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Numerical Response and Vibration Characteristics of Cracked Shaft

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ABSTRACT : *Vibration monitoring is the most effective technique to detect mechanical defect in rotating machinery. Early detection of cracks is essential to avoid a catastrophic event. The vibration signals gives idea about the dynamic behavior of the machinery In this work, a detailed physics-based three-dimensional model of an experimental Machinery Fault Simulator apparatus is developed using NX software. Transverse crack is developed near the disc-1and numerical analysis is carried out for healthy shaft and for damaged shaft. Modal analysis is used to evaluate the natural frequency of the system and mode shapes. Vertical response is obtained from the analysis by giving transverse excitation at different operating speed. Analysis shows that there is significant difference in response of healthy and damaged shaft.*

Keywords : *Crack, Damaged shaft, Frequency response, Healthy shaft, Vibration monitoring.*

I. INTRODUCTION

The influence of transverses crack in the shaft of the rotating machines on the associated dynamic behaviour has been focus of attention for many researchers [1-7]. The rotating machine vibration behaviour depends upon original dynamic stiffness parameter, the operating load and rotational speed of the machine. Vibration behaviour will also depend on existence of crack on the rotor as the crack would modify the machine dynamic stiffness matrix. A shaft crack is a slowly growing fracture of the rotor. If undetected in an operating machine, as a crack grows, the reduced cross section of the rotor will not able to withstand the dynamic loads applied to it. When this happens, the rotor will fail in a fast brittle fracture mode. The sudden failure releases a large amount of energy that is stored in the rotating system, and the rotor will fly apart. This kind of failure may cause serious injury or even death to anyone unfortunately standing near the machine at that moment. Obviously, shaft crack detection is a very serious matter, and machines that are suspected of having a crack must be treated with the utmost caution. Generally, when the crack is approaching to a dangerous depth, it propagates more quickly, with a propagation velocity that increases exponentially. The final growth up to a critical dangerous depth takes sometimes only few days of operation.

II. NUMERICAL WORK

A bench top Machinery Fault Simulator (MFS) made by Spectra Quest was used as the basis for developing the physics-based model. Figure 1 shows the dimensions of the MFS setup. NX8 (Nastran) was used to develop a detailed physics-based three-dimensional (3-D) model of the MFS. Several types of faults in real machinery can be simulated with this numerical model and experimentally validated. To determine the feasibility of detecting cracks in shafts by analyzing the change in natural frequencies and modal shapes due to cracking, the first task with the numerical model was to obtain these modal parameters (natural frequencies and modal shapes) for a system with a healthy shaft and a system with various faults.

To establish coupling of transverses vibrations, a simply supported rotor–bearing is considered. A single transverse surface Crack is assumed just adjacent to the first disc. Crack can be seeded in the shaft near the first disc at thickness of 0.15 mm and length of 5 mm as shown in Figure2.

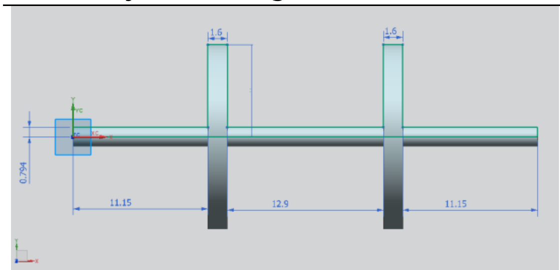

Fig. 1 3D model of MFS with dimension

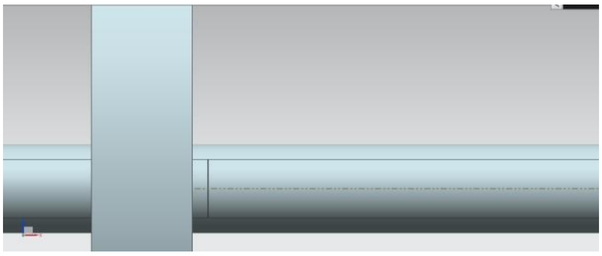

Fig. 2 Transverse crack generated in shaft adjacent to disc-1

Modal analysis by response simulation

Evaluates normal modes and natural frequencies.
Does not consider damping.
The modes correspond to the natural frequency of the body.

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\}$$

For free vibration and undamped system F=0, and C=0. So above equation becomes

$$[M]\{\ddot{x}\} + [K]\{x\} = 0$$

$|K - M\omega^2| = 0$

$$\omega_n = \sqrt{\frac{K}{M}}$$

ω_n =undamped natural frequency

The solution for a single DOF model demonstrates that frequency is proportional to stiffness (K) and inversely proportional to mass (M).

Table 1 Workflow of Response simulation

1.Build the finite element (FE) model	Define the geometry, material properties, mesh, and constraints, as you would for other structural solution types. Also, specify the locations of your excitations and define any static and dynamic loads.
2. Create the NX Nastran solution	Create an NX Nastran SEMODES 103 – Response Simulation solution.
3. Solve the model.	NX Nastran generates normal modes, constraint modes, attachment modes, and other modal information.
4. Create the Response Simulation.	After solving the model, create the Response Simulation solution process.
5. Review the mode shapes.	Review the mode shapes in the Post-Processing Navigator or in the Response Simulation Details View subpanel in the Simulation Navigator.
6. Create an event.	Define the type of response simulation you will perform, such as transient or frequency. The event combines the modal model and your excitation functions.
7. Create excitation functions.	Excitations define the loading for the response Simulation.

Table 2 Parameters selected for Response simulation

Elements	3D tetrahedral or hexahedral solid elements 1D bar, beam, rod, rigid link, and spring elements 0D concentrated mass elements
Material	Isotropic Shaft –steel

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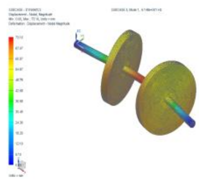
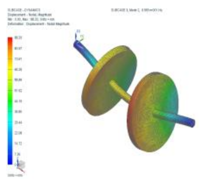
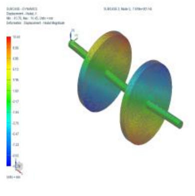
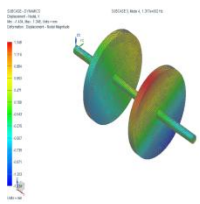
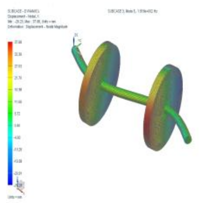
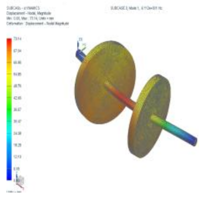
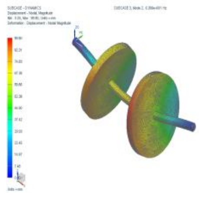
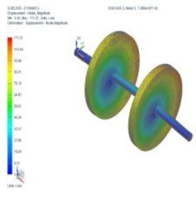
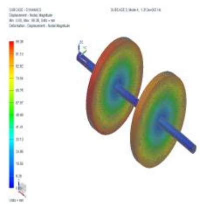
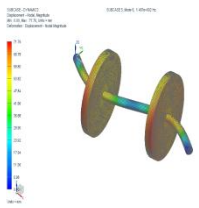
	Disc- aluminum
Material properties	Steel / Aluminum Density(kg/m^3): 7800 / 2800 Modulus of elasticity(Gpa): 200 / 70 Poisson's ratio: 0.3 / 0.345
Mesh type	3D Swept mesh
Boundary conditions	Displacement constraint Enforced motion location: The location of an enforced motion excitation on the model.
Dynamic Load	Centrifugal inertia
Operating speed	1200 rpm

Reviewing the model analysis results
Natural frequencies and mode shapes are the primary results for a modal solution.
The results are ordered by frequency, with the lowest natural frequency being the first mode shape, the next highest being the second mode, and so on.
The normal modes represent dynamic states in which the elastic and inertial forces are balanced when no external loads are applied. The magnitude of the mode shapes is arbitrary.
The amplitude of the displacement is not significant, but the relative displacement of the nodes is significant.
Mode shapes help you determine what load locations and directions will excite the structure.

Mode-1 61.48 Hz Mode- 2 65.51 Hz Mode- 3 75.80 Hz Mode- 4 131.7 Hz Mode- 5 151.8 Hz

Mode- 1 61.12 Hz Mode- 2 63.99 Hz Mode- 3 73.89 Hz Mode- 4 131.2 Hz Mode- 5 149.7 Hz

Fig 3. Numerically generated mode shapes and natural frequencies (modes 1–10) for a healthy shaft and a cracked shaft,

III. TRANSVERSE EXCITATION

Transverse excitations on bearing were applied in the healthy and damaged shaft and the coupled vibratory response was analyzed on the opposite bearing. A transverse excitation of 100 sin (107*time) Nm is applied at the first node of the uncracked rotor and cracked rotor. The response obtained is shown in figure. At the enforced motion location the transverse excitation is applied and the vertical response can be obtained from the bearing-2.shaft is rotating at 1200 rpm

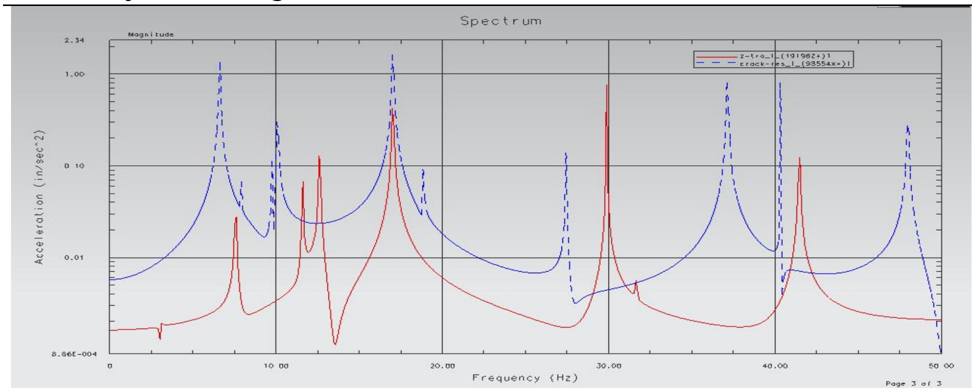


Fig. 4 Change in response of healthy shaft and cracked shaft, red line=response of healthy shaft, blue line=response of cracked shaft

IV. CONCLUSION

Coupled transverse vibrations have been studied using the finite element model of a cracked rotor using NX software. The coupling is studied with a response-dependent crack model accounting the partial crack closing. The model is useful in analyzing the response of cracked rotor to any type of excitation encountered in a rotor-bearing system, steady or transient

From the present numerical study following conclusion can be made.

The most important aspect of the on-line simulation result was the notable transient response about the Z-axis for all the cases considering a cracked shaft with an excitation in any orientation

This result was useful from a diagnostic point of view because the transient response amplitudes for the crack responses differ from the case for a healthy shaft. The response is always larger for the cracked shaft than the healthy shaft.

Due to crack there is reduction in bending stiffness, so natural frequency of the system will also reduces for the cracked shaft as compared to the healthy shaft.

Modal analysis will give us the modal parameter such as natural frequency and mode shape. The modal analysis result indicates that there is significant change in natural frequency and mode shape due to the crack in the shaft.

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