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Fault detection of rotor system due to disc mass imbalance and misalignment in Flange coupling

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Abstract— Misalignment is an important cause of vibration problems in rotating machinery. Misalignment in the context of this thesis includes any deviation from the ideal case in which a straight shaft rotates in perfectly aligned bearings. Misalignment can apply to a single piece of rotating equipment with a bent shaft or a straight shaft forced to rotate in three or more bearings which are not coaxially aligned. It can apply to two or more pieces of coupled machinery whose shafts are not properly aligned. In this paper a survey is made of the various types of misalignments often encountered in rotating machinery and the vibrational characteristics which arise due to these misalignments are examined. In particular lateral, axial and angular coupling misalignments will be studied as a function of flexible coupling vibrational behavior. This discussion will include bent or bowed rotors, close-coupled systems, constant and non-constant velocity flexible couplings and axial motion couplings.A specific coupling problem could yield vibrations in one or all of the above directions. In the specific discussions which follow directionality will be specified. The organization which follows will take the shaft coupling problem through an evolving chronology from simple to complex geometric shaft arrangements.
Index Terms— Misalignment, rotating machinery, fault detection of rotor, flange coupling systems, vibrations, Disc mass imbalance,

1 INTRODUCTION

A single uncoupled rotor may exhibit vibration problems for a variety of reasons. Sources of these vibrations may include shaft bearing misalignments, bowed rotors and mass unbalance. Bearing-shaft misalignments may from the vibrational analysis point of view be the hardest to detect. A shaft-bearing misalignment occurs when the bearing Journal placements are not axially concentric and a straight shaft is forcefully fitted into the bearings. Figure depicts an extreme case of a shaft distorted by bearings not axially aligned. The forces exerted by each of the bearing foundations cause the shaft to bend. As this forcefully bent shaft rotates, the forces exerted by the foundations remain constant as the shaft rotates at its operating angular velocity.

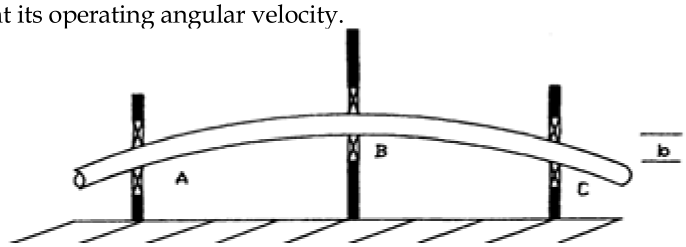

Fig.1 Shaft distorted by bearings connected with the help of suitable coupling.

It is mandatory to align the shafts within this permissible margin before the machinery is put in service. All the shafts have some form of deviation from their longitudinal axis because of self-weight of the shaft, thus the shaft are not straight, that is why the location where the two shaft can be compared is only at the point of power transfer i.e. from one shaft to the next

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2 VIBRATION ANALYSIS – FAULT DETECTION FAULT MODE ANALYSIS.

The machinery diagnostics technique viewed here is based on a technique known as "fault mode" analysis. This technique utilizes the fact that specific mechanical events, such as unbalance, misalignment, looseness, bearing defects, aerodynamic and hydraulic problems, and gearbox problems usually generate vibration frequencies in specific patterns. The frequency, amplitude and pattern of the peaks in a vibration spectrum can be a telling indication of the type of problem being experienced by the machine. The following chart summarizing specific machinery faults and their vibration patterns

2.1 Principles of "fault mode" analysis

Measurement of mechanical faults such as unbalance and misalignment generate mechanical vibration in a well-defined frequency pattern.

Comparing the vibration levels and vibration spectra on similar types of machines will help establish the severity and cause of a vibration problem.

2.2 Fault identification of unbalanced and misaligned rotor system

Fault Diagnosis is utilized as an advance tool in any industry to increase the productivity, reliability and endurance capability of any machine. The general procedure of model based fault diagnosis can be roughly divided into generation of residuals (i.e., functions that are emphasized by fault vector), and identification of faults. There are few differential forms of equation in which we can represent the presence of faults in any vibrating system

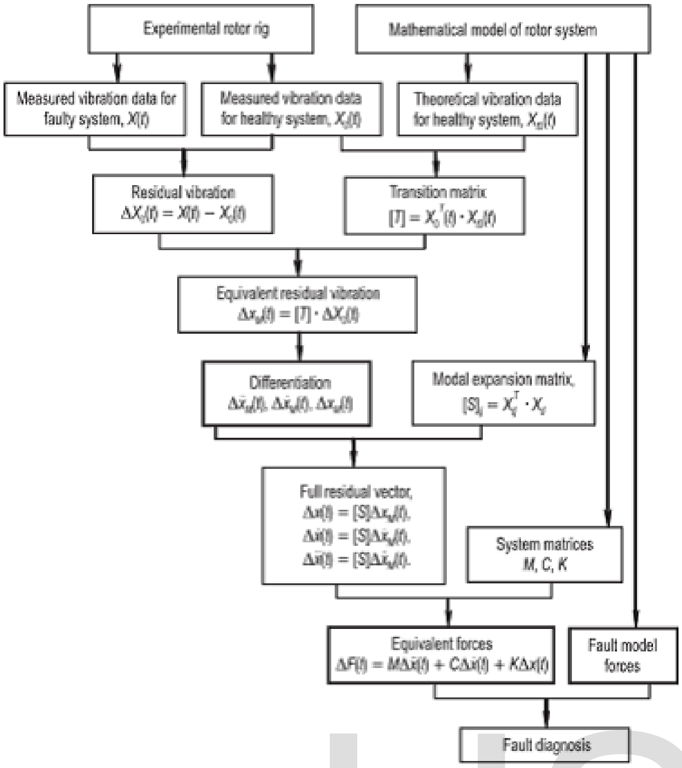

Fig. 2 Flowchart to develop model

2.3 Modeled System

The complete modeling of the entire system is done by assuming the system to be comprising of 17 nodes in total, out of which the first three nodes are governed by the coupling. The Disc is supported in between two roller bearing supports at node positions 5 and 17. Thus, the disc fault model is to be applied at node 11. The general approach that is to be followed in solving the equation is to first discretize the system which is done by considering it to be made up of 17 node system. For every node in the FEA model, the nodal degree of freedom that has been considered is 4. Thus, the elements of the FEA model is considered to comprise of 16 beam elements which have four nodal degrees of freedom, one in vertical plane and the other in horizontal plane of reference. The figure below explains the fea model in further detailing:

The above model has 4 nodal degree of freedom (two translation and two angular), thus making the system to have a total of 68 degrees of freedom. Since, we have segregated our study to horizontal and vertical planes, our force unbalance values will change accordingly and we will have different values at nodal points for the two planes under consideration. A Matlab Code is written in order to accommodate all the required force unbalance and following assumptions are made as per Reference [10] to initiate the code.

Inputs for Code:

1. Unbalance Mass = 6.14 gm
2. Eccentricity = 70 mm from center of shaft
3. Rotor Speed Frequency = 30 Hz
4. Parallel Misalignment Considered = 1mm for vertical and horizontal plane of reference

Literature[10] has the experimental work and analytical work which we validated using matlab and found that it is in good agreement with that of literature.

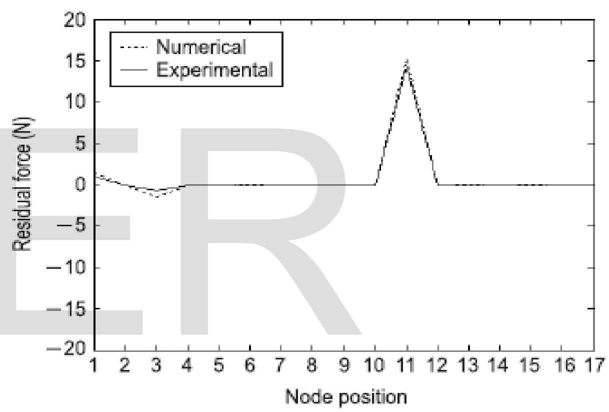

Fig. 3 Developed FEA model

3 Results For Our Code

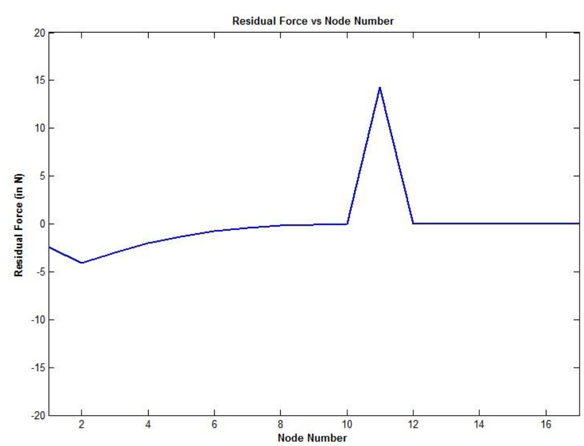

Fig. 4 Results for our code

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3.1 Variation of Residual Forces with varying Rotor Frequency (MATLAB)

The residual forces provide us basis for detecting any abnormality in coupling operation which can be involved in any kind of power transmission system. The above methodology can be easily interpreted and modified based on the kind of assembly. These residual forces must be as low as possible to attain a fault free operation from any of the components involved in the entire system such as the coupling, disc and bearings. From the plot obtained above, it can be easily concluded, that mass imbalance in any system comprises of higher values of residual forces, and however, coupling misalignment can also be seen to vary with varying rotor speed frequency.

Thus, in this section we will study on how the varying rotor frequency affects the residual forces that are being generated. For this observation to be made, we will consider rotor frequencies of 30, 50, 100, 150, 200, 250 Hz respectively and observe the trend of the results. At node no. 2, rise in residual force is observed because of misalignment in flange coupling whereas rise in residual force at node 11 is observed because of the disc or rotor mass imbalance. On increasing frequency, magnitude of residual force keeps on increasing at the nodes.

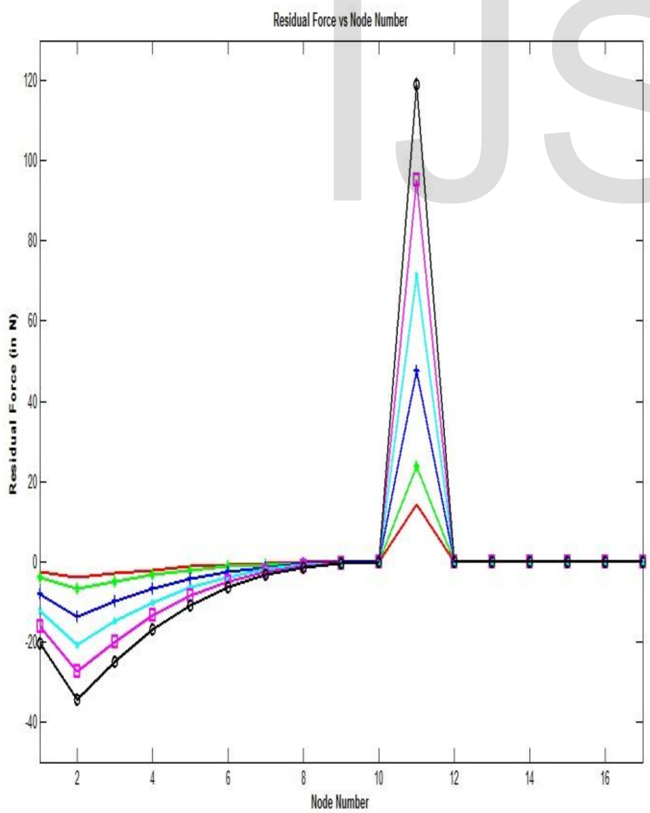


Fig. 5 variation of rasidual force vs node number

3.2. Conclusion

Thus, we can see that with increasing rotor frequency the residual forces keep on increasing at a linear rate. It can be clearly observed that the change in rotor frequency impacts the residual forces developed due to unbalanced mass much more than that developed due to the coupling misalignment. Thus, with these plots any fault seen in the respective vibration components can be easily rectified.

4. Static And Modal Analysis

Static and modal analysis of solid crank shaft is used to determine stresses and displacement while modal analysis is done to obtain vibration characteristics namely natural frequency and mode shapes to calculate crankshaft stiffness and response of rotor system

Design of pin type flexible coupling for shaft alignment

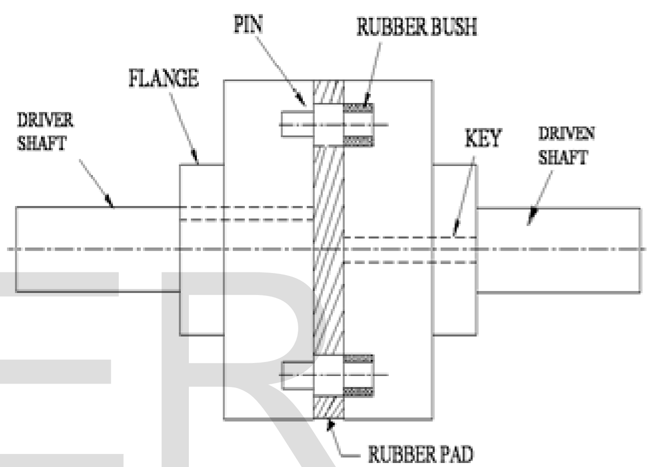


Figure 6 shaft coupled through flange

Table 1 Model Dimension

Part specification	**Dimensions in mm**
Hub diameter (dh)	120
Hub length (lh)	90
Bolt circle diameter (D)	180
Flange thickness (t)	30
Web thickness	15
Diameter of spigot and recess (dr)	90
Outside diameter of flange (Do)	270

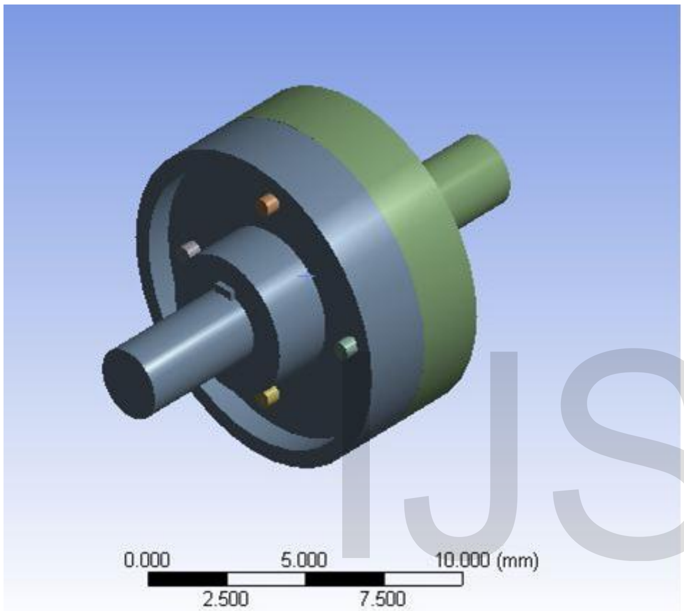
Table 2 Material used

Item Name	Material
Shaft	Plain Carbon Steel- 40C8
Flange	Grey Cast Iron
Key, Nuts, Bolts	Plain Carbon Steel- 40C8

Shaft, nuts and bolts are made up of plain carbon steel with young's Modulus 210GPa and density 7850 kg/m3 contains approximately 0.05–0.25% carbon. While Flange is made up of grey cast iron with 280MPa and density of 7200 kg/m3

Mesh Generation

Tetrahedron elements were assigned to the bodies in order to avoid non uniform mesh. The relevance was kept at 10(more the relevance finer the mesh).While relevance center and span



angle center (how related the mesh are) was kept to be medium with medium smoothing (uniformity of mesh along

Figure 7 Rigid Flange coupling model isometric view

edges or closed surface) of mesh. The initial mesh consists of 105113 elements and 163964 nodes

After the Data declaration is done, we proceed to Structural Analysis of the coupling System which is followed by Modal Analysis. Different Boundary Conditions that are shown in fig.

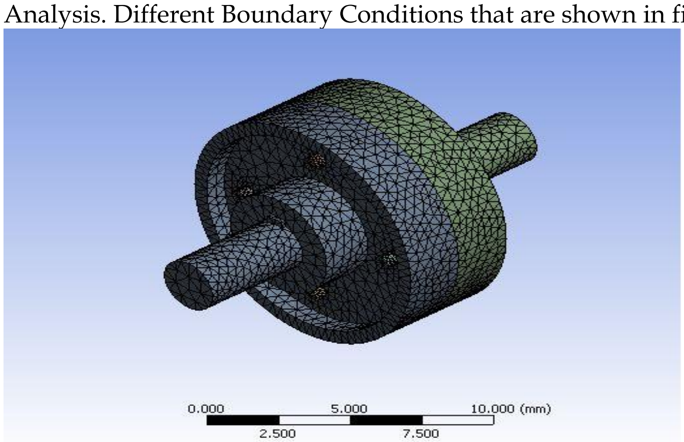


Figure 9 Mesh generation

Boundary condition for frictional and moving bolts and nuts

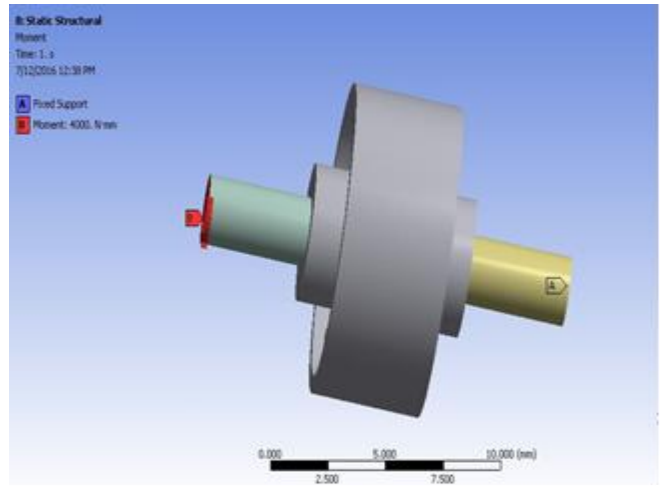


Figure 10 Boundary conditions

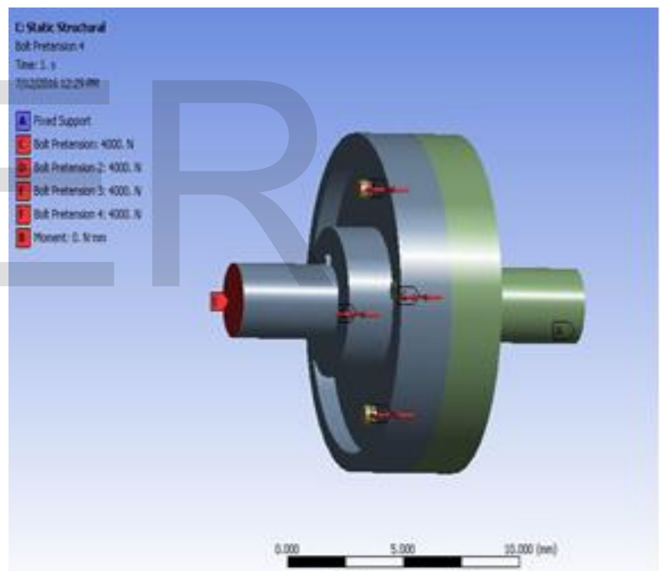


Figure 11 Boundary condition for friction and moving bolts

From above study we get following results which are then compared with literature

Table 3 Results

Parameters	From Literature		From Present Study	
	Maximum	Min-imum	Maximum	Minimum
Deformation (Z-axis)mm	1.68906187 e-3	-1.71915849 e-3	1.34E-03	-1.24 e-3
Deformation (Y-axis)mm	0.48747	0.48715	0.48315	0.48315
Deformation (X-axis) mm	0.48741	0.48715	0.48315	0.48315
Total Deformation (mm)	0.48741		0.48315	
Normal Stress (X- axis)(Mpa)	78.854	110.57	78.854	110.57
Normal Stress (Y- axis)(Mpa)	78.854	110.57	78.854	110.57
Normal Stress (Z- axis)(Mpa)	78.854	110.57	78.854	110.57
Equivalent (Von-Mises Stress) (Mpa)	380.24		326.11	

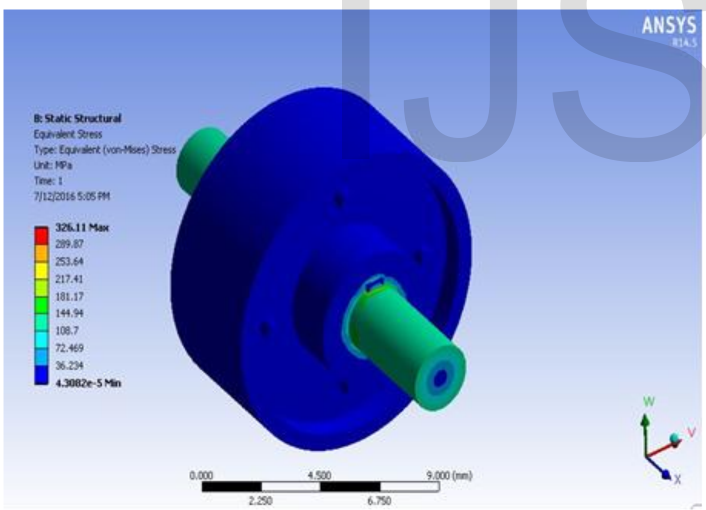


Figure 12 Equivalent Stress

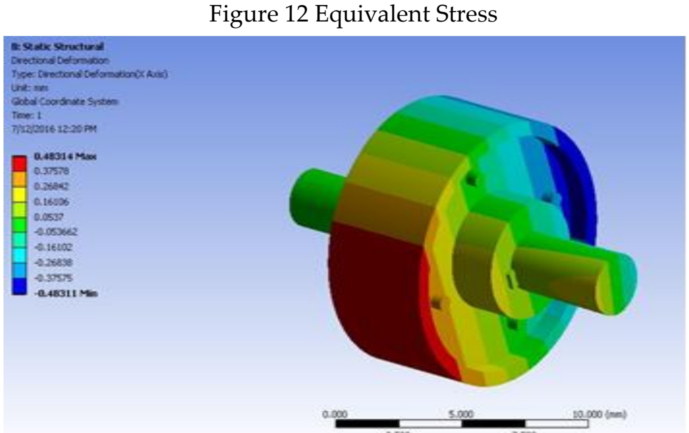


Figure 13 Deformation along X axis

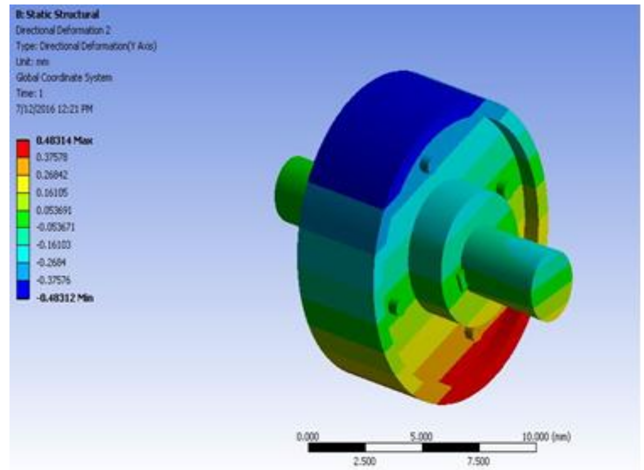


Figure 14 Deformation along Y axis

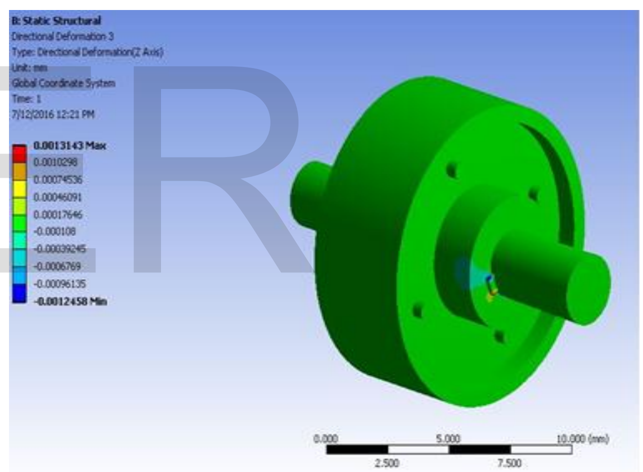


Figure 15 Deformation along Z axis

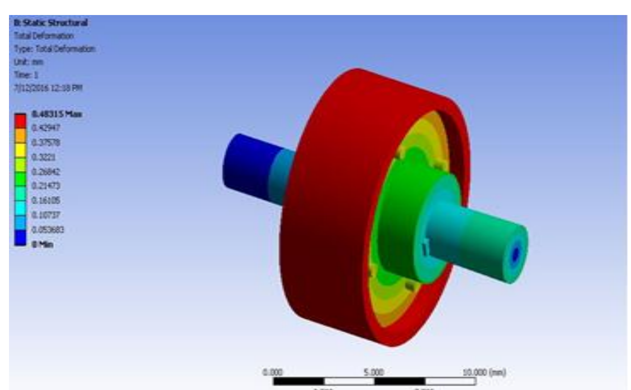


Figure 16 total deformation of flange

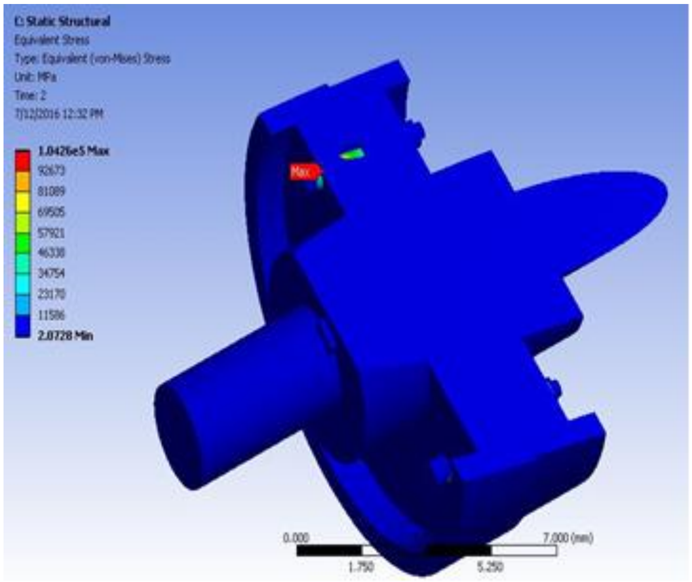


Figure 15 Equivalent stress when shaft bolts and nuts were given frictional contacts (unhealthy model)

Comparison of Modal frequencies of normal flange coupling and misaligned coupling

Table 4 Comparison

Frequency Flange(with bonded nuts), Hz	Frequency Flange(with frictional nuts), Hz
3358.4	**4868**
3358.6	**4873**
5676.8	**5145**

5. Conclusion

1) A study on dynamic vibration in coupling systems is carried out following based approach using MATLAB where mass imbalance and parallel misalignment of shaft is considered in Flange coupling Model. The results are in good agreement with the literature [10]. Moreover total residual force was plotted against node numbers to know the response of the system where fault occurs. Thus we conclude that one can find the effect of various faults on response of the coupling system. So, accordingly we can design the coupling based on requirements.

2) This method may be useful for large systems like in motors, turbines, gearboxes and the like. The present approach allows for on-line fault identification effectively.

3) We have also performed in ANSYS three dimensional analysis of Flanged coupling model to determine misalignment due to change in modal frequencies. Result is validated with literature[9]. Any misaligned shaft due to disc mass imbalance or shaft weight can cause the stresses and vibrations in coupling.

4) A faulty system of another coupled model was run in Ansys with frictional contact between nuts and bolts to capture the misalignment caused due to their movements in coupling. Thus Stress and modal frequencies of faulty system was found to be different from healthy system

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