Experiments on the Dynamic Behavior of a Supercritical Rotor

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Tests have been performed on supercritical rotors to determine the sensitivity to unbalance and the suitability of balancing techniques. Results are presented for a rotor with an overhanging disk and supported on two rolling element bearings in series with squeeze-film dampers. The rotor has two flexural modes with high relative strain energy in the speed range up to 55,000 rpm. After completion of the balancing exercise the rotor could be run to maximum speed and was found to be stable and free from half-frequency whirl instability, depending on the oil inlet pressure of the dampers. Pressurization of the dampers and increasing the clearance of the dampers had a very desirable effect on the stability behavior and the unbalance response.

Introduction

The increasing demand for higher speeds and lighter structures in turbomachinery encourages the use of rotors that operate in the supercritical speed range. For these rotors, design criteria must be established regarding the damping and balancing requirements that will ensure safe operation, especially at speeds near and at the bending critical speeds in the operating range. In addition to these requirements, there may be limitations related to instabilities.

Sufficient external damping is essential in the application of supercritical rotors. The damping suppresses the potentially large deflections at the critical speeds. It may also be required to overcome other dynamic phenomena occurring in supercritical rotors. Among these are subharmonic resonance due to the non-linear character of the damper forces and instabilities due to internal friction, for example, from shrink fits, sleeves, bolted connections, splines, and flexible couplings.

The balancing requirements for supercritical rotors also play an important role in the design phase of aircraft turbine engines. In the first place, balancing limits for production rotors must be attainable with available balancing equipment and methods. Additional problems must also be resolved, such as how to ensure the balance condition of rotors with components that have been replaced in the field.

In order to gather experience with regard to the dynamic behavior of supercritical rotors, a number of test rotors has been run in a research program. It is the purpose of this paper to present test results for one of these rotors and to explain some of the aforementioned problems in the light of the data.

Related Work

The tests have been conducted in an available rig as an extension of a program of earlier investigations related to squeeze-film dampers[1-3]. In these investigations design details of dampers were considered, such as oil supply to the damper, end seals to approximate the "long bearing" condition, no end seals, squirrel cage or retainer springs to center the damper journal, etc. Similar efforts have been reported in the literature. Theoretical predictions of the behavior of closed and open ended dampers are given in [4,5]. A numerical solution of pressure distributions in dampers with arbitrary boundary conditions, surface discontinuities, and variable inlet pressure is reported in [6]. The identification of subharmonic resonances relative to the critical speeds of flexible rotors is discussed with reference to the detailed rotor motion in [7]. Instability problems have been investigated both theoretically and experimentally and references of immediate interest in this regard are [8-10]. The balancing technique is discussed in [11]. A theoretical prediction of the stability behavior of flexible rotors on squeeze-film dampers is given in [12].

Test Rotor

The objective of the research program is to establish design criteria for supercritical rotors with respect to critical speeds and unbalance response. This is particularly important in the design of power turbine (PT) rotors which, in the compact arrangements of small aircraft turbine engines, are preferably chosen to be co-axial through the gas generator. Consequently, subcritical PT rotors generally present severe constraints on the design of the engine, especially on the bearing span, diameter and speed of the PT rotor, and on the bore and the length between the PT bearings of the gas generator rotor. The use of a supercritical PT rotor provides some relief on these constraints and, therefore, allows improvement in other basic engine design limits, such as cost, weight, and performance.

A number of test rotors, at a scale suitable for testing in an available rig after a minimum of modifications, has been manufactured and is in the process of being tested. The present paper concerns one of them, a rotor on two bearings with an overhanging disk and a balancing collar on the shaft

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Contributed by the Vibration and Sound Committee and presented at the Design Engineering Technical Conference, Hartford, Conn., September 20-23, 1981, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Manuscript received at ASME Headquarters, June 10, 1981. Paper No. 81-DET-59.

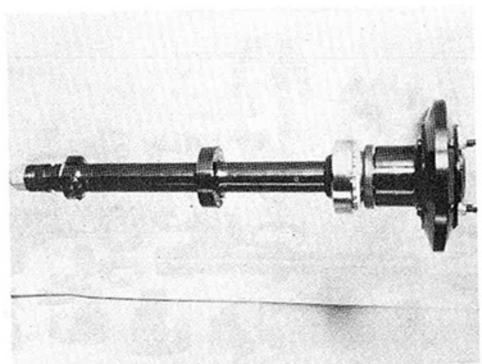

Fig. 1

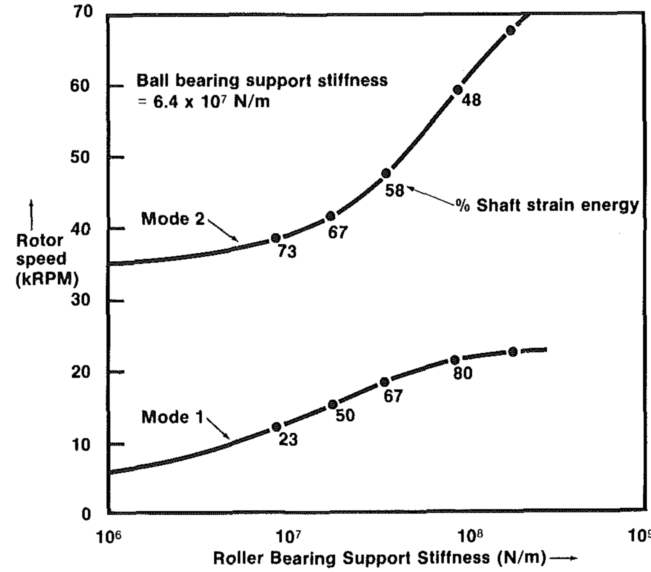

Fig. 2

Table 1

Mode	Critical speed (RPM)	Shaft strain Energy (%)
1	18,500	66.5
2	48,000	57.5

Table 2

Mode	1 g. cm Unbalance at	Deflection (mm) of		Bearing load (N) at	
		Disc	Collar	Roller	Duplex
1	Disc	.095	.004	680	280
2		.010	.017	190	120
1	Collar	.033	.011	250	120
2		.011	.216	2600	2100

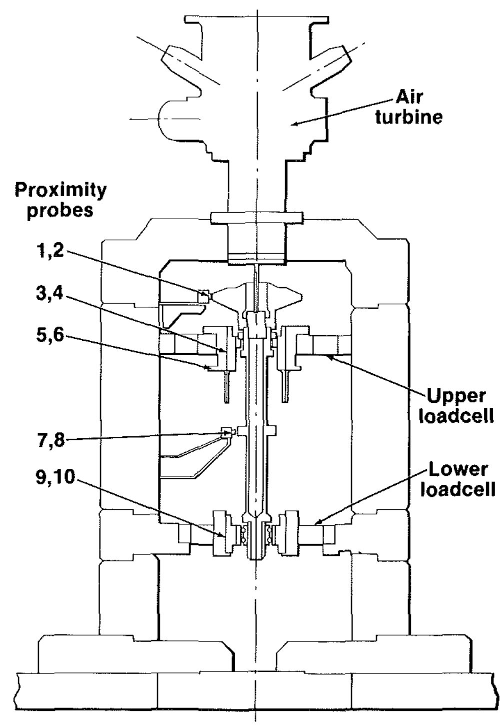

Fig. 3

between the bearings (Fig. 1). No attempt was made to simulate engine hardware in detail. The total length of the shaft and disk is 38 cm (15 in.) and the total weight is 2.9 kg (6.5 lb). The disk is the same as one used in the tests of references [1-3]. The rotor cross-section is shown in Fig. 3.

The design of the rotor included routine analysis by means of existing, transfer matrix type, critical speed, and unbalance response programs. The rotor is supported on a roller bearing in series with a squirrel cage retained oil-film damper at the disk end and on a duplex ball bearing in series with an oil-film damper at the other end. The radial stiffness of each half of the duplex bearing with the oil-film damper has been calculated by means of finite element methods to be 64 10^6 N/m (367,000 lb/in.). Similarly, the roller bearing with the oil-film damper has a radial stiffness of 35 10^6N/m (200,000 lb/in.). The designs of the various rotors differ in the critical speeds and the relative amounts of strain energy in the rotors at the critical speeds. The predicted results for the present rotor are given in Table 1. In the first mode, the disk has substantially more deflection than the collar; in the second mode, there is more deflection at the collar than at the disk. Predictions for bearing support stiffnesses are generally known to be relatively inaccurate. As a consequence, predictions for critical speeds are also inaccurate. Figure 2 shows how the predicted critical speeds and the relative amounts of shaft strain energy depend on the assumed stiffness of the roller bearing support. Unbalance responses can also be predicted and are presented in Table 2 for an arbitrary unbalance of 1 g. cm either at the disk or at the collar.

The theoretical predictions are based on linear models in which the stiffness and damping of the oil-film dampers are assumed to have constant values independent of load and deflection amplitudes. The damper stiffnesses are taken into account in the assumed support stiffnesses. The damping values are similarly represented by average damping coefficients for the support structures. Separate analyses can be conducted to predict the nonlinear damper behavior as a function of bearing load.

Test Arrangement

The rig on which the rotor was tested is the high speed rig

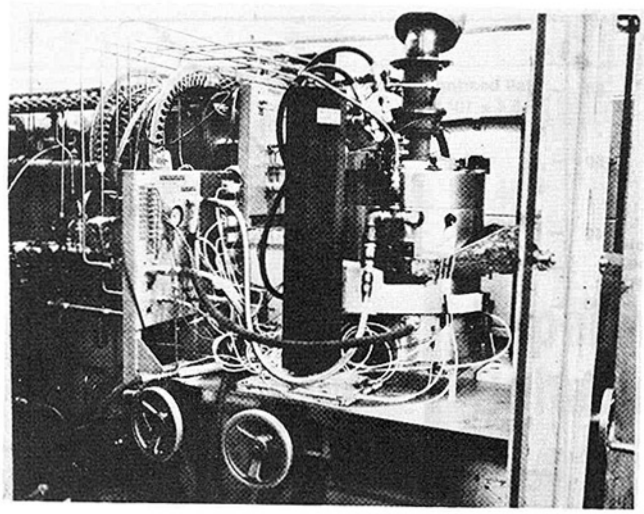

Fig. 4

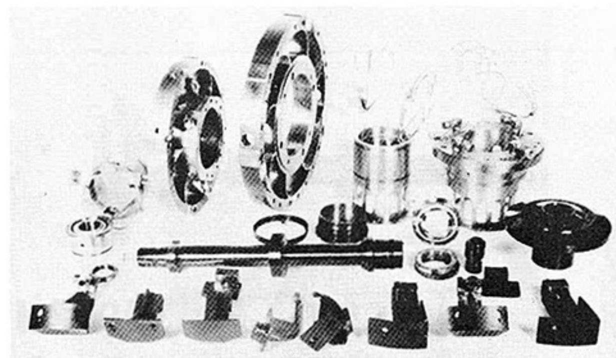

Fig. 5

Table 3

Runs	Purpose
1 to 4	Single plane balancing.
5	'As is' rotor vibration data up to 40,000 RPM.
6	Disc unbalance of 0.5 g, speed to 40,000 RPM.
7	Collar unbalance of 0.3 g, speed to 40,000 RPM.
8	'As is' rotor vibration data up to 55,000 RPM.
9	Disc unbalance of 0.5 g, speed to 55,000 RPM.
10	Collar unbalance of 0.5 g, speed to 55,000 RPM.
11	Reduced radial clearance in lower damper, inlet pressure 345 kP (50 psi).
12	Reduced radial clearance in lower damper, inlet pressure 689 kP (100 psi).

used for earlier damper tests[1-3]. The rig could accommodate in the previous tests a vertical rotor on a single duplex bearing which was mounted in series with a damper in a load cell. The rig was modified to accept a rotor on two bearings (Fig. 3), the lower bearing support being unchanged, by adding the top bearing support in the form of a load cell which carries the damper and the squirrel cage retainer spring. Additional containment rings were provided, as well as the required instrumentation and lubrication features. Figure 4 provides a general view of the test rig. Figure 5 shows various parts of rotor, bearing supports, and instrumentation brackets.

The outer races of the bearings are shrunk fit inside the damper journals. These have complete lateral freedom of movement limited only by the damper forces and, in the case of the roller bearing, by the squirrel cage stiffness. The dampers are continuously supplied with oil under pressure. The outer damper races are mounted in the housings of the load cells. The load cells are instrumented with strain gages and have been calibrated to measure the load in two, mutually perpendicular, radial directions.

The containment casing can be evacuated to 40 Torr in order to minimize the required driving torque necessary to overcome the windage loss. The rotor is driven by a commercial air turbine fed by shop air and connected to the rotor by a quill shaft. The maximum speed attained with this rotor is 55,000 rpm. The rotor bearings are lubricated continuously through spray nozzles.

Rotor motions are monitored by means of inductance type proximity probes. Two probes are mounted mutually orthogonally in each of five planes (Fig. 3). A marker signal is obtained from an additional probe facing the upper disk neck which has a small slot. The marker signal is a once per revolution pulse. In addition to the bearing loads that can be measured in each load cell in two directions, there are also two thermocouples which measure the temperature in each damper bearing race. The oil flow to the dampers is provided by separate lubrication circuits with provision for measurement of the flow in and out of each damper. The lubricant inlet pressures are controlled with regulating valves.

The speed of rotation is read on an electric counter driven by a signal from a magnetic pickup which faces a wheel on the drive shafts with 60 teeth. Data reduction of all measurements involves recording on FM tape, transcription to digital tape, and processing by mini computer. The variables recorded are rotor speed, deflection amplitudes, bearing loads, damper and lubricant temperatures, and lubricant supply pressures and flow rates. Additional information on the rig and the instrumentation can be found in [1-3].

Test Results

A number of runs were performed with the rotor. A selection of those is summarized in Table 3. The experiments can be divided into three series. The first consisted of runs 1-4 necessary to carry out single-plane balancing in order to be able to cross the first mode at about 18,000 rpm. The second series of runs (5-10) constituted the multiplane, multispeed (MPMS) balancing procedure. Runs 11 and 12 were done to investigate the effect of some parameters.

Run 1 was done to gather the "as is" vibration data for the rigidly balanced rotor. Rigid balancing means low speed balancing of the rotor on a balancing machine, using two supports, the bearing locations, and two planes, the collar and the disk. Measurements taken in run 1 are shown in Fig. 6 as curve A. In this run, the speed could not exceed a maximum of 15,000 rpm because of excessive deflections at the disk. Runs 2 and 3 were done with trial weights at the disk and these constituted a single plane balancing exercise "in situ" necessary to correct apparent inaccuracies in the rigid balancing procedure. Curves B and C in Fig. 6 show the measurements taken in these runs. The deflection and relative phase data from runs 1-3 were used to calculate a single plane correction weight which was tried in run 4 (curve D, Fig. 6). This time the rotor could be run to a speed of about 40,000 rpm (Fig. 7). Above that speed, the collar deflections became excessive.

A multiplane, multispeed (MPMS) balancing exercise was performed in runs 5-8. Runs 5-7 (Table 3) provided the basic

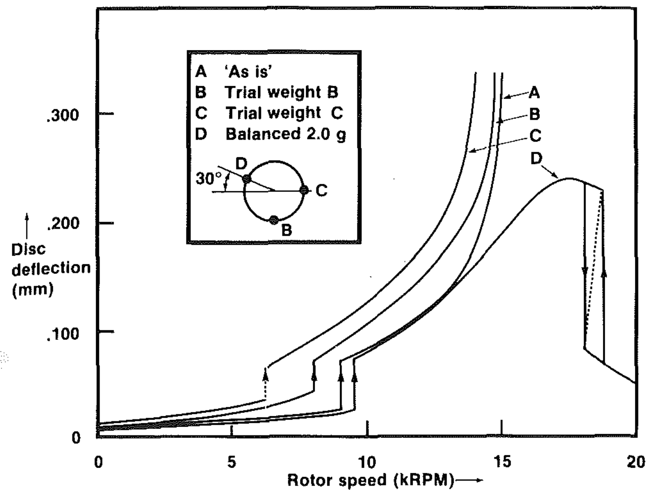

Fig. 6

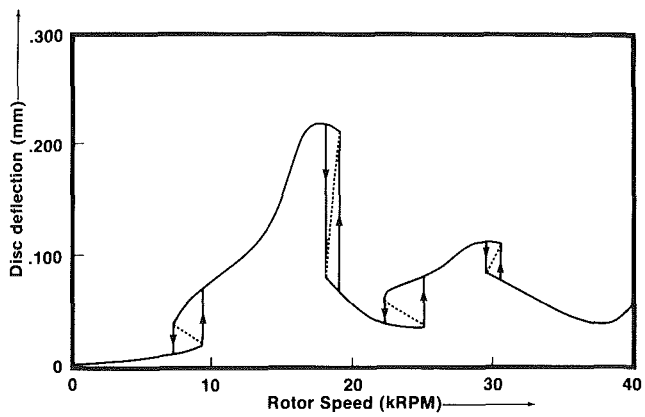

Fig. 7

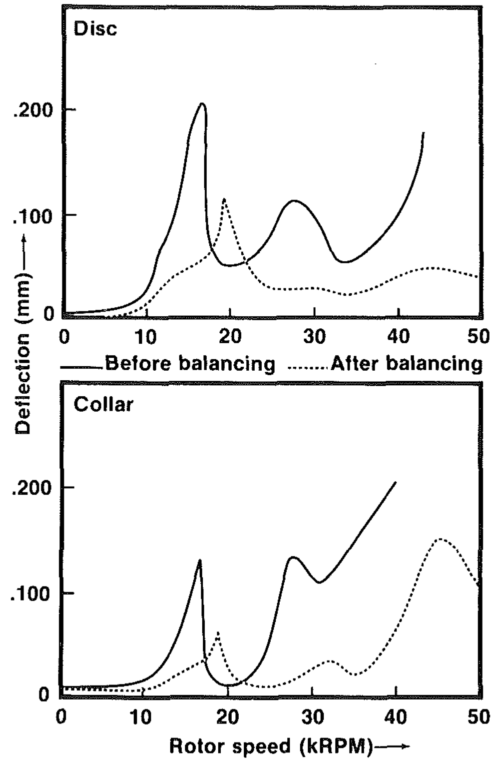

Fig. 8

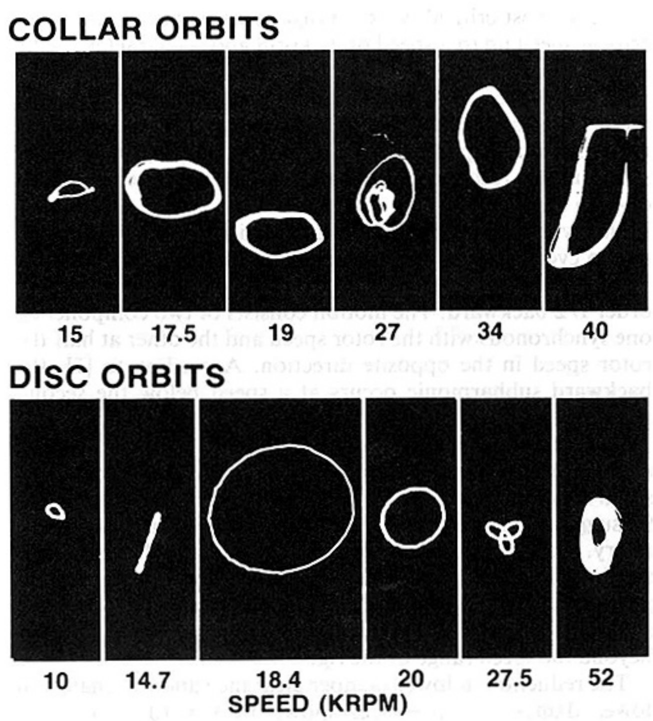

Fig. 9

information needed to calculate the amount and location of the required correction weights at disk and collar. The calculation was done with the least-squares method described in [11]. The procedure is as follows:

(*a*) Measure the "as is" rotor vibration readings.

(*b*) Measure the trial mass data with trial masses in each balance plane in turn, at all speeds of interest.

(*c*) Calculate the response coefficients, i.e., the responses to a unit correction mass in each plane.

(*d*) Use the response coefficients to compute the correction masses that will eliminate the "as is" rotor vibrations.

(*e*) Install the correction masses and take a new set of "as is" readings.

The first cycle of MPMS balancing allowed the rotor to be run to the maximum speed of 55,000 rpm (run 8). The disk and collar responses are shown in Fig 8. Some selected orbits taken from the oscilloscope are presented in Fig 9. Since it was now possible to gather trial mass data at speeds above 40,000 rpm, a new cycle of MPMS balancing was performed in runs 8–10. This reduced the deflection further at high speeds.

Runs 11 and 12 were done to determine the effects of changes in lower damper clearance as well as oil inlet pressure. Some of these results are shown in Figs. 10–12.

Discussion of Test Results

The rotor dynamic behavior can be explained readily by recognizing four speed ranges associated with each critical speed.

1 The subcritical speed range, extending to 12–15 krpm in Fig. 8, is characterized by small deflections. In this range, the dampers of a balanced rotor may not have lifted off. As a result, the support stiffness seen by the rotor may be asymmetric, resulting in small nonsymmetric whirling orbits.

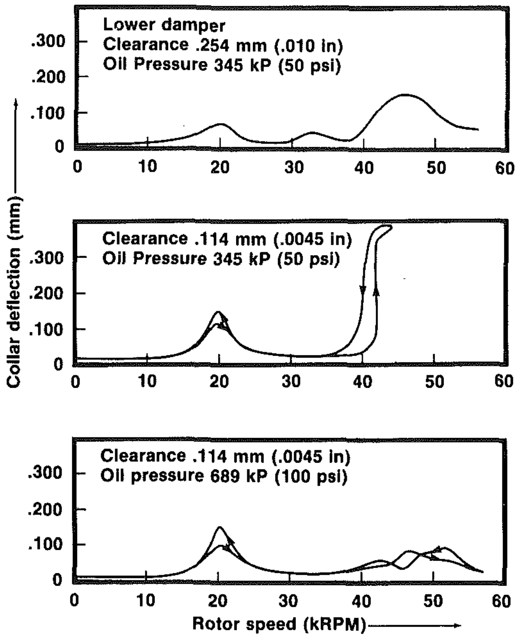

Fig. 10

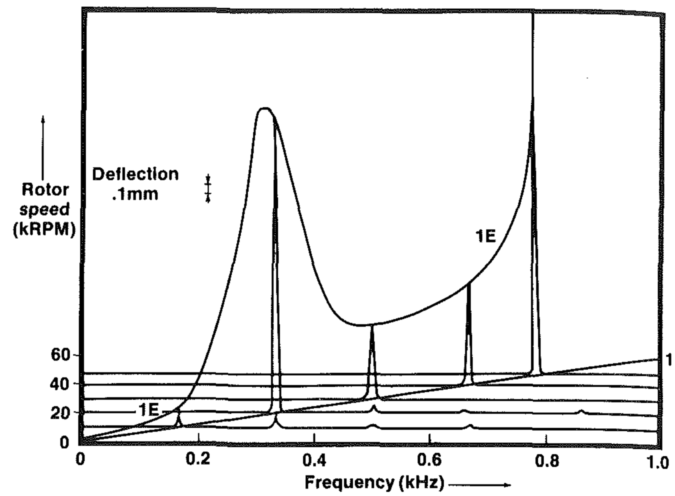

Fig. 11

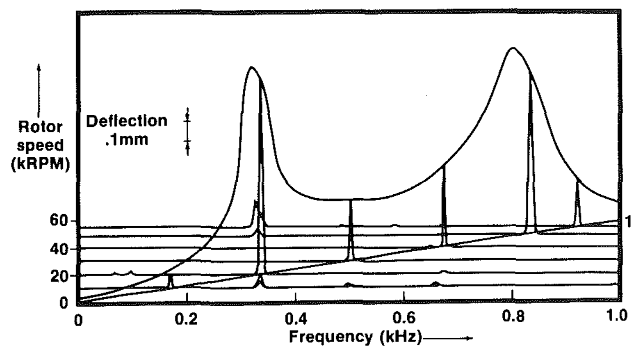

Fig. 12

2 The critical speed range starts around 16 krpm and is characterized by large circular orbits. The first mode critical speed appears to depend on the balance condition, an indication of nonlinearities in the system.

3 The post-critical speed range occurs above the first critical speed up to a speed of 27 kprm and is characterized by small circular orbits.

4 The supercritical speed range associated with the first mode extends from 27-32 krpm. This range is characterized by nonsynchronous precession as shown in Fig. 9. The cause is subharmonic resonance [7] due to nonlinear properties of the oil-film dampers.

Nonsynchronous precession is undesirable because it gives rise to cyclic stresses in the rotor. The particular orbit shown in Fig. 9 for the disk at 27.5 krpm can be indentified to be order 1/2 backward. The motion consists of two components, one synchronous with the rotor speed and the other at half the rotor speed in the opposite direction. According to [7], the backward subharmonic occurs at a speed below the second multiple of the critical speed.

The subcritical speed range associated with the second mode starts at about 35 krpm and is characterized again by small synchronous orbits. The collar orbits in this range (Fig. 9) suggest the presence of bearing support stiffness asymmetry. The second critical speed range starts at 43 krpm with maximum deflections occurring at 47 kprm. The second postcritical speed range shows again smaller orbits. The supercritical speed range associated with the second mode is beyond the speed range of the rig.

The reduction in lower damper clearance and the change in lower damper oil pressure, shown in Fig 10, were tried especially to determine their effect on the stability behavior of the rotor in the supercritical speed range. The well balanced rotor with the larger clearance exhibited a stable response even with the lower oil pressure (Fig. 10(*a*)). With the reduced clearance, the response was found to have an onset speed of instability at 42.5 krpm (Fig. 10(*b*)), similar to what was found for the collar deflection in Fig. 8. Spectrum analysis of the deflection (Fig. 11) indicated that the deflections were synchronous only. This response is not a subharmonic resonance, but may be an example of the bistable operation or jump phenomenon described in [12]. It is proved theoretically in [12] that pressurization of the damper has a very desirable influence on the stability and may eliminate bistable operation. This appears to be confirmed in the results of run 12, as shown in Fig. 10(*c*), where the unstable behavior has been largely eliminated. The spectral components of the collar deflection are shown in Fig. 12. It can be seen that the first mode is excited at high speed to a minor extent in the case of high oil pressure to the lower damper.

Figure 7 illustrates clearly another aspect of nonlinear behavior, i.e., that responses differ when the rotor speeds increase or decrease.

Further Work

Experiments are in process on rotors similar to the one discussed in this paper, but with different critical speeds and strain energy distributions. The control of nonsynchronous deflections by means of changes in the geometry and oil pressure of the oil-film dampers requires more attention. It is necessary to develop practical methods for the prediction of the occurrence and the severity of instabilities in actual rotor designs.

Conclusions

The experimental investigation has been successful in showing the effectiveness of the MPMS balancing method for modal balancing of a rotor with two critical speeds with high strain energy in the running range. It was found necessary to perform several cycles of MPMS balancing before the rotor could be run over the whole speed range.

Nonlinearity in the rotor-bearing-damper system, due to the nonlinear damper properties, has given rise to subharmonic resonance and to bistable operation. As far as the intensity of the response is concerned, bistable operation seems more important than subharmonic resonance. However, the latter produces undesirable fatigue stresses in the rotor.

Nonlinearity in the system also caused changes in the rotor responses over the speed range as compared to the linear system. The actual critical speeds appeared to depend to a minor degree on the balance condition of the rotor. The responses as a function of speed were different for a speed increase as compared to a speed decrease.

Dampers with a small clearance may not be able to prevent unstable vibration. On the other hand, an increase in oil inlet pressure to the dampers may reduce unstable vibration. The threshold of bistable operation depends on the amount of external damping and this depends, in the non-linear system, on the balance condition of the rotor.

Acknowledgment

The experimental work was made possible by the contributions of many individuals at P&WAC. The present program is supported in part by the Department of Industry, Trade and Commerce of Canada.

References

1 Botman, M., "Experiments on Oil-Film Dampers for Turbomachinery," ASME *Journal of Engineering for Power,* Vol. 98, No. 3, July 1976, pp. 393–400.

2 Botman, M., and Sharma, R. K., "Experiments of the Transient Response of Oil-Film Dampers," ASME *Journal of Engineering for Power,* Vol. 100, No. 1, Jan. 1978, pp. 30–35.

3 Sharma, R. K., and Botman M., "An Experimental Study of the Steady-state Response of Oil-film Dampers," ASME Journal of Mechanical Design, Vol. 100, Apr. 1978, pp. 216–221.

4 Mohan, S., and Hahn, E. J., "Design of Squeeze Film Damper Supports for Rigid Rotors," ASME *Journal of Engineering for Industry,* Aug. 1974, pp. 976–982.

5 Taylor, D.L., and Kumar, B. R. K., "Non-linear Response of Short Squeeze Film Dampers," ASME Paper No. 79-Lub-24.

6 Marmol, R. A., and Vance, J. M., "Squeeze Film Damper Characteristics for Gas Turbine Engines," ASME Paper No. 77-DET-18.

7 Tondl, A., "Notes on the Indentification of Subharmonic Resonances of Rotors," *Journal of Sound and Vibration,* Vol. 31, No. 1, 1973, pp. 119–127.

8 Gunter, E. J., Jr., and Trumpler, P. R., "The Influence of Internal Friction on the Stability of High Speed Rotors with Anisotropic Supports," ASME Paper No. 69-Vibr.-2.

9 Marmol, R. A., Smalley, A. J., and Tecza, J. A., "Spline Coupling Induced Non-synchronous Rotor Vibrations," ASME Paper No. 79-DET-60

10 Vance, J. M., and Lee, J., "Stability of High Speed Rotors with Internal Friction," ASME Paper No. 73-DET-127.

11 Goodman, T. P., "A Least Squares Method for Computing Balance Correction," ASME Paper No. 63-WA-295.

12 Rabinovitz, M. D., and Hahn, E. J., "Stability of Squeeze Film-Damper Supported Flexible Rotors," ASME Paper No. 77-GT-51.