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## Performance Tests on Six-Inch Tilting Pad Thrust Bearings

*Laboratory tests were conducted on six-in-dia tilting pad thrust bearings to compare operating characteristics of several pad facing and backing material combinations and also to study the effects of different oil feed and discharge arrangements. Tests were conducted at various speeds and loadings, including a series of failure tests. Results of these tests are presented.*

### Introduction

Tilting pad type thrust bearings are used in many types of rotating machinery. These bearings are available in catalog listed sizes and are widely applied in these standard designs. The pads of these bearings normally consist of a facing of tin base babbitt on a steel backing with the pivot located centrally to permit the bearing to be used for either or both directions of shaft rotation.

For many applications these bearings have been and will continue to be entirely satisfactory. For other applications improved bearing designs are required for one or more of several reasons. Higher allowable temperature capabilities, higher load capacities, smaller space requirements, and reduced power losses are more frequently being required of thrust bearings by new rotating machinery designs.

Modifications to the standard designs to provide improved performance are in use, primarily in the form of high strength copper backing as a replacement for the steel, and/or the use of circumferentially offset pivots. The purpose of these tests, from which the data reported here were obtained, was to study the effect of these and other modifications to provide guidelines for their application.

The modifications tested included the following:

- a radial shift in pad pivot position
- b circumferential shift in pad pivot position
- c pads of high strength copper backing with tin babbitt facing
- d pads of solid bearing aluminum
- e pads with graphite fiber composite facing on a steel backing

- f pads with copper-lead facing on a steel backing
- g pads with mechanical scrapers to reduce hot oil carryover
- h bearing with lube oil supplied through jets within the pads
- i throttled versus open tangential discharge of the bearing oil
- j reduced number of bearing pads.

Tests were run over a range of speed and load conditions, and a series of failure load tests were also included.

### Test Equipment

A general view of the setup for this test work is shown in Fig. 1. Two six inch tilting pad thrust bearings are contained within the cylindrical shell located between the two journal bearing pedestals. The shaft is rotated by a variable speed drive through a stepup gearbox. Shaft speed is measured with the use of an electronic counter and a magnetic pickup. Lubricating and cooling oil is supplied from one of the lube oil systems in the lab, located to the left and behind the test bearings in the Fig. 1 photo. Pumps, a filter, cooler, heater, associated piping, valves and controls are assembled atop a 1450-liter (380-gal) reservoir suspended into a pit in the floor. Light turbine oil was used in all of the tests reported here except as specifically noted.

A cross sectional view of the test bearing arrangement is shown in Fig. 2. Two thrust bearings are loaded back to back against integral shaft collars by a ring of opposing hydraulic pistons. Thus, no net thrust is transmitted beyond the test portion of the housing or the shaft. The cylindrical test housing is supported in a hydrostatic bearing. This permits the housing torque to be measured by means of a torque arm and a strain gauge type load cell and provides a check on the bearing power losses as determined by oil flow and temperature rise.

Oil is fed individually to the test bearings in a conventional manner at the outside diameter of each bearing opposite the thrust pads. From here the oil flows radially inward to the shaft, axially to the runner, out across the pads, and then is discharged tangentially in the direction of rotation horizontally at the top. In all tests, except those specifically noted, this discharge was not restricted. The flow rate to each bearing was measured with rotameter type flow meters specifically calibrated for the conditions used.

Thrust load was applied to the test bearings by hydraulically

<sup>1</sup> Numbers in brackets designate References at end of paper.

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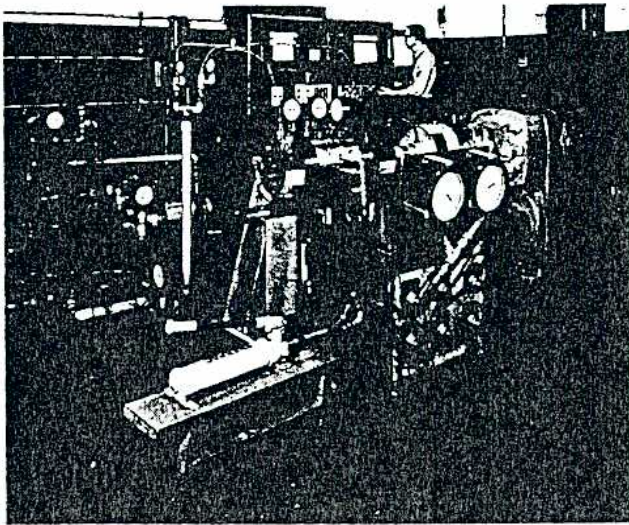


Fig. 1 Thrust bearing test setup

pressurizing the loading pistons. This hydraulic pressure was indicated on test gauges from which the total bearing load was then determined.

The basic test bearings were 152 mm (6 in.) OD, 76 mm (3 in.) ID, with six 51-deg pads. Load equalization between pads was provided by a conventional leveling link construction. For those tests on nonequalized bearings the hydraulic pistons provided for alignment of the bearing. A net surface area of 116 cm<sup>2</sup> (18 in<sup>2</sup>) is provided in this bearing. A photograph of a test bearing with babbitted, steel backed pads is shown in Fig. 3.

Temperatures were measured with copper-constantan thermocouples and logged on multipoint recorders. The pad temperatures presented here were all taken from the center of the outer trailing edge quadrant of the bearing pad (so called 75-75 percent point). In all but the graphite fiber composite faced pads these thermocouples were located 0.8 mm (0.03 in.) below the pad face. With the GFC facings it was necessary to mount the thermocouple tip flush with the face of the pad.

### Test Procedures

In order to limit the number of variables involved in these tests several factors were held constant, as a general rule. Exceptions did exist, and these are noted wherever they occurred. In all other cases, the following conditions were used:

- 1 lubricant—light turbine oil
- 2 oil inlet temperature—49 deg C (120 deg F)
- 3 oil flow per bearing at test speed
  - a 15.1 l/min (4 GPM)—1780 RPM
  - b 18.9 l/min (5 GPM)—6000 RPM
  - c 22.7 l/min (6 GPM)—8000 RPM
  - d 28.4 l/min (7½ GPM)—10000 RPM
  - e 32.2 l/min (8½ GPM)—12000 RPM

Tests were normally conducted with the bearing load being the only operating variable. All input factors were held constant at a specified load until temperatures were stable. A reading of all



Fig. 3 Test bearing with babbitted, steel-backed pads

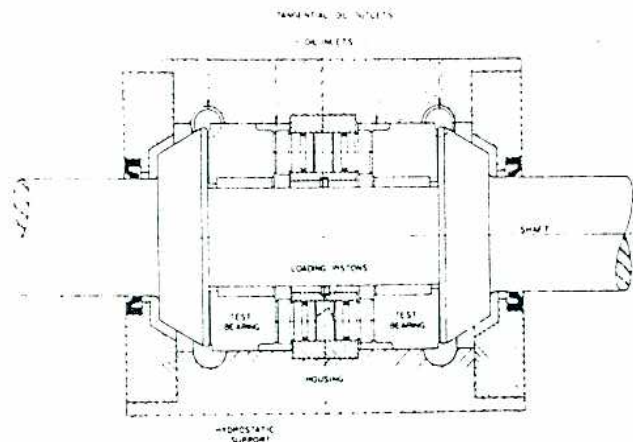


Fig. 2 Cross section of test housing, bearings, shaft, and loading arrangement

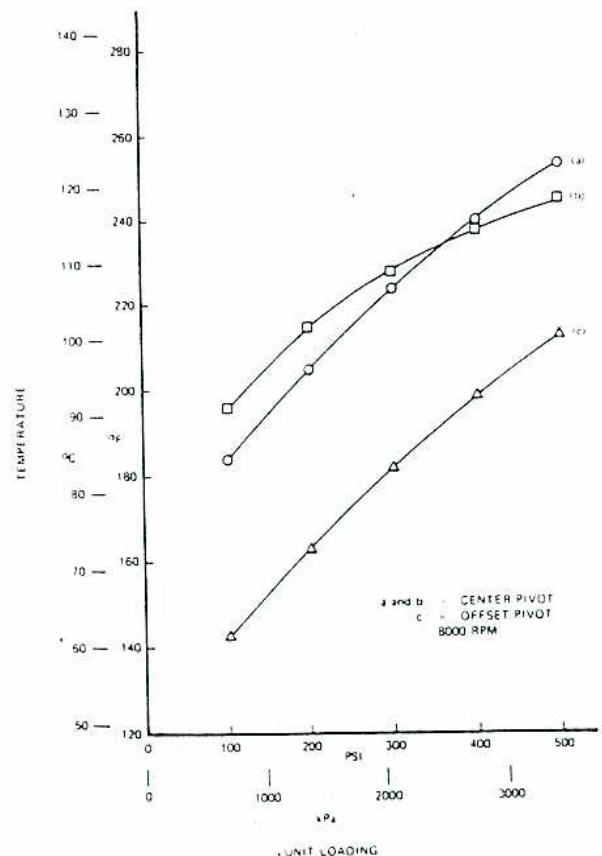


Fig. 4 Influence of pivot position on pad temperature for babbitted, steel-backed pads



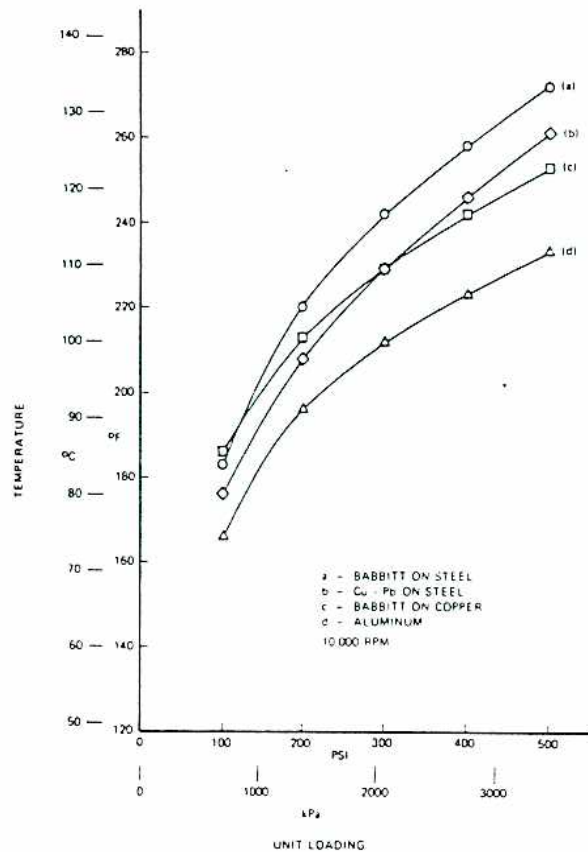


Fig. 5 Temperatures for center pivot pads of different materials

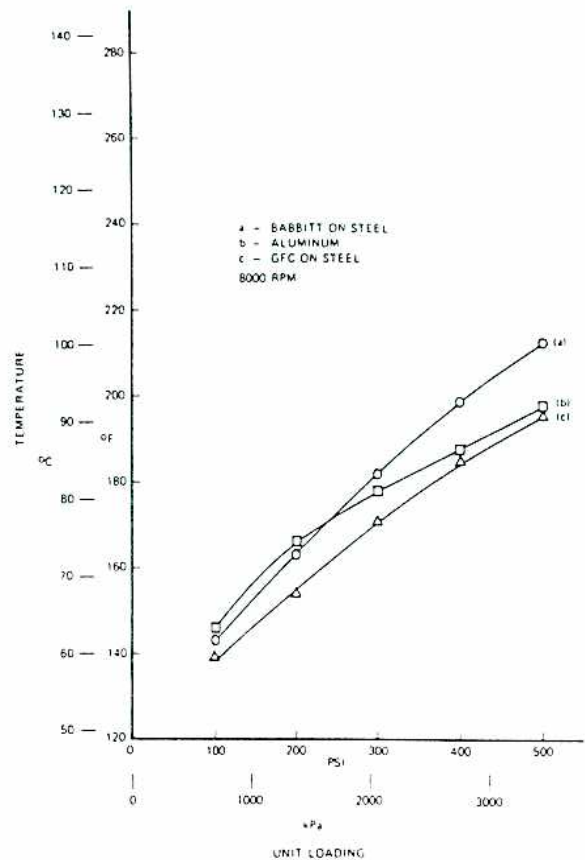


Fig. 6 Temperatures for offset pivot pads of different materials

data was then made. The load was then changed to a new value and another reading taken, etc.

Tests were also run with the bearing discharge pressure as the variable. In these, the discharge pressure was increased by closing (as needed to establish the pressure desired) the gate valves in the discharge lines.

The failure tests were all made under the same operating conditions of speed, oil flow, and inlet temperature with the load increased until failure occurred. The failure point was observed as that load at which the pad temperature rose abruptly and rapidly indicating pad to runner contact. At this time the power to the drive motor was cut, and the thrust bearing load was dumped. In all cases, evidence of pad to runner contact was apparent on inspection after a failure. There was no doubt in any of the failures that continued operation would not have been possible. On other occasions of pad inspection, after normal data runs, minor surface disturbances were sometimes found but, in general, the pads were in essentially "new" condition.

### Test Results

The data presented here is, of necessity, only a small portion of that obtained during this series of tests. It is representative, however, and provides a sampling of the results of tests of the various modifications. In most all cases relative performance characteristics were confirmed by repeat tests and by tests at other operating conditions (as at other speeds).

Fig. 4 is a plot of pad temperature versus loading for babbitt faced, steel backed pads showing the influence of pivot location. The two upper curves compare two radial locations for a pivot in the center of the pad circumferentially. The radial location for the pivot in curve (b) was determined by computer analysis as the point at which the pad would exhibit zero tilt radially. This analysis includes thermal and pressure caused distortions of the pad in addition to the effects of viscosity variations over the pad face.

The radial location for the pivot in curve (a) was 1.65 mm (0.065 in.) outward from that in curve (b). This results in the pad tilting radially to result in a greater film thickness at the inner diameter than at the outer diameter, providing a geometry similar to that in a taper land thrust bearing with compound tapers. The advantage which the (b) construction appears to exhibit at the higher loadings in this plot was confirmed by the failure tests (see Fig. 15). Curves (a) and (b) in Fig. 4 are for the same pad construction as curves (a) and (b), respectively, in Fig. 15.

The advantage in using a pivot offset circumferentially in the direction of runner rotation was again confirmed as evidenced by curve (c). The primary disadvantages of this construction are that the bearing is unidirectional and special care must be taken.

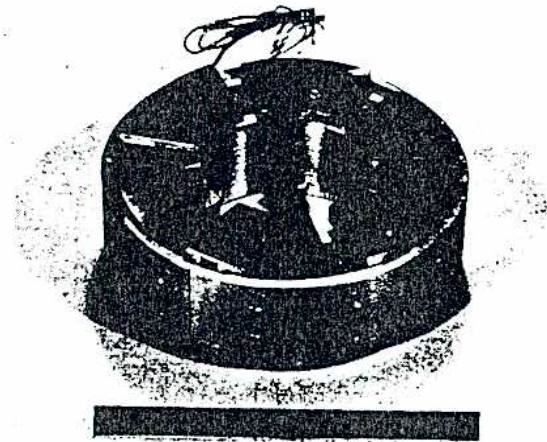


Fig. 7 Test bearing with graphite fiber composite facing

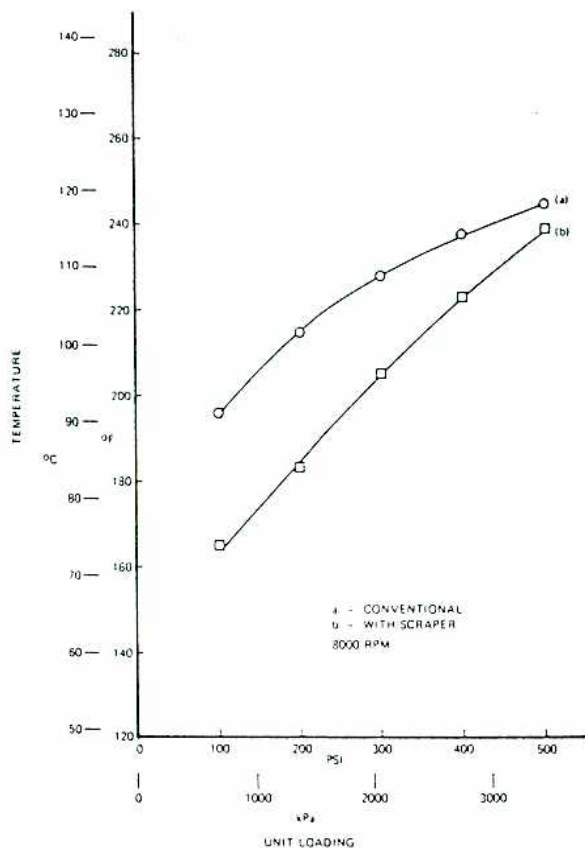


Fig. 8 Influence of trailing edge scraper on pad temperature for babbitted, steel-backed pads with center pivot

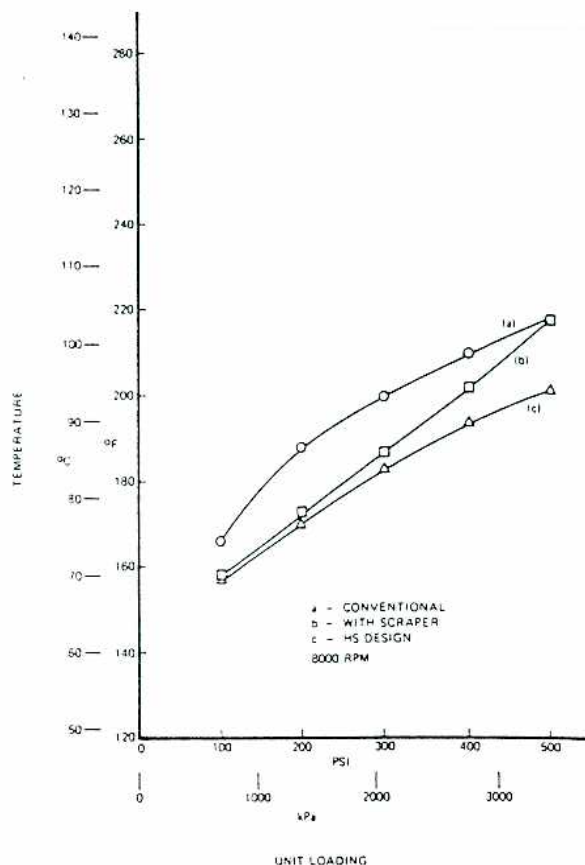


Fig. 10 Temperatures for center pivot aluminum pads of three different constructions

particularly with double thrust bearings, to insure proper assembly. In this test, the circumferentially offset pivot was located at 60 percent of the total pad angle.

Fig. 5 compares temperatures of four different center pivot pad constructions. The babbitt on steel construction is conventional and the babbitt on copper is relatively common. The copper alloy used was 99 percent Cu, 1 percent Cr. This provides substantially greater strength than 100 percent Cu while maintaining high thermal conductivity. The construction used in the aluminum pads was a solid pad of 6 percent tin aluminum. A hardened steel support button was inserted in the back face for the pivot, but this was true of all the pad constructions. Thus, the aluminum

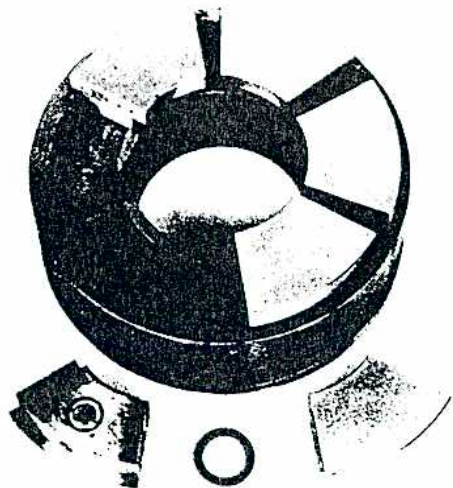


Fig. 9 HS design test bearing with aluminum pads

served as both the facing and the backing material. The Cu-Pb material was sprayed onto the steel backing using a 40 percent Pb content powder manufactured under a patented process [1] which insures uniform distribution of the lead throughout the copper matrix. This material is marketed commercially under the tradename Estalloy.

With center pivots, a measure of pad crowning works in conjunction with the viscosity variations to provide substantial load capacity where theory shows that a center pivot pad with a perfectly flat surface (no crowning) and isothermal conditions in the lubricant will have no load capacity [2, 3]. Pad crowning results from thermal and pressure gradients in the pad (and may, of course, be included as part of the manufacturing process). The relative magnitudes of the deflections resulting from these two factors will depend on the operating conditions in addition to the pad geometry. In general, at high speeds and light loads the thermal effects will dominate, while at low speeds and heavy loads the pressure effects will be more significant. Thus, when considering thrust pad construction, the modulus of elasticity and coefficient of thermal expansion of the backing material, and the thermal conductivity of the facing plus the backing are important factors. Although some distortion is desired in a centrally pivoted pad, the amount is small and the problem generally is to reduce the distortion to approach this limited amount.

By substituting copper for steel as the backing material, thermal distortions are reduced while pressure distortions are increased (for the same pad geometry, as used in these tests). In high speed bearings where thermal distortions may be large (with a steel backing) as compared to the pressure distortions, this substitution is beneficial. A further step in this direction is the substitution of bearing aluminum for both the facing and backing material. This is effective in further reducing thermal distortions as compared to a babbitt and copper combination, due to the elimination of the babbitt layer. The pressure distortions are increased slightly with the aluminum, but the net result is a de-



crease in total distortion as compared to babbitt and copper.

The primary advantage of Cu-Pb as the facing material is its ability to withstand higher temperatures than babbitt, although it also exhibited lower operating temperatures than babbitt faced pads in these tests. An operating limit of 177 deg C (350 deg F) may be considered for Cu-Pb where a comparable figure for tin babbitt is 149 deg C (300 deg F) [4]. This limit for babbitt has recently been studied and found to vary with the load imposed [5]. And for static conditions Cu-Pb can withstand temperatures approaching the melting point of lead 327 deg C (621 deg F). This can be of value where soakback temperatures present a problem with babbitt.

The use of Cu-Pb or aluminum as a substitution for babbitt is made with some loss of the excellent bearing properties of babbitt, of course, and this must be recognized [4].

Fig. 6 compares temperatures of three different offset pivot pad constructions. The operating temperatures are substantially lower than with comparable center pivot pads (see also Fig. 4), and the reduction in temperature realized by using aluminum in place of babbitt and steel is less than with center pivot pads. Of particular interest was the performance of the graphite fiber composite facing on a steel backing in this offset pivot construction. A photograph of this bearing is shown in Fig. 7. At the time of these tests the availability of these composites suitable for this application as a facing material was limited. Three different matrix materials were used; polypropylene, epoxy, and nylon. The data in Fig. 6 is from the composite using polypropylene as the matrix. The relatively low strength of this material limited its use. The combination of temperature and pressure at conditions much beyond those shown in Fig. 6 apparently deformed this matrix material as failures developed.

The relatively low operating temperatures obtained with these GFC materials as facings on steel backings were of interest, however. Due to the relatively low thermal conductivity of GFC perpendicular to the face (across the plane of the fibers), it was

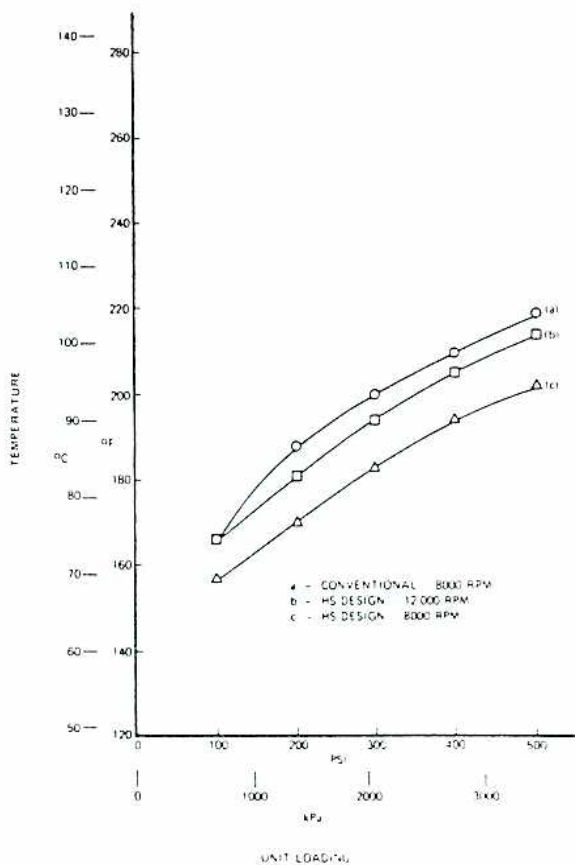


Fig. 11 Comparison of pad temperatures of HS and conventional designs; aluminum pads with center pivots

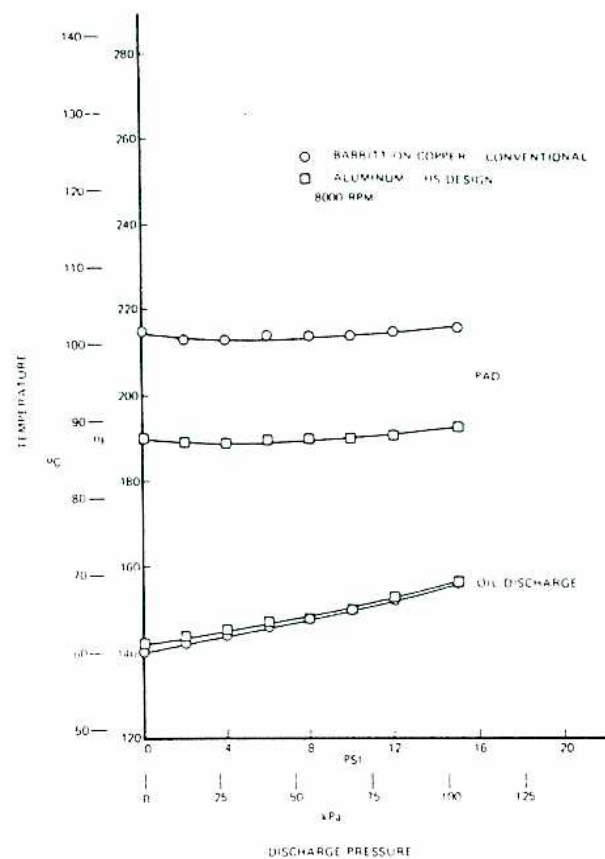


Fig. 12 Influence of discharge oil pressure on temperatures; center pivot pads, constant loading

found necessary to mount the pad thermocouple flush with the pad face and exposed to the oil film. This low thermal conductivity is believed to be a major contributing factor to the low pad temperatures obtained, however. The facing thus acts as an insulator for the backing with the end result that a high temperature drop develops across the facing thickness and a very low temperature difference then exists across the backing. This was confirmed by test by locating thermocouples at the pad back face and at the interface of the facing and the backing, in addition to the bearing face thermocouple. The thermal distortions of the backing are thus low. The overall distortions were thus the lowest of any pad construction tested. This is desirable for an offset pivot construction and believed to be the basis for the good performance, in this respect, obtained in the tests.

The GFC material used in the failure tests (Fig. 16) used epoxy as the matrix material. This performed best of the three tested with respect to maximum load capacity.

The problem of hot oil carryover in tilting pad thrust bearings and its influence on groove temperature (and thus bearing performance) has previously been discussed and studied [3, 6, 7]. To further study this phenomena in this test work, mechanical scrapers were secured to the trailing edges of test pads in an effort to physically remove the layer of hot oil adhering to the runner. These scrapers were made from 0.038-mm (0.0015-in.) shim stock to provide flexibility. They were cut to the radial length of the pad and mounted to the trailing edge with a portion (0.13 mm (0.005 in.)) extending above the pad surface. As the pad came within this distance of the runner, the scraper made contact with the runner and was flexed in the direction of rotation. This arrangement was not intended as a practical construction for production bearings but only as a test bearing to obtain information on the possible benefits. Temperature data comparing babbitt faced, steel backed pads with and without these scrapers is plotted in Fig. 8.

At the lower loadings the reduction in pad temperature ob-

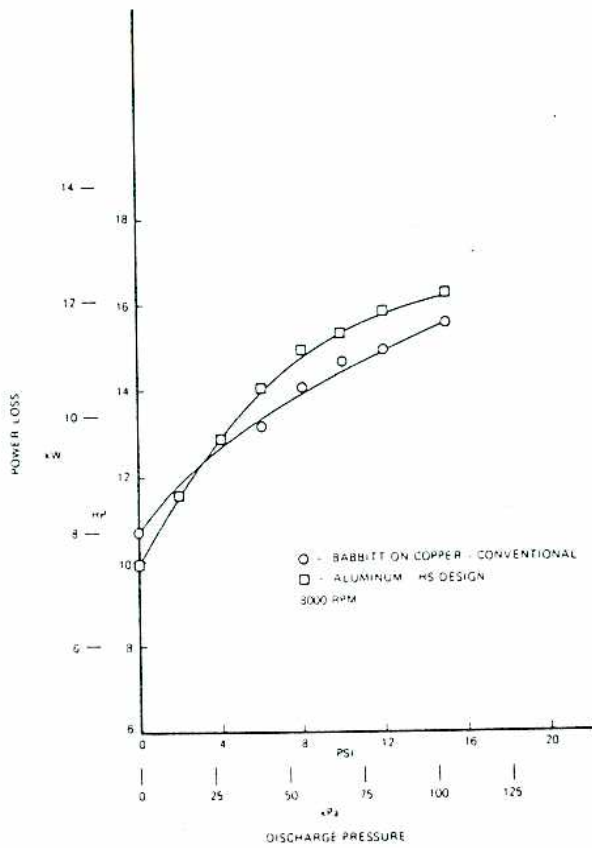


Fig. 13 Influence of discharge oil pressure on bearing power loss: center pivot pads, constant loading

tained by use of the scrapers was substantial. As the bearing load is increased, however, this advantage is reduced. It is theorized that at these higher loadings the portion of the oil being carried past the scraper represents a larger percentage of the trailing edge film, thus approaching the standard configuration. This test did, however, further substantiate that gains in performance are available by reducing the hot oil carryover.

Pursuing this further, the bearing shown in Fig. 9 was developed for test. In this bearing, the oil is supplied through passages within the retainer to the back face of each pad. The pad to retainer interface is sealed by an O-ring to permit the oil to transfer to the drilled passages within each pad. The O-ring provides the necessary seal and provides flexibility to allow the pad to tilt and to transmit the thrust load through the hardened button (in the

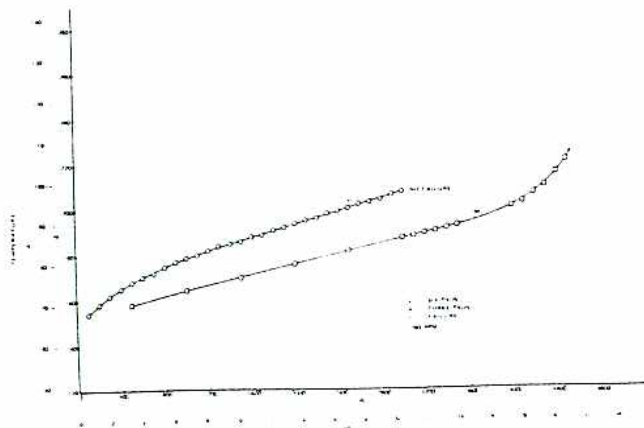


Fig. 14 Comparison of pad temperatures for 6 pad and 3 pad bearings, babbitt on steel, center pivot

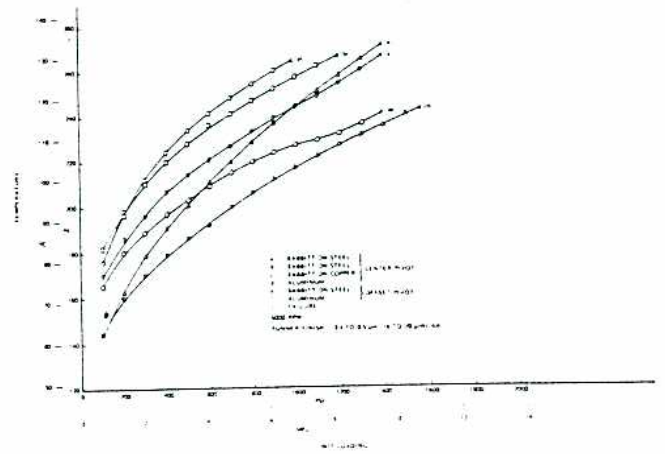


Fig. 15 Failure test results

back face) to the retainer. The outlet holes within the pads are sized and directed to provide jets of oil to the runner in the spaces between pads. These jets physically disturb the film of hot oil coming from each pad face, provide cool oil to replace this for the next pad, cool the runner, and also cool the pads internally.

Fig. 10 is a comparison of pad temperatures from three different pad configurations, all in aluminum: (a) conventional, (b) with the trailing edge scrapers, and (c) the design just described, referred to as the HS (high speed) design. Fig. 11 provides further evidence of the benefit of the HS design.

Tests were run with the discharge oil pressure as the variable to determine the influence of this on temperatures and power loss. Data from these tests with the copper backed, babbitted pads and with the HS design (aluminum pads) are plotted in Figs. 12 and 13. All showed similar characteristics. Oil discharge temperatures and power losses increased with increasing discharge pressures, while pad temperatures showed only a few degrees change. The zero discharge pressure condition corresponds to the open tangential discharge used in all other tests reported here. The maximum attainable discharge pressure was limited due to seal leakage. It is anticipated that the power loss would level off at higher discharge pressures; however, as the housing became completely flooded. The increase in loss indicated in these plots is believed to be primarily from the added shear and turbulence loss generated at the runner OD.

Comparison of the power loss figures for the two bearings at the zero discharge pressure condition (as at the other conditions also) shows only a small difference. The use of an open discharge has been relatively common in the United States while European

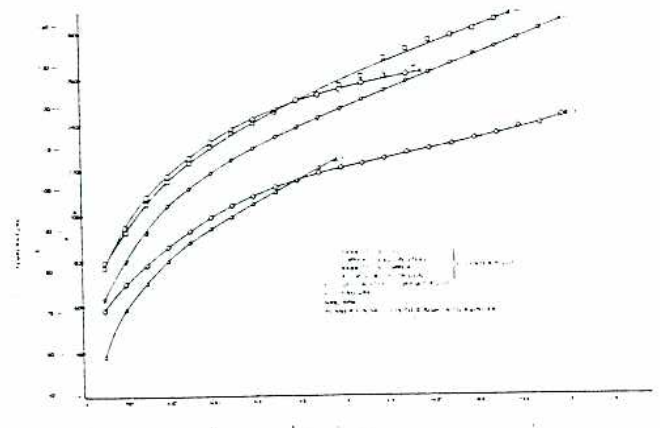


Fig. 16 Failure test results



practice has primarily been to flood the bearing housing by throttling the outlet. In high speed bearings this latter practice leads to unnecessarily high power losses. A comparison of three systems of lubricating tilting pad thrust bearings has recently been made [8]. In this work the flooded system (throttled outlet), throttled (orificed) inlet system, and a directed flow system are compared. This latter is similar to the HS design discussed here. It appears that the throttled inlet system reported in [8] did not incorporate a tangential discharge, as in these tests, however.

From this test data the primary advantage of the HS design over a conventional bearing with open tangential discharge is seen to be reduced operating temperatures and the associated increase in load capacity, as both operate with essentially the same minimized power loss.

The data in Fig. 14 was obtained from a portion of some test work done under contract to Union Carbide Corporation, Nuclear Division, of Oak Ridge, Tennessee. It must be noted that the operating conditions were different from the other tests discussed here. A shaft speed of 1780 rpm and the use of a heavy medium turbine oil were the primary changes. The thrust pads were conventional, steel backed, babbitted with center pivots. Of specific interest were the relatively high psi loadings obtained without failure at these operating conditions as compared to those at higher speeds, Figs. 15 and 16. Data is also shown for a 3 pad bearing (every other pad removed) which reflects reduced temperatures with this arrangement as compared to the full complement of six pads, at equal unit loadings.

Data from the failure tests are given in Figs. 15 and 16. Two plots were necessary as the runner finish was improved for the Fig. 16 tests. Failure tests on two of the pad constructions were made on both runner finish conditions: 1) steel backed, babbitted, and 2) copper backed, babbitted (curves (a) and (c) of Figs. 15 and 16 correspond). Comparison shows the same percentage increase in ultimate load capacity for both of these when used in conjunction with the smoother runner finish.

## Discussion

The use of offset pivots provides for significant reductions in pad operating temperatures at normal loadings. It is interesting to note, however, that although increased ultimate load capacities also result, the increases are not substantially larger than with comparable center pivot construction.

The use of pads of solid aluminum construction is believed to be somewhat unique. In addition to the test work reported here, tilting pad bearings of this type have been in field use in turbo-expander applications with over two years trouble free service on the initial installations. In addition to lower temperatures and/or

higher load capacity this pad construction can be of value where the presence of copper in the system is restricted.

Copper-lead as a bearing material has been in use for many years. The metal spray method of application of this as a facing makes it economically feasible for low quantity and special design bearings. And with the Estalloy powders, high lead contents can be used without segregation problems.

The HS (high speed) design arrangement for supplying lube oil individually through the pads and in jets to the runner can be incorporated with any of the pad material combinations to provide additional reduction in temperatures.

This HS design, the GFC construction, and a pad retaining means are developments from this test program which are the subject of a U. S. Patent [9] and a patent application, currently allowed but not yet issued.

As evidenced by the test results at lower speeds, bearing load capacity is strongly influenced by surface velocity. Care must be used in translating load capacity data from a specific set of test conditions to a different set of conditions for an application.

## Acknowledgment

The author wishes to express his appreciation to the Union Carbide Corporation, Nuclear Division, of Oak Ridge, Tennessee, for permission to publish the data contained in Fig. 14.

## References

- 1 U. S. Patent No. 3,556,779, *Copper-Lead Alloy*.
- 2 Raimondi, A. A., and Boyd, J., "The Influence of Surface Profile on the Load Capacity of Thrust Bearings With Centrally Pivoted Pads," *TRANS. ASME*, Vol. 77, 1955, pp. 321-330.
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- 9 U. S. Patent No. 3,814,487, *Thrust Bearing With Pad-Retaining and Lubrication Means*.

## DISCUSSION

### R. C. Elwell<sup>2</sup>

From a practical standpoint, this paper extends the useful range of measured data on a very important bearing type. Because they are used in thousands of critical applications, we need this type of detailed knowledge. This now makes available metal temperature and other data for the following standard bearing sizes:

Outside Diameter		Reference
mm	In.	
152.4	6	Gardner, this paper
266.7	10 <sup>1/2</sup>	Gregory, reference [10]
342.9	13 <sup>1/2</sup>	Elwell, reference [6]
381.0	15	Elwell, reference [6]

In all cases, light turbine oil was the lubricant. Oil groove temperatures, power loss, failure loads and other data are not consistently given. Nevertheless, this is a significant improvement in di-

rectly available performance data over the situation only five years ago.

On the subject of failures, I would like to request clarification of the author's position on allowable babbitt temperature limits at the 75-75 percent reference position. This location was recommended as a standard in reference [6], so that data could be compared between various tests on a realistic basis. It will not be the maximum temperature location except in some combinations of loads and speeds. Figs. 15 and 16 show several failures at babbitt temperatures in the range of 126 to 139°C (260 to 282°F). Since an "operating limit" of 149°C (300°F) is suggested in the section entitled "Test Results," should this not be modified to a lower *indicated* temperature at the 75-75 location?

Further on failures, the statement is made that evidence of pad to runner contact was apparent. Could this contact be described? Was there gross melting, appearance of blackened spots over the pad pivot locations, runner scoring, or what? We are all faced with

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analyzing occasional high speed failures and do not often have them as well documented as this. A photograph or two would be useful.

I would like to request two items of basic data: viscosity versus temperature for the two oils (this is probably available in a convenient reference), and the mechanical properties of the "high strength" copper backing. It is described only as 99 percent copper and 1 percent chrome.

The author's experience with circumferentially offset pivots generally agrees with ours. Fig. 4 is typical of the improvements we have also obtained in reduced babbitt temperatures in the past several years. To be sure of an important detail, would the author confirm that these tests were done on six-pad, rather than eight-pad bearings? I think this improvement in performance is generally worth the extra care needed in assembly. The author is correct in pointing out the vulnerability of double bearings.

One of the topics on which further knowledge is especially needed is the oil film entering temperature as functions of speed and load. The term "hot oil carryover" is used in this paper, tending to perpetuate the theory that the entering film is largely made up of "used" oil from the previous pad. Probably the most work of significance on this subject has been done by C.M.M. Ettles, either by himself or with others.

A major revision in this theory was unfortunately published obscurely [11].<sup>3</sup> In their closing paragraph there, Ettles and Cameron state "The factor which dominates hot oil carry-over is the temperature of the moving surface.—Lubricant which approaches this surface becomes hot and is drawn into the film. The term "carry-over" is thus slightly misleading, but preheating of the lubricant entry charge with consequent loss in viscosity still takes place." The topic is pursued in still further detail by Ettles in reference [12].

A reliable physical explanation for the oil film entering temperature is needed to improve the accuracy of the computer programs with which detailed bearing film behavior is computed.

The author is to be complimented on the data on runner surface finish in Figs. 15 and 16. It is unique to my knowledge, and again is a very practical result.

### Additional References

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11 Ettles, C. M. M., and Cameron, A., Author's Closure to Discussion of "Considerations of Flow in a Bearing Groove," JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Vol. 91, Series F, No. 1, Jan. 1969, page 212.

12 Ettles, C. M. M., "Hot Oil Carry-Over in Thrust Bearings," *Proceedings of IME*, Vol. 184, Part 3L, Proceedings in Tribology Convention 1970, pp. 75-81.

### K. J. Smith<sup>4</sup>

As a representative of a user company of this type of thrust bearing, the data presented is interesting and worthy of consideration for product improvement programs. Some of the concepts promoted by Mr. Gardner are familiar to us and have been used on our product compressors. Relevant comments and questions are based upon our experiences during shop and field testing and are more directed to application than theory.

<sup>3</sup> Numbers 10-12 in brackets designate Additional References at end of discussion.

<sup>4</sup> Director-Rotating Product Engineering, Cooper-Bessemer Co., Mount Vernon, Ohio.

One of the usual requirements placed upon the design of petrochemical and gas industry machinery is to provide leveling-links in the thrust bearings. The premise behind this criteria is that any minor deviations in individual pad-to-runner distance or parallelism will be compensated by adjustment thru the linkage. If appreciable pad distortions due to thermal and pressure effects occur, as reported in this paper and which corroborate measurements C-B has made, one has to question whether leveling-links really provide the intended benefits. This is especially true where high rotary shaft speeds are used. Would you comment on this thought please, Mr. Gardner? Also, if deformation occurs, aren't the benefits of offset pivots negated or greatly reduced?

The test using wipers to rid the thrust collar (runner) of hot oil was an interesting idea. The results indicate the beneficial effects of properly supplying the bearing oil both in regards to quantity and location. Mr. Gardner has hypothesized that the drop in effectiveness of the scraper as bearing loads are increased represents a larger leakage flow past the scraper as compared with the film thickness. Could it be that the oil path undergoes a change in direction? It has been amply demonstrated that the thrust collar (runner) is a reasonably effective pump with negative pressures often occurring near the shaft bore and large positive pressures at the OD. The established pressure field in conjunction with the boundary layer shearing action dictate the "new oil" path. As the oil film thickness decreases due to increased axial load on the runner, both the pressure field and shearing force change, the latter somewhat more I would suspect.

We have employed both systems of oil supply—throttled (inlet orifice) and backpressured (discharge orifice) in our equipment. By observation, the discharge pressure rise capability exceeded 70 psi in a 12-inch bearing at 10,000 rpm for the discharge orifice system. This power adsorption is appreciable and certainly favors the use of inlet orificing whenever possible. As Mr. Gardner has also reported, an appreciable discharge oil temperature decrease occurred in a recent development test at C-B when every other pad of a conventional thrust bearing was removed under the same load condition.

One of the fundamental problems faced in the field with our machinery customer requirements is start-up of rotation under high thrust loading. This leads to the need for either oil retention in the pad face until the thrust collar has turned sufficiently to build the oil wedge, or a small natural lubricity in the bearing face material to circumvent instantaneous contact. I would be quite concerned about the use of Al in this instance because of its galling tendencies. A ranking of the different materials used in the tests on this basis would be interesting. Also, the HS type of bearing may be decidedly deficient in this respect.

The directed lubrication, or HS type of bearing, holds much promise as a provider of very high unit load capability. Oil directed to and through recognized hot spots on the pads in combination with selected pad face materials gives appearances of potentially doubling design unit loads. (Subject to solving or proving a solution to the start-up problem mentioned in the previous paragraph.) The ability to achieve these increased unit loads results in friction HP reductions because of smaller size for a given load as indicated by Mr. Gardner. If the bearing oil is used more efficiently through directed lubrication, it would also seem probable that it would take less oil and less friction HP to do the same job as with a conventional bearing with open tangential discharge. From the testing procedure, I am led to believe the same oil flow was used at the same speed regardless if of HS or conventional design. Were tests run with reduced oil flows to the HS bearing? Some investigations made at C-B would indicate you can appreciably decrease flow to the directed lubrication type bearings without an apparent rapid increase in pad temperature or bearing failure and thus promote an even greater FHP reduction. Would Mr. Gardner comment on this, please?



The author appreciates the reviews and comments by Mr. Elwell and Mr. Smith. With respect to questions raised in these discussions and requests for clarification in certain areas, I will respond first to those of Mr. Elwell.

The allowable babbitt temperature limit of 149°C (300°F) noted in the paper was given as listed in reference 4, primarily for comparison to the limit for Cu-Pb of 177°C (350°F) given in this same source. It was also noted that this limit varies with the operating conditions and it is well to emphasize this point. And, as Mr. Elwell points out, when discussing pad temperatures it is important to associate them with a location as they vary substantially over the face of a pad.

The pad damage generally found after a "failure" is best described as light wiping. The damage was limited to this extent because the test setup allowed a good visual indication of the pad temperature and quick dumping of the load when this temperature exhibited the sharp increase associated with pad to runner contact. Runner damage was thus avoided in all cases except one failure of aluminum pads where operator response was not sufficiently quick.

The viscosity vs temperature data requested is noted below:

Oil	Viscosity @ 38°C (100°F)	Viscosity @ 99°C (210°F)
Light	4.3 microreyns	.58 microreyns
Heavy Medium	9.0 microreyns	.93 microreyns

The "high strength" copper used as a pad backing material has a tensile strength of 53,000 PSI and an elongation of 17 percent. It has a modulus of elasticity of about 17 million psi.

The tests of the offset pivot pads were made on six pad bearings, similar in all other respects to the center pivot pads.

With respect to the use of the term "hot oil carryover", I would suggest that the term "heat carryover" may more accurately describe the situation. However, I believe that our tests with the mechanical scraper indicate that some "hot oil" is "carried over" in a conventional bearing as the scraper did result in lower pad operating temperatures but not through any cooling means other than physically redirecting the oil.

Mr. Smith questions whether leveling links actually provide the benefits intended in view of pad distortions due to thermal and pressure effects. The intended benefit of the leveling (equalizing)

links is to equalize the load on all the pads. This equalizing action is provided by the links between pads, these being a simple arrangement of mechanical levers. Equalization does occur, except under certain specific operating conditions, and strong evidence of this uniformity of loading was provided in the remarkably uniform damage patterns from pad to pad in the failure tests reported here. The two conditions which limit the equalizing action are: (a) a change in shaft (collar) to bearing angular alignment while under load and, b) a runout of the collar face with respect to the axis of shaft rotation. Both restrict the equalizing action due to the mechanical nature of the leveling links. Tests have been made<sup>5</sup> showing that the friction between links under load limits the equalization for the condition in a). This, plus the mass of the link elements prevents equalization under condition (b) for most shaft speeds encountered in rotating machinery. These are not normally serious restrictions in obtaining the benefits of the equalizing system, however. Thus, the pad distortions influence the load capacity of the bearing, but generally not the load equalization.

The effect of pad deformations on pads with offset pivots is to reduce the load capacity and offset in part the advantage of using such a design. It is undoubtedly these deformations that reduce the ultimate load capacity of offset pivot pads to something less than might be expected from viewing temperature reductions in such pads (as compared to center pivot pads) at normal loadings. However, as these tests showed, there are benefits, including ultimate load capacity, in using offset pivots.

With respect to the problem of starting thrust bearings under load it should be noted that this particular problem was not studied in these tests and thus actual data cannot be supplied to substantiate any comparison. I believe, however, that the three primary materials tested would be ranked in the following order with respect to suitability for starting under load: 1) Babbitt, 2) Copper-Lead and, 3) Bearing Aluminum.

I don't believe that the HS type bearing would be deficient in starting under load as compared to a conventional bearing, as oil is available at the pad leading edge in either construction assuming, of course, that oil pressure is supplied prior to shaft rotation.

With respect to the oil flows used, for any given speed they were the same regardless of bearing type. However, I would anticipate, as Mr. Smith indicates, that flows could safely be reduced with an HS type bearing as all of the oil is directed where needed by the bearing.

<sup>5</sup>Gustafson, R. E., "Behavior of a Pivoted-Pad Thrust Bearing During Start-Up," JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 89, No. 2, Apr. 1967, pp. 134-142.