I-A2049-3

ANNUAL REPORT ONR PROJECT A-2049

by

Dudley D. Fuller

July 15, 1958

Prepared under

Contract Nonr-2342(00)
Task NR 097-343

Jointly supported by

DEPARTMENT OF DEFENSE
ATOMIC ENERGY COMMISSION
MARITIME ADMINISTRATION

Administered by

OFFICE OF NAVAL RESEARCH
Department of the Navy

TABLE OF CONTENTS

INTRODUCTION

This is a summary report on the activities of the research program conducted for the Office of Naval Research under Contract Nonr-2342(00), "Research on Gas Lubricated Bearings", for the first year of effort, April 1, 1957 to April 1, 1958.

The stated objective of this investigation is to conduct a broad and basic program of analytical and experimental research into the fundamental behavior of gas-lubricated bearings and to establish general design criteria for these bearings.

The problems that need solution were known to be exceedingly complex and perhaps too difficult to be handled by any single approach, either experimental or analytical. It was concluded, therefore, that the greatest chance for success would come from a program where several parallel and complementary efforts were being made at the same time. Accordingly, work was initiated in both the theoretical and experimental phases of hydrodynamic and hydrostatic (externally pressurized) lubrication phenomena, supplemented by a digital computer program. In order to fully implement these plans, the staff of the Friction and Lubrication Section was augmented by the addition of three qualified investigators, and also by a consultant on theoretical fluid mechanics and heat transfer.

With the group organization established, one of the early tasks was to survey the field of published literature on gas-lubricated bearings and gather together and abstract the information that became available in this manner. An associated effort was to establish liason with other groups working on similar problems, so that the program would benefit from the information that might, as a consequence, be made available, and also, equally important, duplication of effort would be avoided. Through personal acquaintance and through correspondence, these contacts were made. The response was remarkable — almost without exception, one of complete and enthusiastic cooperation. As a result, there is now continuing contact with many research personnel conducting gas-bearing investigations in this country and abroad. This effort has been greatly assisted by the quarterly meetings of the Advisory Group consisting of members of supporting government agencies associated with this program. In effect -- a network of lines of communication has been established through The Franklin Institute.

The survey of literature in this field, both publically and privately published, was made more complete as a result of the assistance received from members of the Advisory Group and from guests who have been present at several of these Advisory Group meetings, as well as the help that was given by those people doing gas-bearing research in various parts of this country. A bibliography and collection of abstracts was published as part of the program in December 1957. This bibliography should be considered as an interim effort, since additional reference material is being constantly collected. A final report on the literature in this field should be ready near the end of 1958. The interim bibliography contains references to 102 papers, 16 patents, and several books. The bibliography also contains cross reference systems so that information can be obtained in terms of the year of publication, authors' names and on the basis of subject content.

It might be mentioned that a somewhat related program is under way at Columbia University in which the gas-bearing literature is being critically reviewed. This work is being sponsored by WADC and it is believed that both the Franklin program and the Columbia University effort have benefited through an interchange of information.

In addition to making a survey of literature, the effort on this program for the past year has been directed toward hydrodynamic lubrication, analytical and experimental; hydrostatic lubrication, analytical and experimental; and toward establishing a digital computer program.

HYDRODYNAMIC LUBRICATION

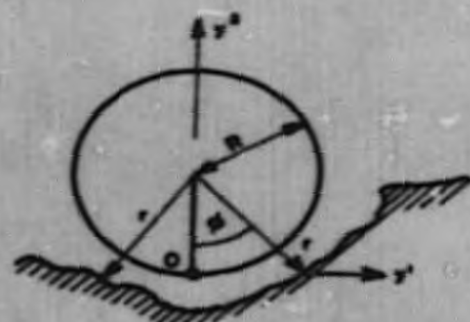
Analytical

Evaluation of Classical Incompressible Theory

In this area, Professor Elrod from Columbia University initiated a critical re-evaluation of the basic hydrodynamic equations. These are the classical equations that have been accepted for many years. He has rederived these equations with complete generality.

The first work was concerned with a derivation of the basic equations for hydrodynamic lubrication with a fluid having constant properties. The analysis that was developed applies to both journal and slipper bearings of arbitrary film geometries (Fig. 1).

In the treatment a "natural" coordinate system for the bearings was introduced



$$\zeta_1 = \frac{R}{L}\phi$$

$$\zeta_2 = \frac{y^2}{L}$$

$$\zeta_3 = \frac{R-r}{h(\zeta_1,\zeta_2)}$$

Figure 1

where:

- R = shaft radius
- r = radial coordinate within the fluid film
- L = characteristic dimension of bearing
- ϕ = angle
- h = any film thickness
- h_o = a particular film thickness
- y^2 = coordinate normal to y^1 y^3

The Navier-Stokes equations are written in this system by means of tensor notation. In all practical applications of hydrodynamic bearings, the ratio of film thickness to a characteristic bearing dimension is very small compared to unity. The author therefore expands his unknown quantities as power series in $\varepsilon = \frac{h_0}{L}$ and applies a perturbation technique to solve the Navier-Stokes equations. The final equation turned out to be Reynolds' equation with correction terms. The equation with the first correction term is given below:

$$\frac{\partial}{\partial x}\left\{h^3\left(1 - \frac{h}{D}\right)\frac{\partial p}{\partial x}\right\} + \frac{\partial}{\partial z}\left\{h^3\left(1 + \frac{h}{D}\right)\frac{\partial p}{\partial x}\right\} = 6\mu U\frac{\partial}{\partial x}\left\{h\left(1 - \frac{h}{3D}\right)\right\}$$

where:

D = shaft diameter
p = fluid pressure
U = shaft surface velocity
x = distance around the shaft in direction of rotation
z = distance parallel to shaft axis
μ = fluid viscosity

This equation shows that the curvature of the fluid film may be neglected whenever $\frac{h}{D} \ll 1$. In the case of a slipper bearing $D \to \infty$ or $\frac{h}{D} \to 0$.

Pertubation equations of $O(\varepsilon^n)$, $n > 1$, are given and discussed in principle. Theoretically, any degree of approximation to pressure and velocity distribution in the fluid film may be obtained with the method used by the author. This information in the form of a paper has been submitted for publication in a technical journal and will also appear as part of a technical report under the ONR contract.

Influence of Mean Free Path

A program was initiated to try to explain certain discrepancies existing between experimental results and theoretical predictions for hydrodynamic gas-lubricated bearings. To date, the preliminary investigations have shown three factors which may contribute to the discrepancies mentioned:

(a) the gas is a non-continous flow medium,

(b) bearing surfaces are relatively rough and wavy,

(c) Reynolds' equation does not apply accurately along the outer edges of a bearing.

Detailed theoretical investigations were made considering the molecular properties of a gas. The present theory of gas-lubricated bearings assumes that the gas can be considered as a continuous fluid. A closer analysis has shown however, that this assumption is no longer valid when the bearing gap width becomes comparable in magnitude to the molecular mean free path of the gas.

One example where the influence of molecular mean free path should be taken into consideration is the case of hydrodynamic gas bearings operating at low ambient pressures.

As soon as the film thickness in the bearing becomes comparable to the molecular mean free path, effects occur which can no longer be explained by continuum flow theory. For small ratios, (<<1) of the mean free path and the film thickness, the flow is known as the so-called "slip flow". This "slip flow" phenomenon is accounted for by modifying the boundary conditions of the gas in the bearing clearance and instead of assuming vanishing velocities at the walls, slip velocities are introduced. The temperature profile could also be modified by assuming a temperature discontinuity at the boundary between the wall and the gas but this has been shown to be so small that it can be neglected. The magnitude of the slip velocity can be evaluated from the kinetic theory of gases. Neglecting higher order terms, the slip velocity is given by:

$$U_{slip} = \sigma \times \frac{(2 - f)}{f} \lambda \frac{\partial u}{\partial z} \cdot \ . \ . \ . \qquad (1)$$

where: σ is a numerical constant,
f is reflection coefficient,
λ is molecular mean free path, inches.

$$\lambda = \frac{1}{\sqrt{2}\, nS}$$

where: n is number of molecules per in^3
S is mutual collision cross section in^2

The differential equation for the pressure distribution in a slider bearing operating under "slip flow" conditions has been derived, using a simplified form of the Navier-Stokes equation and the continuity equation, leading to:

$$\frac{\partial}{\partial x}\left\{\rho\left[\frac{h^3}{12\mu}\frac{\partial p}{\partial x}\left(1+6\frac{\lambda}{h}\right)-\frac{U_r h}{2}\right]\right\}+\frac{\partial}{\partial y}\left\{\rho\,\frac{h^3}{12\mu}\frac{\partial p}{\partial y}\left(1+6\frac{\lambda}{h}\right)\right\}=0 \qquad (2)$$

where:

h = the film thickness, inches,
ρ = the mass density, lb sec^2/in^4
μ = the viscosity, lb sec/in^2
U_r = the velocity of the runner, in/sec

Because the viscosity is nearly independent of the pressure in the range considered, and since the temperature variations across the gas film are small, the viscosity can be assumed to be constant. This yields:

$$\frac{\partial}{\partial x}\left[ph^3\frac{\partial p}{\partial x}\left(1+6\frac{\lambda}{h}\right)\right]+\frac{\partial}{\partial y}\left[ph^3\frac{\partial p}{\partial y}\left(1+6\frac{\lambda}{h}\right)\right]=6\mu U_r\frac{\partial}{\partial x}(ph) \qquad (3)$$

Equations (2) and (3) are modified forms of Reynolds' equation of lubrication applicable in the "slip flow" region $(0<\frac{\lambda}{h}<<1)$.

Equation (3) has been converted to a dimensionless form and has been solved for the plane slider bearing with no side flow and for the Rayleigh step bearing, also with no side flow, using a small parameter, perturbation technique. A solution could also be obtained for the journal bearing by replacing the x coordinate by $r\phi$.

A typical curve is shown below indicating the variation in pressure generation for a straight slider bearing as a function of "m", where "m" is the ratio of the molecular mean free path to the film thickness at the entrance of the bearing. (Fig. 2)

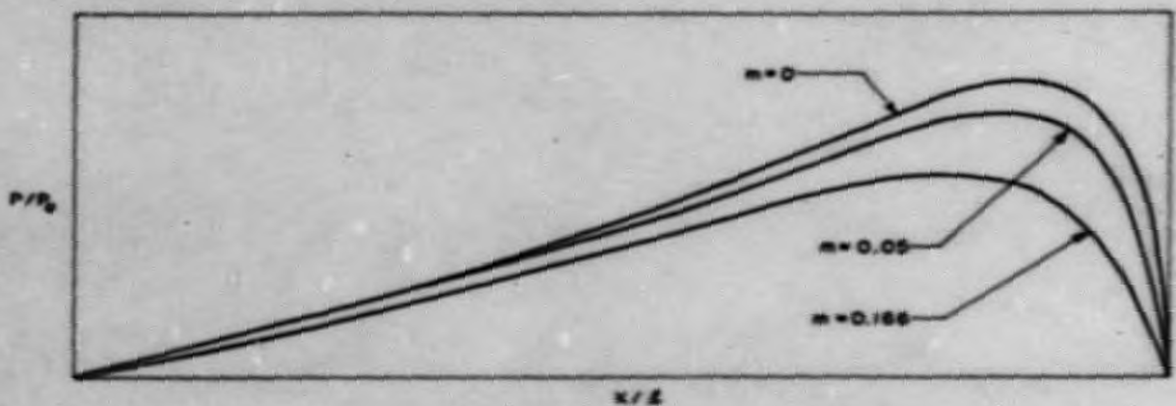


Figure 2. Pressure Distribution for Compressible Fluids along a Straight Slider Bearing

It is concluded that where values of "m" are greater than 0.01 a noticeable effect on bearing pressure and load-carrying capacity may be expected.

Typical values of mean free path at atmospheric pressure are:

Hydrogen	4.43×10^{-6} inches
Helium	6.77×10^{-6} inches
Air	2.32×10^{-6} inches

Thus for m = 0.01 for air at atmospheric pressure, the film thickness at the entrance to the bearing would be of the order of

2.32×10^{-4} inches. Such values are not uncommon for hydrodynamic bearings. For low ambient pressures however, the effect would be more marked.

The influence of mean free path on friction loss has also been evaluated.

This information will be presented as a technical paper at the Joint Conference on Lubrication of the ASME and ASLE in October 1958 and has been issued as Interim Report I-A2049-2 on the ONR program.

The influence of surface deviations on bearing performance has not yet been studied sufficiently to permit drawing any final conclusions. However, the work shows that a wavy shaft centered in a journal bearing will have the same average pressure distribution as an ideal round one.

Errors introduced by applying Reynolds' equation to regions close to the edges of the bearing have not yet been investigated.

At the present time, most effort is being directed toward developing a theory of whirl instability for hydrodynamic, gas-lubricated bearings.

Experimental

First might be mentioned the gas-lubricated rotor for the proposed closed-cycle gas turbine. This was constructed previously under Contract DA-44-ENG-2562 for the U.S. Army Engineer Research and Development Laboratories. Permission was received to use this rig under the ONR contract and, although it is a combination of hydrodynamic and hydrostatic lubrication it is described, in part, in this section of the annual report. Although only limited tests have been conducted on this machine, the journal bearings proved to be completely adequate for the expected operational speed of 18,000 rpm. The rotor was actually run accidentally at a top speed of 21,000 rpm. The journal bearings are tilting-pad bearings with two tilting pads under the shaft and one at the top of the shaft or journal. The two lower tilting pads are pressurized so that the bearing is essentially a combination hydrostatic-hydrodynamic type. There are two such journal bearings. One is lightly loaded with a force of about 15 pounds, the other is more heavily loaded and carries a force of about 50 pounds.

It was observed that lift on the lightly-loaded journal bearing was achieved with a supply pressure of from 20 to 25 psi. On the heavily loaded bearing, a pressure of 50 to 65 psi was required to attain initial lift. Normal running pressure to these bearings was 35 psi and 75-80 psi respectively. When operating at rated speed the supply pressure to the lightly-loaded bearing was cut off and that bearing ran hydrodynamically. The hydrodynamic action of the bearing established its own film, and pressure was measured to be from 8 to 14 psi. The heavily-loaded radial bearing would not operate hydrodynamically without supplemental hydrostatic pressure. A vibration of the top shoe (non-hydrostatic) was observed to occur at about 5,000 rpm. This was not severe and did not seem to cause any difficulty. This mild form of instability will be looked into at a later date.

The thrust bearing was originally designed to operate hydrodynamically but the thrust load of about 300 pounds would produce an average unit pressure on the shoes of the thrust bearing approaching 19 psi. This is somewhat beyond the limit of safe hydrodynamic operation so that the bearing has been converted to a hydrostatic type. The design is not yet completely satisfactory and will have to be revised and improved. The full thrust load has not yet been carried on this bearing but with suitable modification this should be accomplished. The drive on this unit is presently an air turbine.

The EHDL has recently advised that the specifications on this gas-turbine rotor have been changed resulting in larger bearing loads in most cases.

Specifically, the bearing loads have been changed as follows:

Bearing	Original Design	Revised Design
Compressor End Journal	13.6 lbs	3 lbs
Turbine End Journal	49.7 lbs	92 lbs
Thrust (Normal Power)	325 lbs	1134 lbs
Thrust (125% Power)	-	1628 lbs
Journal Bearing Diameter	2.25 in.	3 in.

A new rotor test rig will be designed and built in terms of these revised specifications and this should be ready for test by the end of 1958.

. . . .

A high-speed journal-bearing testing machine has been designed but is not yet in production. This device has been considered for operation up to 150,000 rpm with journal diameters up to one inch and shaft lengths from about 6 to 12 inches. It is expected that both hydrodynamic and hydrostatic theory and design can be investigated on this test rig. It will also provide an opportunity to determine the results of the interaction of the dynamics of the high-speed rotating system and the gas bearings that are involved.

. . . .

A simple screening test device has been acquired consisting of high-speed, air-lubricated and air-driven, grinding spindles. One is capable of operation at 50,000 rpm and the other at 100,000 rpm. Some preliminary tests have been made using a simple, single sleeve bearing supported by the cantilever shaft of these spindles. This bearing can be loaded directly by means of a small dead weight and information obtained on load-carrying capacity and whirl instability.

HYDROSTATIC LUBRICATION

Analytical

Externally-Pressurized Circular Thrust Bearings

The mathematical analysis of an externally-pressurized, circular thrust bearing leads to equations that are rather cumbersome to use for design purposes. The emphasis on this phase of the work was to present these equations in non-dimensional, graphical form in order to reduce the amount of labor required for their solution. These curves are for the case of laminar flow in the film. Inertia effects on flow are not included. A subsequent analysis is being made which will include inertia effects on flow and pressure development in the bearing.

For non-recess bearings, (those bearings that do not have a high-pressure cavity to help support the bearing load), three graphs have been evolved. The first graph relates the load-carrying capacity of the bearing to the pressure ratio across the bearing sill. Once a pressure ratio for a given bearing is known, the load can be easily computed, or conversely, if given load must be supported by a given bearing, the graph will show the pressure ratio that must be maintained, in order to support the load.

The second graph relates the flow through an isentropic nozzle to the pressure ratio across the nozzle. In the case of a non-recess bearing the nozzle consists of the narrow annulus around the circumference of the inlet hole of height equal to the film thickness in the bearing. By assuming a pressure ratio, a flow parameter is found which when used in conjunction with a constant obtained from the third chart, the flow, film thickness, and supply pressure can be calculated with minimum effort. If the results from the first calculation are not satisfactory, a new pressure ratio can be selected from the second chart, new values of the flow parameter found, and a new flow and film thickness obtained.

The recess-type bearing creates a new problem in that the assumption is made that a second nozzle is added in series with the first. The first nozzle is now defined either by the entrance hole diameter alone, or by the annulus around the circumference of the inlet hole of height equal to the film thickness plus the recess depth, the choice being governed by the relative magnitude of either one, the smallest being used in the calculations.

The second nozzle is described by the annulus around the circumference of the recess and of height equal to the film thickness.

The assumption of two nozzles in series leads to two new graphs, one the relationship of pressure ratios to area ratios of the two nozzles in series, the other, a new load parameter-pressure ratio relationship.

The calculations for the design of a recess bearing become somewhat more involved than for a non-recess bearing. An assumption must be made for the pressure ratio across either the first or second nozzle, along with an area ratio. An assumption of pressure ratio across the bearing sill is made, and from the load parameter chart a load quantity is found that is substituted into an equation that relates the actual bearing load to the bearing geometry, the pressure ratio and the load parameter. A trial-and-error solution results in a fixed pressure ratio.

With the assumptions of nozzle ratios and the use of the chart for two nozzles in series, all pressures can be found and the remainder of the problem can be solved exactly as a non-recess bearing. It is expected that this information will become part of a technical report to be issued on this type of hydrostatic thrust bearing.

Instability of Hydrostatic Type Bearings

A second, but independent, effort is being made concerning the instability of hydrostatic type bearings. This form of instability has been called pneumatic instability. The approach to the problem has been to establish general equations for stability and apply these equations first to the incompressible case. The analysis is made for a unidirectional, circular thrust plate having a central recess. This is shown diagrammatically in Figure 3.

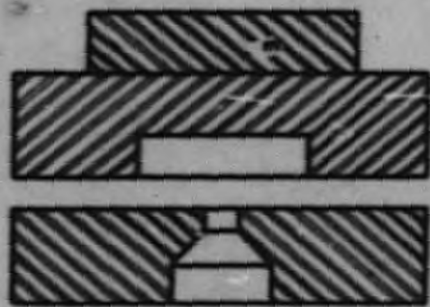


Figure 3

This analysis can be extended to thrust bearings capable of supporting loads in both directions and having different pocket arrangements. It is assumed that the bearing is restrained to permit only axial relative motion and that a truly incompressible fluid of constant viscosity is supplied from a constant-pressure reservoir. External drag effects are not included in the equation of motion.

A simplified version of the Navier-Stokes equation (as applicable to laminar, viscous flow) in conjunction with the continuity equation describe the flow in the annular and recess regions of the bearing. After the pressure terms are eliminated there results a non-linear differential equation in terms of the dimensionless displacement H and its time derivatives. $(H = h/h_o)$

$$H^3 \left[C_1 \ddot{H} + C_2 \frac{\dot{H}}{H^3} + C_3\right] = \left[C_4 + C_5 \frac{\dot{H}}{H^3} + C_6 \ddot{H}\right]^{1/2} \qquad (1)$$

Here the C's represent constants which include the bearing and fluid parameters. Their explicit expressions are included in the complete paper of which this is a very brief abstract.

This equation has been solved for a number of cases using an electric-analog computer as shown below. However, in order to provide more insight into both the nature of the problem and the expected method of solution for the more complicated compressible case, equation (1) was linearized yielding a second order differential equation in terms of the perturbation displacement Δ and its time derivatives.

Thus,

$$h = h_o + \delta \qquad (\delta \ll h_o)$$
$$H = 1 + \Delta \qquad (\Delta \ll 1)$$
$$\dot{H} = \dot{\Delta}$$
$$\ddot{H} = \ddot{\Delta}$$

Introducing Δ and its derivatives in equation (1) and neglecting their squares and cross-products as quantities of higher order, equation (1) reduces to:

$$\ddot{\Delta} + \left(\frac{2C_2 C_3 - C_5}{2C_1 C_3 - C_6}\right)\dot{\Delta} + \left(\frac{9C_3^2 - 3C_4}{2C_1 C_3 - C_6}\right)\Delta = \frac{C_4 - C_3^2}{2C_1 C_3 - C_6} \tag{2}$$

but since $C_4 = C_3^2$, there results

$$\ddot{\Delta} + \beta\dot{\Delta} + \theta\Delta = 0 \tag{3}$$

The dimensional equivalent of equation (3) is

$$m\frac{d^2\delta}{dt^2} + b\frac{d\delta}{dt} + k_D\,\delta = 0 \tag{4}$$

The conditions for stability are that the coefficients of equation (3) be positive, or $\beta > 0$ and $\theta > 0$. Upon examination it is shown that this is true and the bearing will consequently be stable for all ratios of radii and pressures. Stability is defined by saying that small oscillations will not grow with time but will instead reduce in amplitude.

Figure 4 shows some typical analog computer solutions using non-linear Equation (1) and these are compared to the results from the linearized Equation (3). The similarity between the graphs representing the two solutions is rather remarkable. In addition the linearized equation appears to represent the motion of the bearing quite well even beyond the region of small displacements.

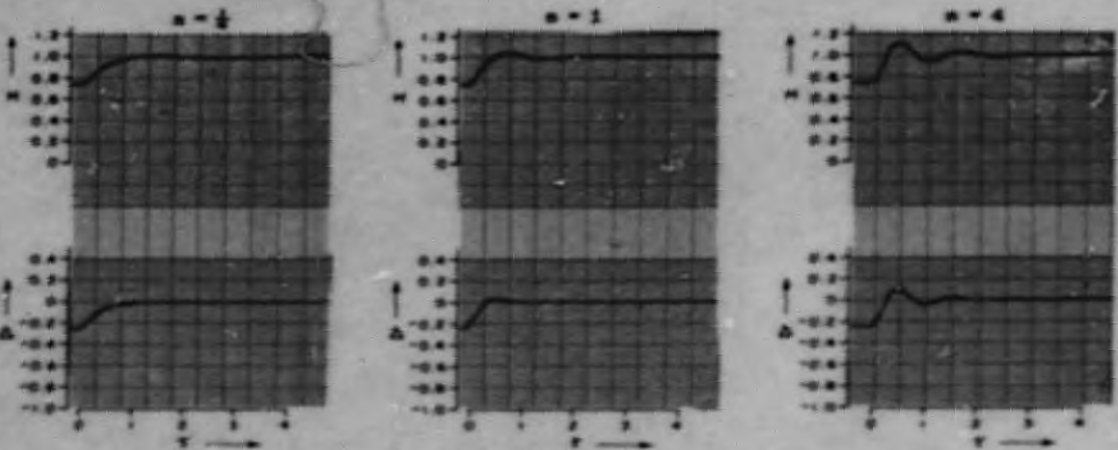


Figure 4

Although this work was preliminary to the attempt to analyze the more general problem of the gas-lubricated thrust bearing, it is felt that it has yielded background information on methods of solution, and has also provided a means for finding local stiffness, damping constants and dynamic response of hydrostatic bearings operating with liquids. The complete paper is scheduled for presentation at the Joint Conference on Lubrication of ASME and ASLE in October of 1958 at Los Angeles, and will also be a part of a technical report on this phase of the work, to be issued as part of our project activity.

Related Investigations

As an extension of the analytical approach to hydrostatic lubrication, a program of investigation is now being planned on both the theoretical and experimental level, to explore the basic phenomena associated

7

with compressible flow through narrow slots and passageways and especially at the entrance to such flow restrictors. This kind of information appears to be basic to any further extension of hydrostatic theory, thinking in terms of both the flat thrust type of bearing and the cylindrical journal bearing.

Experimental data taken from our test rig indicate pressure drops at the entrance to the sill of the bearing which are larger than one would expect from nozzle theory. It is believed that these large pressure drops may be caused by undeveloped flow at the entrance to the sill area of the bearing. An experimental rig is being designed to aid in our understanding of this phenomenon. Theoretical studies have also been started.

Experimental

A test rig for the circular type of thrust bearing, 5 1/2 inches in diameter, was built early in the year and has been used to gather data of pressure distribution and flow. Supply pressures up to 800 psig have been used. Maximum film thickness was 0.0035 in. Our experience with this machine has led us to the realization that in order to check out precisely our hydrostatic theories for gases we would need to have a greater degree of rigidity of the test faces, less distortion of flow from the bearing, and a more exact way for obtaining a continuous pressure profile across the bearing sill. Accordingly, a new design has been made and is presently being fabricated.

The ERDL high speed gas turbine rotor model mentioned above has also served as a test unit for our experimental hydrostatic work.

DIGITAL COMPUTER PROGRAM

The ultimate objective of these efforts is to develop a digital computer solution for the pressure distribution, and associated quantities, in bearings using compressible lubricants. As a preliminary step in this direction, it was decided to develop such a solution for incompressible bearing lubricants.

The present status of these efforts is as follows: A digital computer program has been developed, for the incompressible case, for each of these types of bearings: journal, slider and sector-shaped thrust. The three programs are similar in nature and differ only in the way they handle quantities and variables peculiar to each of the three bearing types. These programs are being checked out and are not yet to the point where they can be used to obtain data. It is reasonably certain that the slider program will be completed very shortly and the journal and thrust program soon thereafter.

While these routines have been in the checking-out stage, work has proceeded on formulating or flow-charting the method of machine solution for the compressible case. This will be a more general solution, considering variable density, and will allow choice of grid size limited only be the cost of machine time which would be required for solution. The basic plan of solution has been completed and the next step will be to translate it into machine language or code. It is expected that this second development should be ready to begin turning out data, for one type bearing, in several months.

CONCLUSIONS

In conclusion, it appears that the first year of effort on this program has been productive in a number of ways.

A network of communications has been established with many fellow investigators of gas bearings in this country and in other parts of the world.

A central file of technical papers on gas bearings has been established which includes many translations of foreign language papers.

An extensive (interim) bibliography has been prepared and issued including an abstract of each referenced item.

Three papers have been written and are expected to be published in the technical literature. They will also be issued as technical reports as part of the operation of this contract.

The investigations of a number of other phases of our general program are well under way and they have already benefited from the experience gained during the first year's work.

Dudley D Fuller
Dudley D. Fuller
Principal Scientist

Approved by:

George M. Robinson
George M. Robinson
Branch Head
Friction and Lubrication

N H Smith
for Francis L. Jackson
Technical Director

Nicol H. Smith
Nicol H. Smith
Director, Laboratories
Research and Development

SUPPLEMENT TO ANNUAL REPORT

COMMENTS ON PHASES OF GAS BEARING LUBRICATION THAT NEED INVESTIGATION
AND EXTENT OF PRESENT RESEARCH ACTIVITY

COMMENTS ON PHASES OF GAS BEARING LUBRICATION THAT NEED INVESTIGATION AND EXTENT OF PRESENT RESEARCH ACTIVITY

Through the contacts that have been made by members of The Franklin Institute, the Advisory Group, and others, it appears that there are about 20 companies in this country that are either engaged in research dealing with gas bearings or are contemplating such activity. News of research in England and on the continent also indicates some degree of activity in these areas.

From this information, and also from a study of the available literature, an impression can be gained regarding the extent of the problems that still need solution in the field of gas bearing lubrication. Some of these ideas have been suggested by Dr. Gilbert Boeker as a result of his literature investigation on a WADC program at Columbia University. It appears that the following theoretical and experimental phases of gas bearing lubrication need to be investigated.

Hydrodynamic Lubrication

a. Experimental verification is needed of thrust bearing theory for the spiral-grooved type, the herringbone-grooved type, the inclined-plane type and the Rayleigh-plate type of gas-lubricated thrust bearing. The experimental determination of side-flow coefficients for the latter types of bearings is needed unless these coefficients can be obtained by direct analytical or digital computer methods. Experimental verification of the theory is needed when the pressure is not atmospheric nor the gas air. The determination of the lowest pressure at which the theory begins to fail should be made. This point should be predictable on a theoretical basis, as for example, as has been suggested by one of the papers developed in our program. Some of the work of a preliminary nature has appeared in British publications. All of the above can be done with a single piece of apparatus. It is to be considered the fundamental experiment, since it provides a critical test of the hydrodynamic theory of gaseous lubrication without the complications that arise from journal bearings.

A divergence between theory and fact with journal bearings may well be due to the mathematical approximations required by its application to the journal geometry and may not be due to the fundamental theory itself. Also, it is more difficult to manufacture, measure and maintain small clearances in journal bearings to the degree of accuracy required for exact verification of theory.

The current program at The Franklin Institute is fairly extensive. A number of the tasks outlined in this summary are being undertaken, or will be investigated in the future. It must be recognized, however, that there are many problems in establishing what might be called a gas-bearing technology, and that our program is not large enough to get the complete answers to all of these problems in a short time. However, the group here is on the alert for information from other sources which may serve to supplement the work that is currently under way or even reduce the number of tasks that remain to be done.

In the next year or so, we contemplate no work in the hydrodynamic area (a) described above.

b. Tests on the performance of single, pivoted shoes with gas lubrication should be made. These would be for pivoted-shoe journal bearings. From such tests of single shoes the bearing characteristics of shoe journal bearings with multiple shoes can be predicted. Some of this information may be made available from a British paper, presently classified.

This type of bearing should be the most useful of the hydrodynamic types at high speeds and low loads because of its greater stability.

Awaiting the release of information on the work that has already been done on this type of bearing, we are not planning to begin any further research unless it is found to be necessary.

c. Tests of plain full journal bearings and grooved journal bearings should be made. Both of these types can be tested with the same apparatus. The most significant tests, practically speaking, would be on the grooved journal bearings, since they tend to be far more stable than the plain full journal bearing at high speeds of rotation.

The tests on plain full journal bearings would, however, have some interest in themselves, and would also be of value as proof or disproof of some work in this field that appears to be controversial. The full journal bearing is the object of some activity on the part of several companies in the U.S.A. and some research has also been done on this bearing in England. However, no comprehensive publication on this bearing has as yet appeared.

Besides the theoretical work described in the Annual Report, a completely instrumented test rig has been designed for journal bearing evaluation. This should serve to provide accurate test data for hydrodynamic journal bearings.

Two supplementary journal bearing test devices without extensive instrumentation, are also available.

d. The most important tests of hydrodynamic bearings, for high-speed, lightly-loaded operation, should be concerned with self-induced, half-frequency whirl characteristics. A general criterion for stability should be established and the range of validity of this criterion should be experimentally determined. We are making a start on this phase of the work, and if successful, should produce a stability criterion for gas-lubricated hydrodynamic journal bearings, with a rigid journal.

It has been predicted that even gas-lubricated shoe journal bearings will whirl if the angular speed and rotor weight are sufficiently small. This prediction should also be tested.

A machine should be designed with a gas-bearing-supported shaft which should be stable at say 50,000 rpm or even higher when run in air at atmospheric pressure when the shaft is vertical or the journal bearings are unloaded. The bearing characteristics and hence the stability characteristics of such a machine could then be changed by varying the lubricating gas, the ambient gas pressure or the orientation of the shaft so as to load the journal bearings. The changing theoretical and experimental stability characteristics should then be compared.

The instrumented journal bearing test rig described above in Section (c) will be used for collecting whirl instability data to supplement our theoretical investigation of this form of instability.

Hydrostatic Lubrication

Basic information is needed on the flow of gases through narrow slots and channels and through orifices and other nozzle-type restrictors. Essentially what is needed is a thorough study of what might be called "microaerodynamics". This is needed on both the theoretical level and the experimental level in order to understand and be able to predict the action of gases in hydrostatic or externally-pressurized bearings of various kinds, and to be able to analyze these bearings for their operating and performance characteristics. This is essential for a fundamental approach to the problems listed below.

We are planning to attack this problem in a comprehensive fashion as described in the Annual Report.

a. A self-excited vibration has long been recognized as being quite common in hydrostatic gas bearings and has been called pneumatic instability or pneumatic hammer. Several papers have recently been concerned with this form of instability so that more light is being shed on it. A concentrated effort is being made at The Franklin Institute to break through with a rigorous analytical solution supported by some carefully obtained experimental data. If successful, this will be an important contribution in itself and also for the light shed on associated aspects of film spring stiffness and damping. However, this work, and that described in the papers that were just mentioned, concerns itself only with the flat circular disc type of hydrostatic thrust bearing. Additional effort will be needed on journal bearing geometry, and our efforts will be directed toward this form of bearing probably within the next year.

b. The slots, channels, orifices, etc., mentioned above will cause the gas flow to induce a variety of pressure gradients. These in turn will establish the load-carrying capacity of the bearing. The rate of change of pressure at a given stationary point in the bearing, as the film thickness changes, will determine the stiffness of the gas film in the bearing. With this information, the response of the bearing due to dynamic disturbances may be predicted. The establishment of parameters for load-carrying capacity and for bearing film stiffness will be important for both thrust-type and journal-type bearing. A portion of this work on a limited scale is being done by one company. Some background information may be obtained from one or two British papers, but what is now known on this phase of the subject must be considered as very rudimentary. The evaluation of damping in such bearings must be determined as well.

Our efforts along these lines have been described in the Annual Report. Our approach will be on both the theoretical and experimental level. Some results may also be forthcoming, of an experimental nature, incidental to the operation of the high-speed journal bearing rig that has been designed.

c. Fluid-film whirl with hydrostatic bearings is essentially a complete unknown. It seems, on the basis of some limited work done by one company, that the half-frequency type of whirl which has been recognized in hydrodynamic bearings also exists in hydrostatic bearings, but to complicate the picture, it appears to begin over a rather wide spectrum of frequencies and also appears to have various ratios of its own orbital frequency to that of shaft speed, not just one half as is so often the case with hydrodynamic bearings with rigid shafts. A machine of the type described briefly in Section (d) above would probably suffice very nicely to provide an experimental testing device.

The hydrostatic type of journal bearing should prove to be a very practical type of bearing, in that it separates the rubbing surfaces before starting, can carry a relatively high unit load, and can be made to be a stable bearing and suppress whirl and other similar instabilities, but it has many unknown aspects in both theoretical and experimental phases. These will have to be understood before one can expect any better than a trial-and-error solution for each bearing configuration that is to be considered.

It is hoped that these comments convey some idea of the extent of the work that needs to be done in the gas-bearing field and also provide some idea of the extent of present research activity.