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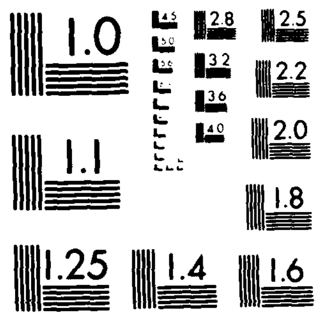


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high speeds can combine to create conditions in practical applications where fluid inertia and viscoelastic effects may become significant, violating the assumptions under which Reynolds equation can be applied. The analysis shows that the results of lubrication theory can be greatly in error with regard to phase effects between bearing forces and displacements, which may have profound implications regarding critical speed and forced response behavior.

Very large deviations from lubrication theory have been predicted in a series of analytical papers under this contract. These deviations have been observed in laboratory experiments by the present investigator, and, independently, by several industrial research groups. In the present study 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 the improved theory which includes fluid inertia forces.

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THE EFFECT OF FLUID INERTIA AND
VISCOELASTICITY IN SQUEEZE-FILM
DAMPER BEARINGS

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TABLE OF CONTENTS

## LIST OF TABLES



## LIST OF FIGURES

## NOMENCLATURE

$c$ - Radial clearance

De - Deborah number, Eq.(5)

$h, h_o$ - Film thickness, reference film thickness

$O(\ )$ - Order of magnitude

$R$ - Damper radius

Re, Re′ - Reynolds number for inertia effect in squeeze films, journal bearings; Eqs.(1), (2). Also called modified or reduced Reynolds number

$t$ - Time

$u$ - Fluid velocity in film-wise direction

$U$ - Surface sliding velocity, reference film-wise velocity

$V$ - Reference squeeze velocity

$W, W_r, W_t$ - Bearing load (force); amplitude, radial direction, tangential direction; see Eq.(6) and Figure 1

$W_{lub}$ - Load predicted by lubrication theory

$\dot{\gamma}$ - Shear rate of lubricant

$\epsilon, \bar{\epsilon}$ - Eccentricity ratio of damper; instantaneous value, time average value

$\nu$ - Kinematic viscosity

$\phi$ - Phase angle between inner ring velocity and load, Figure 1 and Eq.(6)

$\omega$ - Oscillatory frequency

$\Omega$ - Attitude angle, Figure 1

$\theta$ - Fluid relaxation time

ABSTRACT

In the modeling and analysis of rotor dynamic systems, the behavior of squeeze film damper bearings is normally predicted by the Reynolds equation of hydrodynamic lubrication. Large bearing gaps and high speeds can combine to create conditions in practical applications where fluid inertia and viscoelastic effects may become significant, violating the assumptions under which Reynolds equation can be applied. The analysis shows that the results of lubrication theory can be greatly in error with regard to phase effects between bearing forces and displacements, which may have profound implications regarding critical speed and forced response behavior.

Very large deviations from lubrication theory have been predicted in a series of analytical papers under this contract. These deviations have been observed in laboratory experiments by the present investigator, and, independently, by several industrial research groups. In the present study 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 the improved theory which includes fluid inertia forces.

# I. INTRODUCTION

This document is a proposed final report of ARO Contract No. DAAG 29-80-K-0064, "The Effect of Fluid Inertia and Viscoelasticity in Squeeze-Film Damper Bearings." In this Introduction, the background and motivation for the research is briefly outlined, with important references. Later, a progress report on the research conducted to date is presented.

## A. Technical Background

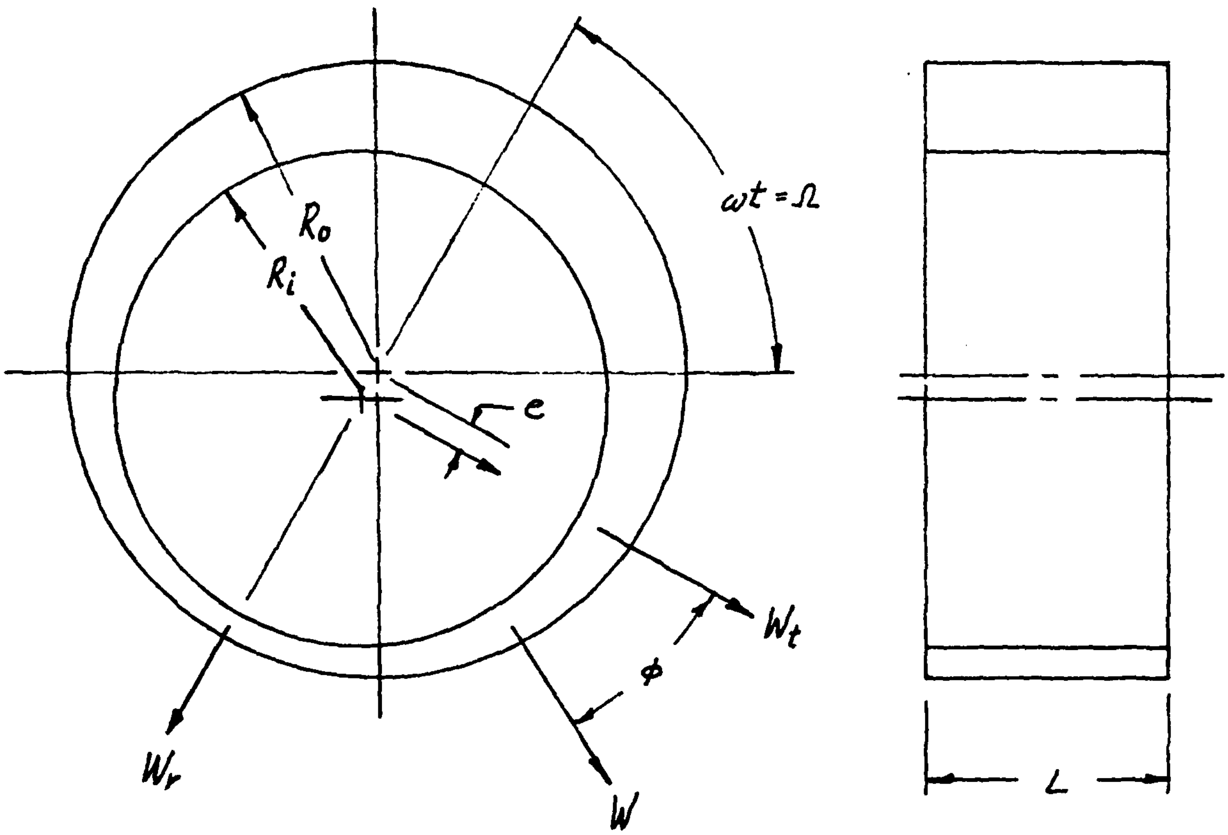
The hydrodynamic lubrication theory has been used as a basis for design and analysis of fluid film bearings (such as squeeze-film dampers) for many years. There is no question that the theory has been adequate for the vast majority of practical applications when the "lubrication assumptions" are operative. In recent times, however, the tendency to very high-speed turbomachinery applications, and the use of high molecular weight polymer additive lubricants, call two of these assumptions into serious question: (1) noninertial or low Reynolds number flow, and (2) the Newtonian fluid.

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 almost no damping to the rotor dynamic system, increasing susceptibility to vibration 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 [1].

The hydrodynamic squeeze-film damper 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.

This report addresses the effects of fluid inertia and viscoelasticity on the behavior of high-speed squeeze film bearings, where the lubrication theory may break down [2-4]. Purely viscous non-Newtonian effects, such as viscosity variation, are not considered _per se_. It is important to note at this point that viscoelasticity and inertia effects are considered here as similar phenomena in that both come into play at high speeds and both introduce similar phase shifting behavior into lubrication problems. Lubrication theory requires that an imposed local surface velocity and the resulting pressure be exactly in-phase.

The squeeze-film bearing forces are nearly always obtained from the Reynolds equation of hydrodynamic lubrication. For computational simplicity it is most common to use either the "short" or "long" bearing approximation, with the so-called $\pi$ and $2\pi$ films representing the cavitating and noncavitating cases, respectively. Both cavitation and finite length effects can cause deviation from the idealized predictions, and both effects have been the subject of recent study [5-7]. In these instances Reynolds equation is still in force but perhaps difficult to apply. In the case of fluid inertia and viscoelastic




$c = R_o - R_i$

$R \cong R_o \cong R_i$ $\qquad$ $\varepsilon = e/c$

Figure 1  Squeeze Film Damper Geometry

effects, Reynolds equation itself may not be applicable. Discrepancies from lubrication theory have been established analytically, and measured in the laboratory as part of this project. In addition, detailed independent experimental measurements on squeeze-film dampers from Mechanical Technology, Inc. [8] and Bently Nevada Corporation [9] strongly point to fluid inertia effects as a cause of large deviations from Reynolds equation. After years of skepticism, th effect of fluid inertia in practical applications of squeeze-film damp is becoming widely accepted by the gas turbine industry.

The design and selection of squeeze-film dampers are often based on computer modeling of entire rotor dynamic systems. Typically, the equations of motion for a segment of rotor are considered. The rotor forces may be due to the squeeze-film bearings, rotor weight, rotor unbalance, etc. The rotor may be rigid or flexible. In the latter case, elastic compliance between the rotor segments is considered. In these computer studies, very detailed and sophisticated programs are developed to predict machine behavior, but the analyses may be no better than the damper models used. The design of rotor dynamic systems with squeeze-film dampers is known to be tricky. Substantially worse performance can result if improper values of the bearing parameters are used [10,11]. Clearly, accurate knowledge of damper bearing behavior is required. Large errors in phase angle may have profound implications on critical speed and forced response analysis.

Lubrication theory predicts that damper force is in-phase with the local squeezing direction, i.e., $\varphi = 0$ on Figure 1. The present study shows that phase angle moves forward up to $90^{\circ}$ relative to lubrication theory predictions. This phenomenon has been termed "negative stiffness" or "fictitious mass" by rotor dynamicists. For circular centered motion the direction of the fluid inertia force is opposite to that of the hydrodynamic bearing stiffness force ($\varphi = -90^\circ$) and the same as the rotor mass inertia ($\varphi = 90^\circ$).

## 1. The Fluid Inertia Effect

As shown in Refs.[2] and [3] the appropriate Reynolds number governing the ratio of inertia to viscous forces in oscillating squeezing flow is

$$Re = \frac{\omega h_o^2}{\nu} = \frac{\omega c^2}{\nu} \quad \text{(for dampers)} \tag{1}$$

See the Nomenclature for a list of symbols used. In steadily operated journal bearings, the corresponding Reynolds number is the so-called "modified" or "reduced" Reynolds number [12,13]:

$$Re' = \frac{Uh^2}{\nu R} \tag{2}$$

Since for journal bearings $U = \omega R$, the two expressions are equivalent. In fact for both squeeze-film and journal bearings $u \approx 0(\omega R)$. In the former case $u \approx 0(VR/c) \approx 0(\omega R)$, where V is a reference squeeze velocity.

For steady journal bearings, at a particular Reynolds number $Re'$, the inertia correction to lubrication theory for load is small, e.g.

$$W \cong W_{lub}(1 + 0.0066\ Re') \tag{3}$$

at $\epsilon = 0.5$. Equation (3) follows from Eq.(12-30) and Table 12-2 of Pinkus and Sternlicht [13]. For bearing conditions $\omega = 1000\ s^{-1}$

($\sim$ 10,000 rpm), $c = 0.05$ cm, $R = 30$ cm, $\nu = .10$ St; $Re' = 25$. Hence the inertia correction is only, say, 5%. For squeeze film flows the author has shown [3]

$$W \cong W_{lub}(1 + k\ Re) \qquad (4)$$

where k is a positive number between, say, 0.1 and 0.3. For the same conditions (typical for squeeze-film dampers) the inertia correction is 250%! In both cases inertia increases load. Higher pressures are required to accelerate the fluid, as well as to overcome viscous shear.

The fact that the inertia effect in steady journal bearings is known to be so small is one reason workers in squeeze-film damper technology have been slow to recognize its importance. However, approximate inertia corrections for simple squeeze films have been around for many years [14,15]. If such large effects are present, recognition of the proper role of fluid inertia in dampers can greatly affect the design process.

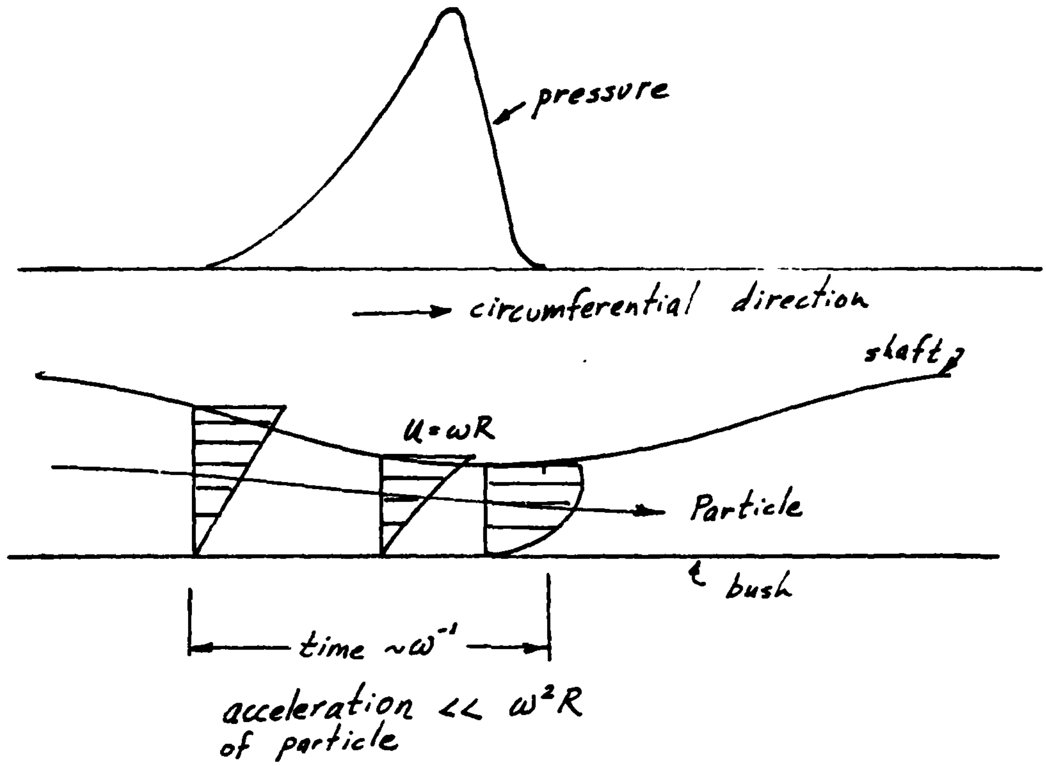
It is natural to ask why, physically, this large difference exists. In a very lightly loaded journal bearing the flow is entirely Couette (linear velocity profile). Apart from centrifugal instabilities which tend to occur in much thicker films, there would be no inertia effect at all. The fluid particles move about the circumference in parallel laminae at a constant velocity. A more highly loaded bearing has a pressure-driven Poiseuille flow contribution (parabolic profile). As the pressure field increases and decreases along the circumference the Poiseuille flow velocity component points in one circumferential direction and then the other. This effect, experienced by fluid particles as they move along the gap, causes the acceleration or inertia effect. One may note from Table 12-2 in Pinkus and Sternlicht, that the inertia effect increases as loading (eccentricity) increases. However, the acceleration is only sensed by fluid particles within the relatively small circumferential angle of the pressure wave.

For squeeze-film dampers (or any squeezing flow) the flow is almost entirely Poiseuille. Acceleration effects are present in both the unsteady and convective sense (i.e., at each spatial point, and following the particle, respectively). In the case of oscillations, all fluid particles are constantly undergoing rapid changes in flow direction as the surface performs its cyclic oscillations. These differences between the journal bearing and squeeze-film damper are depicted in Figure 2.

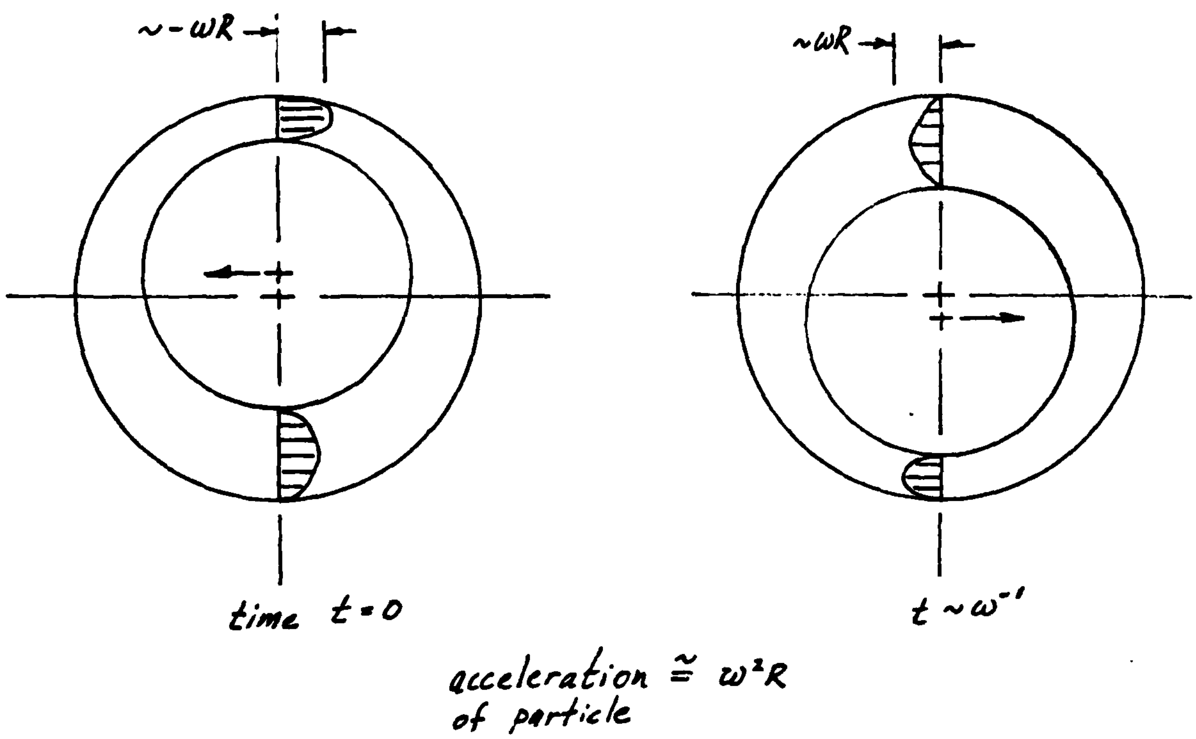
The phase angle between the instantaneous squeezing direction and the resultant fluid film force ultimately shifts forward $90^{\circ}$. For the lubrication theory case, $\varphi = 0^{\circ}$. Curiously, the bearing filmwise dimension has no effect on the fluid inertia effect in squeeze-film bearings, cf. Eq.(1). Hence a large squeeze-film bearing will tend to have large inertia effects regardless of the aspect ratio c/R or c/L. These basic physical concepts are explained in a paper recently presented by the author at the 1983 ASME Gas Turbine Conference [16].

## 2. The Viscoelastic Effect

It is well known that modern lubricants are significantly viscoelastic due to the addition of high molecular weight polymer additives. The highly unsteady nature of damper applications also tends to increase




a) Velocity profiles, "unwrapped" steady journal bearing




b) Velocity profiles, squeeze film damper

Figure 2 Squeeze Film and Journal Bearing Inertia Effect the likelihood of significant viscoelastic effects, as will be seen below. Viscoelastic behavior is distinguished from non-Newtonian purely viscous behavior where the viscosity may vary with shear rate, for example. For Newtonian fluids subjected to a simple shearing flow, the shear stress is directly proportional to the shear strain rate through a constant viscosity. Viscoelastic fluids differ in two essential ways: (1) normal stresses (other than isotropic pressure) may be produced by steady shearing motion, and (2) relaxation or time-dependent effects may be displayed when shearing stress (or motion) is suddenly applied. For purely viscous fluids these two effects are absent although the shear stress-strain rate relation may be nonlinear. Viscoelastic models (constitutive equations) must be written in tensor form to handle shearing and elongational motions, and shear and normal stress. In addition the models must obey certain rules of invariance such that any observer in any coordinate frame would calculate the same stress field. The study of viscoelastic materials is part of rheology, the science of deformation and flow.

For nearly two decades, a controversy has existed concerning the possible effects of lubricant viscoelasticity in hydrodynamic lubrication. Various aspects have been discussed in the literature for many years, see for example Refs.[17-21]. Most attention in this area has focused on load-carrying capacity in steady applications, i.e., do the viscoelastic properties of a lubricant increase or decrease load in various bearing applications? Although the final verdict is uncertain, it appears that the overall effect is not large, i.e., the loads predicted by the (Newtonian) lubrication theory do not differ from those obtained with a viscoelastic fluid by more than, say 30% for steady flows.

The case of dynamic lubrication problems where phase shifting effects are important may be quite different. The Reynolds equation is fluid mechanically steady, and time-dependent effects enter only through the boundary conditions. This means that the fluid velocity must respond instantaneously to the boundary motion. Thus the Reynolds equation is inherently ill-equipped to handle unsteady flows. In reality, both fluid inertia and viscoelasticity introduce a time shift into the fluid response. Due to the high rotational speeds in many turbomachinery applications, predictions of Reynolds' equation may be significantly in error.

In the unsteady flow of a viscoelastic fluid, the most important governing dimensionless group is the Deborah number De, a ratio of the fluid characteristic response time (relaxation time $\theta$) to the characteristic process time $\Delta t$. For oscillatory flow, $\Delta t \sim \omega^{-1}$, hence

$$De = \omega\theta \,. \tag{5}$$

For Newtonian fluids $\theta \rightarrow 0$, and for steady flow $\Delta t \rightarrow \infty$; in either case $De = 0$. For polymer additive lubricants typically $\theta = 10^{-4} - 10^{-3}$ s. In high-speed dampers if $\omega = 1000\ s^{-1}$, $0.1 < De < 1.0$.

## II. PROGRESS REPORT

All of the essential research findings of this project have been reported in published papers listed below. The interested reader is referred to the appropriate paper for the details. A brief description of each is provided below.

A. Analysis

The research of five analytical papers has been supported under this contract. In all five papers the support of ARO is acknowledged exclusively. These papers are:

[16] "The Effect of Fluid Inertia in Squeeze-Film Damper Bearings: A Heuristic and Physical Description," ASME Paper No. GT 83-GT-177, 1983.

[24] "Effects of Fluid Inertia and Viscoelasticity on Squeeze-Film Bearing Forces," ASLE Trans., v.25, n.1, January 1982, pp.125-132.

[25] "Effects of Fluid Inertia and Viscoelasticity on Forces in the Infinite Squeeze-Film Bearing," ASLE Paper 83-AM-3E-1, 1983.

[26] "An Approximate Analysis of Fluid Inertia Effects in Axisymmetric Laminar Squeeze-Film Flow at Arbitrary Reynolds Number," Applied Scientific Research, v.37, n.4, 1981, pp.301-312.

[27] "Effects of Fluid Inertia and Viscoelasticity on Squeeze-Film Bearing Forces at Large Vibration Amplitudes," Wear, v.76, 1982, pp.69-89.

The first paper is a physical description of the fluid mechanical processes occurring in a squeeze-film damper in the presence of viscous and inertia forces. It was presented at the 1983 Gas Turbine Conference at the behest of several researchers in squeeze-film damper technology. The conference attendees by-and-large were interested in rotor dynamics application of dampers rather than hydrodynamic lubrication phenomena *per se*. The second and third papers, Refs.[24] and [25] are relatively straightforward applications of the theory developed in [2-4]. If the amplitudes are sufficiently small, the theory is "exact." Simple correction formulae for the lubrication theory damping coefficients are developed, in terms of the mean eccentricity ratio, and Reynolds and Deborah number. Large deviations from lubrication theory predictions due to the improved theory are found for a practical range of bearing kinematic variables and lubricant properties. Only the noncavitating case is included, since the pressure disturbances are very small in principle.

In Ref.[26], a method is sought, through suitable approximation, to remove the small amplitude restriction. It is also desirable to not rely on low Reynolds or Deborah number perturbation methods, since they are strictly valid only for values $\ll 1$. The approximation method is to linearize the convective terms through an Oseen-type approximation [28]. Only the Newtonian fluid case is considered, and results are presented for the parallel surface squeeze film. The predicted high Reynolds number behavior deviates only slightly from low Reynolds number behavior. Good agreement was found to existing analytical solutions and experimental results in limiting cases ($Re \ll 1$, $Re \gg 1$, $\epsilon \ll 1$).

Reference [27] seeks to apply this linearization technique to the squeeze-film damper problem with the second-order viscoelastic fluid at large vibration amplitudes. The short bearing with a circular centered orbit is considered. A rather complicated solution is developed in terms of a series in $\epsilon^n \epsilon^{ni\omega t}$. The first-order term is identical to that of the first paper in the case $\bar{\epsilon} \to 0$, $\epsilon \to 0$. Only the noncavitating case was solved due to the complexity involved. Again large deviations from lubrication theory are found, although the amplitude does not have a large effect on the deviations.

B. Experiment

Most recent efforts on this project have focused on the experimental aspects. After some early delays due to procurement difficulties, the experimental device was completed by an outside machine shop, in July 1981. A schematic of the apparatus is shown in Figure 3.

There are two papers concerned with the experimental phase of this project:

[29] "Measurements of Squeeze Film Bearing Forces to Demonstrate the Effect of Fluid Inertia."

[30] "Measurements of Squeeze Film Bearing Forces and Pressures at High Reynolds Number."

The former has been accepted as an ASME pamphlet and will be presented at the 1984 Gas Turbine Conference. The latter is being prepared for submission to the 1984 ASME/ASLE Joint Lubrication Conference, and for publication in the ASME Journal of Tribology.

The purpose of the experimental rig is to allow measurement of the squeeze-film bearing forces. In particular, discrepancies from lubrication theory predictions (amplitude and phase shift) are to be noted. Amplitude and phase angle are defined as follows:

$$|W| = (W_r^2 + W_t^2)^{1/2} \qquad \phi = \tan^{-1}\left(\frac{W_r}{W_t}\right), \tag{6}$$

see Figure 1. From lubrication theory, the inertia effect should occur as speed increases for a given fluid and geometry configuration. The rheological effect should be observed as a Newtonian fluid (low molecular weight oil) is replaced by a viscoelastic fluid (oil with polymer additive) with the same viscosity at the test shear rate.

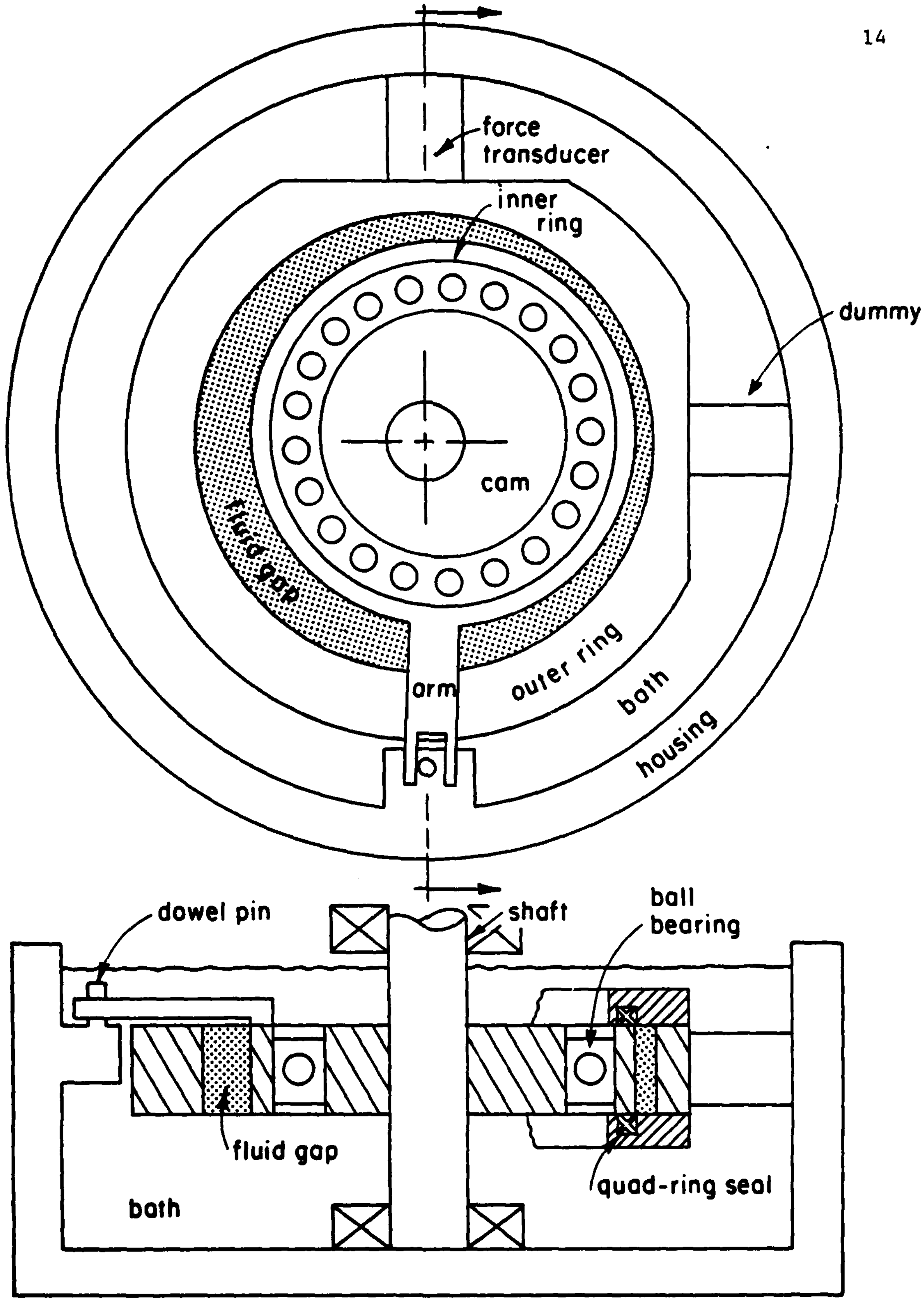



Figure 3 Schematic of Experimental Apparatus

The experimental device imposes a fixed circular centered orbit on the nonrotating inner ring, through an off-center cam. Force on the outer ring is measured by the piezoelectric transducer shown. The outer ring assembly has a very high natural frequency relative to the imposed inner ring motion as measured by a noncontacting fiber-optic probe. Three cams and three belt-and-pulley drive combinations allow a wide range of kinematic variables, see Table 1. Only shaft speeds up to 5000 rpm have been tested.

Conditions are sought which isolate the two effects to be studied, and do not introduce intervening variables such as viscous heating or cavitation. For this reason, the pressure fluctuations generated are much less than those of real dampers, and the kinematic conditions much less severe. The emphasis is to create an experimental situation where the basic thesis of this contract can be tested, rather than to reconstruct the more complex "real-world." The key dimensionless variables Re and De are to be kept in the range applicable to actual dampers although the physical parameters ($\rho, \mu, c$, etc.) may be quite different.

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

TABLE 1

RANGE OF EXPERIMENTAL VARIABLES

<u>Apparatus</u>

d.c. motor speed: 100 - 2000 rpm

Belt and pulley drive combinations: 1:2, 2:1, 9:1

Resultant shaft speed (damper squeezing rate): $N = 50 - 18{,}000$ rpm.
$\omega \cong 5 - 2000$ rad/s

Outer ring radius: $R_o = 4.166$ cm

Inner ring radius: $R_i = 4.064$ 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

<u>Fluid Properties</u>

Absolute viscosities: $\mu \cong .10 - 500$ p

Relaxation time (estimated): $\theta = 10^{-8} - 10^{-3}$ s

<u>Typical Dynamic Variable Range</u>

Reynolds number: $Re = \omega c^2/\nu \cong .01 - 30.$

Deborah number (estimated): $De = \omega\theta \cong 0 - 2$

Shear rate: $\dot{\gamma} \cong \varepsilon\omega c \cong 1 - 200\ s^{-1}$

sufficiently low (see Table 1) such that viscous heating problems do not occur.

A fiber-optical displacement probe is used to determine the phase shift. Two methods have been used: (1) measuring the inner ring displacement, or, more simply, (2) optical detection of a timing mark on the cam shaft. Since the inner ring assembly motion has been shown to be essentially rigid (through dynamic displacement measurement) the two methods are equivalent. The inner ring kinematics is entirely specified by the shaft rotation.

## 1. Experimental Results

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

$$W^* = \frac{W}{W_\ell} . \tag{7}$$

The value of $W^*$ represents a correction factor to the lubrication theory load. The results for $\epsilon = 0.2$ and 0.5 are shown in Figure 4 from Ref.[29].

In Ref.[30], similar data are being analyzed for the case $\epsilon = 0.8$ along with pressure measurements.

The existence of the fluid inertia force is clearly demonstrated by experiments of 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.

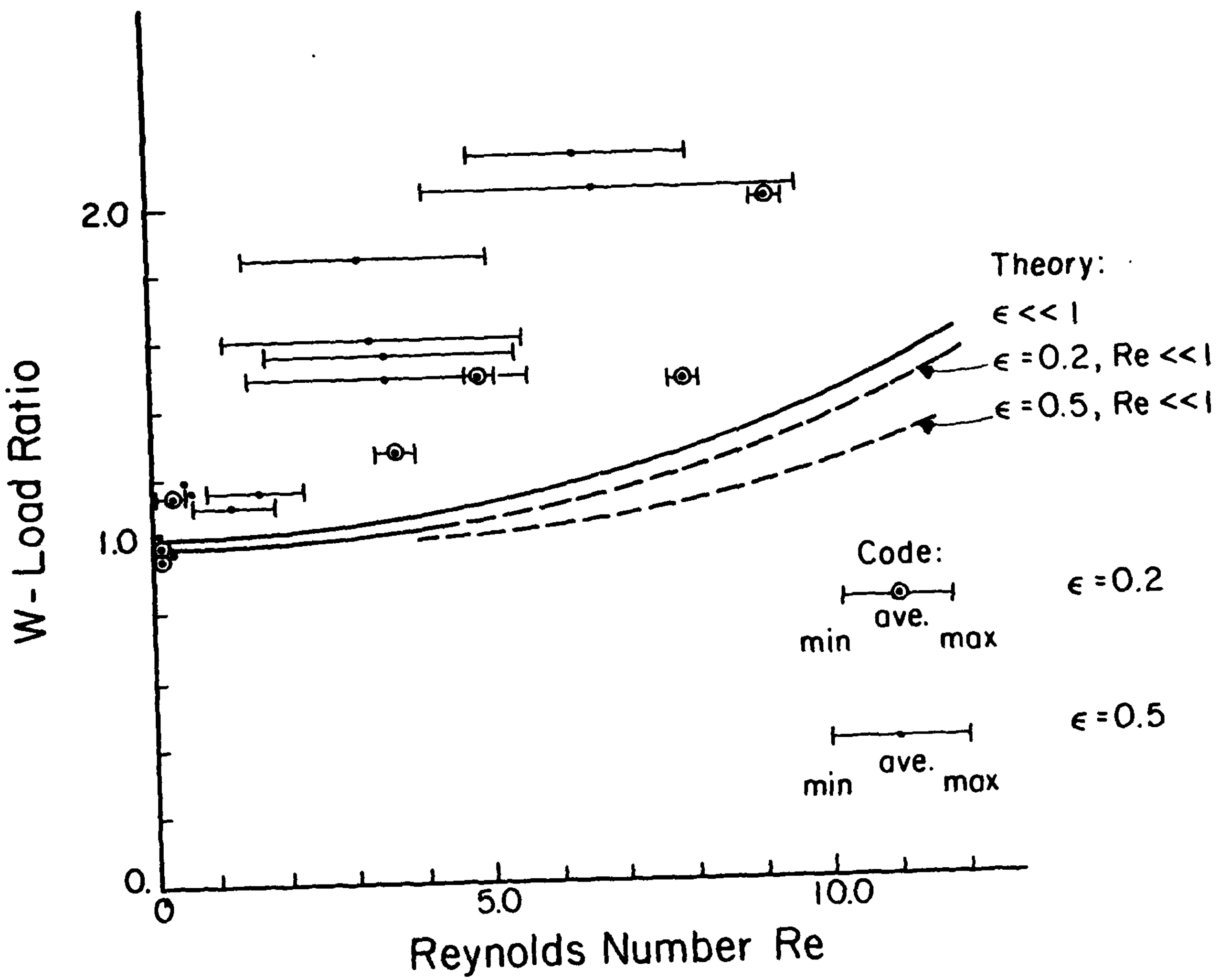



Figure 4 Summary of Experimental Data, Newtonian Fluids, Eccentricity Ratio = 0.5

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