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INFLUENCE OF BACKUP BEARINGS AND SUPPORT STRUCTURE DYNAMICS ON THE BEHAVIOR OF ROTORS WITH ACTIVE SUPPORTS

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Progress to Date

Substantial progress has been made toward the goals of this research effort in the past six months. The tasks that have been accomplished to date are:

1. A simplified rotor model with a flexible shaft and backup bearings has been developed. The model is based upon the work of Ishii and Kirk. Parameter studies of the behavior of this model are currently being conducted.

2. A simple rotor model which includes a flexible disk and bearings with clearance has been developed and the dynamics of the model investigated. The study consists of simulation work coupled with experimental verification. The work is documented in the attached paper.

3. A rotor model based upon the T-501 engine has been developed which includes backup bearing effects. The dynamics of this model are currently being studied with the objective of verifying the conclusions obtained from the simpler models.

4. Parallel simulation runs are being conducted using an ANSYS based finite element model of the T-501 developed by Dr. Charles Lawrence.

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1. Flowers, G.T., and Wu, Fangsheng, "Disk/Shaft Vibration Induced by Bearing Clearance Effects: Analysis and Experiment," to be presented at the **Second Biennial European Joint Conference on Engineering Systems, Design, and Analysis (ESDA)**, July 4-7, London, England; submitted to *ASME Journal of Vibration and Acoustics*, February, 1994.

DISK/SHAFT VIBRATION INDUCED BY BEARING CLEARANCE EFFECTS: ANALYSIS AND EXPERIMENT

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Abstract

This study presents an investigation of the dynamics of a rotor system with bearing clearance. Of particular interest is the influence of such effects on coupled shaft/disk vibration. Experimental results for a rotor system with a flexible disk are presented and compared to predictions from a simulation model. Some insights and conclusions are obtained with regard to the conditions under which such vibration may be significant.

Nomenclature

e = imbalance eccentricity
f_{n1}, f_{n2} = nonlinear bearing forces
r_1 = gyroscopic mass influence ratio
r_2 = disk mass influence ratio
x_1, x_2 = shaft degrees of freedom
x_3, x_4 = disk degrees of freedom
Δ = bearing clearance
Ω = rotor speed
ω_1 = natural frequency of rotor support in x_1 direction
ω_2 = natural frequency of rotor support in x_2 direction
ω_3 = natural frequency of rotor disk
τ = Ωt
ξ_1 = damping ratio of rotor support in x_1 direction
ξ_2 = damping ratio of rotor support in x_2 direction
ξ_3 = damping ratio of rotor disk
$(\dot{\ }) = \frac{d}{d\tau}$

Background and Motivation

Rotor systems typically consist of a shaft with one or more bladed disks attached. Disk and blade vibration are issues of important concern for stress analysts. Complex finite element models are developed to assess the dynamic stresses in such components and insure the integrity of the design. However, models for rotordynamical analyses are typically developed assuming that disk flexibility effects are negligible. It is generally presumed that the disk is sufficiently rigid so as to not significantly impact rotor vibration over the operating region.

There is a fairly large body of work documented in the literature concerning studies of disk flexibility on rotors and turbomachinery. A discussion of this previous work with regard to linear rotor systems is presented in Flowers and Ryan (1991). An excellent source for a comprehensive review of work in this area is presented by Davis (1989).

Previous investigators into the area of coupled rotor/disk vibration have noted that disk flexibility has little effect on critical speeds but that it may significantly influence higher natural frequencies of the rotor system [Chivens and Nelson (1975)]. One can draw the conclusion from these results that synchronous vibration due to imbalance will be little affected by disk flexibility. However, there are sources of higher frequency excitation that could serve to excite these natural frequencies. Perhaps the most obvious are multi-synchronous effects corresponding to the blade pass frequency (from fluid forces impinging on the rotor blades). Another potential source is from nonlinearities that may be present in the system. For example, nonlinear bearing forces due to clearance effects may result in supersynchronous rotor vibration. There are quite a number of studies in the literature concerning the effects of bearing clearance (and the related phenomena of rubbing) on rotordynamical behavior. Some of the works that have most influenced the current study were conducted by Johnson (1962), Black (1968), Ehrich (1966), and Childs (1978), and Muszynska (1984).

Advanced designs for many types of high speed rotating machinery that use magnetic bearings for support have been proposed and are in the development and construction stages. There are a number of such systems already in commercial use. Rotor systems supported by magnetic bearings must have backup bearings to provide support under overload conditions or if the magnetic bearing fails. Backup bearings are characterized by a clearance between the rotor and the bearing such that contact does not occur under normal operating conditions. As a result, issues related to the effects of bearing clearance on rotordynamical behavior are of current concern. The objective of the present work is to develop an understanding of possible coupling between the dynamics of disk and shaft that may be induced by bearing clearance effects and to provide guidance to designers concerned with such systems.

Experimental Model

In order to investigate whether bearing clearance can lead to significant coupling between rotor and disk vibration, experimental tests were performed with a rotor test rig. A drawing of the test rig is shown in Figure 1.

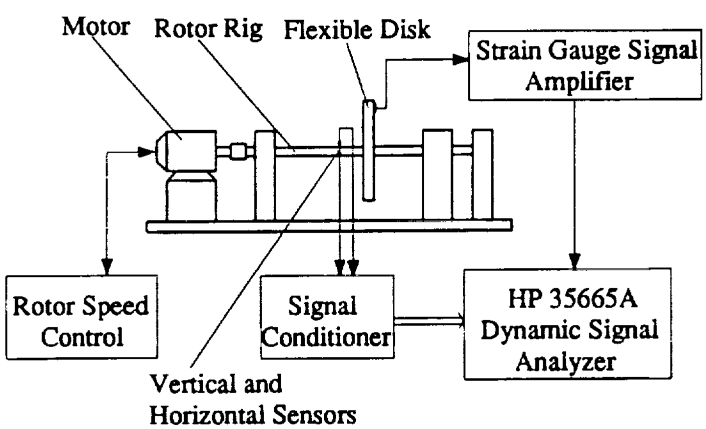

Figure 1 Experimental Test Apparatus

The rotor used in this study has two basic components: a flexible disk and a shaft. The steel shaft is 0.375 in. diameter and 18 inches long. The flexible disk is a circular aluminum plate 0.0125 inches thick and 14.0 inches in diameter. The natural frequency of its lowest one nodal diameter bending mode is about 40 Hz. It is attached with bolts to a 2.0 inch diameter hub that is fixed on at

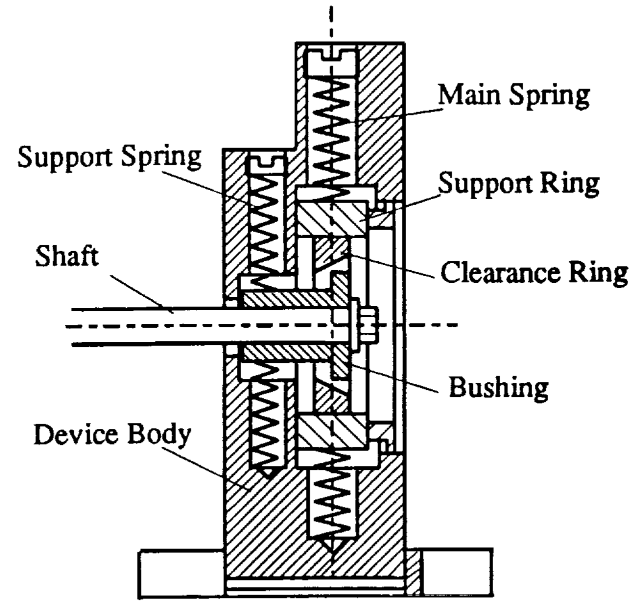

Figure 2 Shaft Right End Support Device

the center of a 0.375 inch diameter steel shaft 16.0 inches in length.

The rotor shaft is supported by bushings at two ends. The left end of the shaft is placed directly in the bushing base. This support provides a force but only a minimal couple so that the shaft is effectively free to pivot about this point. The right bearing is supported by a special device, which is designed to provide stiffness and to simulate bearing clearance. A diagram for this device is shown in figure 2. It has two sets of springs. The softer springs are to support the weight of the rotor, and the harder springs act as the main stiffness for the system. The stiffness in the x and y directions of each spring set may be different. The clearance is adjustable by turning the inner tapered ring in or out. In this experiment, the spring constant of the soft spring is 1.9 lb/in. The spring constant of the hard spring is 8.1 lb/in. in the x_1 (horizontal) direction and 19.7 lb/in. in the x_2 (vertical) direction.

The rotor is driven by a speed adjustable motor. The speed of the rotor can be controlled by turning the speed control knob on the front panel of the control box. The measurement and analysis system comprises proximity displacement sensors, sensor conditioners and a signal analyzer. A vertical and a horizontal displacement sensor were mounted to pick up the displacement signals to form orbit plots. A third transducer was used to sense the disk vibration. During the test, displacement signals of the shaft were sent to the signal analyzer, where the orbit trajectories were recorded and the frequency components of the vibration were analyzed.

Simulation Model

The simulation model is similar to that developed in an earlier study [Flowers and Wu (1993)]. The primary differences are that rotational stiffening of the disk have been taken into account and that damping is included in the bearing support forces. A schematic diagram of the model is shown in figure 3. It consists of a rigid shaft, a rigid hub, a flexible disk and a support with a symmetric clearance. The equation development is based on the following considerations:

(1) The disk is assumed to flex only in the lateral direction.
(2) Only rotational vibration is considered. This is because only the rotational motion is coupled with the lateral disk vibration. One nodal diameter disk vibration is assumed.
(3) The rotor speed is constant.

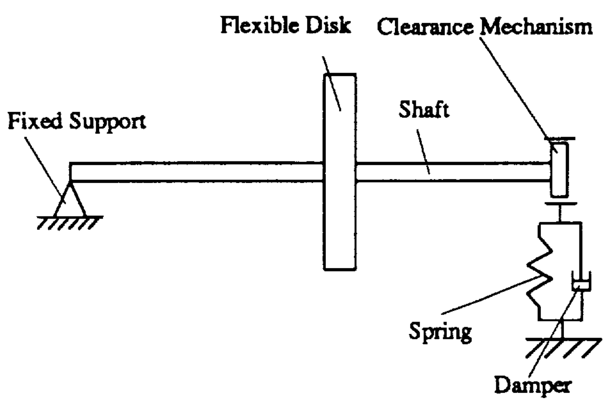

Figure 3 Simulation Model

After some mathematical manipulations, the dimensionless equations of motion for the whole system can be obtained as

$$\ddot{x}_1 + r_1\dot{x}_2 + r_2\ddot{x}_3 + 2\dot{x}_4 + f_{n1} = \cos\tau \qquad (1)$$

$$\ddot{x}_2 - r_1\dot{x}_1 + r_2\ddot{x}_4 - 2\dot{x}_3 + f_{n2} = \sin\tau \qquad (2)$$

$$\ddot{x}_3 + 2\dot{x}_4 + 2\xi_3\omega_3\dot{x}_3 + (\omega_3^2 - 1)x_3 = -(\ddot{x}_1 + 2\dot{x}_2) \qquad (3)$$

$$\ddot{x}_4 - 2\dot{x}_3 + 2\xi_3\omega_3\dot{x}_4 + (\omega_3^2 - 1)x_4 = -(\ddot{x}_2 - 2\dot{x}_1) \qquad (4)$$

$$f_{n1} = -\phi\left[(1 - \frac{\delta}{\sqrt{x_1^2 + x_2^2}})(\omega_1^2 x_1) + 2\xi_1\omega_1\dot{x}_1\right]$$
$$f_{n2} = -\phi\left[(1 - \frac{\delta}{\sqrt{x_1^2 + x_2^2}})(\omega_2^2 x_2) + 2\xi_2\omega_2\dot{x}_2\right]$$
$$\delta = \frac{\Delta}{e}$$

$$\phi = \begin{cases} 1 & \text{if} \sqrt{x_1^2 + x_2^2} > \delta \\ 0 & \text{otherwise} \end{cases}$$

In the above equations, x_1, x_2, x_3 and x_4 are scaled by e, the imbalance eccentricity. Time is scaled by the rotor speed, Ω. f_{n1} and f_{n2} are the nonlinear moments due to contacting between the rotor and the bearings.

The effective stiffness of a spinning disk can be strongly influenced by the spin speed. Based upon earlier work [Wu and Flowers (1992)], the lowest natural frequency of such a disk can be written as

$$\omega_3^2 = \frac{19 + \nu}{16} + \frac{\omega_{30}^2}{\Omega^2}$$

where ω_{30} is the natural frequency of the nonspinning disk. This expression is used in equations 3 and 4 to account for spin stiffening effects.

Comparison of Experiment and Simulation

A number of tests were conducted using the experimental setup described in the previous section. In addition, studies for differing parametric configurations have been conducted using the simulation model described above. The basic results are very similar to those obtained by Flowers and Wu (1993). The results presented here are typical. The following discussion is directed at comparing the predictions of the simulation model with experimentally observed responses, with the objective of obtaining insight into the behavior of flexible disk rotor systems with bearing clearance effects.

First, the linear characteristics of the system were examined. Figure 4 shows experimentally obtained data of the natural frequencies for the coupled disk/shaft vibration mode as a function of rotor speed. Using the data from such tests and additional measurements and calculations, the linear stiffness, damping, and mass characteristics of the experimental rotor system were identified. The numerical values are shown in Table 1.

Parameter	Value
r_1	0.5
r_2	0.1
ξ_1	0.06
ξ_2	0.06
ξ_3	0.01
ω_1	0.55
ω_2	0.85
ω_3	1.8
ν	0.3
δ	1.2

Table 1 Simulation Model Parameters

Figure 4 also shows the natural frequencies as a function of rotor speed obtained from the simulation model. These results agree relatively well with the experimentally obtained data and serve to validate the structural parameters selected for the simulation model. Ω, 3Ω, and 5Ω lines are also shown on the figure. The intersections of these lines with the natural frequency curve provides significant insight into the nonlinear behavior of the flexible disk rotor system, as is discussed below.

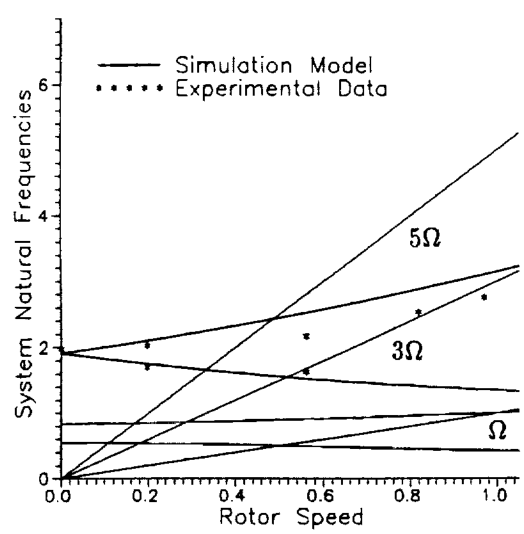

Figure 4 Rotor System Natural Frequencies

Next, the nonlinear behavior of the system was investigated. A clearance value was selected and the rotor speed was slowly increased over a range of rotor speeds, with the resulting response amplitudes and frequencies measured and recorded.

It is important to note that many vibration frequencies are possible. During the course of the study, 1Ω – $6\ \Omega$ frequency components were observed for certain parametric configurations. In addition, frequency components that are fractions of the rotor speed were observed for certain speed ranges and initial conditions. However, for a rotor with a symmetric clearance and minimal frictional effects, the odd integer multiples of the rotor speed appear to generally be the primary supersynchronous frequency components. The current discussion will concentrate on those frequencies. However, the basic conclusions should be directly applicable to other supersynchronous vibration frequencies as well.

Figure 5 shows the experimentally measured amplitudes of the Ω, 3Ω, and 5Ω components of the shaft response for various rotor speeds. Simulation results are also shown on figure 5. These results were obtained using the harmonic balance method and were verified at selected points through direct numerical integration of the equations of motion. There is relatively good agreement between the simulation results and the corresponding experimental data. Certainly the basic trends match quite well.

An important observation that can be made from examination of these figures is that the peaks of the respective components correspond to the the intersections of the Ω, 3Ω, and 5Ω lines with the natural frequency curves. It appears that the nonlinear effects are serving to excite the coupled modes of the rotor/disk system. This is true for both the forward and backward whirl modes of the rotor system.

The peaks occurring at a rotor speed of 0.51 correspond to the intersections of the 3Ω line with the natural frequency curve. They relate to a backward whirl mode. The peaks occurring at rotor speeds of 0.33 and 0.485 correspond to the intersections of the 5Ω line with the the natural frequency curve. Those occurring at 0.33 relate to a backward whirl mode and those occurring at 0.485 relate to a forward whirl mode. Note that the amplitudes of the supersynchronous components associated with the forward whirl mode of the disk/shaft vibration are much lower than those associated with the backward whirl mode. This result was observed for all the parametric configurations examined in the course of this study.

The Ω, 3Ω, and 5Ω components contribute (more or less significantly) to the overall response at the corresponding intersection point. However, it is important to note that the Ω component is increased rather dramatically at the intersection points as a result of the presence of disk flexibility. Apparently, the nonlinear coupling between the various frequency components serves to produce this effect.

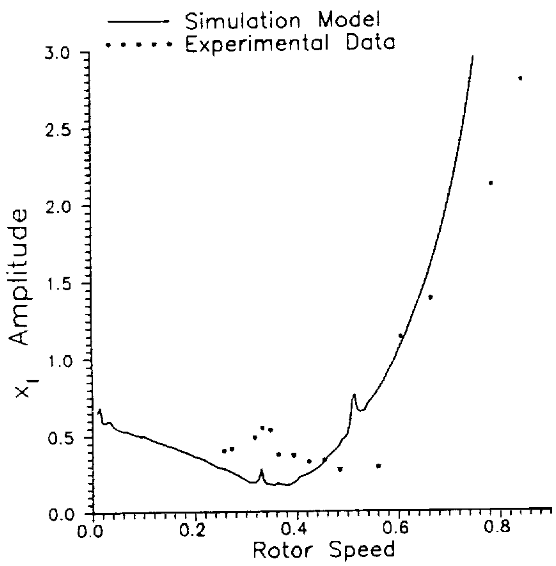

Figure 5.a Ω **Component of Shaft Vibration Response**

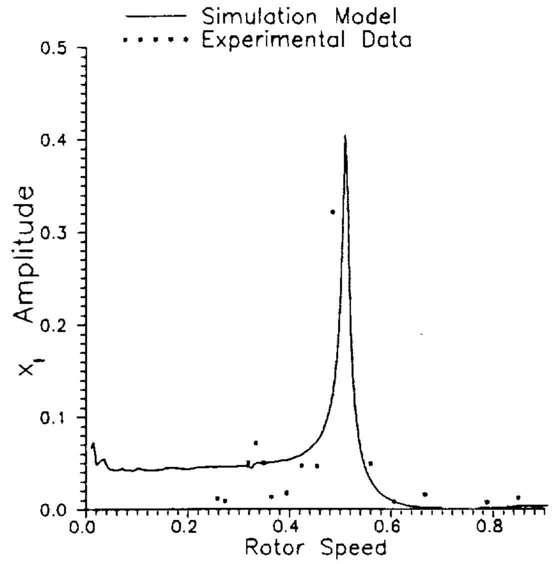

Figure 5.b 3Ω **Component of Shaft Vibration Response**

Conclusions

A study of disk/shaft vibration induced by bearing clearance effects has been presented. Both experimental tests and simulation results have been presented. The responses predicted by the simulation model and those observed experimentally agree quite well. The behavior of both the experimental test rig and the simulation model was quite sensitive to changes in parametric configuration. As a result it is quite difficult to predict exactly how a certain system is going to behave in the presence of a bearing clearance effect, i.e., whether or not coupled disk/shaft vibration will be significantly excited. However, a few general conclusions and guidelines can be drawn from this study.

1. An understanding of the mechanism for coupled disk/shaft vibration induced by clearance effects has been obtained. The nonlinear effects have served to produce superharmonics that excite coupled disk shaft modes. Conversely, the additional degree of freedom pro-

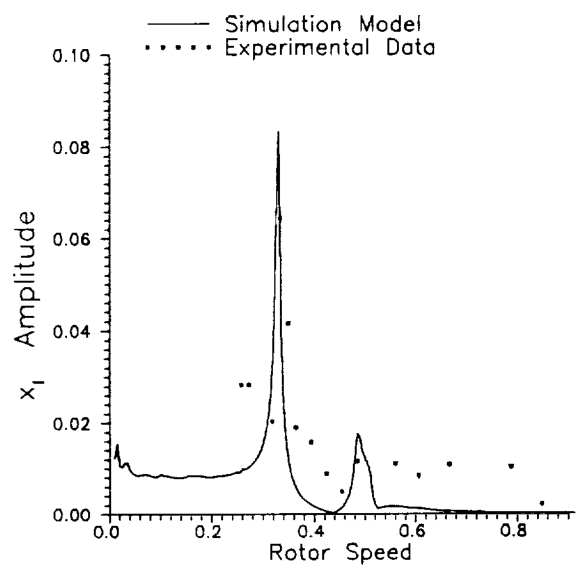

Figure 5.c 5Ω **Component of Shaft Vibration Response**

vided by the disk flexibility has served to exaggerate the frequency components of the response, resulting in higher amplitudes at the rotor speeds corresponding to the respective natural frequency/multi-synchronous line intersection points.

2. Bearing clearance effects primarily serve to excite backward whirl modes for coupled disk/shaft vibration. Forward whirl modes may be excited but the associated amplitudes appear to be much lower.
3. The bearing support stiffnesses used in this study were not symmetric. These nonsymmetries, together with the clearance, have served to produce the supersynchronous vibration that excites the coupled disk/shaft vibration modes.
4. It appears to be relatively difficult to excite coupled disk/shaft vibration with bearing clearance effects. The occurrence of such behavior was very sensitive to rotor speed, as judged by the relative difficulty encountered during the experimental work in obtaining the peak amplitude responses. This conclusion is verified by the simulation work that indicates that the behavior occurs only over a very limited ranges of rotor speed. The supersynchronous frequency must almost exactly coincide with a natural frequency in order to excite the behavior.
5. From a design perspective, the Campbell diagram is a useful tool to predict when such behavior may occur. The intersections of the corresponding supersynchronous line with the coupled disk/shaft vibration mode frequency curve will indicate at what rotor speeds peak responses for that frequency component are likely to occur. Whether or not such behavior will actually occur and the relative significance of such effects depends very strongly on the imbalance, clearance, damping, and stiffness values.

Acknowledgement

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