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THE UNBALANCE RESPONSE OF A FLEXIBLE ROTOR MOUNTED ON UNCENTRALIZED SQUEEZE-FILM DAMPER BEARINGS

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ABSTRACT

This paper deals with the steady-state response of a single disc flexible rotor supported on two uncentralized squeeze-film damper bearings and subjected to varying degrees of unbalance. The measured responses+ on an experimental rotor operating to speeds exceeding its first bending critical show that the uncentralized squeeze-film damper bearing has a very desirable effect under the abusive conditions of large unbalance. The unbalance level was increased by a factor of three in some tests and yet the corresponding vibrational amplitude of the disc, at the first bending critical speed, was increased by less than 30 %. The results predicted from a computer simulation study, where the equations of motion are integrated forward in time until steady-state orbits are reached and automatically plotted, show good qualitative agreement with the experimental results.

INTRODUCTION

The life, reliability and general mechanical integrity of high speed rotor systems can be seriously reduced by the presence of mechanical vibration. One method for reducing the effect of vibration is the inclusion of squeeze-film damper bearing as proposed by Cooper [1]. A squeeze-film damper bearing can be simply an annular space surrounding the outer race of a rolling contact bearing. The fluid within the clearance annulus is the oil employed in lubricating the rolling contact bearing. The outer race of the rolling contact bearing is not allowed to rotate within the clearance circle but it can move with fixed coordinates (orbit) within this space. It is this movement which produces the squeeze-film from which the device gets its name.

Two main types of squeeze-film damper bearing exist. One type allows the bearing to settle freely to the bottom of the clearance circle until unbalance causes it to lift-off. The other type has the bearing outer race supported by a set of cantilever springs (known collectively as a squirrel-cage) so that the bearing is held centrally within the clearance space. The two design variants are shown schematically in Fig. 1. The first of these two types, which is often refered to as an uncentralized squeeze-film

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damper bearing, is the subject of the present investigation.

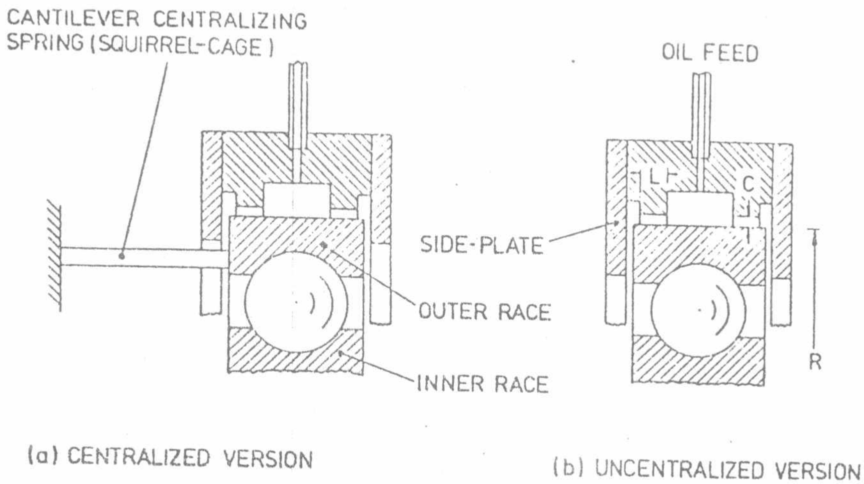


Fig.1. Centralized and Uncentralized Squeeze-film Damper Bearing.

Previous work of the authors [2] showed that the uncentralized squeeze-film damper bearing is an effective means of reducing the rotor vibrational amplitudes and of attenuating the vibratory forces transmitted to the structure surrounding an unbalanced flexible rotor. Also, it was shown in [3] that the effectiveness of the device is a function of the design variables such as the radial clearance, land width, fluid viscosity ... etc. However, the work reported in [2-3] is limited to the case of low unbalance level.

In view of the fact that the unbalance level, initially set to very low limit by the manufacturer, may increase in service due to thremal distortion producing shaft bow, corrosion of the compressor and turbine blades, wear of some components and from many other possible reasons; it was felt necessary to extend the study reported in [2-3] to higher unbalance levels. The purpose of this paper is to present the response of a single-disc flexible rotor mounted on uncentralized squeeze-film damper bearings to varying degrees of unbalance.

FLEXIBLE ROTOR MODEL

Fig.2. illustrates the flexible rotor model which is used in the present investigation. It can be clearly seen that the rotor is an extended Jeffcott model where a single disc is positioned mid-way between the bearings.

The following assumptions are made for the rotor model :

1- The rotor is symmetric with part of its mass lumped mid-way between the bearings and the remainder is evenly divided and lumped at the bearing stations.
2- No significant exciting forces are introduced by the rolling contact bearings.
3- There is no gyroscopic effect.
4- Unbalance can be defined in one plane at the disc location.

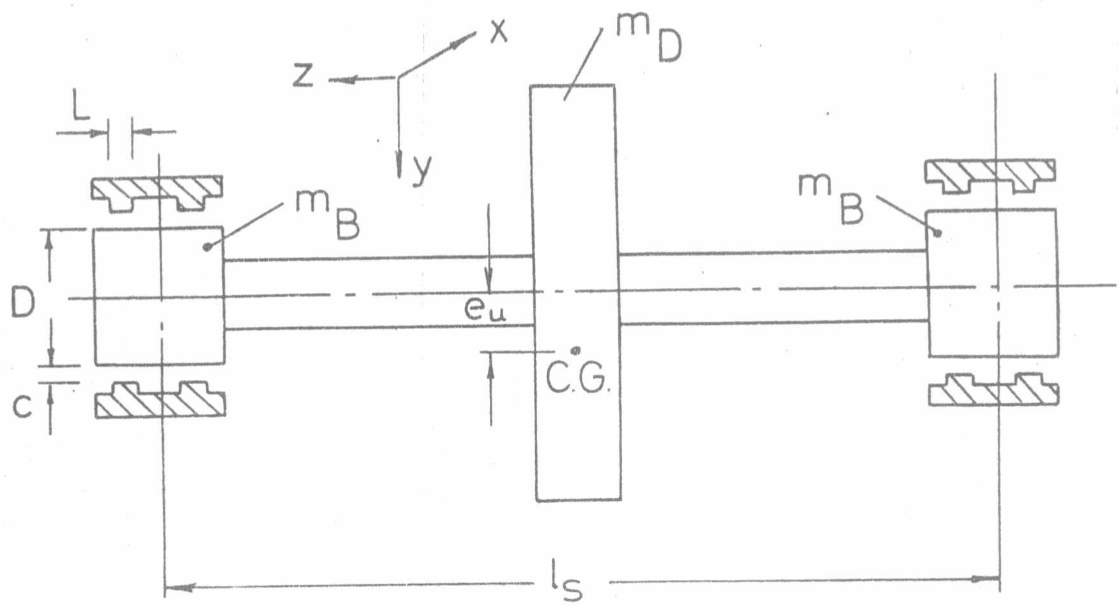


Fig.2. Rotor Model

Under the previous assumptions, the system equations of motion were derived in [4] ; it is shown that the system response depends on the following nondimensional parameters:

Bearing parameter	$B = \mu R\, L^3/m_B\, \omega_c c^3$
Gravity parameter	$\bar{W} = g/c\ \omega_c^2$
Unbalance parameter	$U = e_u\ /c$
Frequency ratio	$\Omega = \omega/\omega_c$
Damping ratio	$\zeta = C_d/C_c$
Mass ratio	$\alpha = m_B/m_D$

where,

C_c	critical damping coefficient
C_d	damping coefficient
g	gravitational acceleration
μ	lubricant absolute viscosity
ω	angular speed of rotation
ω_c	pin-pin critical speed

Other terms are shown in Fig. 2. The subscripts B and D stand for the bearing and disc, respectively.

EXPERIMENT

A schematic representation of the apparatus used in the experimental investigation is shown in Fig. 3.

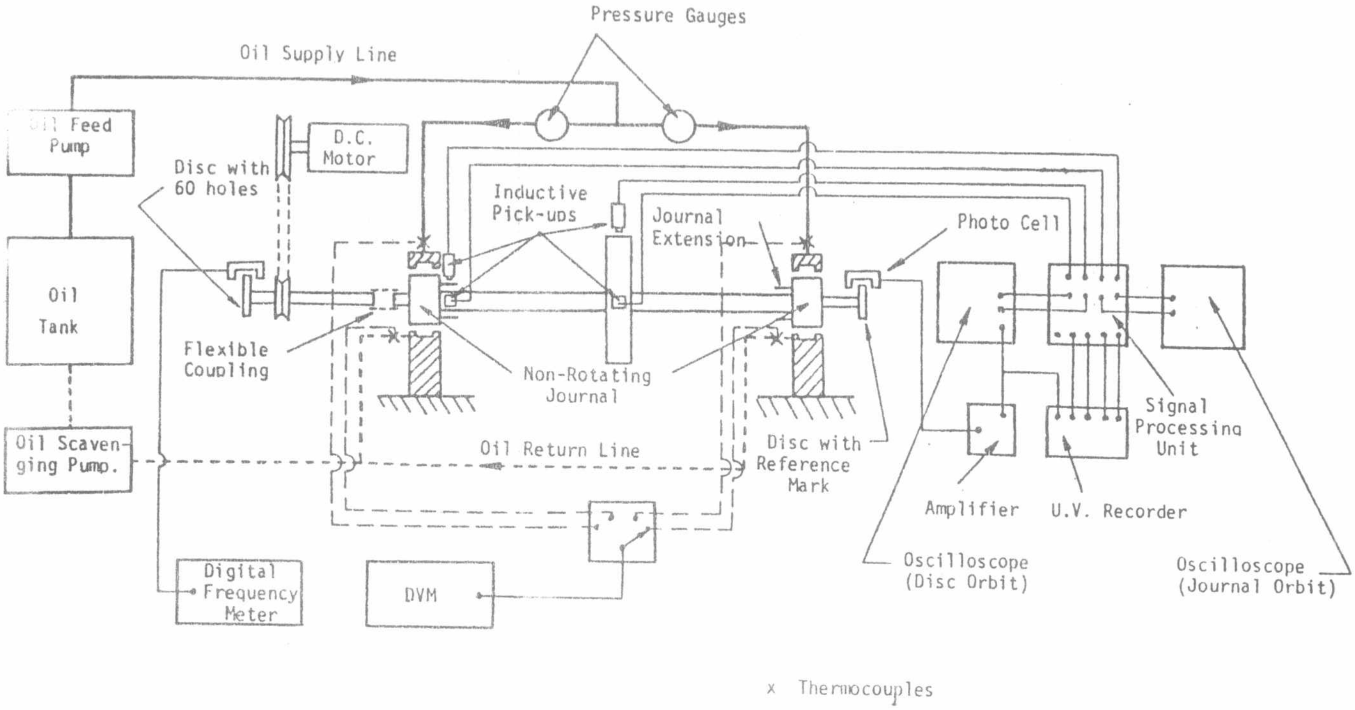


Fig.3. Schematic of the Experimental Equipment

The experimental rotor consists of an integral steel disc and shaft. The disc is 0.0508 m thick and 0.127 m in diameter. It is located at the mid-span of a uniform cylindrical steel shaft of 0.0381 m diameter, which is supported in two rolling contact bearings having a distance of 0.6096 m between centres.

The outer race of each of the rolling contact bearing is located with an interference fit in a cylindrical steel ring which serves as the squeeze-film bearing journal. The oil film is located between the journal and an annular ring which has a circumferential groove to which the oil is supplied via an inlet hole. The outer ring is located with a force fit in a massive support pedestal. The two pedestals are rigidly connected to a large steel plate which is fixed on top of a massive concrete block.

Four of the system nondimensional parameters can be varied on the test rig, namely : the bearing parameter B, gravity parameter $\overline{W}$, unbalance ratio U and the frequency ratio Ω.

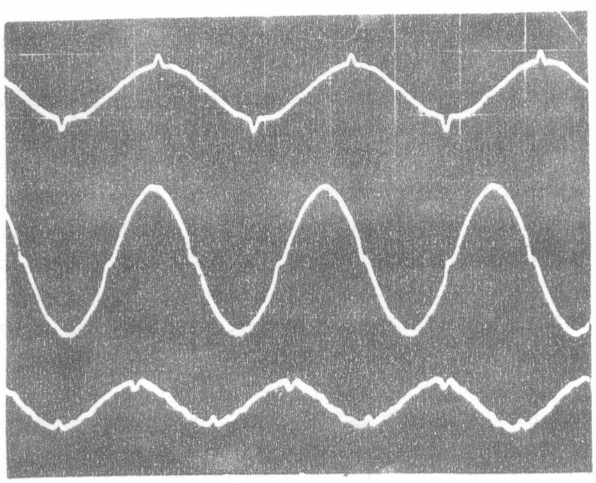
Variation of the gravity and bearing parameters is achieved by using a number of interchangeable squeeze-film journals and outer rings with different geometrical dimensions. For running the rotor on rigid supports, the damper journal and the outer ring are replaced by one solid piece which can be located with force fit in the bearing housing.

A. D. C. motor controlled by a closed loop control system and coupled to a tacho-generator, which is used as a feed back signal generator, is used to drive the rotor to a maximum speed of 17000 rpm through a step-up belt-pulley-shaft drive system. This maximum speed exceeds the first pin-pin critical speed of the rotor which equals 7248 rpm. Two modes of operation are possible : either the rotor speed is automatically increased, at a constant rate in all tests, until it passes through the critical speed and then decreases to the stop position, or the speed is manually adjusted at any required level of the frequency ratio .

The motion of the disc is detected by means of two perpendicular pairs of non-contacting inductive transducers. Each pair consists of two opposite transducers working in a"push-pull" arrangement. A similar arrangement is employed for detecting the journal motion. The motion signals, for both of the disc and the journal, are displayed on X-Y oscilloscopes in order that the complete orbit can be photographed. At the same time, the signals are recorded on a U-V oscillograph.

For accurate determination of the critical speed, a thin foil is glued to the disc outer surface just above the position of the unbalance mass. When this thin foil passes under any of the inductive transducers, a corresponding mark appears on the motion trace indicating the position of the unbalance force. Thus, it is possible to detect the variation in the phase angle between the disc displacement and the unbalance force when the rotor passes through a critical speed as may be seen in Fig. 4.

Vertical Motion of the Disc:



before the critical speed

at the Critical speed

after the critical speed

Fig.4. Phase Angle Variation Through the Critical Speed.

Out-of-balance masses can be added to the integral shaft/disc assembly by means of two rings which are firmly attached to either side of the disc. The added masses can be attached to the rings through the medium of equi-spaced tapped holes. Although the rotor was machine balanced, further fine balancing was carried out by trial and error until the maximum peak-to-peak amplitude of the disc in the first critical speed was only about 0.05 mm. Further reduction in the residual unbalance was not possible. However, it was estimated that the residual out-of-balance was less than 4 % of any of the added unbalance values employed during the experimental tests.

EXPERIMENTAL RESULTS AND ANALYSIS

Initial tests were carried out with the rotor bearings rigidly supported. The measured peak-to-peak amplitude of the disc, at the first bending critical, was 0.289 mm for an unbalance value of 1.016 gm cm for this rigid support test. When the unbalance magnitude was doubled to reach a value of 2.032 gm cm, the disc vibrational amplitude was considerably increased to a value of 0.502 mm. The responses of the rigidly supported rotor, always gave sharp resonance peak at the critical speed indicating that the external damping of the system was very low. The damping ratio of the rotor was estimated to be in the order of 0.005. No further rigid support tests were carried out at higher unbalance levels because it was thought inadvisable to risk permanent deformation of the shaft. Experimental tests were then carried out with squeeze-film fitted to the bearing. The two small unbalance levels, previously applied in the rigid support tests, were again applied. No appreciable improvement in the vibrational amplitudes of the disc at the critical speed was observed as such an unbalance level is too small to generate a sufficient dynamic force to lift the journal. Hence, under these conditions of low unbalance levels the journal is lying at the bottom of the clearance circle and the squeeze-film bearing is not operating.

Increasing the applied unbalance magnitude, for example to a value of 17.0 gm cm which is equivalent to U= 0.1 as in case (a) Fig.5., the character of the rotor response was completely changed. The rotor mid-span response became flat and smooth and no longer showed the sharp resonance peak at the critical speed. The journal began to lift-off producing measurable orbital motion at a value of Ω about 0.6, while the built-up of the mid-span excursions was limited. It should be noted that the measured disc amplitude at the first bending critical with an unbalance of 2.03 gm cm, acting on the system with rigidly mounted bearings, was about twice as great as that experienced with the same system when squeeze-film dampers are fitted and with unbalance magnitude more than eight times greater as may be seen in Fig. 6.

With further increase of the unbalance level to U=0.2, no proportional increase of the disc vibrational amplitudes was observed while at the same time the journal motion increased significantly as shown in case (b) Fig.5.

Fig.7. shows the journal and disc orbits, for unbalance of U=0.1,0.2 and 0.3, observed at the same rotor speed of 7172 rpm which is the measured critical speed for this test. It is worth noting that although the unbalance level is increased three times, form U=0.1 to 0.3, the maximum amplitude of the disc motion is increased by less than 30 %. Note also the significant increase of the size of the journal orbit with the increase of the unbalance level.

The variation of the disc amplitudes with the unbalance level, shown in Fig.6. for different values of the bearing parameter B, is an impressive example of the possible reduction in the sensitivity of the rotor vibrational amplitudes with respect to unbalance when uncentralized squeeze-film dampers are used.

Case (a) U=0.1

Case (b) U=0.2

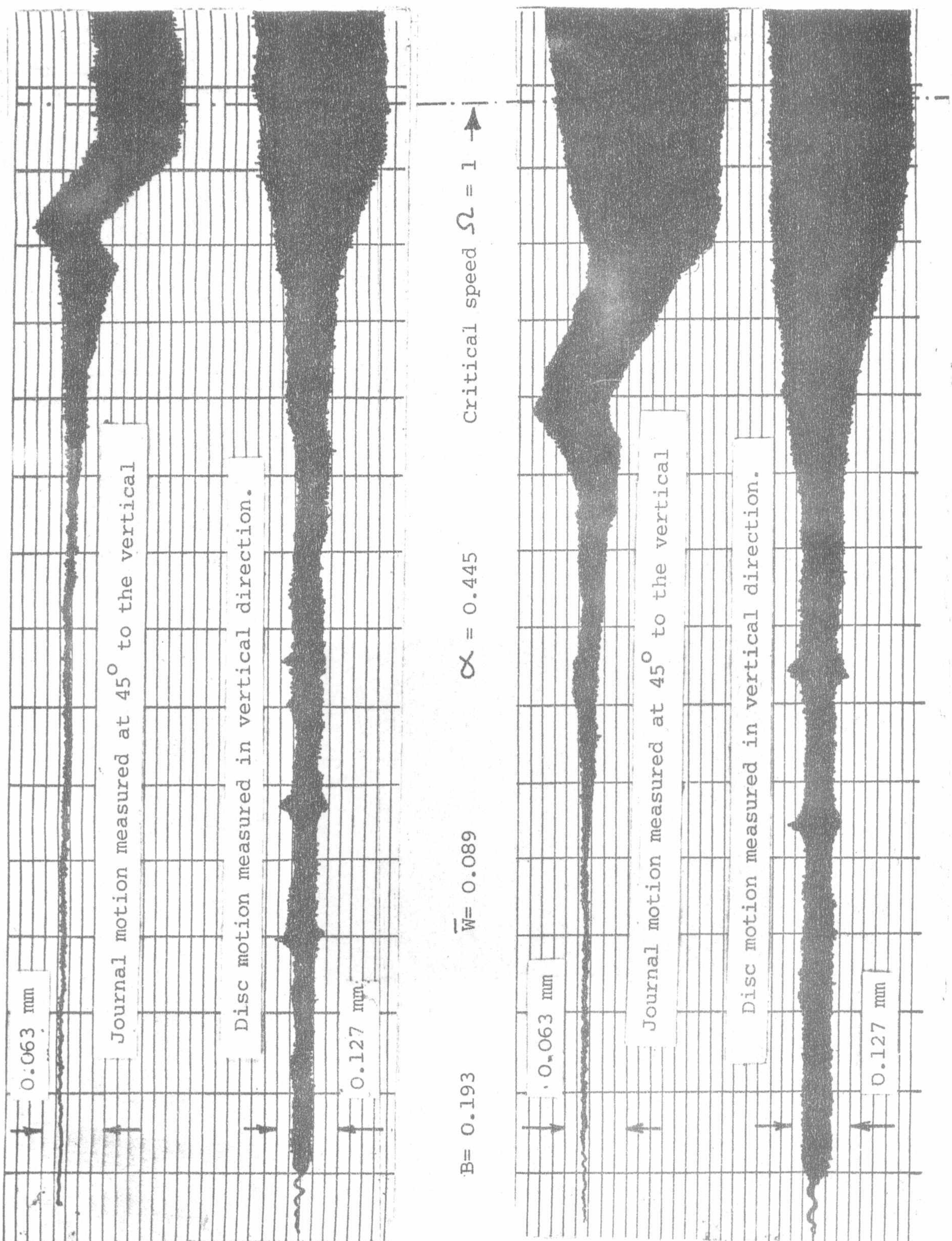


Fig.5. Oscillograph Traces of the Rotor Responses at Two Unbalance Levels.

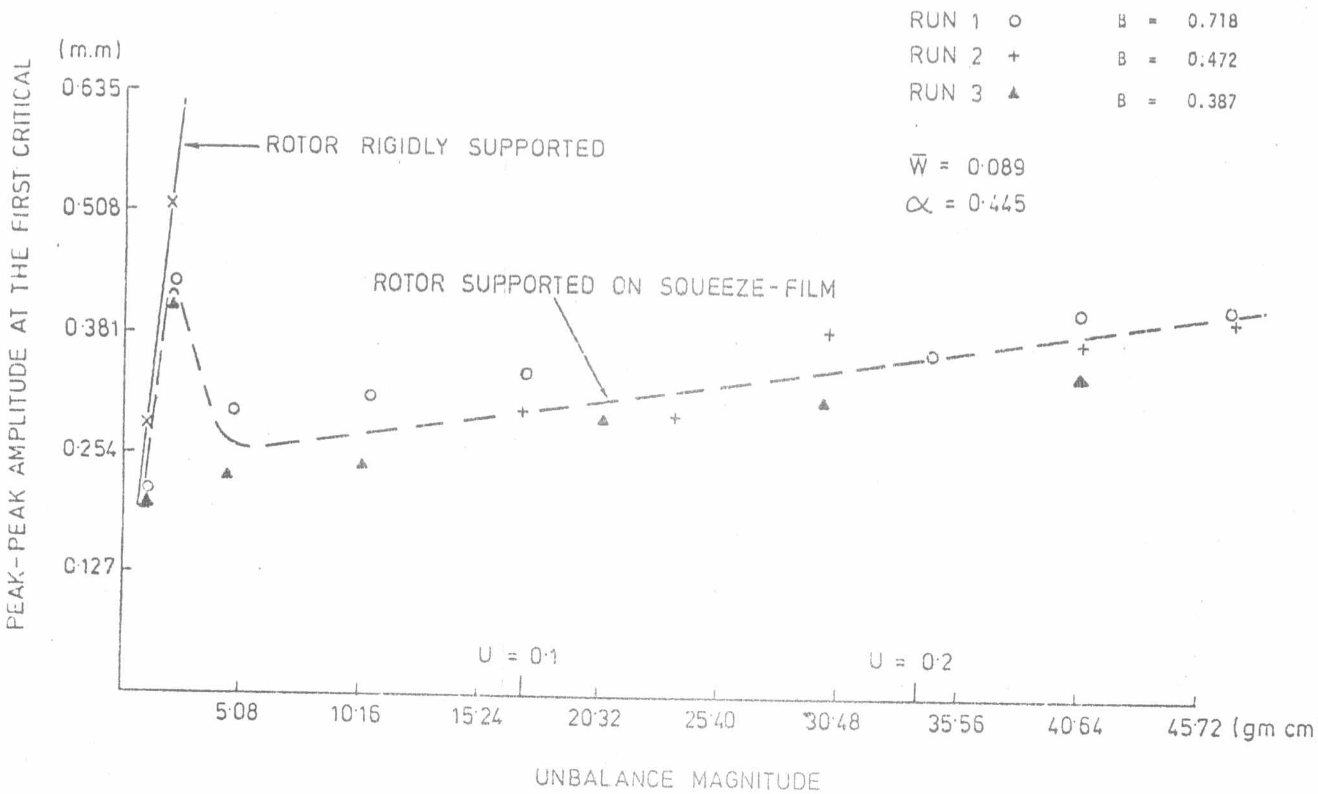


Fig.6. Variation of the Disc Amplitude With Unbalance

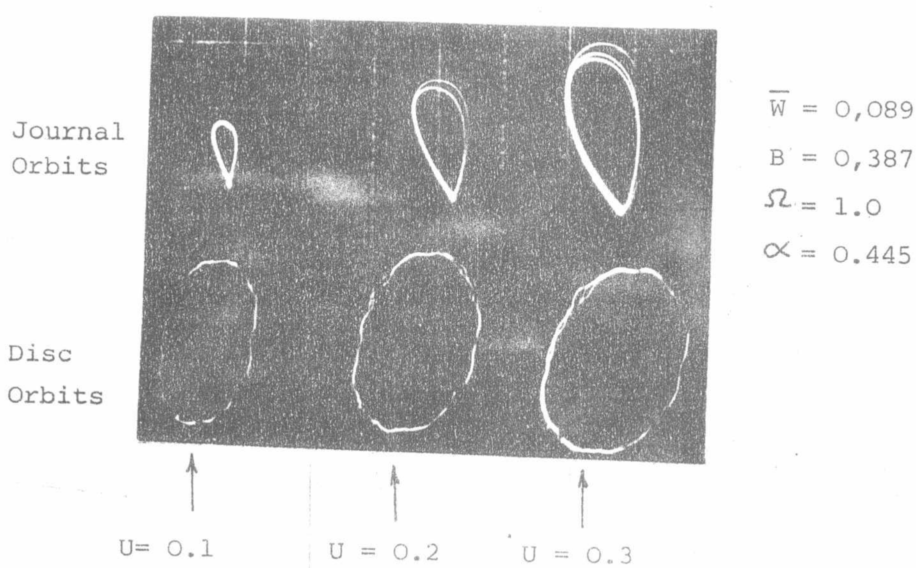


Fig.7. Disc and Journal Orbits for Varying Degrees of Unbalance

COMPUTER SIMULATION

The equations of motion for the rotor-bearing model, derived in [4] , are a set of four simultaneous nonlinear differential equations. A computer program was written for their numerical solution based on a time marching technique. The equations are integrated forward in time until steady-state orbits are reached. The hydrodynamic forces can be calculated using different fluid-film models [5] .

To reduce the considerable effort required for analysing the motion orbit data, the steady-state orbits of the journal and disc centers are plotted automatically with the aid of a Calcomp graph plotter. A typical computer output is shown in Fig.8. for a set of system parameters similar to those of the experimental test rig. Fig.8. depicts that the predicted maximum amplitude of the disc at an unbalance level U=0.2 is only about 1.1 times greater than the maximum amplitude at U=0.1. This result agrees well with the experimental observations.

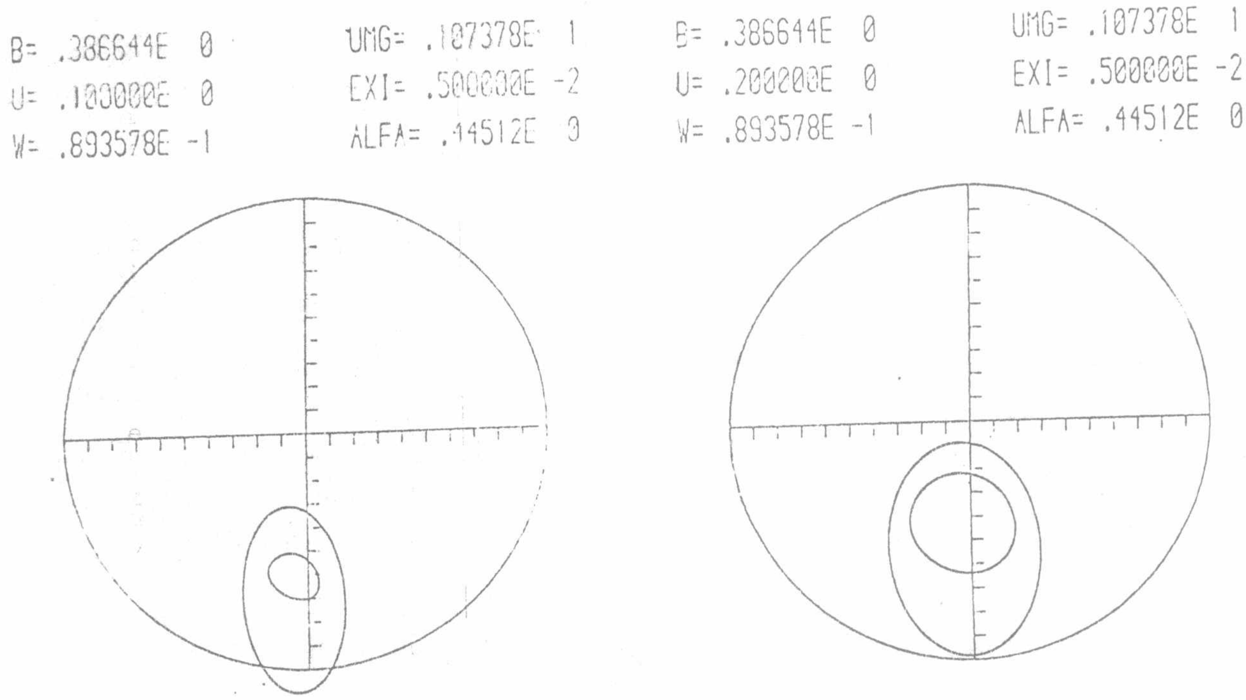


Fig.8. Computer Simulation Plots of the Disc and Journal Orbits.

In general, the predicted rotor responses show good qualitative agreement with those obtained experimentally, both in the overall shape of the response curve and in the effect of the unbalance parameter. However, the absolute values of the predicted vibrational amplitudes of the disc are smaller than those measured experimentally while those of the journal are greater. This indicates that there is a need for a more realistic film model for accurate evaluation of the hydrodynamic forces and consequently of the rotor vibrational amplitudes .

DISCUSSION AND CONCLUSIONS

From the work reported in this paper, it would appear that the vibratory effects caused by unbalance in a flexible rotor- bearing system can be significantly reduced by supporting the bearings on uncentralized squeeze-film dampers.

It is found that for unbalance level U=0.1, which is roughly equivalent to the level of unbalance normally encountered in gas turbine engines and similar machines, the rotor vibrational amplitudes are considerably smaller than those of the same rotor on rigidly mounted bearings and with smaller unbalance. With increasing the unbalance level, as may be expected during operational life, the vibrational amplitudes increase but with a much smaller rate than what would be expected for a rigidly mounted rotor.

The level of unbalance represented by U=0.3 is quite large, roughly equivalent to the case where a turbomachine sheds a blade, and yet the disc excursion is only about 30 % greater than that experienced with normal unbalance.

All of the results described in this paper, both theoretical and experimental, have been for the steady-state conditions and do not represent the transient state when a sudden unbalance is applied to the system (as in the case of a blade-loss). Work in this direction is required.

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