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Whirl Analysis of an Overhung Disk Shaft System Mounted on Non-rigid Bearings

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Abstract

Eigenvalues of a simple rotating flexible disk-shaft system are obtained using different methods. The shaft is supported radially by non-rigid bearings, while the disk is situated at one end of the shaft. Eigenvalues from a finite element and a multi-body dynamic tool are compared against an established analytical formulation. The Campbell diagram based on natural frequencies obtained from the tools differ from the analytical values because of oversimplification in the analytical model. Later, detailed whirl analysis is performed using AVL Excite multi-body tool that includes understanding forward and reverse whirls in absolute and relative coordinate systems and their relationships. Responses to periodic force and base excitations at a constant rotational speed of the shaft are obtained and a modified Campbell diagram based on this is developed. Whirl of the center of the disk is plotted as an orbital or phase plot and its rotational direction noted. Finally, based on the above plots, forward and reverse whirl zones for the two excitation types are established.

Keywords

Vibration; Disk-shaft system; Campbell diagram; whirl analysis;

1. Introduction

A disk-rotor system is a basic building block of many mechanical systems that find application in automotive transmissions, differentials, and accessory drives, electrical motor output shafts, etc., to name a few. Understanding vibration characteristics of a rotating disk-rotor system can help us refine the system level design by addressing certain resonance conditions or *NVH* issues. Before personal computers became a widespread commodity, researchers mainly relied on analytical formulation to solve vibration analysis problems of various disk-rotor systems. However, this came at the cost of accuracy due to certain assumptions and/or simplifications.

In literature, vibration analysis was addressed for systems consisting of a flexible and/or rigid shaft carrying flexible or rigid single or multiple disks. Some of the earlier methods used lumped parameter approximation. This approach mainly used transfer matrix methods [1, 2]. Lumped parameter approach simplifies the problem by dividing the system into a series of mass/inertia (nodes) connected by spring-damper elements. However, a true physical system having infinite degrees of freedom cannot be accurately represented by this approach. Natural frequencies and critical speeds of even a simple disk-shaft system obtained by this approach is slightly off compared to those obtained by more accurate methods. The problem of a continuous flexible non-rotating shaft carrying a rigid disk was solved by Srinath et al. [3]. Similar system for the non-rotating case was solved by Eshleman et al. [4]. In both the cases, the thickness of the disk was ignored, and characteristic equation method was used to solve the system eigenvalues. Influence of disk flexibility on the natural frequency of bending and critical speeds of a rotating disk-shaft system was solved by Chivens et al. [5]. Nataraj [6] developed a mathematical model that investigated the interaction between the torsional and flexural deflections of a uniform shaft rotating at a constant speed. Chiu et al. [7] analytically solved coupled vibration among shaft torsion and blade bending of a multi-disk rotor system with grouped blades by ignoring shaft bending in the formulation. Chowdhury et al. [8, 9] analytically examined the coupled vibration of high speed geared and pulley systems mounted on short, rotating, flexible shafts. Galerkin's method was used to discretize the discrete-continuous systems. For a low speed geared shaft system, Chowdhury et al. [10] used assumed modes method to find the effect of shaft flexibility and gear mesh stiffness on the coupled vibration modes and responses.

Upon enhancement of digital computers, dynamic finite element-based methods have become popular to solve such systems. Taplak and Parlak [11] used a program named

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While for force excitation frequency up to 80 *Hz*, disk node rotational direction is opposite to the shaft rotation, the same frequency for base excitation is 66 *Hz*. Knowing true vibrational dynamic characteristics such as the above can help us design better rotor dynamic systems by avoiding resonance conditions.

4. Conclusions

Shaft speed is kept constant while the excitation frequency is varied.

1. For the gyroscopic disk shaft system, Campbell diagram obtained in frequency domain from FE tool, Ansys and dynamic analysis tool, AVL Excite (*shaft modeler*) is very similar and differ from plots based on simplistic analytical equations. The difference comes due to realistic calculation of shaft bending stiffness.
2. In Absolute coordinate system, for a degenerate mode, frequency of forward whirl mode is higher than the corresponding mode with a reverse whirl. The relationship gets reversed in Relative coordinate system. This is because, the frequency of the forward whirl gets reduced by the rotational frequency of the shaft while that of the reverse whirl gets increased by the same amount.
3. Campbell diagram based on response in time domain showed that resonance frequencies for both forward and reverse whirls are slightly lower than the values obtained in frequency domain. This is because in the frequency domain nonlinearities due to bearing clearance is ignored.
4. For both the force and base excitations, the highest peak observed corresponds to the forward whirl condition. Although extremely rare, a slight peak is observed for reverse whirl condition as well.
5. For the magnitude of excitations chosen, base excitation is more predominant than the force excitation. These values are representative of an IC engine. The magnitude of the force corresponds to the force coming on the crank pin while magnitude of the base displacement corresponds to the displacement of the engine on the mounts. As such, engine vibration can be more damaging than cylinder pressure to the durability of the crankshaft.
6. Even though the cut off frequencies are not exactly the same, the orbital diagrams show interesting trends for both the force and base excitations:
   a) At lower frequencies, the orbital plots are aligned to the excitation direction and at higher frequencies, these are perpendicular to the direction of excitation.
   b) At lower frequencies, the whirl direction is opposite to shaft rotation and at higher frequencies the direction is along to the shaft rotation.
   c) At resonance, the orbit is circular and in Absolute and Relative coordinate systems these overlap.
7. Response to base excitation is more intricate than the force excitation.
   a) At the resonance frequency for reverse whirl, the radial displacement is twice the magnitude of base excitation indicating that the motion of the disk-center is synchronous and opposite in phase to the base motion.
   b) In-between the forward and reverse whirl resonance frequencies, the shaft is bent while the disk makes small wobble around it.
   c) As the Relative coordinate is fixed translationally, unlike the force excitation, orbits in Absolute and Relative coordinate systems differ at higher frequencies.

Nomenclature

Ω	Angular velocity of disk rotation
ω_1, ω_2	Forward and reverse whirl frequencies
I_1	Moment of inertia of the rotating shaft-disk system about the diametric axis through O
I_2	Moment of inertia of the rotating shaft-disk system about the shaft axis
k	Bending stiffness of the cantilever shaft at the point of support assuming cylindrical shaft
Absolute coordinate system	Coordinate system that is fixed to the ground
Relative coordinate system	Coordinate system that rotates with the shaft. However, no translational motion is allowed.

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