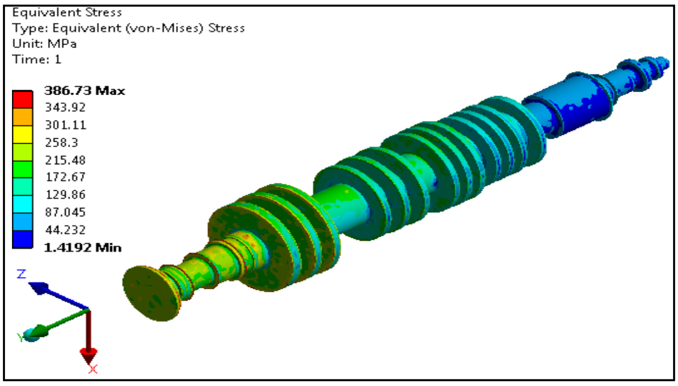


Fig 5.9, total deformation is 0.29.63 mm multi stage rotor

Thermal analysis



Fig,5.7 max equivalent stress in multi stage rotor

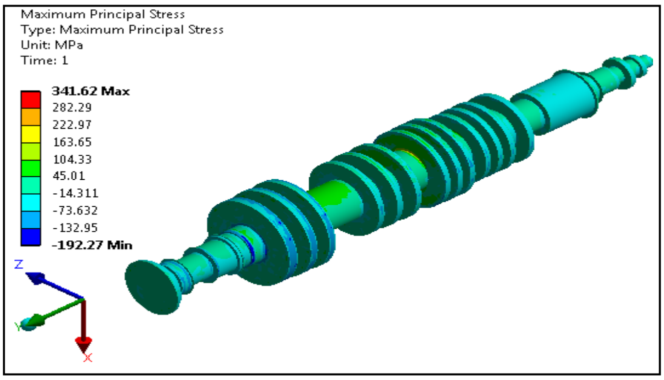
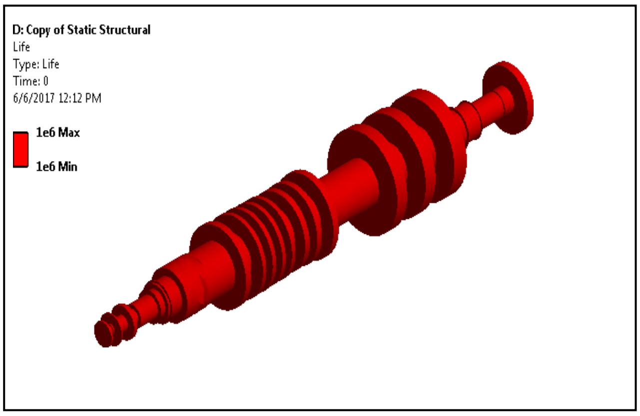


Fig 5.8. maximum principal stress is 341.82 Mpa in multi stage rotor

Life estimation of multi stage rotor



Fig, 5.9, Life estimation is 1000000 cycles multi stage rotor

Goodman diagram

Mean Stress can be calculated from,

$$\sigma_{mean} = \frac{\sigma_{von}}{2}$$

= 330.5 mpa

Where

σ_{von} = Equivalent von-Misses Stress

$$\sigma_a = \frac{\sigma_1 - \sigma_2}{2}$$

=

=140.965 mpa

Where

σ_1= Maximum Principal Stress

σ_2= Minimum Principal Stress

s/no	Rotor	Equivalent stress(Mpa)	Total deformation (mm)
01	**solid section**	356.97	**0.388**
02	**I section**	344.83	**0.368**
03	multi stage rotor	**310.76**	**0.0290**

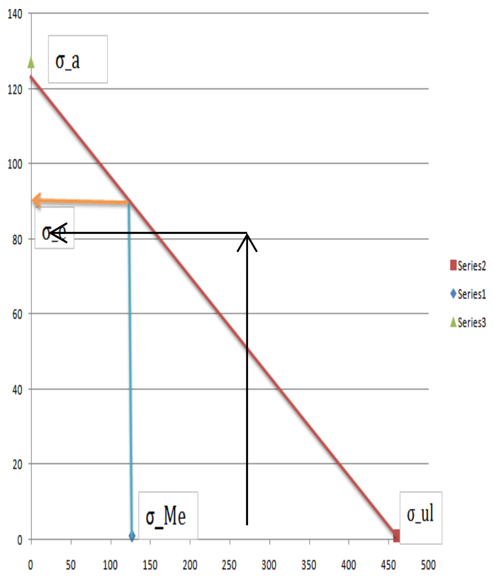


fig 4.4 Goodman Diagram

Number of cycles:

$$N_f = \left\{\frac{\left[\sigma_{ult} - \sigma_{ult}\left(\frac{1}{fos} - \frac{\sigma_a}{\sigma_e}\right)\right]}{\sigma_a}\right\}^{\frac{1}{0.08}}$$

Where,

N_f=Fatigue life

σ_{ult}=Ultimate stress

fos=Factor of Safety

σ_e=Endurance limit

b = Fatigue strength exponent

σ_a=Alternating stress

$$N_f = \left\{\frac{580 - 580\left(\frac{1}{1.4} - \frac{122}{79}\right)]}{122}\right\}^{\frac{1}{0.08}}$$

$$N_f = 1.01 \times 10^6$$

7. CONCLUSIONS

1. Linear static structural analysis has been carried out to estimate the maximum stress, strain and deformation at blade, disc and fillet regions. It is found that peak stress of 310.45 Mpa, total deformation of 0.02mm is obtained along in multi stage rotor, hence the design is safe.

2. Fatigue analysis of multistage rotor was carried and the fatigue life estimated obtained more than 100000 cycles

3. Weight of 120.1 kg is reduce in multi stage rotor which is Optimization of in multi stage rotor to increase the life and efficiency

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