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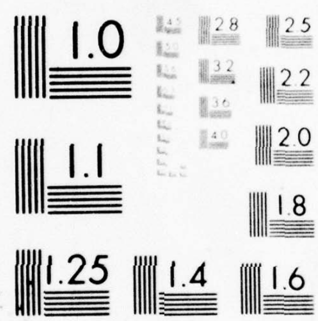

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THE SWING-PAD BEARING - A NEW CONCEPT IN SLIDING-SURFACE BEARINGS

DAVID W. TAYLOR
NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER
Bethesda, Md. 20084

THE SWING-PAD BEARING - A NEW CONCEPT IN SLIDING-SURFACE BEARINGS

By
J. Greene

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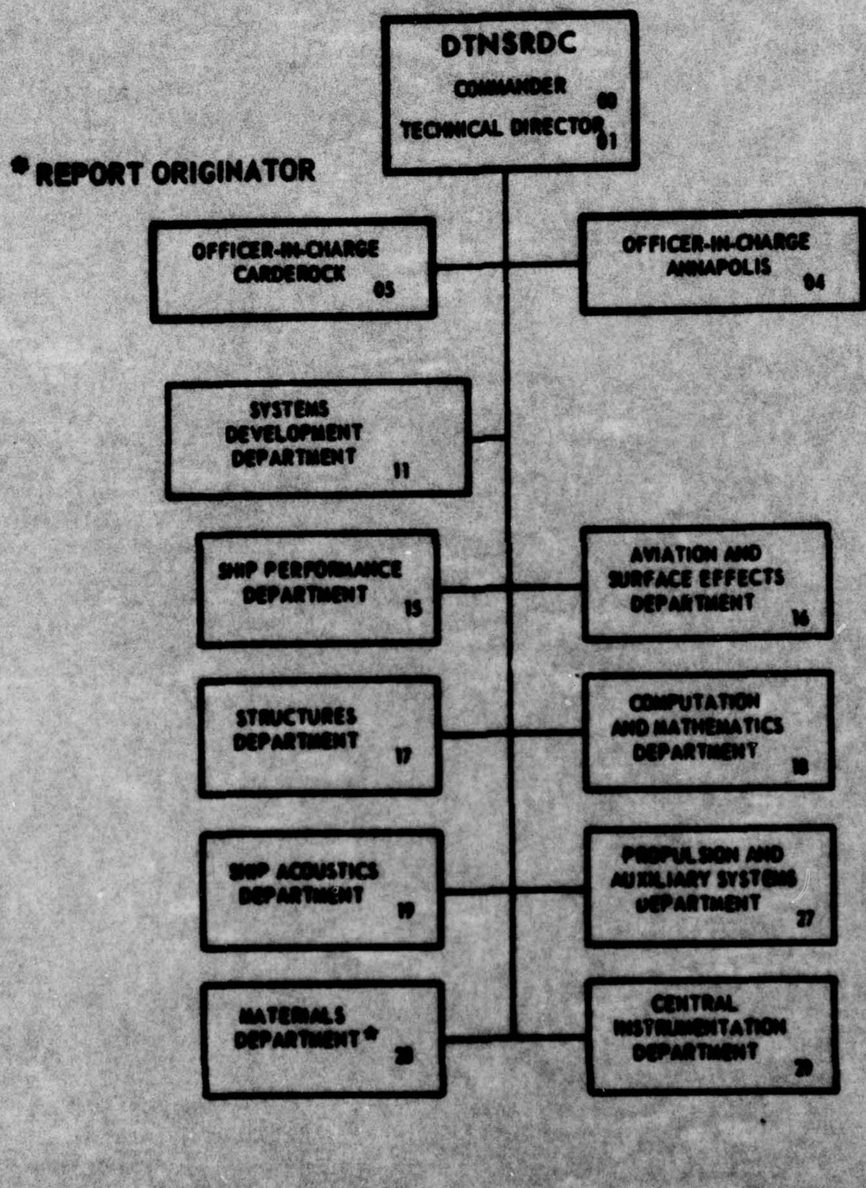

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This report introduces a new bearing concept, the swing-pad bearing. The concept is a sliding-surface lubricated bearing in which the stationary bearing surface is segmented into pads that can displace through a motion resembling a swing to form a wedge-shape converging in the direction of motion with the rotating member. Viscous friction force moments are in the direction to

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Abstract (Cont)

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(Author)

ADMINISTRATIVE INFORMATION

The swing-pad bearing investigation described in this report was conducted under the Laminated Elastomer Advanced Design Program, Work Unit Summary SF54-543-501, Task 17465, Work Unit 2832-160, 21 May 1973. The program was terminated at the conclusion of fiscal year 1974 under Work Unit Summary SF54-543-706, Task 17465, Work Unit 2832-160, 1 May 1974. United States Patent 3,930,691 on the swing-pad bearing concept was issued on 6 January 1976.

The Laminated Elastomer Advanced Design Program was sponsored by the Naval Sea Systems Command (SEA 035). The Naval Ship Engineering Center (SEC 6101E), Hyattsville, Maryland, was the Technical Agent for the program.

LIST OF ABBREVIATIONS

e.g.	- for example
°F	- degrees Fahrenheit
°C	- degrees Celsius
in/sec	- inch per second
lb/in^2	- pounds per square inch
mm	- millimeters
MN/m^2	- meganewtons per meter square
rms	- root mean square
r/m	- revolutions per minute

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ABSTRACT

This report introduces a new bearing concept, the swing-pad bearing. The concept is a sliding-surface lubricated bearing in which the stationary bearing surface is segmented into pads that can displace through a motion resembling a swing to form a wedge-shape converging in the direction of motion with the rotating member. Viscous friction force moments are in the direction to form the converging wedge-shape; fluid pressure force moments oppose wedge formation. The pad bearing surface stabilizes at a position where these moments are in balance. Exploratory experimental investigations conducted on model elastomer surface swing-pad bearings indicate exceptional performance characteristics. These characteristics are: (1) low coefficients of friction representative of fluid lubrication under load/speed/fluid viscosity combinations that would produce boundary lubrication in conventional bearings; (2) operation at rotational start-up and at extremely low speeds with applied loads that would cause stick-slip motion and squeal in conventional bearings.

INTRODUCTION

It is the purpose of this report to introduce a new concept in sliding-surface lubricated bearings, the swing-pad bearing. Exploratory experimental investigations, initiated under the Laminated Elastomer Advanced Design Program, indicate performance characteristics that are exceptional in the art of lubricated bearings.

The swing-pad bearing is a pad-type lubricated bearing with the pads supported by curved alternate laminates of elastomer and metal. Figure 1 shows the general construction of the journal and thrust versions of the swing-pad bearing.

A brief review of conventional hydrodynamic bearings and their performance characteristics is presented as an introduction to this new hydrodynamic bearing concept.

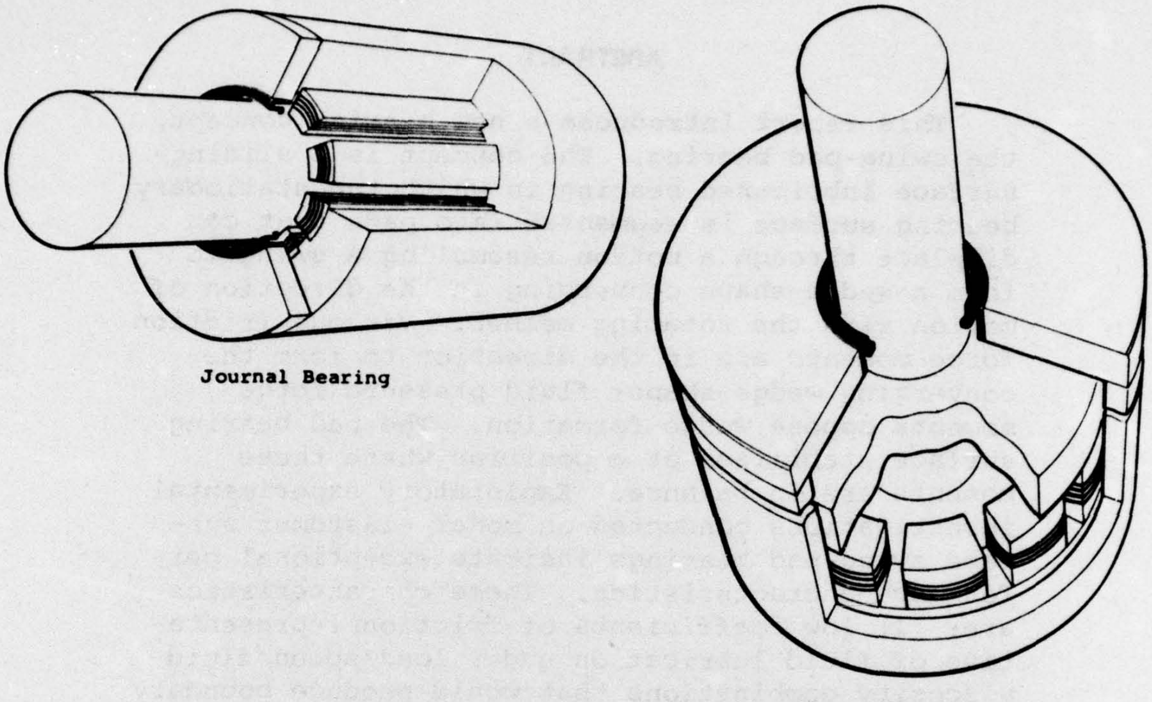


Figure 1
Swing-Pad Thrust and Journal Bearings

THE HYDRODYNAMIC BEARING

The three types of lubrication between two sliding surfaces in loaded contact are boundary lubrication (complete support of the load through contact of surface asperities), mixed film lubrication (partial support of the load on a fluid film), and fluid lubrication (complete fluid separation of the surfaces). Fluid lubrication avoids surface damage and wear. Mixed film lubrication may result in damage to the surfaces, e.g., at the start of the sliding motion. Boundary lubrication is usually accompanied by some degree of wear.

Fluid lubrication between two sliding surfaces is attained by generating pressures within the lubricant film to transmit the applied load and maintain surface separation. The two basic methods for attaining load-supporting fluid pressures are application of external hydrostatic pressure and generation of the pressures hydrodynamically by the geometry and motion of the surfaces.

The hydrodynamic bearing generates fluid pressures by the rotating member drawing the fluid into a film formed between itself and the stationary member. The film for bearings with rigid surfaces - the journal bearing, figure 2; the tapered-land thrust bearing, figure 3; the tilting-pad thrust and journal bearings, figures 4 and 5; has a converging wedge-shape.

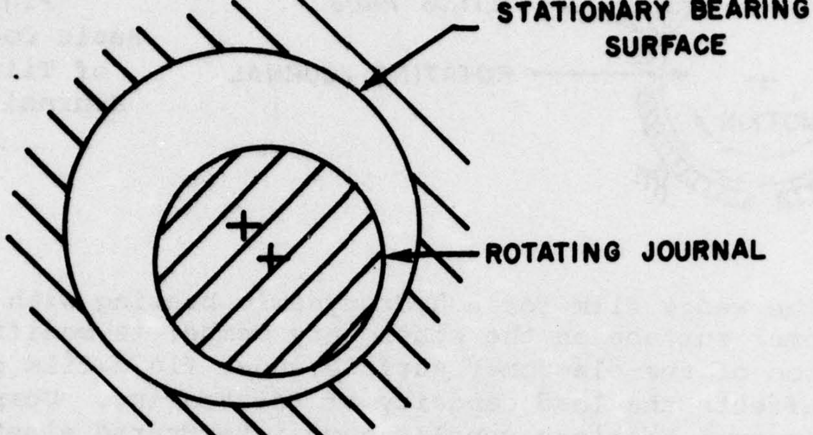


Figure 2
Typical Journal Hydrodynamic Bearing
(Bearing Clearance Exaggerated)

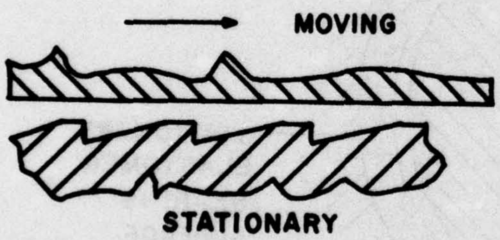
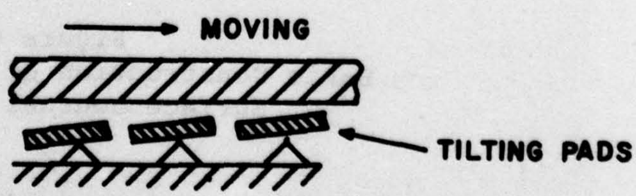


Figure 3
Basic Construction of the Tapered-Land Thrust Bearing

Figure 4
Basic Construction of Tilting-Pad Thrust Bearing



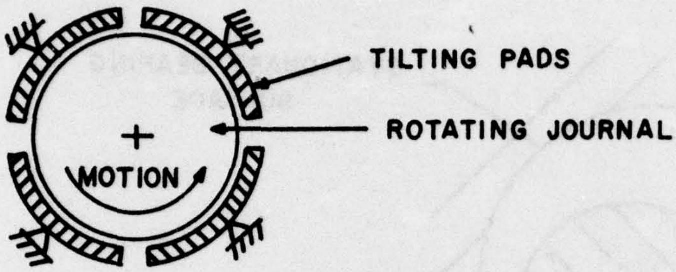


Figure 5
Basic Construction
of Tilting-Pad
Journal Bearing

The wedge film for a hydrodynamic bearing with a compliant elastomer surface on the stationary member is modified by compression of the elastomer surface under fluid-film pressure. This affects the load capacity of the bearing. Compliant surface journal bearings usually have interrupted elastomer surfaces, creating in effect a set of partial bearings sharing the load, figure 6. Compliant surface thrust bearings are not commonly used.

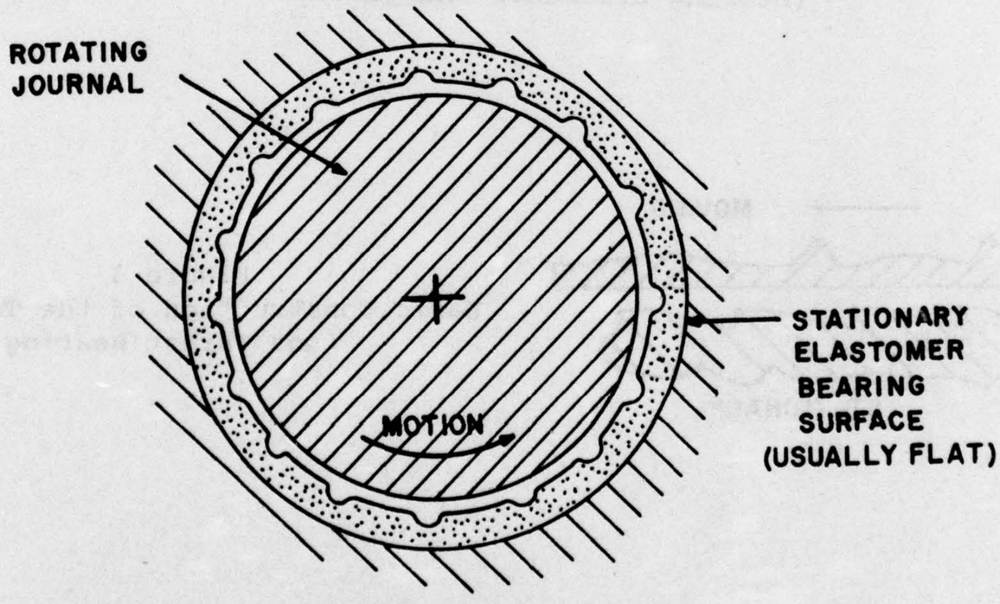


Figure 6
Basic Construction of the Compliant
Surface Journal Bearing

The performance characteristics of the rigid surface bearings are:

- Load capacity of the journal and the tapered-land thrust bearing is directly proportional to speed and fluid viscosity. The tilting-pad bearing adjusts to increase load capacity at off-design speeds.

- Load capacity at design speeds is high.

- Extended life is normal.

- Hydrodynamic load capacity at rotational start-up is zero. Start-up can occur without serious bearing surface damage if a lubricant with good boundary lubrication characteristics or with a high viscosity is present within the bearing and if the load applied before start is limited.

- Stick-slip occurs at low speeds (boundary or mixed film conditions) particularly with low viscosity lubricants (water).

- Sensitivity to misalignment is high.

The performance characteristics of the compliant elastomer surface bearings are:

- Load capacity is a complicated function of the speed, fluid viscosity, and the stiffness of the elastomer surface.

- Load capacity is low as compared with rigid surface bearings.

- Load limit for start-up with loads applied before start is low.

- Stick-slip predominates at low speeds under boundary or mixed film conditions.

THE SWING-PAD BEARING

The swing-pad bearing is similar to the tilting-pad hydrodynamic bearing in that the nonrotating bearing surface is segmented into pads that displace to form a converging wedge shape with the rotating member. The difference between the bearings is that the tilting-pad surface tilts about a center located behind the pads, whereas the pad surface on swing pads

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*Definitions of abbreviations appear on page i.

displaces through a swing-type motion about a center located in front of the pad surface. Bearing friction, therefore, tends to close the lubricating wedge of the tilting-pad bearing, whereas it tends to open the swing-pad bearing wedge.

Figures 7 and 8 show the construction of the thrust and journal versions of the swing-pad bearing. The material of the bearing surface on the pad is one conventionally used in sliding-surface bearings, e.g.,* babbitt metal for oil lubrication, elastomer for water lubrication. The bearing surface is supported by curved segments of bonded alternate metal-elastomer laminates, with the center of curvature located above the bearing surface on the center line of the pads. The curvature of the laminates in a thrust bearing is spherical and in a journal bearing cylindrical or spherical, The bearing surface swings in the direction of the curvature.

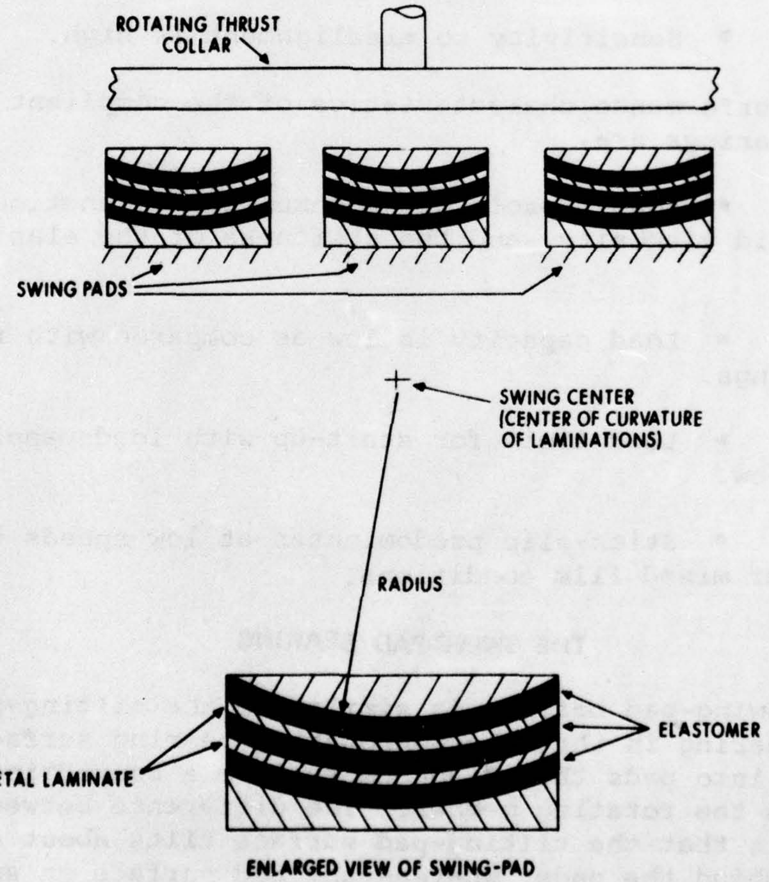


Figure 7
Basic Construction of Swing-Pad Thrust Bearing

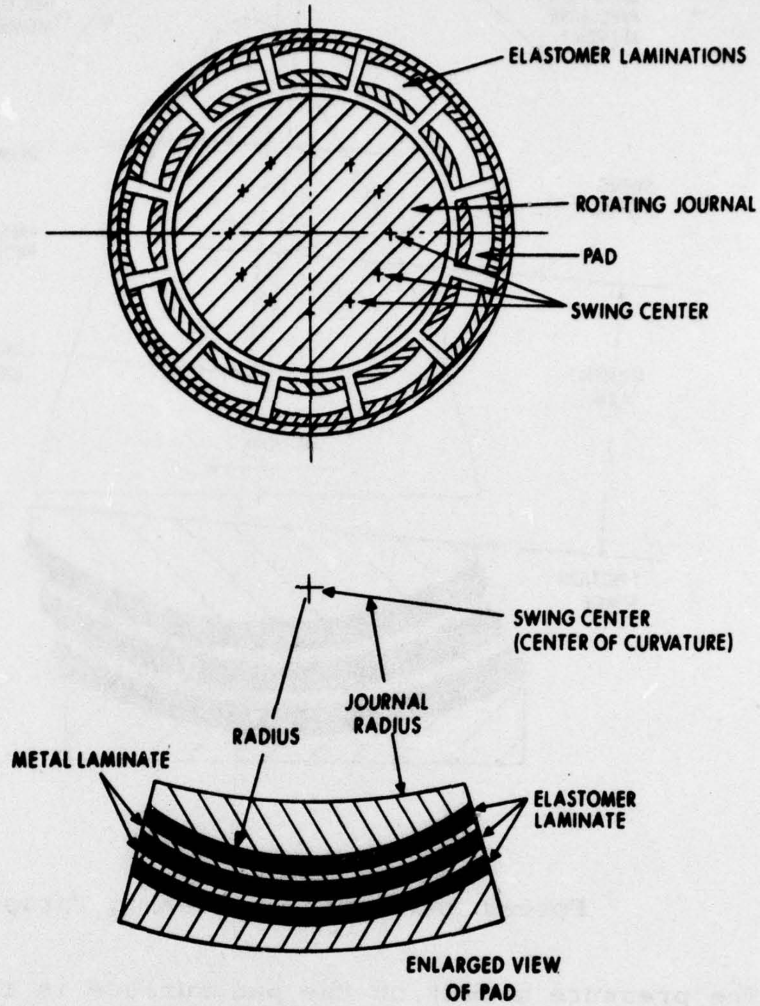


Figure 8
Basic Construction of Swing-Pad Journal Bearing

As wedge-shape angles are minute, the elastomer shear displacement and the related shear resisting forces to form these angles are correspondingly minute and can be ignored in the operation and the analysis of the bearing. The laminated elastomer support returns the pad surfaces to the level neutral position when the applied load is removed, ensures that the pad surfaces displace in unison, promotes equal load sharing between the pads by elastomer compression, and completely supports the pad surface.

Figure 9 shows the fluid pressure plus friction force and moment vectors on a swing thrust pad during rotation. Forces and moments on a journal pad are similar.

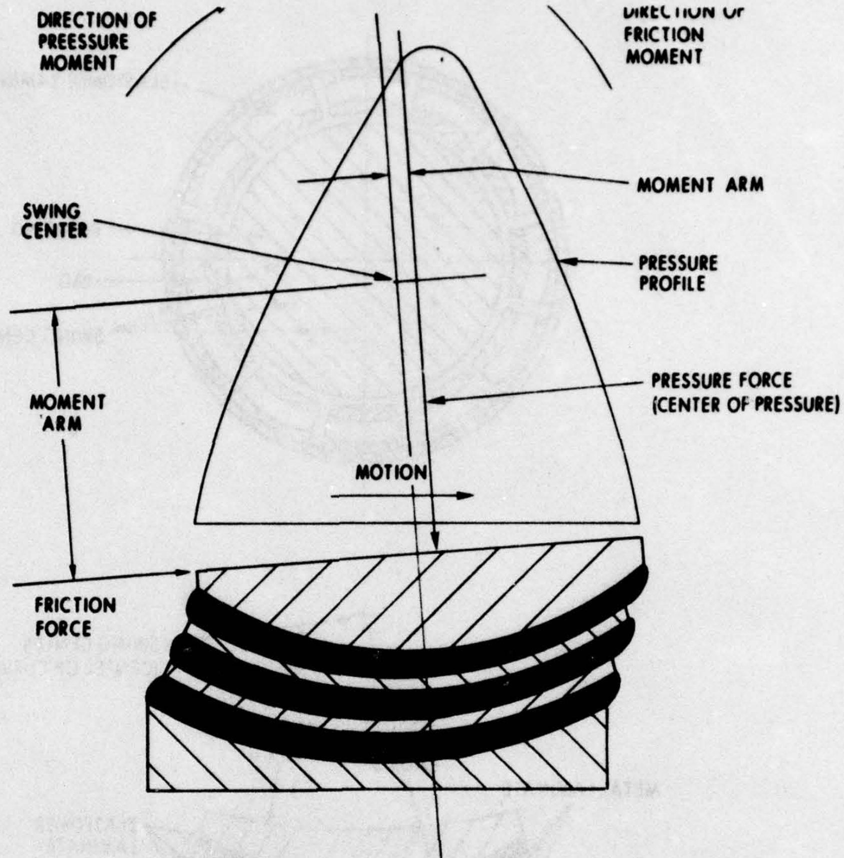


Figure 9
Forces and Moments on Swing Thrust Pad

The pressure moment on the pad surface is in the direction to decrease the wedge-shape angle. The friction force moment on the pad surface is in the direction to increase the wedge-shape angle. A converging wedge shape in the direction of motion is always formed. The pad surface swings and stabilizes at a converging wedge angle where the moments are in equilibrium.

Figure 10 shows the wedge-shape formed by the friction force moment at a rotational start-up with an applied load. The swing motion forms the wedge-shape prior to sliding, the lubricant enters, and sliding draws the lubricant between the bearing surfaces.

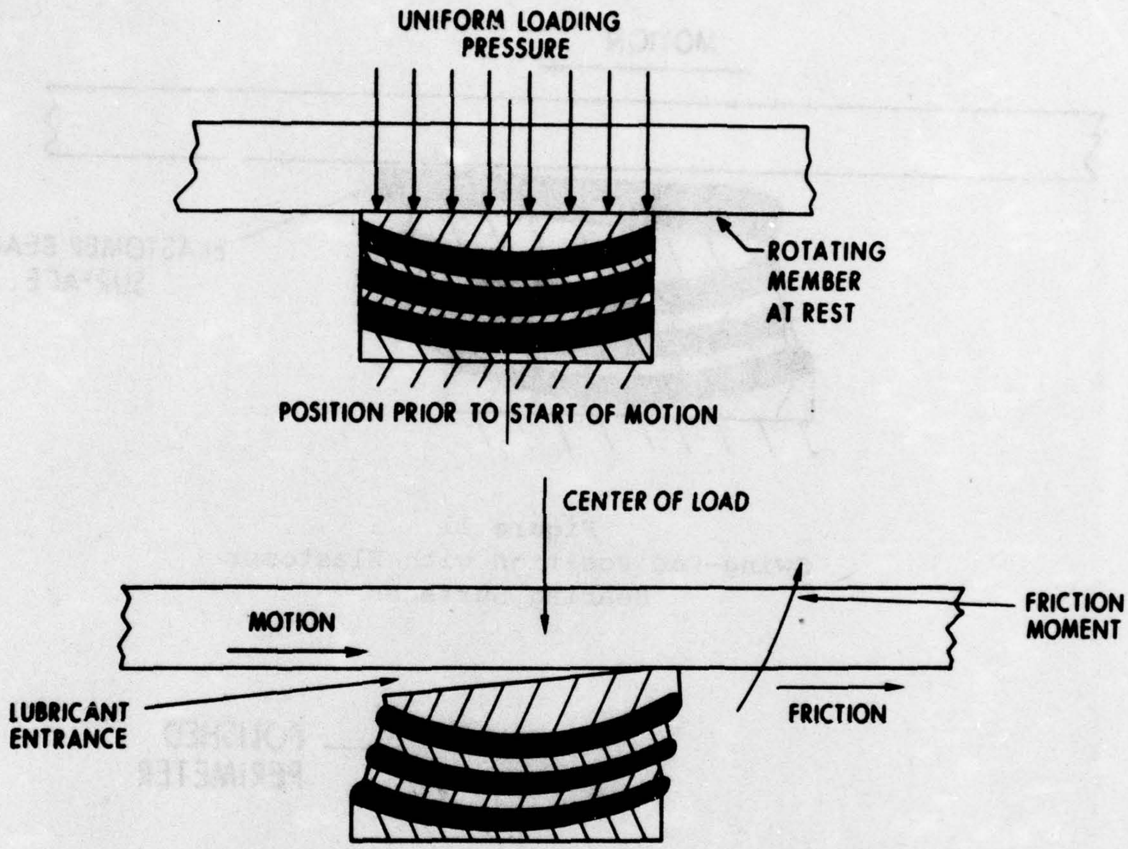


Figure 10
Swing-Pad at Rotational Start-up

It is expected that an elastomeric bearing surface, as with inverse hydrodynamic bearings, will subtract from the swing-pad's ability to form the converging wedge, figure 11. However, substantial loading can be supported. The swing of the bearing surface alters the surface deformation at the fluid inlet and outlet edges. Experiments have demonstrated that, depending on the surface stiffness and the magnitude of the wedge angle, fluid will enter between the sliding surfaces and form a pressure pocket in the elastic surface, figure 12. Post-test inspections have shown the pocket surrounded by a raised perimeter, indicated by polished surface of the rubber. However, the mechanism by which the pocket is formed is not clearly understood.

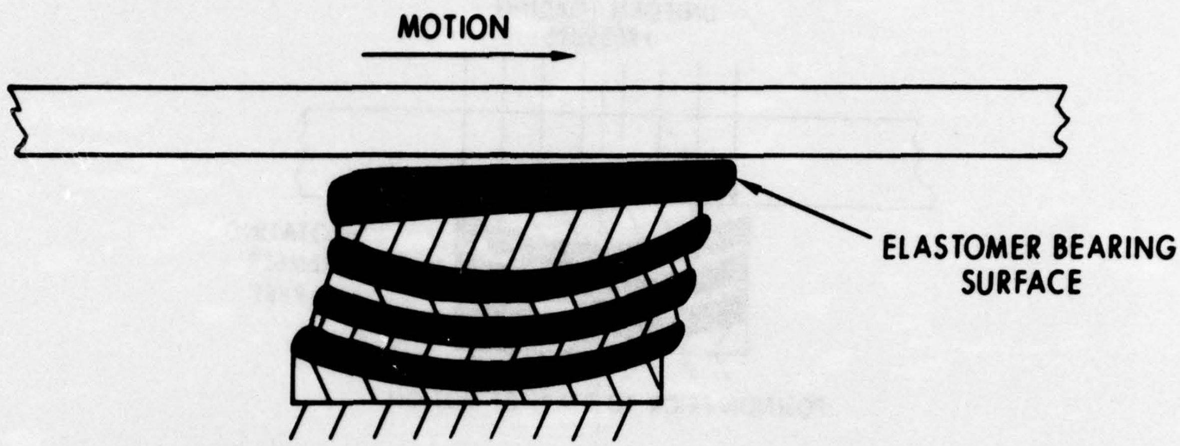


Figure 11
Swing-Pad Position with Elastomer Bearing Surfaces

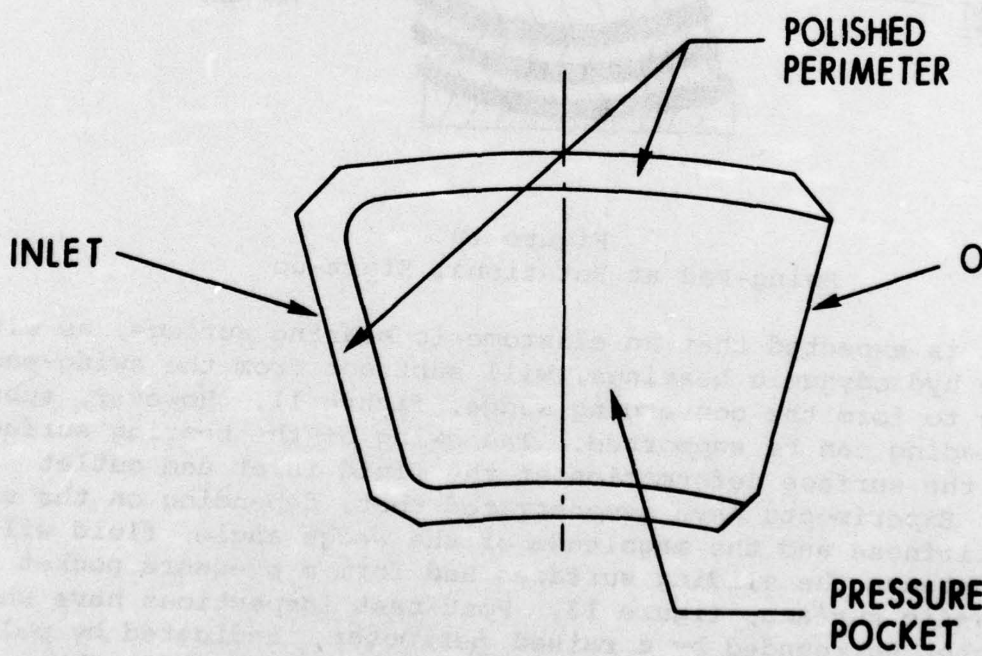


Figure 12
Fluid Pressure Pocket in Thrust Pad Elastomer Bearing Surface

The design parameters and performance predictions of conventional hydrodynamic bearings are based on the Navier-Stokes theory of fluid flow between rigid fixed boundaries and on Reynolds' simplifying equation for solutions of this theory. A modification of the theory is required for application to the swing-pad bearing to account for the wedge-forming actions. In order to evaluate the bearing concept, exploratory experimental investigations were conducted on model bearings with arbitrarily selected designs.

EXPERIMENTS AND RESULTS

EXPERIMENTS APPROACH

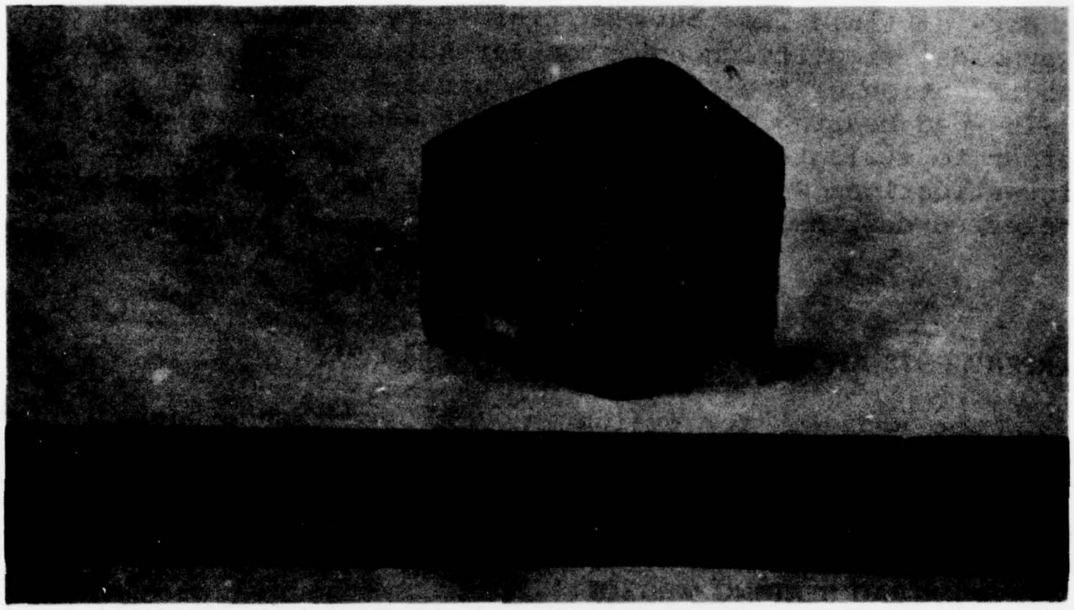
The approach for evaluating the performance characteristics of the swing-pad bearing was to conduct identical experiments on swing-pad, tilting-pad, and fixed-pad thrust and journal bearings and to compare performance.

THE SWING-PAD BEARING MODELS

The model swing thrust pad is shown in item (a), figure 13. The laminated elastomer was a nitrile elastomer and consisted of four laminations, 0.04-inch-thick (1.02 mm), 55 durometer hardness, and curved to approximately 2.25-inch (57.0 mm) spherical radius. Oil lubrication was planned. The bearing surface was aluminum bronze (8 rms surface finish). When the lubricant was changed to water, the bearing surface was changed to nitrile elastomer. This surface was applied to the pads by bonding the elastomer to a thin steel backing and pinning this adapter to the pad. Adapters with three different elastomer thicknesses were fabricated: 0.01-inch (0.25 mm), 1/8-inch (3.2 mm), and 1/4-inch (6.3 mm). The elastomer hardness was 55 durometer. The model thrust bearing assembly is three pads, equally spaced, mounted on a supporting plate, item (b), figure 13.

The model swing journal pad is shown in item (a), figure 14. The journal diameter was 6 3/4 inches. The laminated elastomer geometry consisted of five laminations, 0.04-inch-thick (1.02 mm), 55 durometer hardness, and curved to approximately 2-inch (51.0 mm) cylindrical radius. The bearing surface can be changed by a pinned adapter. The initially fabricated bearing surface was 0.03-inch-thick (0.76 mm) dressed nitrile elastomer, 55 durometer hardness. The model journal bearing assembly is shown in item (b), figure 14.

Item (a) - Model Thrust Pad



Item (b) - Model Thrust Bearing (with Two Laminations)

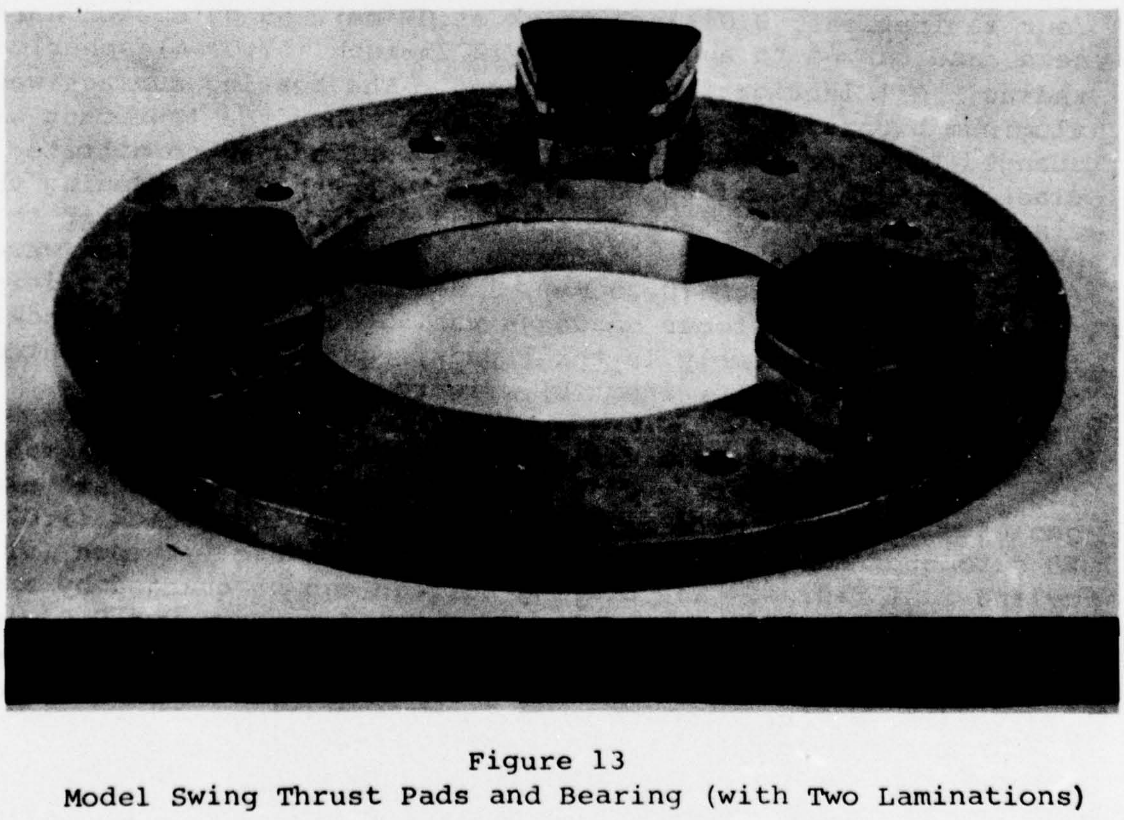
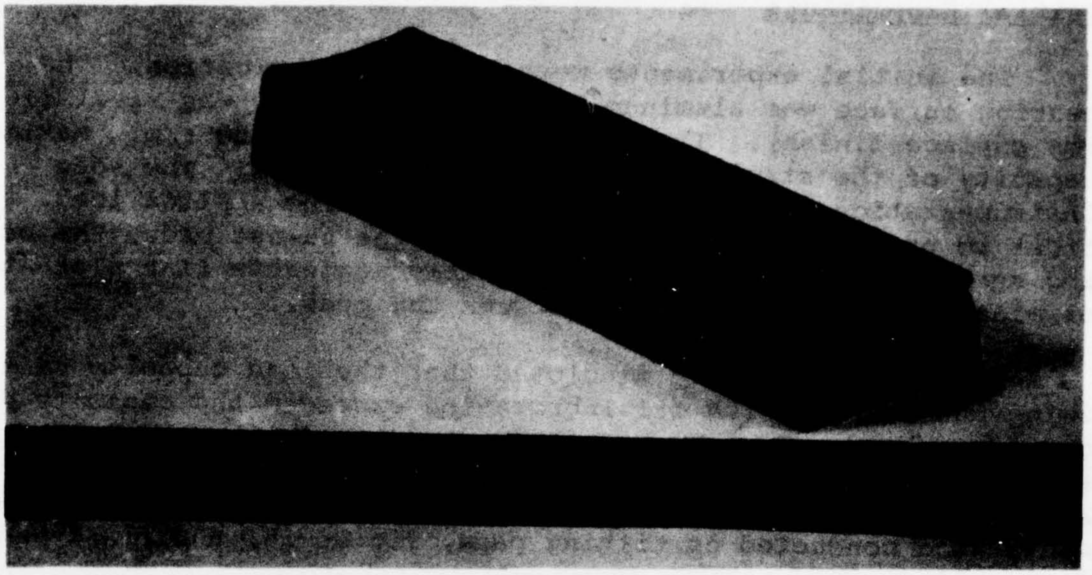


Figure 13
Model Swing Thrust Pads and Bearing (with Two Laminations)

Item (a) - Model Journal Pad



Item (b) - Model Journal Bearing (with Three Laminations)

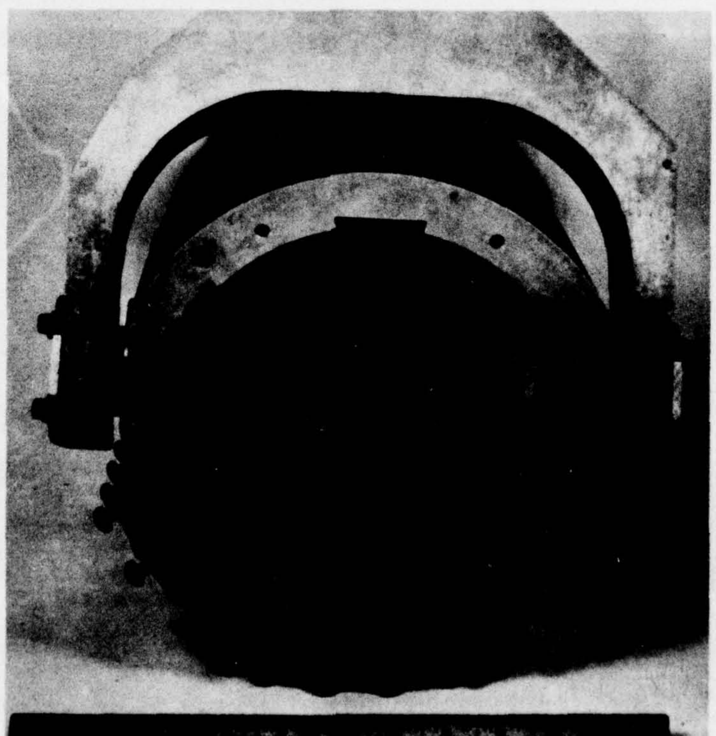


Figure 14
Model Swing Journal Pad and Bearing (with Three Laminations)

THRUST BEARING EXPERIMENTS WITH STEEL RUNNER

Initial Experiments

The initial experiments were with oil lubrication. The bearing surface was aluminum bronze. The runner was steel (16 rms surface finish). The speed was a constant 150 r/m. Maximum capacity of the stand was 600 lb/in^2 (4.1 MN/m^2). The sole instrumentation was a pressure gage indicating applied load. In order to promote equal load sharing by the thrust pads, the bearing surfaces were sanded after assembly against a flat plate to eliminate height differences between the pads.

Experiments quickly disclosed that the load capacity of the swing-pad bearing with oil lubrication exceeded the capacity of the test stand. To induce failure, the lubricant was changed to tap water, and the bearing surface was changed to nitrile elastomer. The experiments were continued. Identical experiments were conducted on tilting pads, figure 15, and fixed-pads, figure 16.

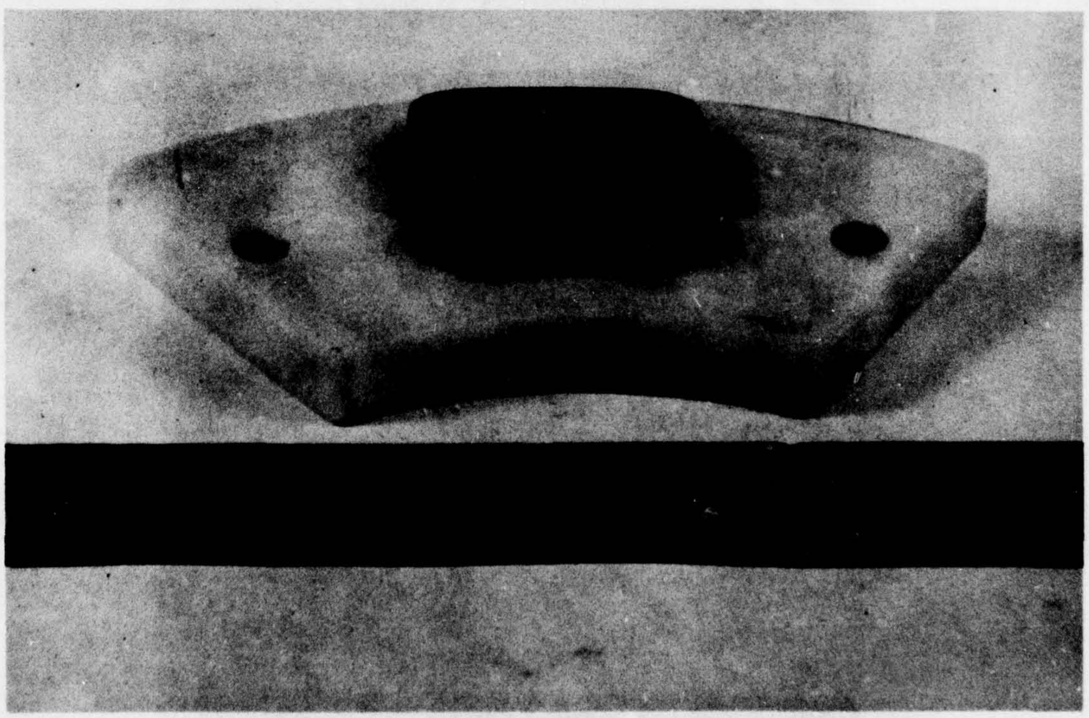


Figure 15
Model Tilting Pad

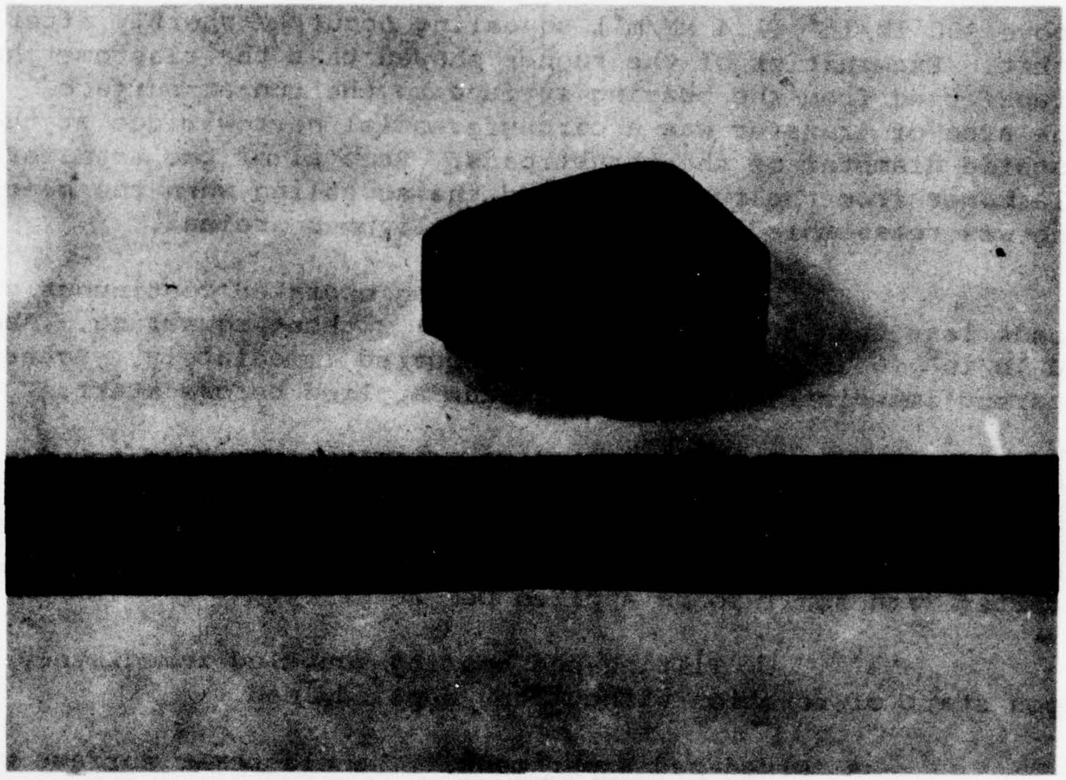


Figure 16
Model Fixed Pad

The following lists the applied load and performance observations. The time duration of the experimental runs was 1 hour or until apparent failure.

• <u>0.01-inch-thick (0.25 mm) Elastomer Surface</u>.

. Swing-pad bearing started rotation with 600 lb/in^2 (4.1 MN/m^2) load applied before start. After operation the surface showed progressive polishing.

. Tilting-pad bearing started rotation with 600 lb/in^2 (4.1 MN/m^2) load applied before start. After operation the surface showed considerable roughness and deterioration.

• <u>1/8-inch-thick (3.2 mm) Elastomer Surface</u>.

. Swing-pad bearing operated continuously with 600 lb/in^2 (4.1 MN/m^2) load with surface polishing. It started rotation with a 200 lb/in^2 (1.4 MN/m^2) load applied before start.

Above 200 lb/in^2 (1.4 MN/m^2), squealing occurred shortly after start. Examination of the runner showed that the elastomer had transferred from the bearing surface to the runner surface. The area of transfer was a circumferential narrow ridge at the outside diameter of the thrust pads. Removal of the transferred elastomer from the runner stopped the squealing when the bearing was reassembled and restarted at lighter preload.

. Tilting-pad bearing operated continuously with loads less than 50 lb/in^2 (0.35 MN/m^2) without squealing. Above 50 lb/in^2 (0.35 MN/m^2) squealing occurred immediately. Squealing started immediately with small loads applied before start.

- <u>1/4-inch-thick (6.3 mm) Elastomer Surface</u>.

. Swing-pad bearing operated continuously with loads up to 100 lb/in^2 (0.69 MN/m^2) without squealing. Squealing started with small loads applied before start.

. Tilting-pad bearing squealed immediately upon start of rotation without any applied load.

- <u>Fixed-pad Experiment</u>. The elastomer surface was flat with rounded edges, thickness was 5/8-inch (15.9 mm), 85 durometer hardness. Upon start of rotation and with no load, noticeable heating of the water began immediately. At a loading of 50 lb/in^2 (0.35 MN/m^2) water temperature was considered excessive (120 °F (49° C) without being stabilized), and the experiment was stopped. Examination of the bearing surface showed deep circumferential scores. The thrust runner was stained brown, indicating pad material transfer as a result of the heating.

<u>Additional Experiments</u>

The experimental stand was modified by installing a variable speed hydraulic drive, by adding a torquemeter to measure friction torque, and by increasing its load capacity. The elastomer thickness of the fixed pad was changed from 5/8-inch (15.9 mm) to 3/8-inch (9.5 mm).

The experiments were continued. Figures 17 and 18 show rotational coefficients of friction versus pressure loads at various speeds. Data are shown for the swing pads with the three elastomer surface thicknesses mentioned above. Data for the tilting pads with a 0.01-inch-thick (0.25 mm) elastomer surface also are shown.

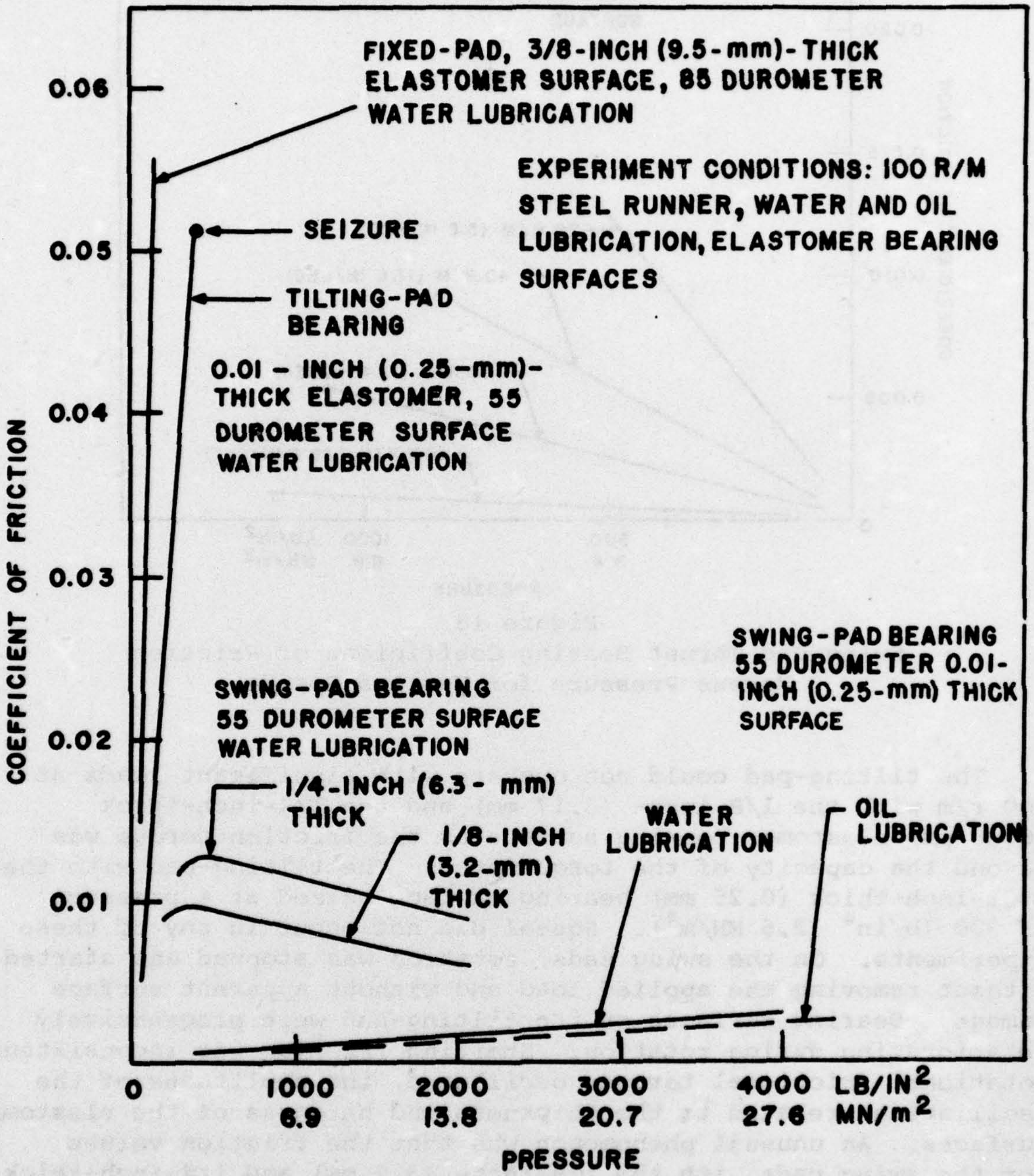


Figure 17
Comparison of Coefficient of Friction
Versus Pressure for Swing-Pad, Tilting-
Pad, and Fixed-Pad Thrust Bearings

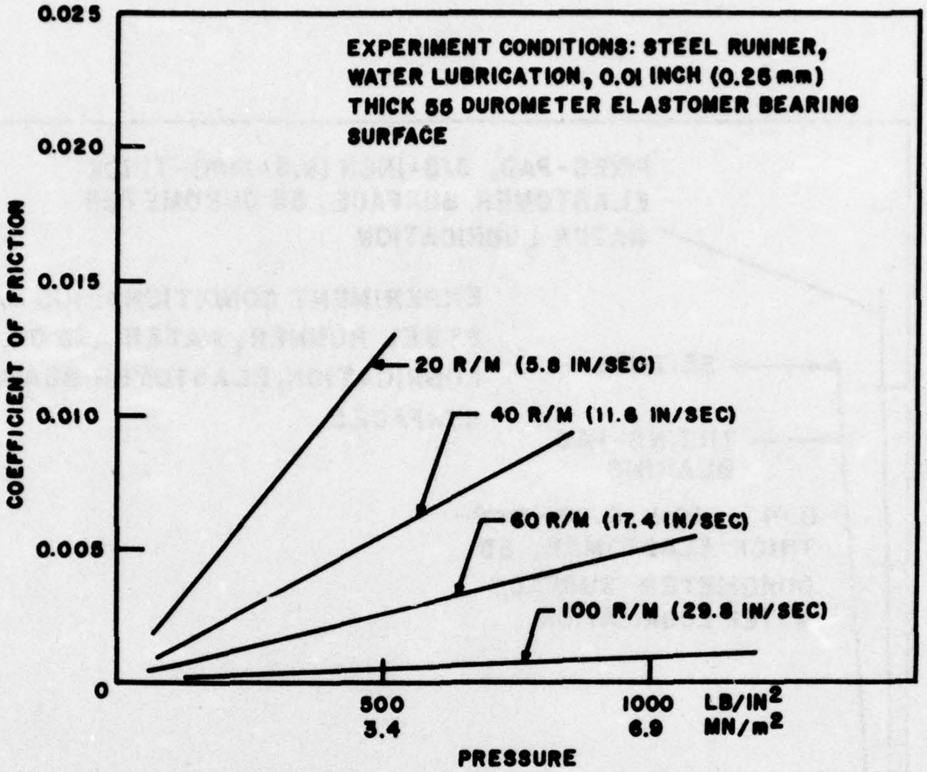


Figure 18
Swing-Pad Thrust Bearing Coefficient of Friction Versus Pressure for Various Speeds

The tilting-pad could not operate with significant loads at 100 r/m with the 1/8-inch- (3.17 mm) and the 1/4-inch-thick (6.36 mm) elastomer bearing surface as the friction torque was beyond the capacity of the torquemeter. The tilting-pad with the 0.01-inch-thick (0.25 mm) bearing surface seized at a pressure of 380 lb/in^2 (2.6 MN/m^2). Squeal did not occur in any of these experiments. On the swing pads, rotation was stopped and started without removing the applied load and without apparent surface damage. Bearing surfaces on the tilting-pad were progressively deteriorating during rotation. Starting friction was inconsistent. Rotational frictional torques oscillated, the amplitudes of the oscillations related to the thickness and hardness of the elastomer surfaces. An unusual phenomenon was that the friction values for the swing pads with the 1/8-inch- (3.2 mm) and 1/4-inch-thick (6.3 mm) elastomer surfaces oscillated as the loading pressures were increased until a pressure was reached at which the oscillations ceased and friction torque remained practically constant with increasing pressures. Examination of the bearing surfaces revealed distinct marks of the fluid wedge pocket and the raised perimeter, see figure 12.

The most interesting and unusual phenomenon is that the coefficient of friction of the swing-pad bearing in these experiments was practically equal with water or oil lubrication. An experiment with the 0.10-inch-thick (0.25 mm) elastomer surface with oil or water lubrication at 100 r/m showed approximately equal coefficients of friction. Apparently the wedge film thickness adjusts to the viscosity and results in equal coefficients. Figure 17 shows these coefficients of friction.

Breakaway coefficients of friction on the model thrust bearings with a steel runner and water lubrication were measured by use of a torque wrench. On the swing-pad bearing, sliding between the bearing surfaces started and continued smoothly, and the breakaway torque was clearly indicated. On the tilting-pad bearing, the breakaway torque was difficult to discern as the motion between the bearing surfaces was an erratic combination of slipping and grabbing. This motion is typical of stick-slip.

Figure 19 shows breakaway coefficients of friction versus pressure for both the thrust swing-pad and tilting-pad bearings with the 0.01-inch-thick (0.25 mm) elastomer surface. The friction coefficients for the tilting pads are approximated due to the erratic motion. At 1000 lb/in^2 (6.4 MN/m^2), the friction torque on the tilting pads became excessive for manual measurement, and the experiment was stopped. Pressures were applied up to 4000 lb/in^2 (27.6 MN/m^2) on the swing-pad bearing without stick-slip. Figure 20 shows breakaway coefficient of friction versus pressure for the swing pads with the different elastomer surface thicknesses. Breakaway coefficients of friction for the tilting pads with 1/8-inch- (3.2 mm) and 1/4-inch-thick (6.3 mm) surface could not be obtained because of stick-slip.

THRUST BEARING EXPERIMENTS WITH GLASS RUNNER

To observe the thrust pad bearing surface, experiments on the swing pads and the tilting pads were conducted with a transparent glass runner. To avoid risk of breaking the glass, surface pressures were limited to 200 lb/in^2 (1.38 MN/m^2).

The first series of experiments with the glass runner consisted of rotational experiments on the stand at the constant speed of 150 r/m and without a torquemeter for measuring friction torque. Observations were as follows:

- <u>0.01-inch-thick (0.25 mm) Elastomer Surface</u>.

    . <u>Swing-pad</u>. Elastomer surface imperfections appeared to cause a small, stable, cloudy fluid wake that converged in the direction of motion.

. Tilting-pad. A cloudy fluid wake at the fluid inlet edge of the pad appeared and disappeared at frequent intervals.

- 1/8-inch-thick (3.2 mm) Elastomer Surface.

. Swing-pad. A narrow wake of constant area remained stable at the fluid outlet edge of the pad.

. Tilting-pad. A cloudy fluid wake appeared at the fluid inlet edge of the pad, blossomed to cover one-eighth of the pad area, disappeared, and reappeared at frequent intervals.

- 1/4-inch-thick (6.3 mm) Elastomer Surface.

. Swing-pad. The cloudy fluid wake was identical to that on the 1/8-inch-thick (3.2 mm) surface except that it covered one-quarter of the pad area.

. Tilting-pad. Operation at zero load showed the cloudy fluid wake was identical to that on the 1/8-inch-thick (3 mm) elastomer surface, except that it covered half of the pad area.

It is to be noted that squeal did not occur in any of the experiments. As the stand did not have a torquemeter, the significance of the fluid wakes was not understood at this point.

A second series of experiments was conducted to help clarify the significance of the cloudy fluid wakes. These experiments were conducted with the 1/4-inch-thick (6.3 mm) elastomer bearing surface. Rotational speeds were 100 r/m; loads were less than 50 lb/in^2 ($0.34\ MN/m^2$). On the tilting-pad, when the runner rotated, a cloudy fluid wake appeared at the fluid inlet edge of the pad, blossomed toward the center of the pad, suddenly disappeared, reappeared at the inlet edge, and repeated. This is interpreted as stick-slip. On the swing-pad, when the runner rotated, the cloudy wake appeared near the fluid exit edge of the pad and remained stable and constant in size. Figure 21 shows these cloudy wake areas. Figure 22 is a hypothesis to explain the absence of stick-slip in the swing-pad bearing and its presence in the tilting pad bearing. Under this hypothesis, friction on the swing-pad bearing surface swings the surface in the direction in which the trailing edge will contact the thrust runner and, due to friction, motion will pull and extend the trailing edge. The trailing edge bearing surface remains essentially flat. Friction on the tilting-pad bearing surface tilts the surface in the direction in which the leading edge will contact the thrust runner and motion will push and compress the leading edge. Compression raises a ridge on the surface, the surface releases and snaps back and this repeats (stick-slip). Observation of the tilting-pad bearing surface tends to confirm this by exhibiting a roughened leading edge.

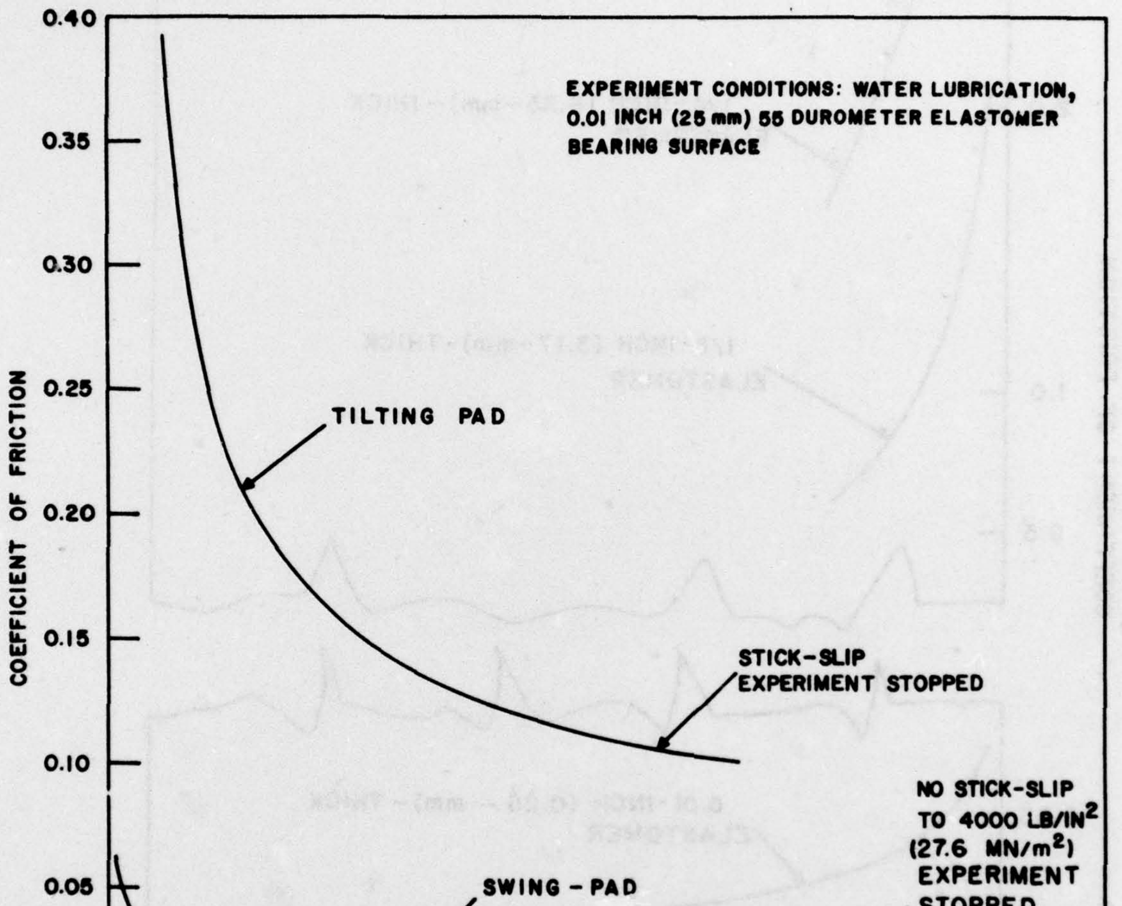


Figure 19
Breakaway Coefficient of Friction Versus
Surface Pressure Swing-Pad and Tilting
Pad Thrust Bearings

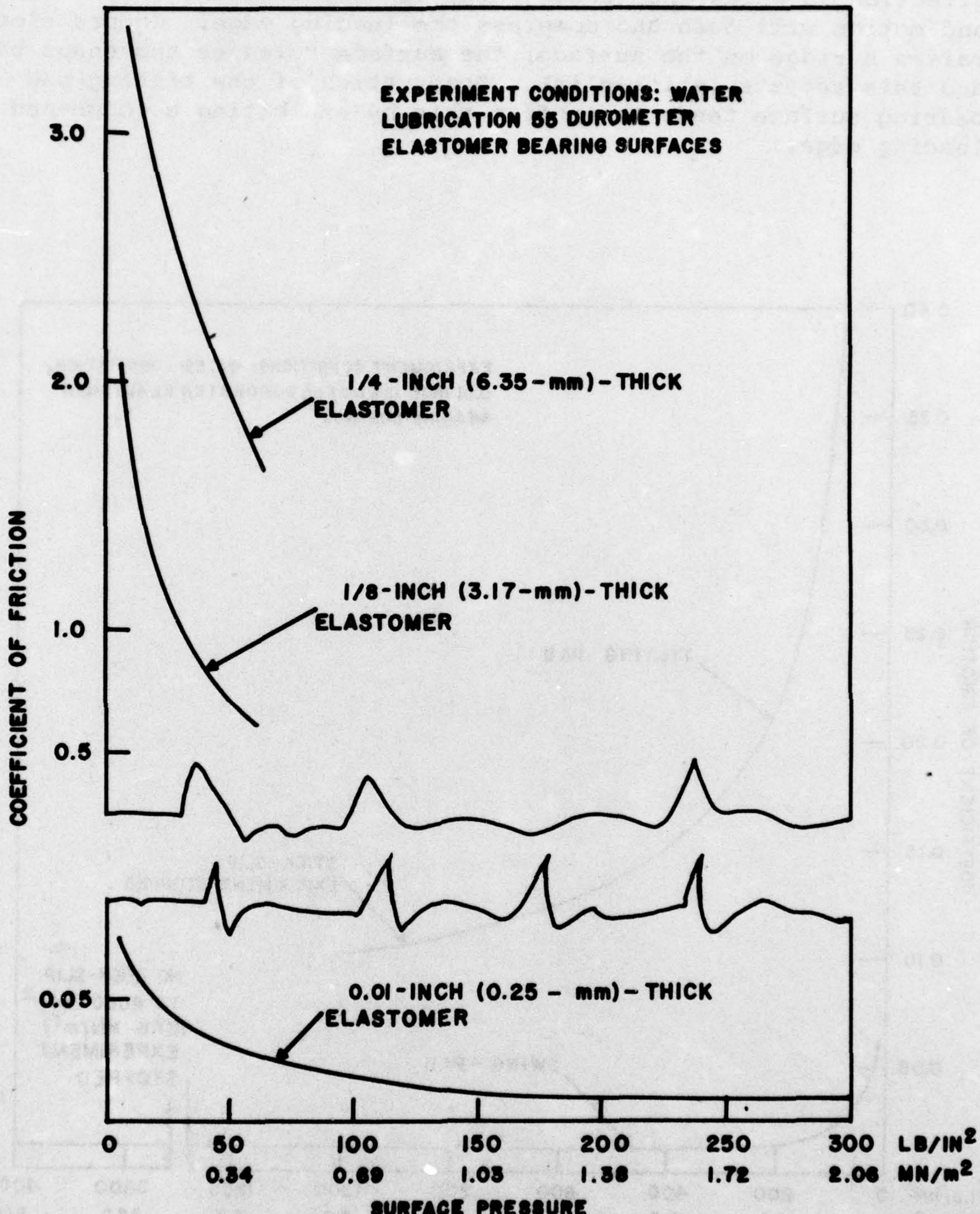
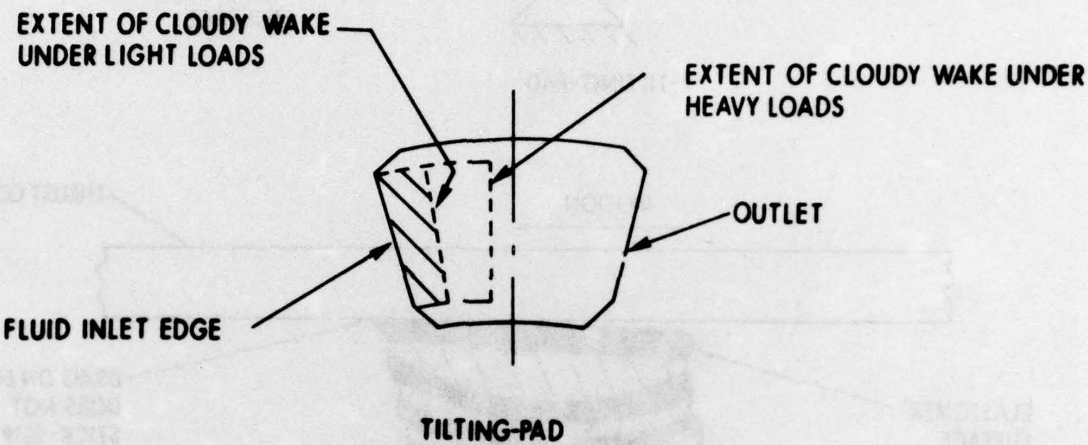


Figure 20
Breakaway Coefficient of Friction Versus Surface Pressure, Swing-Pad Thrust Bearing



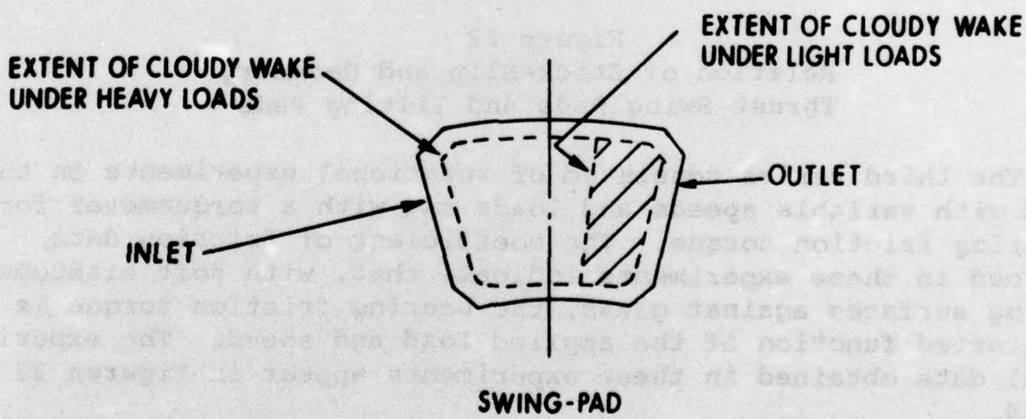


Figure 21
Fluid Pressure Areas of Swing Pads and Tilting Pads

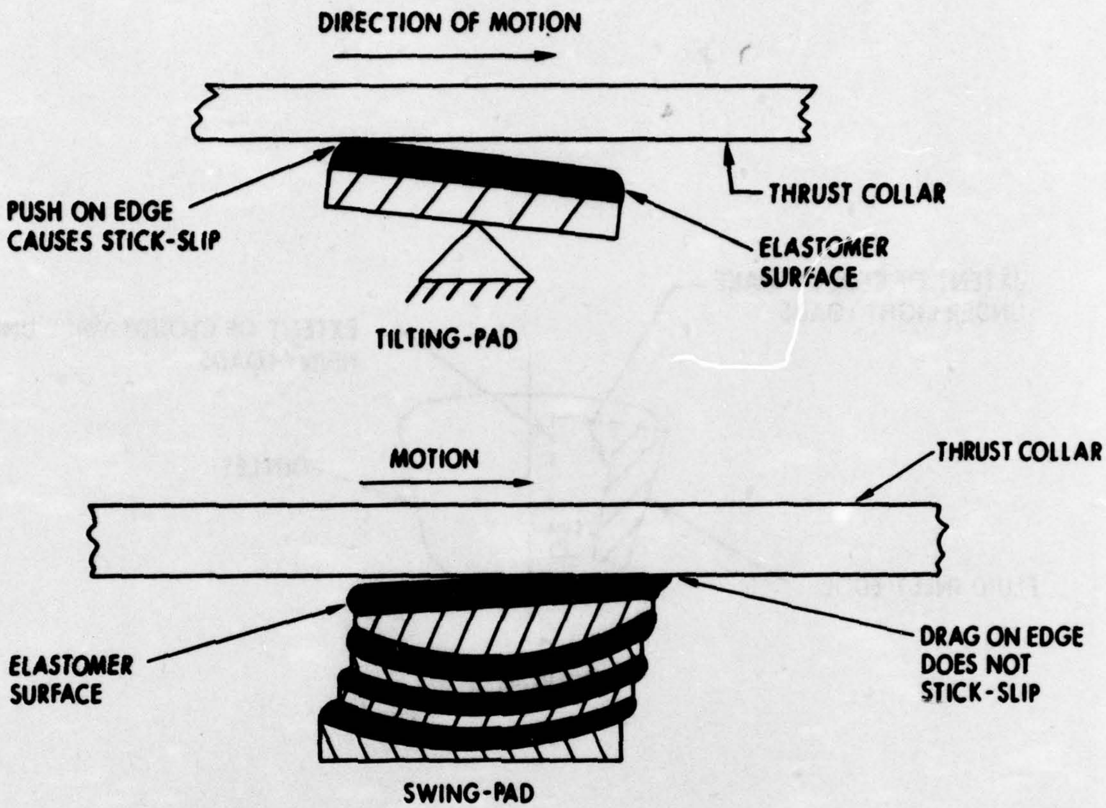


Figure 22
Relation of Stick-Slip and Geometry
Thrust Swing Pads and Tilting Pads

The third series consisted of rotational experiments on the stand with variable speeds and loads and with a torquemeter for measuring friction torque. The coefficient of friction data obtained in these experiments indicate that, with soft elastomer bearing surfaces against glass, the bearing friction torque is a complicated function of the applied load and speed. The experimental data obtained in these experiments appear in figures 23 and 24.

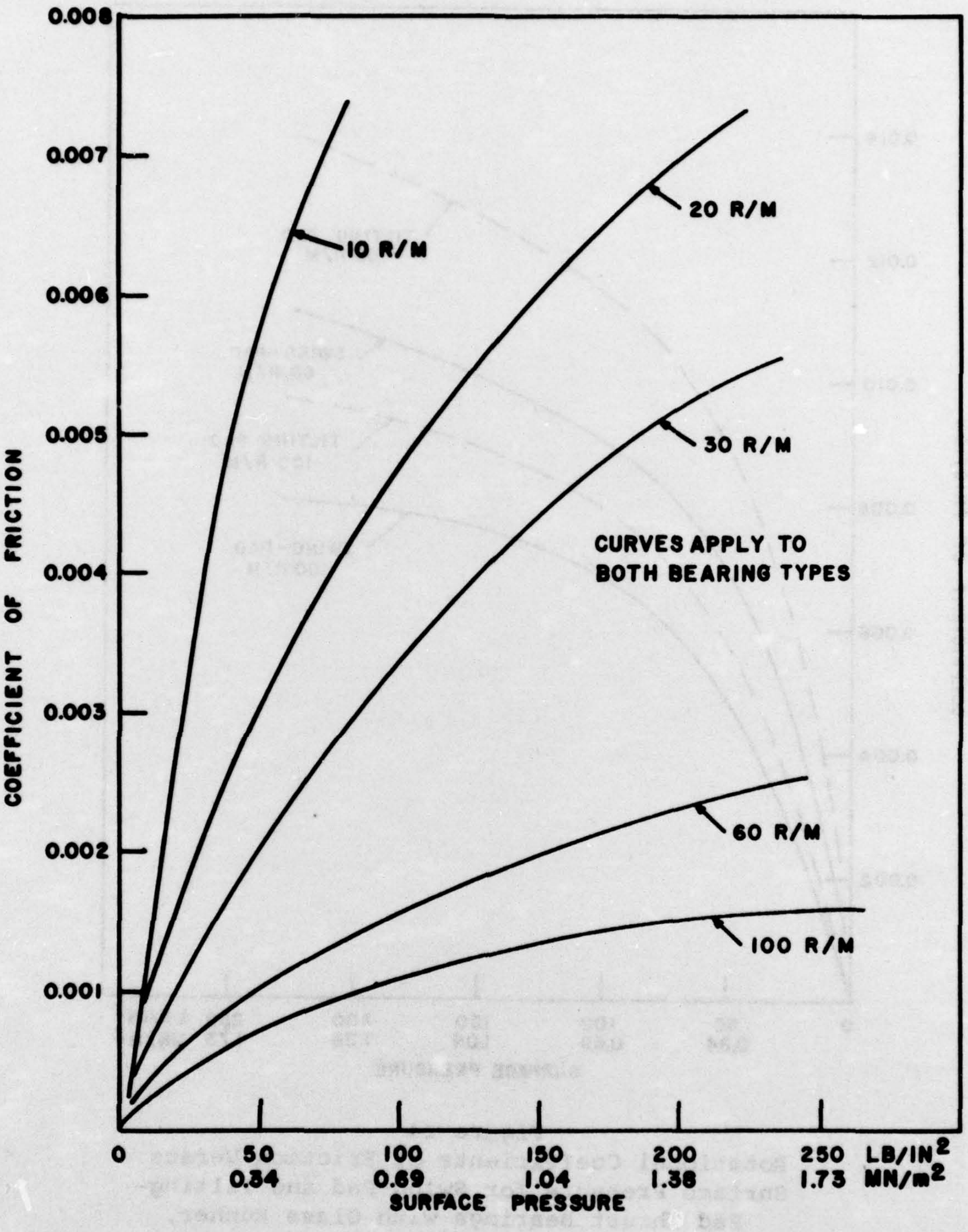


Figure 23
Coefficients of Friction Versus Surface Pressure for Swing-Pad and Tilting-Pad Thrust Bearings with Glass Runner (0.01-inch-thick (0.25 mm) Elastomer Bearing Surfaces)

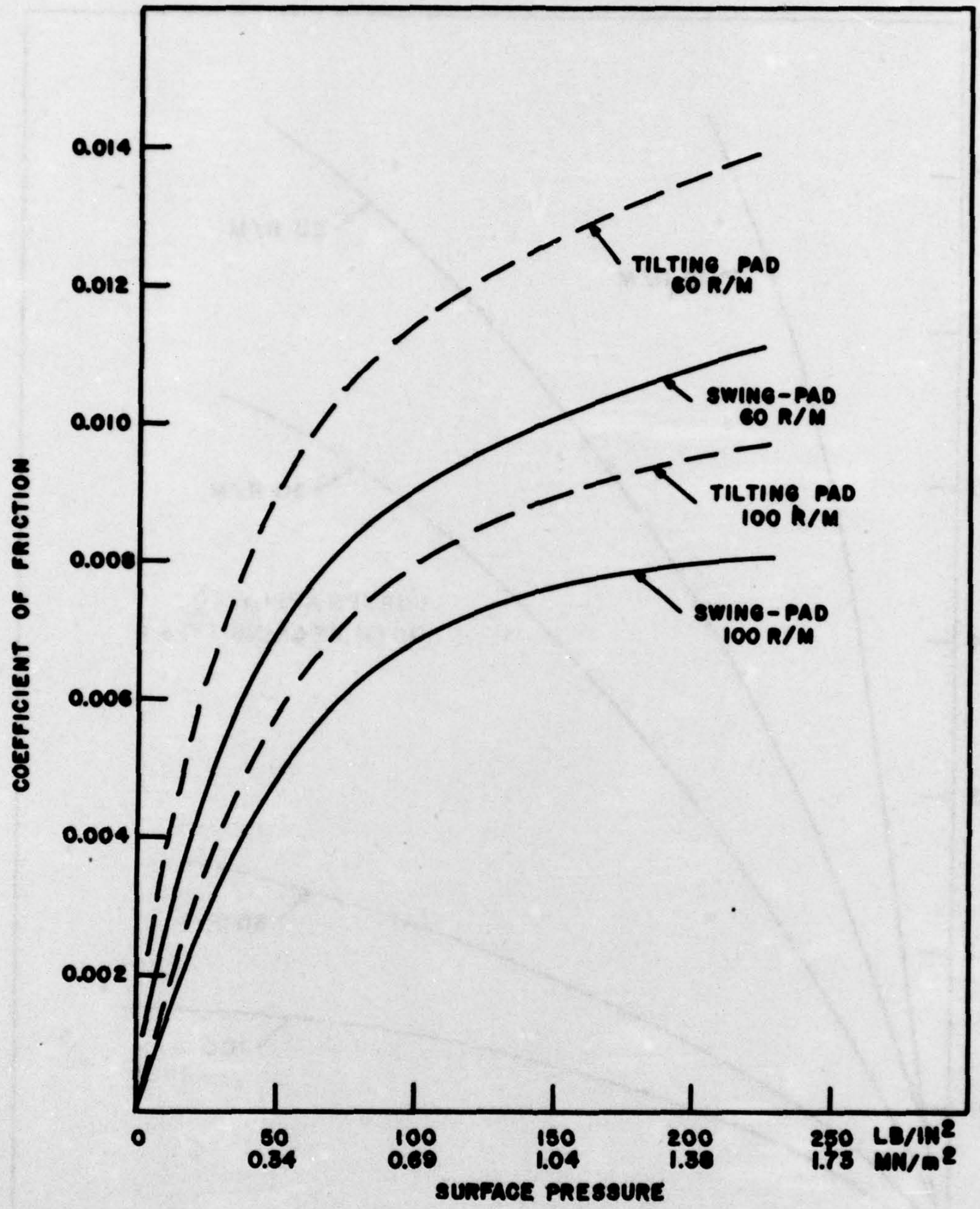


Figure 24
Rotational Coefficients of Friction Versus Surface Pressure for Swing-Pad and Tilting-Pad Thrust Bearings with Glass Runner, (1/8-inch-thick (3.2 mm) Elastomer Bearing Surfaces)

Figure 23 shows coefficient of friction versus speed and pressure for the swing pads and the tilting pads with the 0.01-inch-thick (0.25 mm) elastomer surfaces. With this thin elastomer, the friction coefficients for the swing pads and the tilting pads are equal, figure 23. The ability to operate at extremely low speeds indicates the absence of boundary lubrication. Figure 24 shows the coefficient of friction versus pressure with the 1/8-inch-thick (3.2 mm) elastomer surface. With this thicker elastomer, the operational differences between the swing-pad and the tilting-pad are shown by the differences in the friction coefficients. Squeal occurred with both pads below 20 r/m. Experiments with the 1/4-inch-thick (6.3 mm) elastomer surface showed that the tilting pads could operate only with zero loads at 100 r/m and squealed continuously. The swing pads supported 100 lb/in^2 (0.69 MN/m^2) before squealing began.

JOURNAL BEARING EXPERIMENTS

The bearing surface of a pad-type bearing is noncontinuous. Individual pads are spaced into a housing. The journal bearing experiments were conducted without machining the bearing surfaces to equalize their height precisely. Pads were inserted only in that half of the housing which carried the load. Five pads were used.

Experiments were conducted on: (1) a swing-pad with a 0.03-inch-thick (0.76 mm), 55 durometer hardness elastomer bearing surface, curved to mate with the journal (25 rms surface finish); (2) a 3/8-inch-thick (9.5 mm) 85 durometer hardness fixed elastomer pad with a flat bearing surface; (3) a 0.030-inch-thick (0.762 mm) 55 durometer hardness elastomer bearing surface curved to mate with the journal rigidly supported on a metal base. Two different elastomer hardnesses were used because it was desired to explore the effect of elastomer hardness. It is planned to repeat these experiments using an elastomer hardness of 85 durometer for all bearing surfaces.

Figure 25 shows the rotational coefficients of friction versus pressures and speeds for the swing-pad and the fixed elastomer pads, items (1) and (2) above. The coefficients of friction of the rigid metal base with the 0.03-inch-thick (9.5 mm) curved elastomer surface (item 3 above) are not shown as this pad was inoperable.

The coefficients of friction of the swing-pad are indicative of fluid-film lubrication, and those of the solid elastomer pad are indicative of boundary lubrication. Examination of the bearing surface on the swing-pad shows progressive polishing and on the solid elastomer pad shows progressive destruction. The solid elastomer pad could not operate below 40 r/m due to excessive friction and the associated stick-slip.

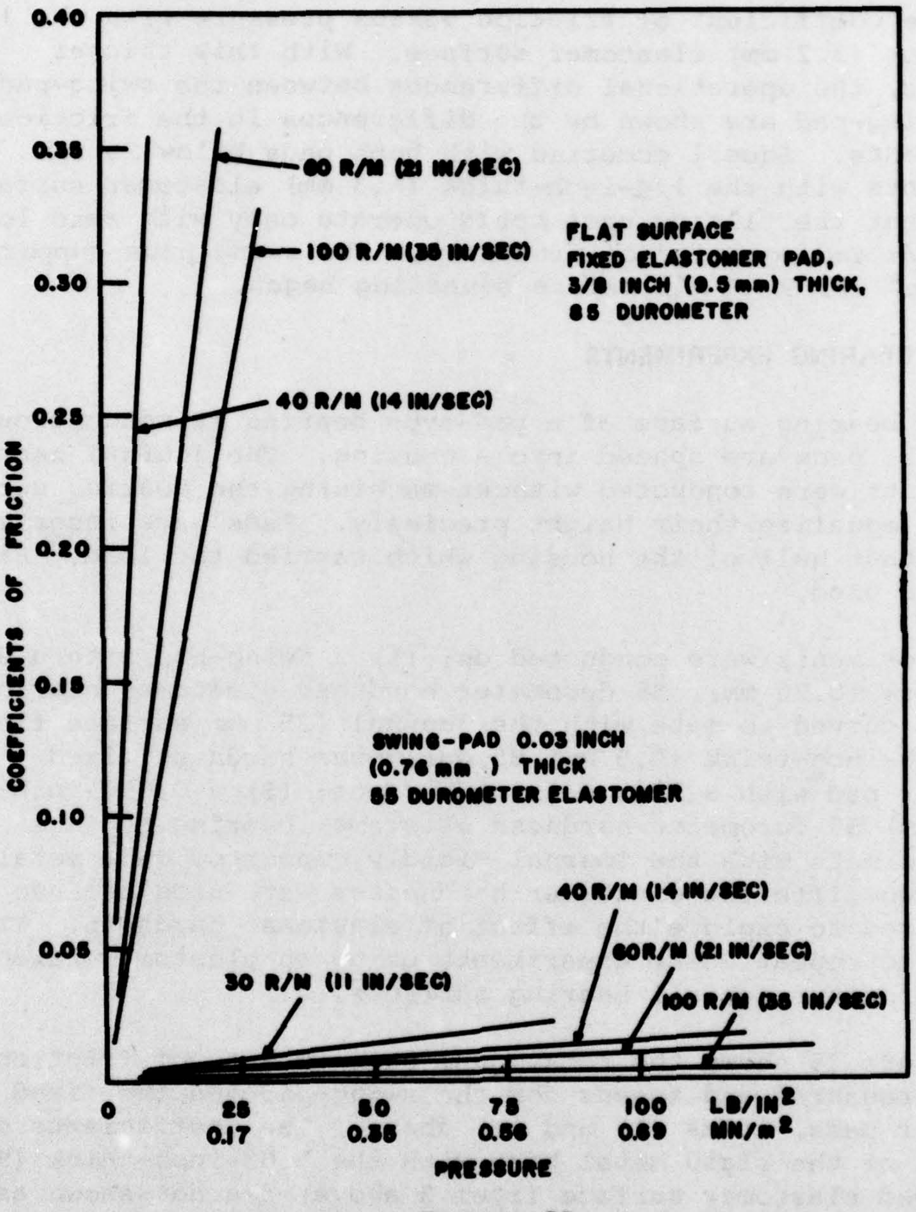


Figure 25
Rotational Coefficients of Friction Versus Surface Pressure for Swing-Pad and Fixed-Pad Journal Bearings

Experiments were conducted to determine breakaway coefficients of friction with load applied before start of rotation. Figure 26 shows the coefficients of friction versus surface pressure for the journal swing-pad. These are comparable with those on the thrust swing-pad. Coefficients of friction for the solid elastomer pad are not shown as stick-slip resulted in erratic friction torque measurements that could have damaged the torquemeter and the recording instrument if the experiment had been continued.

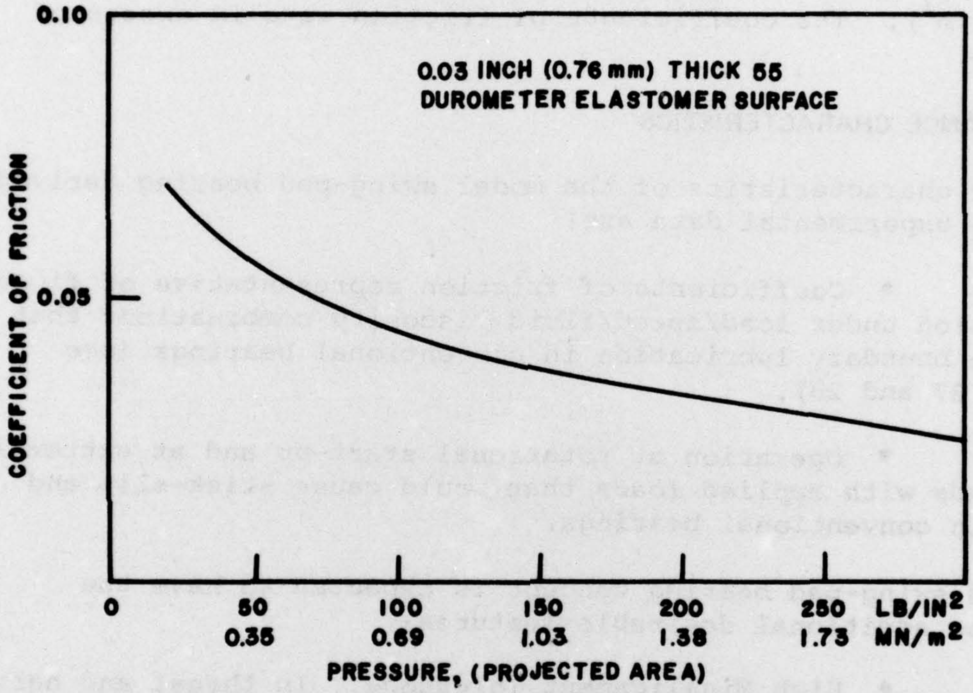


Figure 26
Breakaway Coefficients of Friction Versus Surface Pressure for Swing-Pad Journal Bearing

SUMMARY OF COMPARATIVE PERFORMANCE DATA

The model swing-pad thrust bearing with the 0.01-inch-thick (0.25 mm) elastomer bearing surfaces supported a loading pressure of 4000 lb/in^2 (27.6 MN/m^2) at 100 r/m with water lubrication. The coefficient of friction was 0.004. (The experiment was stopped at 4000 lb/in^2.) In an identical experiment with the tilting-pad bearing, the bearing seized at a pressure of 380 lb/in^2 (2.6 MN/m^2). The friction coefficients of the tilting-pad bearing were more than 50 times those of the swing-pad bearing.

The model swing-pad thrust bearing with 1/4-inch-thick (6.3 mm) 55 durometer hardness elastomer bearing surface supported loading pressures of 400 lb/in^2 (2.80 MN/m^2). The coefficient of friction was less than 0.010. In identical experiments, the tilting-pad bearing was inoperable.

The model swing-pad journal bearing with 0.03-inch-thick (0.76 mm) elastomer bearing surfaces supported 250 lb/in^2 (1.7 MN/m^2). The coefficient of friction was 0.003 at 100 r/m. The model fixed-pad journal bearing with approximately equal surface stiffness had excessive surface destruction at 40 lb/in^2 (0.28 MN/m^2). The coefficients of friction were in excess of 200 to 1.

PERFORMANCE CHARACTERISTICS

The characteristics of the model swing-pad bearing derived from the experimental data are:

- Coefficients of friction representative of fluid lubrication under load/speed/fluid viscosity combinations that would be boundary lubrication in conventional bearings (see figures 27 and 28).

- Operation at rotational start-up and at extremely low speeds with applied loads that would cause stick-slip and squeal in conventional bearings.

The swing-pad bearing concept is expected to have the following additional desirable features:

- <u>High Misalignment Tolerance</u>. In thrust and narrow journal swing pads, the spherical curvature of the laminated elastomer allows considerable misalignment. In wide journal bearing swing pads, the cylindrical elastomer laminates will allow limited misalignment. The length of the journal bearing is reduced by the high load capacity of the swing-pad bearing. This considerably increases misalignment capability.

- <u>Environmental Compatibility</u>. The material of the elastomer and metal parts can be selected to suit the environment. The fluid indigenous to the environment can be used as the lubricant, e.g., the bearing can operate submerged in seawater.

• Small Space Envelopes. While the radial thickness is approximately twice that of a typical journal bearing, it is less than half of that of a ball bearing, and the axial dimension can be as small as the radial thickness or as long as feasible to fabricate.

• Simple, Rugged, and Inexpensive Construction. The metal shims in the laminations are simple to form, and close tolerances are not required.

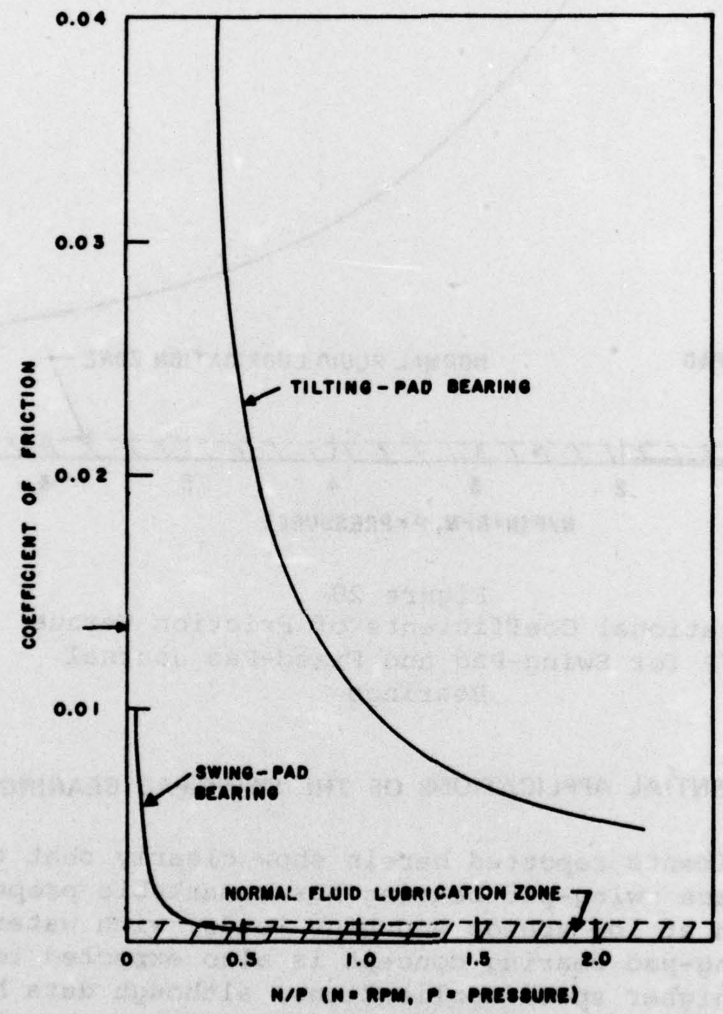


Figure 27
Rotational Coefficients of Friction Versus N/P for Swing-Pad and Tilting-Pad Thrust Bearing with 0.01-inch-thick (0.25 mm) Elastomer Surface

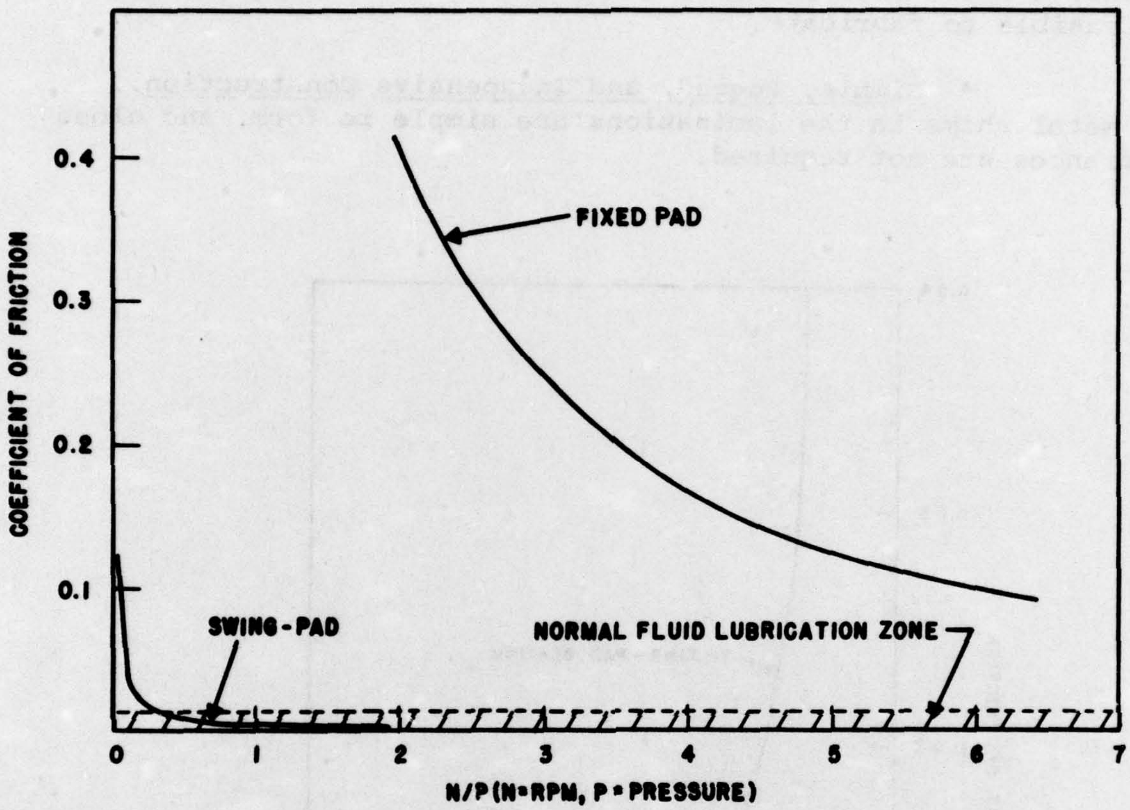


Figure 28
Rotational Coefficients of Friction Versus
N/P for Swing-Pad and Fixed-Pad Journal
Bearings

POTENTIAL APPLICATIONS OF THE SWING-PAD BEARING

The experiments reported herein show clearly that the elastomer surface swing-pad bearing has remarkable properties of low friction at low speeds and high loads, with water lubrication. The swing-pad bearing concept is also expected to be applicable to higher speed applications, although data have not yet been collected to verify this. Applications suggested include process fluid lubricated pump bearings and submersible electric motor bearings.

Only a small amount of evidence has been collected on oil-lubricated hard surface swing-pad bearings. If further work confirms the early indications, the swing-pad should allow considerable increase in load capacity, particularly in start-up load limits. Application in thrust bearings would be especially useful.

CONCLUSIONS

The elastomer-faced swing-pad bearing has been shown capable of very high load capacity with water as lubricant and capable of low-friction operation under conditions of high load, low speed, and low lubricant viscosity that would be expected to produce high friction and bearing distress in conventional bearings.

More work is needed to confirm the projected benefits of applying the concept to higher speeds and to oil-lubricated hard surface bearings.

APPENDIX A

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