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MEASUREMENTS OF SQUEEZE FILM BEARING FORCES TO DEMONSTRATE THE EFFECT OF FLUID INERTIA

John A. Tichy
Department of Mechanical Engineering
Rensselaer Polytechnic Institute
Troy, New York 12181

ABSTRACT

Squeeze film dampers are commonly applied to high speed rotating machinery, such as aircraft engines, to reduce vibration problems. The theory of hydrodynamic lubrication has been used for the design and modeling of dampers in rotor dynamic systems despite typical modified Reynolds numbers in applications between ten and fifty. Lubrication theory is strictly valid for Reynolds numbers much less than one, which means that fluid viscous forces are much greater than inertia forces. Theoretical papers which account for fluid inertia in squeeze films have predicted large discrepancies from lubrication theory, but these results have not found wide acceptance by workers in the gas turbine industry. Recently, experimental results on the behavior of rotor dynamic systems have been reported which strongly support the existence of large fluid inertia forces. In the present paper direct measurements of damper forces are presented for the first time. Reynolds numbers up to ten are obtained at eccentricity ratios 0.2 and 0.5. Lubrication theory underpredicts the measured forces by up to a factor of two (100% error). Qualitative agreement is found with predictions of earlier improved theories which include fluid inertia forces.

INTRODUCTION

The trend to increasing power-to-weight ratios in aircraft engines results in more flexible rotors and structures and higher machinery speeds. Rolling element bearings are almost exclusively used in these applications due to the catastrophic mode of failure in fluid film journal bearings if oil supply is interrupted. However, rolling element bearings provide no damping to the rotor dynamic system, increasing susceptibility to critical speed and instability problems. Hence it is common practice today to fit most such units with squeeze film bearings to introduce system damping and promote stable running of shafts and rotors.

The hydrodynamic squeeze film damper (SFD) is essentially a bearing within a bearing. The inner bearing is usually a rolling element device of which the retainer forms the nonrotating journal of the outer bearing (see Figure 1). A squeezing action between the two surfaces produces hydrodynamic forces which ultimately act on the rotor.

The Reynolds hydrodynamic lubrication theory has been used as the basis for design and analysis of fluid film bearings (such as squeeze film dampers) for many years. There is no question that the theory has been adequate for a vast majority of practical applications. An underlying assumption is that fluid flow in the bearing is very "slow" and the resulting fluid mechanical behavior is quasi-steady. The measure of slowness is the so-called modified or reduced Reynolds number,

$$Re = \rho \frac{U_o h_o}{\mu} \left(\frac{h_o}{\ell_o}\right) = \frac{\text{fluid inertia force}}{\text{viscous force}} , \qquad (1)$$

where ρ and μ are fluid density and viscosity; and U_o, h_o and ℓ_o are reference values of sliding speed, film thickness, and bearing length, respectively. If $Re \to 0$ viscous forces are much larger than inertia forces and lubrication theory applies.

The range of practical application for squeeze film dampers is approximately $10 < Re < 50$. For SFDs the appropriate Reynolds number is

$$Re = \frac{\rho \omega c^2}{\mu} . \qquad (2)$$

Consider the following typical conditions:

R (damper radius) = 10. cm,
c (radial clearance) = 0.03 cm,
ω (circular frequency) = 1500 rad/s (15,000 cpm)
L (damper length) = 2.5 cm
μ (viscosity) = 5 dyne-s/cm^2
ρ (density) = 6.8 gr/cm^3,

then,
Re (Reynolds number) $\cong$ 20.

This value is so high for SFDs because (1) the speed is high and (2) the viscosity is low. The viscosity is low because the fluid is primarily selected as a coolant for the rolling element bearings, and is applied to the damper at elevated temperatures ($\sim 150^\circ$C). It is not reasonable to expect lubrication theory to be accurate in this range of Reynolds number.

The author has published a series of theoretical papers pertaining to the effect of fluid inertia in squeeze film flows [1-3]. In these articles, "exact" analytical solutions to the governing Navier-Stokes equations can be obtained provided the vibration amplitude is small with no further assumptions required. Reinhardt and Lund [4] have developed a small Reynolds number perturbation solution, which could also be regarded as exact for Re $\ll$ 1. Available approximate solutions are due to Szeri [5] and Nelson [6] who use averaging techniques to approximate the fluid inertia terms of the governing equations.

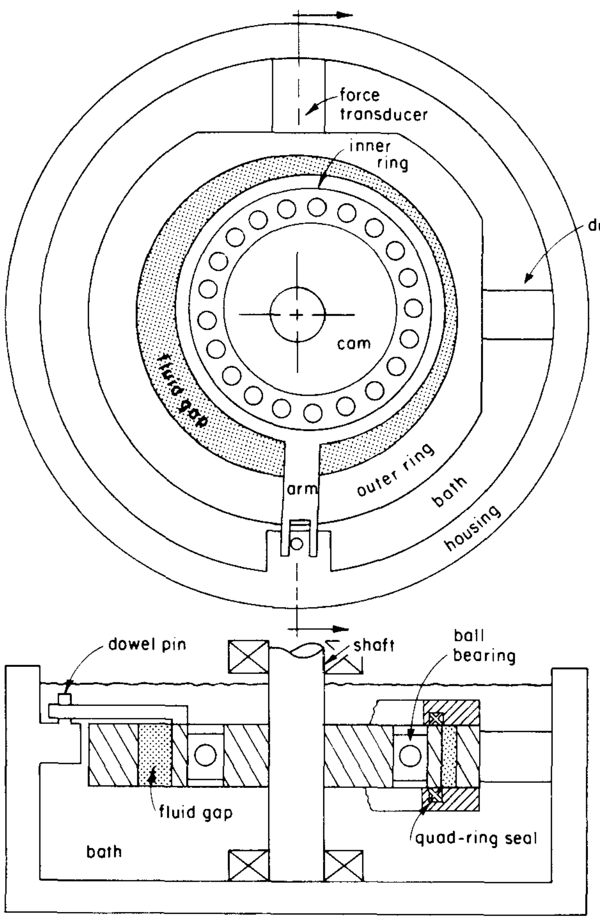


Figure 1 Schematic of Experimental Apparatus

All of the authors listed would agree that for modified Reynolds numbers of ten, a sort of lower bound of practical applications, fluid inertia forces are comparable in magnitude to viscous forces. Nevertheless, until very recently workers in rotor dynamics have been skeptical that lubrication theory could be so greatly in error. Perhaps a reason for such slow acceptance is that in steadily operated journal and thrust bearings the inertia effect in most cases is small. A comparison of inertia in SFDs and journal bearings is made in Ref. [7].

Recent articles by Tecza et al. [8] and Bently and Muszynska [9] have reported experimental results which strongly support the existence of large inertia forces. However, in both cases inferences must be drawn from the behavior of a rotor-bearing system rather than direct measurement of damper forces. Sharma and Botman [10] describe experimental measurements of squeeze film forces but the emphasis is on the effect of film cavitation on rotor response. In the present paper, measurements of damper forces are reported for the first time in a way that the effect of fluid inertia can be directly evaluated.

In Ref. [8] a full-scale damper rig was constructed and fluid inertia was studied as one of several interrelated problems. In the Bently study [9], the main purpose was to develop a rotor dynamics test procedure. By contrast, the sole purpose of the present work is to isolate and study the fluid inertia effect in SFDs. The experiment is designed to vary the bearing parameters to attain sufficiently high Reynolds numbers but at relatively low speeds and forces such that various complications (shaft deformation, cavitation, etc.) do not set in. The range of the experimental variables is given in Table 1. The fluid films are very thick ($\sim$ 0.1 cm) so that they can be easily measured, and so that the experiment is relatively unaffected by small machining inaccuracies and dynamic deflections. The fluid film forces are so low (< 200 N) and the speeds so slow (< 2000 rpm) relative to the natural frequencies of the mechanical components ($\sim$ 1000 Hz) that dynamic deflections are negligible and quasi-steady.

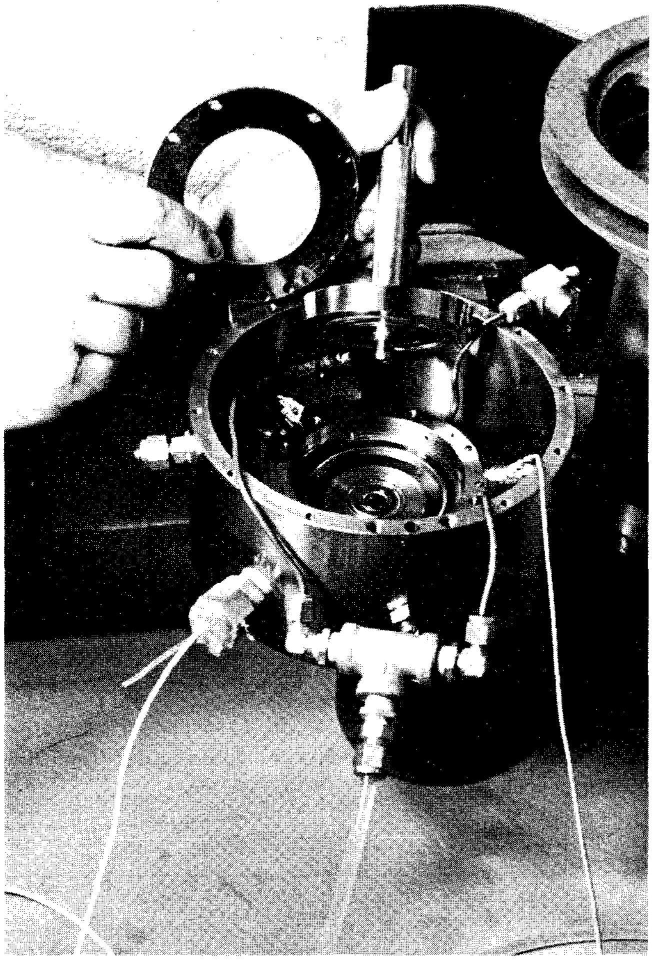
EXPERIMENTAL APPARATUS

A schematic of the apparatus is shown in Figure 1 and photographs in Figure 2. The experimental device imposes a fixed circular centered orbit on the nonrotating inner ring, through an off-center cam. Squeezing of the fluid within the thin gap creates hydrodynamic forces on the outer ring.

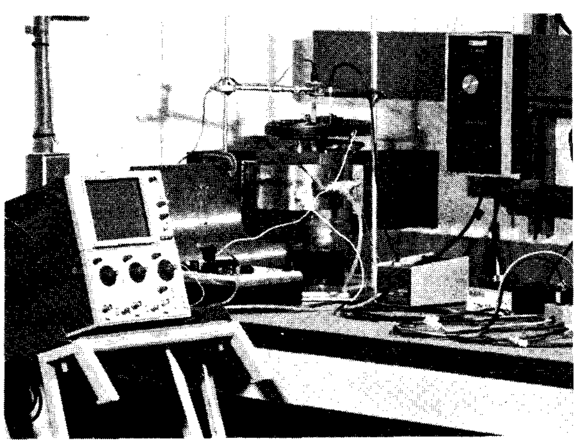
The outer ring is supported by a piezoelectric force transducer fastened to the wall of the cylindrical steel housing. For transverse stiffness of the ring, a dummy transducer also supports the ring 90° away, as shown. The force transducer signal should ideally represent the fluid film force component in the direction of the transducer. The dummy transducer has too little stiffness in bending to contribute significantly to forces acting in the direction of the transducer.

The transducer is a Kistler Model 9300 Quartz Force Link used with a 503D Sunstrand Charge Amplifier. The maximum sensitivity of the system is about 0.4 N/V and the natural frequency is 75 kHz. Displacements were measured with an MTI Fotonic Sensor, a noncontacting fiber optic probe, having a sensitivity of about 10 V/cm and a 50 kHz natural frequency. The probe works by reflecting a beam of light sent through some of the optical bundles off a surface whose displacement is to be measured. Reflected light returns to the sensor by other bundles. The unit can be adjusted to compensate for the reflectivity of various surfaces. The change in the intensity of the reflected light signal is proportional to the change in displacement. The device was statically calibrated *in situ*.

During various tests deflections of the outer ring apparatus were measured and found to be negligible ($< 2.5\times10^{-4}$ cm) relative to the imposed eccentric motion of the cam ($> 2\times10^{-2}$ cm). The inner ring motion was also measured during certain tests by inserting the probe through the housing into the damper film region. No deflection (other than that due to the cam) could be detected in the range of speeds measured. We conclude, therefore, that both the outer ring and the inner ring cam assembly can be treated as rigid bodies.



Damper Rig Components



Test Apparatus

Figure 2 Photographs of Experimental Apparatus

The inner circular off-center cam is driven by a variable speed d.c. motor. The cam outer diameter is pressed into a rolling element bearing bore. The bearing outer race is pressed into the inner ring. The inner ring and bearing outer race assembly is prevented from rotating by the dowel pin shown. The entire assembly fits in a pool or bath of lubricant fluid with temperature control. The bath fluid is identical to the test fluid so contamination of the sample is not a problem. The fluid temperatures are measured during each test and the viscosity determined from prior capillary data. The shear rates are sufficiently low such that viscous heating problems do not occur. Since the apparatus is massive, the system is kept at the operating temperature for several hours before actual testing. Three cams and two belt-and-pulley drive combinations allow a wide range of kinematic variables, see Table 1. Only shaft speeds to 2000 rpm have been tested so far.

TABLE 1

RANGE OF EXPERIMENTAL VARIABLES

Apparatus
d.c. motor speed: 100 - 2000 rpm
Belt and pulley drive combinations: 1:2, 2:1
Resultant shaft speed (damper squeezing rate): $N = 50 - 2000$ rpm, $\omega \cong 5 - 200$ rad/s
Outer ring radius: $R_o = 4.166$ cm
Inner ring radius: $R_1 = 4.064$ cm
Eccentricities: $e = .020, .050, .080$ cm
Eccentricity ratios: $\varepsilon = .2, .5, .8$
Radial clearances: $c = 0.102$ cm
Bearing length: $L = 1.21$ cm
Fluids
Type: Mobil Velocite #10, 20W and 50W commercial engine oils
Absolute viscosities: $\mu \cong .15 - 500$ p
Typical Dynamic Variable Range
Reynolds number: $Re \cong .01 - 10.$
Shear rate: $\gamma \cong \varepsilon\omega c \cong 1 - 200\ s^{-1}$
Temperature: 24 - 54°C
Force: 10 - 200 N

Conditions are sought which isolate the effect to be studied, and do not introduce intervening variables such as viscous heating, rotor dynamic deflections, cavitation, etc. For this reason, the pressure fluctuations generated are much less than those of real dampers, and the kinematic conditions much less severe. The apparatus allows for the film to be pressurized to avoid cavitation but this was not required to perform the series of tests reported here. The emphasis is to create an experimental situation where the basic thesis can be tested, rather than to reconstruct the more complex "real-world." The key dimensionless variable Re is to be kept in the range applicable to actual dampers although the physical parameters (ρ, μ, c, etc.) may be quite different.

Initially the apparatus was built as a "short" open-ended bearing, directly submerged in the bath. After a year or so, this configuration was abandoned due to problems with cavitation and low signal-to-noise ratio. Later, the present sealed configuration was designed and built, which produces a much stronger signal. The fluid film is sealed on the ends (i.e., top and bottom) by two annular plates which are rigidly fastened to the outer ring and contain a channel for a "quad-ring" seal, see Figure 1. The "quad-ring" allows relative sliding motion of the inner ring with a small friction force, but prevents leakage of the fluid in the gap.

EXPERIMENTAL METHOD

A long debugging phase was required for this study as with most experimental projects. Initially there was insufficient stiffness in the outer ring assembly, which was resolved by the dummy transducer. At the beginning of the project, the outer ring could not be properly centered and very strange force signals were obtained. If the outer ring is centered, the problem is steady with respect to coordinates which rotate with the cam, and the force transmitted to the stationary transducer is sinusoidal (as produced by a rotating force vector of constant magnitude).

This situation was eventually resolved by a dummy shaft set-up which can only be removed if the outer ring alignment is perfect. The dummy shaft is a single piece consisting of a shaft and a concentric disk segment at midlength. The shaft fits exactly in the housing where the inner ring assembly normally goes. The disk outer diameter is extremely close to the inner diameter of the outer ring. The dummy shaft is set in place in the housing and the outer ring is loosened slightly to fit around the disk. If there is any slight misalignment when the outer ring is tightened down, the dummy shaft binds up and cannot be turned or removed. The outer ring is infinitely adjustable by special connections between the transducer and the housing.

The cam and inner ring assembly geometry is completely determined by the shaft bearings based on shaft deflection measurements described above. The inner ring inertia does not play a role in data reduction if its motion is considered to be the known problem input and the fluid forces transmitted across the film are the measured output variable.

The force transducer was calibrated dynamically with a vibration test "shaker."

Three Newtonian fluids were used, a very light spindle oil and 20W and 50W commercial automotive oils. Detailed viscosity-temperature curves were taken for each fluid in a capillary viscometer. Tests were performed at a series of temperatures between 24°C and 55°C, to thoroughly cover the viscosity range $0.15\ p < \mu < 30\ p$. The fluid bath (an integral part of the test rig) is very well mixed due to the action of the damper components.

The raw data are two traces on a storage oscilloscope. One signal is the output of the force transducer as the damper is running, which is sinusoidal, as mentioned above. The items of interest are the force amplitude F and the period, which can be obtained from inspection of the scope trace. The other signal is used to obtain the phase angle between the force and the maximum film thickness location. The fiber optical displacement probe signal is used for this task by detection of a timing mark on the cam shaft. When the mark passes the probe, the maximum film thickness point passes the direction of the force transducer and is indicated by a "blip" on the screen.

EXPERIMENTAL RESULTS

Typical experimental data are shown in Figure 3. For each fluid and temperature, two curves (force amplitude vs. speed) were taken. The "curves" are a series of points taken from successive oscilloscope traces as the speed is increased. One curve was taken with the gap empty, but the seal lubricated, and the other with the gap full of test fluid. The "empty" curve represents the sliding friction between the seal and the outer ring. It turns out that both curves are essentially in phase. The algebraic difference between the two curves therefore represents the fluid film squeezing force. The valid test range was considered to be the region where the fluid film exceeds the friction force, until the cavitation region shown. Cavitation can be eliminated by pressurization of the film, however, tests of this type have not been conducted to date. The valid test range varies with fluid viscosity and cam size (eccentricity ratio).

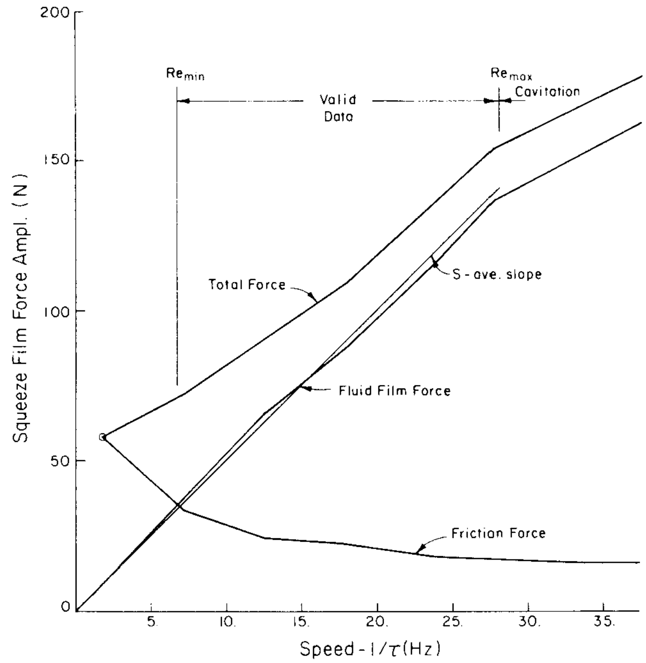


Figure 3 Typical Experimental Data

Table 2 has been prepared for the case $\varepsilon = 0.5$ by evaluation of many curves such as Figure 3. The slope of fluid film force amplitude F vs. $(1/\tau)$ on each curve is is determined.

$$S = \frac{dF}{d\tau^{-1}} \simeq \frac{\Delta F}{\Delta(\tau^{-1})} \tag{3}$$

Due to inaccuracies in constructing these curves only the average slope in the valid test range is used. According to lubrication theory the slope is a straight line starting at the origin over the entire range. If inertia is present, lubrication theory predicts the slope as $(1/\tau) \to 0$. Local deviations of the slope from this value as $(1/\tau)$ increases are corrections to lubrication theory for inertia as Reynolds number increases. Unfortunately, construction of the curves of F vs. $(1/\tau)$ is too crude to yield this sort of detailed information. However, in many cases a concave upwards shape of the curves can be discerned. This turns out to be the proper trend, namely force increasing with Reynolds number.

A dimensionless force group is then constructed:

$$W = \frac{dF/d\tau^{-1}}{\mu(2RL)} \left(\frac{c}{R}\right)^2 . \tag{4}$$

TABLE 2

REDUCED EXPERIMENTAL DATA, $\epsilon = 0.5$

Oil	Temperature ($^{\circ}$C)	Viscosity μ (p)	Slope S (N-s)	Load W (-)	Frequency Range τ^{-1} (Hz)	Reynolds Number Range Re (-)
50 W	26.6	3.29	23.6	42.2	5 - 8	0.074 - 0.118
50 W	37.8	1.79	12.0	39.3	5 - 18	0.136 - 0.493
50 W	43.3	1.33	11.0	48.6	5. - 15.	0.185 - 0.556
50 W	48.9	1.00	8.10	47.6	5. - 15.	0.246 - 0.737
20 W	37.8	0.638	5.05	46.5	10. - 25.	0.733 - 1.83
20 W	43.3	0.534	4.25	46.7	10. - 25.	0.916 - 2.29
20 W	54.4	0.316	3.30	60.9	10. - 35.	1.55 - 5.42
10 W	41.1	0.176	2.50	83.5	15. - 35.	4.12 - 9.62
10 W	46.7	0.151	2.27	87.8	15. - 25.	4.82 - 8.04
10 W	26.6	0.318	3.70	65.8	7. - 37.	1.06 - 5.62
10 W	32.2	0.247	3.20	75.6	6. - 25.	1.37 - 4.90
	Density	$\rho \cong 0.89 \frac{gr}{cm^3}$				

For the infinite length bearing, from lubrication theory,

$$W_{\ell\infty} = \frac{24\pi^2\epsilon}{(2+\epsilon^2)(1-\epsilon^2)^{1/2}} , \qquad (5)$$

which neglects any end leakage or edge effects and also assumes a full film, i.e., no cavitation. In terms of these variables Reynolds number is defined by

$$Re = \frac{2\pi c^2 \rho}{\tau\mu} . \qquad (6)$$

Line 8 of Table 2 represents the data of Figure 3.

The information in Table 2 is then plotted in Figure 4. The limit as Re → 0 is considered to be the lubrication theory value W_ℓ, even though it is less than that predicted by Eq. (5). For $\epsilon = 0.5$, $W_{\ell\infty} = 60.8$, but $W_\ell = 42.1$, from Figure 4. The difference is probably due to edge effects and end leakage, but also due to inability to accurately measure the eccentricity. The accuracy in ϵ is no greater than ± 20% which would account for much of this deviation. The fact that $W_\ell < W_{\ell\infty}$ is in the right direction. Pressure loss at the film ends due to leakage would reduce the force relative to $W_{\ell\infty}$. In addition, the damper is anything but infinitely long, in fact $L/R \cong 0.3$, so the edge effect may be very large in relation to the overall bearing length.

Virtually no phase shift relative to lubrication theory was detected (< 10°). From lubrication theory, the maximum force occurs in the direction of maximum local squeezing rate, which is 90° ahead of the maximum fluid thickness point. A phase shift would be seen as a shift on the timing mark trace relative to the force peaks. This result is troubling since the theory indicates such a phase shift should occur. For Re = 10. the phase shift is 45° according to Ref. [3].

DISCUSSION OF EXPERIMENTAL RESULTS

A ratio of the measured dimensionless load relative to the lubrication theory dimensionless load is formed:

$$W^* = \frac{W}{W_\ell} . \qquad (7)$$

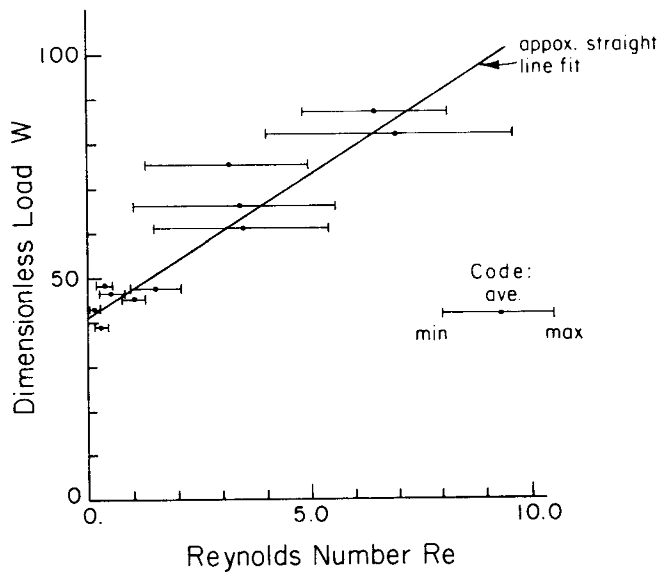


Figure 4 Dimensionless Load Amplitude vs. Reynolds Number, $\epsilon = 0.5$

The value of W^* represents a correction factor to the lubrication theory load. The results for $\epsilon = 0.5$ as discussed above are shown in Figure 5. In a similar way data were prepared for $\epsilon = 0.2$ and this is also shown on the figure. The latter results were somewhat more difficult to obtain because the fluid film forces are smaller, which reduces the range of validity on each curve of the type illustrated in Figure 3.

As this article is prepared, similar data are being analyzed for the case $\epsilon = 0.8$. A preliminary examination shows the same sort of behavior in W^* vs. Re, i.e., the same magnitude of inertia correction. Several theoretical curves are also shown on Figure 5. The curves are for asymptotic limits in the long bearing case: arbitrary Re, $\epsilon \ll 1$, [3]; and arbitrary ϵ, Re $\ll 1$ [4]. It is seen that in all cases the measured inertia effect is clearly greater than the theoretical prediction. It also does not appear that the trend exists for the inertia effect to lessen as ϵ increases according to Ref. [4].

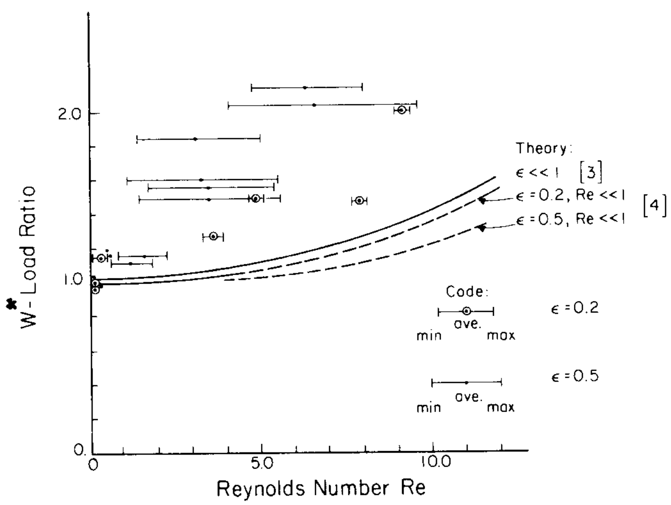


Figure 5 Correction to Lubrication Theory for Fluid Inertia

CONCLUSIONS

Clearly a very substantial effect of fluid inertia in the damping force is seen at quite moderate Reynolds numbers. The relative inertia forces are just as strong at moderate eccentricity ratios ($\epsilon = 0.5$) as at low ratios ($\epsilon = 0.2$). This seems contrary to physical intuition which would say that since viscous forces increase drastically with ϵ, the relative strength of inertia should decrease.

There are probably relatively large uncertainties in the results presented here, mainly due to the procedure in obtaining the fluid film forces, i.e., subtracting the friction in the seals, as illustrated in Figure 3. Fluid pressure in the gap may tend to deform the "quad-ring" and affect the friction force. This situation will be remedied, it is hoped, by adding pressure probes to the apparatus, and checking measured pressures relative to forces, both from the standpoint of magnitude and phase. The estimated uncertainty at present is maximum ± 25% based on the limiting behavior as $Re \rightarrow 0$, see Figures 4 and 5.

The existence of the fluid inertia force is clearly demonstrated in the present study, even allowing for a large uncertainty as to the amount of correction required to lubrication theory. In view of other experimental and theoretical studies, the evidence is overwhelming.

Continued efforts in the present project should (1) elucidate the problem of phase shift in the damper forces due to inertia, (2) establish the behavior at higher eccentricities, and (3) reduce uncertainty in the present range of force amplitude and Reynolds number.

It has been suggested that the apparent lack of a phase shift may be due to nonrigidity of the inner ring and/or outer ring components. While this makes physical sense, we believe it is not so in light of measurements of the outer ring deflection and the inner motion.

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