GAS BEARING APPLICATIONS IN SPACE POWER TURBOMACHINERY

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INTRODUCTION

Since its inception a number of years ago, our space program has demonstrated a steadily increasing need for auxiliary power on both unmanned and manned spacecraft as the weight and complexity of these craft have increased. Initial unmanned satellites required only a few watts of power for operating equipment and for transmitting radio signals. Since that time, however, the manned space flight program and projected space missions for the future have demonstrated a steadily rising demand for nonpropulsive power. Figure 1 illustrates the estimated electric power requirements for various types of missions as noted on the figure. Note that Tiros I back in 1960 required about 20 watts of nonpropulsive power and that the projected power requirements of manned planetary missions in the 1970's are on the order of 100 kilowatts or greater.

While storage batteries were capable of supplying the auxiliary power requirements for early missions, they will not be satisfactory for future missions because of excessive weight. Figure 2 illustrates the specific weight in pounds per kilowatt electric plotted against the applicable electrical power range for various types of systems. Our present concern is with dynamic power conversion units. These include solar, isotope, or reactor powered turbogenerator units. Two types of turbogenerator power systems are presently under development. These include Rankine cycle systems utilizing liquid metals (usually mercury or one of the alkali liquid metals) as the working fluid, and Brayton or gas cycle power systems. Each of these turbogenerator system types has, of course, certain advantages and disadvantages. We will not delve into those today except to say that the gas cycle system may be somewhat easier to develop and to achieve operational status than will the liquid metal systems because of the magnitude of the various technological problems involved. In the lower range of the total electric power range for which turbogenerator systems are thought to be feasible, that is, from about 2 kilowatts up to 10 000 kilowatts of electric power, the Brayton system is competitive weight-wise with the Rankine system. In the higher portion of this power range the Rankine system would probably be significantly superior, weight-wise, to the Brayton system. However, at the present time, we are concerned with the development of power systems which operate in the range of 2 to 15 kilowatts electric.

Regardless of what type system is used, it must be a closed cycle system because it must operate in space. It cannot be an open cycle system such as a turbojet engine on a commercial aircraft because there is no atmosphere available in space. So the working fluid must be retained within the boundaries of the system. Weight and size are extremely important factors in system design. This makes it mandatory to use very high speed rotating units so that the maximum net power output per pound is achieved. A schematic of a typical Brayton

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cycle space power system is shown on figure 3. Here we show a reactor powered system, but it could just as well be powered by an isotope source of energy or solar energy. The actual systems which are under development now utilize solar power for the source of energy. As we can see, the unit consists of a radiator, a power source, a turbine, the compressor and an alternator.

We are concerned directly with the bearing problems associated with the turbine, compressor and alternator components. There are two basic alternatives to be considered in designing the bearing system for a turbocompressor or a turboalternator rotating unit. First of all oil lubricated rolling bearings, which are certainly more within the present state of the art for high speed rotating machinery, are a candidate. The use of conventional bearing types with all the auxiliary equipment required, however, leads to much greater complexity and a considerable penalty in weight. There is also the problem of cooling the bearing to within the useful temperature range for conventional oils. This, of course, would result not only in a penalty to cycle efficiency but might also entail some very difficult design problems. So in order to avoid these complexities we consider the use of fluid film bearings lubricated by the working fluid. Because of the long life requirements of these systems (on the order of 10 000 hr without servicing), it is mandatory to use fluid film bearings. For Brayton systems this means using gas lubricated bearings.

At the present time, NASA has under development a Brayton cycle space power system for delivery of a net power output of 8.5 kilowatts. The operating pressure and temperature ranges of this system are shown in table I. The system uses argon as the working fluid and operates between a compressor inlet temperature of 80° F and a turbine inlet temperature of 1490° F. For the full power output operating condition pressures in the system range from a compressor inlet pressure of 6 psia to a compressor outlet pressure of 13.8 psia. For reduced power output operation pressures range between 2.8 and 6 psia. The approximate pressure in the bearing cavities is 12 psia for the full power operating condition with a 30-foot-diameter solar collector, and 6 psia for reduced power operation with a 20-foot-diameter collector.

The low pressure ratio of the system dictates the use of either self-acting or hybrid bearings. Purely hydrostatic bearings would not have sufficient load capacity unless the bearings were made quite large. The necessity for the power system to operate in a zero-gravity environment at high speeds points up the extreme importance of the stability characteristics of the bearings. This becomes one of the principal problems which must be dealt with in designing a bearing system. The high temperatures which exist in portions of the machine together with the possibility of high temperature gradients makes the tolerance of thermal distortions and stresses in the bearings important and emphasizes the need for careful thermal studies and bearing material choices.

NASA is presently developing and evaluating two turbocompressors, one utilizing a radial flow compressor and a radial flow turbine, and a second unit utilizing an axial flow compressor and an axial flow turbine. A turboalternator

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on a separate shaft is also being developed. These represent three separate pieces of research machinery. The bearing design philosophy, operating conditions and problems peculiar to each one of these machines will be discussed at length.

RADIAL FLOW TURBOCOMPRESSOR

Design

The design arrangement for the radial flow turbocompressor is shown in figure 4. Both the compressor and the turbine are mounted outboard of the two journal bearings with a thrust bearing between the two journal bearings. For aerodynamic reasons, a rotative speed of 38 500 rpm was chosen for this machine. As stated above, the operation of this unit at high speeds and the zero-gravity environment makes the stability characteristics of journal bearings of prime importance.

Journal Bearings

If we consider the use of 360° self-acting plain cylindrical bearings for rotor support, we quickly find that bearings of this type with normal radial clearances will be unstable at the operating speed. The maximum radial clearance permissible for stable operation is on the order of 0.0001 to 0.0002 inch. Such clearances are, of course, impractical so that plain cylindrical journal bearings cannot be considered for applications like these.

Pivoted-pad bearings have long been known for their inherent stability characteristics so it was only logical that such bearings be chosen for the bearing development program. A number of three-pad bearing configurations were evaluated. These utilized either a ball and socket pivot or a pivot system consisting of a fixed stem connected to a flexible diaphragm. Means for introducing external pressurization for use at startup and shutdown was incorporated in all bearings. Two length diameter ratios (0.5 and 0.75) and a number of clearance ratios were investigated. Two schemes for external pressurization, a single orifice at the center of each pad and four orifices located close to the perimeter of each pad, were evaluated. Various combinations of resilient and rigid pad mounts were also evaluated. The final journal bearing pad configuration adopted for the turbocompressor is shown schematically in figure 5 and design data are given in table II. As shown on figure 5, a fitted ball and socket type of pivot is used and high pressure argon is introduced through a small line passing through the pivot to the four orifices located on the surface of each pad. High pressure argon is introduced to float the pads before the shaft begins to rotate, and the bearings are kept pressurized until the rotative speed reaches approximately 20 000 rpm. At this speed the self-acting load capacity of the bearings is sufficient to support the load so that external pressure is no longer required.

The fitted ball and socket type of pivot is used to minimize leakage of high pressure argon through the pivot while the bearings are running in a hydrostatic condition. The use of a single orifice at the center of the pad in the high pressure zone when the bearing is running self-acting causes trouble because of back leakage through the orifices and lines and out through the pivot. This results in a decrease in the self-acting load capacity. Therefore, four orifices, located close to the perimeter of each pad where the self-acting pressure levels are considerably lower than they are in the region of the pivot, are used. The ball portion of the pivot of one pad in each journal bearing is connected to the machine frame through a thin circular diaphragm. The stiffness of this pad mount can easily be controlled by adjusting the thickness of the diaphragm. The mount configuration chosen for the journal bearing pads is shown schematically on figure 6. Two of the pads have their pivots fastened rigidly to the frame of the machine while the third pad is resiliently mounted through the diaphragm. A diaphragm spring stiffness of 2000 pounds per inch is used. The principal purpose of resiliently mounting one of the bearing shoes is to allow the bearing to adjust itself to thermal changes within the machine as it heats up toward the steady-state operating condition.

As illustrated in figure 7, the bearing shoes are assembled with a certain initial preload. This initial preload comes from a diaphragm spring force which actually pushes the pads against the stationary journal in the initial setup of the machine. The magnitude of the initial preload is indicated by point 1. When the shaft is rotating at design speed without any heating effects there is, of course, a finite film thickness at the pivots and this results in an increase in the bearing load from the initial preload at point 1 to point 2. The slope of the line connecting points 1 and 2 indicates the linear spring rate of the resilient mount. As the machine comes up to temperature there may be a differential growth of the shaft and the pads in their mounts which results in an increase in the bearing load from point 2 to point 3. If the pad mounts were perfectly rigid rather than resilient, the resulting bearing loads after differential expansion due to heating and centrifugal effects would be considerably higher than with the resilient mount.

The importance of thermal accomodation in the bearings is illustrated in figure 8, which shows the relative growth between the bearing journal and its carrier as a function of time. It takes approximately 60 minutes of operation for the machine to reach a steady-state temperature distribution. During this time the relative growth between the bearing journal and the pad carrier is significant enough to create bearing overloads with rigid mounts. If the transient thermal conditions are translated into changes in bearing load, the curves shown on figure 9 result. For the turbine bearing the relative load increases about 25 percent after which it starts to diminish once again back toward the original value.

In designing a resilient bearing mount system it must be recognized that the spring stiffness of the mount has an effect on the system critical speeds, and the magnitude of the shaft excursion under a rotating unbalance load. A problem of pad resonances in the radial direction may exist if the mount stiffness is too soft. Shaft excursions are limited by the clearances in labyrinth seals and between rotating blades and shrouds and other close clearance parts. The effect of mount stiffness on system critical speeds is illustrated in figure 10. There are two rigid body criticals which are of concern together with the third or flexural critical. The two rigid body criticals are influenced by the bearing mount stiffness and also by the stiffness of the bearing film. Fortunately these two criticals occur at rather low speeds so that they can be passed through rather rapidly as the unit accelerates up to design speed. It is desirable to have the rigid body criticals either above the operating speed range, or relatively low compared to the design speed so that they can be passed through when the energy input is fairly low. The rigid body critical speeds occur at approximately 7500 rpm and 12 000 rpm. The flexural or free-free critical is about 55 000 rpm which is far enough above the over-speed condition so that amplification of loads is not significant. Figure 11 illustrates approximate bearing loads over the entire speed range for a condition in which a center of gravity eccentricity is approximately 1 mil. This would be an excessive unbalance as can be seen by the fact that the rotating load above the second critical would be about 40 pounds. For a center of gravity eccentricity of 1/10 of a mil this rotating load would be approximately 4 pounds.

Journal Bearing Development Problems

A number of problems were encountered in developing satisfactory journal bearings for the radial flow turbocompressor. In the initial phases of the bearing test program it was noted that shaft crowning occurred in the bearings due to heat generation in the gas film. This resulted in wiping at the axial midpoint of the bearing due to loss of load capacity. To reduce the axial temperature gradient in both the shaft and the pads, a heavy shunt was added to the turbocompressor shaft. This consisted of a rather heavy wall copper cylinder placed inside the hollow shaft. This resulted in an appreciable reduction in the axial temperature gradients and in the crowning of the shaft at the location of the journal bearings.

Some difficulty was also encountered with pad spragging or hanging up of the leading edges of the pads. This can occur if one of the pads is allowed to become completely unloaded. It also has a greater tendency to occur in very tight clearance bearings. Figure 12 illustrates the moment acting on the pad about the pivot for a small and a large clearance bearing operating under hybrid conditions at 20 000 rpm. The sign of the moment, which should remain positive to prevent leading edge spragging, is much more favorable for a large bearing clearance.

Persistent shaft whirls at a frequency equal to the first system critical speed were also encountered in the two bearing test rig and also in operation of the turbocompressor. It has been found that a preload of approximately 8 pounds is required to prevent the occurrence of this shaft whirl. The cause of this whirl is, at this time, unresolved. Theory states that tilting pad journal bearings should not be unstable, but present theory does not take into account pivot restraint on pad motion. Pivot friction may be a factor in promoting this type of shaft whirl.

Journal Bearing Power Loss

Bearing power loss also becomes a problem in this type of machine. In very large high power machines it is of little consequence, but in small machines with a limited power output poor design of the bearings can result in excessive power loss within the bearings with a resulting severe penalty on cycle efficiency. Increasing the machined in clearance between the shoe and the shaft results in decreased power consumption as illustrated on figure 13.

As shown in figure 13 the frictional power loss is decreased as the machined in bearing clearance increases for a specific minimum film thickness. At the same time, the load capacity, as illustrated by the curve for constant minimum film thickness against bearing clearance, does not change markedly. The journal bearings for the radial flow turbocompressor were designed with an initial clearance of 2.3 mils and this decreases to about 2.1 mils at operating speed and temperature. For a minimum film thickness in the region of 0.4 mil this is about optimum as regards load capacity.

While the frictional power loss does not show a sharp decrease with bearing clearance at a specific minimum film thickness, it has been found that the minimum film thickness at which a pivoted pad bearing will operate stably can be increased with increasing clearance. The preload (the ratio of the minimum film thickness to the machined in clearance), required for stable operation is approximately constant so that, as the bearing clearance is increased, the safe minimum film thickness is also increased. As an example, it may be necessary to operate a bearing with a clearance of 1 mil at a minimum film thickness of, say, 0.237 mil which would make the frictional power loss about 37 watts. At the same time a bearing with 2.3 mils clearance might operate stably with a minimum film thickness of 0.475 mil which would decrease the power loss to approximately 17 watts, a very appreciable reduction. The strong effect of minimum film thickness on both load capacity and friction is illustrated in figure 14. The approximate design point minimum film thickness for the journal bearings in the radial flow turbocompressor is 0.3 mil. This makes the power loss per pad about 20 watts, while the bearing is carrying a load of about 18 pounds.

Thrust Bearings

At startup the thrust load in the radial flow turbocompressor is approximately 60 pounds in the direction from the compressor. As the unit comes up to speed the thrust load reverses going toward the direction from the turbine until it reaches a steady-state magnitude of about 30 pounds at design speed. Thus, the machine is equipped with a dual thrust bearing capable of supporting thrust loads in either direction. Both thrust load faces are initially pressurized with argon introduced through a number of orifices. The self-acting or forward thrust bearing face (which is a Rayleigh step bearing) is provided with 4 orifices and the reverse thrust bearing face (which is plain) is provided with 6 orifices. Until the unit reaches design speed the thrust bearing is pressurized and the thrust load is carried on a hydrostatic gas film. When the unit is up to speed, external pressurization is removed and the Rayleigh step bearing carries the 30-pound load. Design features of the forward thrust bearing are shown in table II and also on figure 15. The predicted performance of the Rayleigh step bearing at design speed is shown in figure 16. Under a 30-pound load in a 12 psia ambient pressure environment the minimum film thickness is approximately 0.4 mil, and the bearing power loss is about 150 watts.

Because of misalinement and distortions which result from thermal and mechanical stressing, it is necessary to mount the thrust bearing assembly on a gimbal. The gimbal arrangement, shown schematically on figure 17, includes two sets of 1/4-inch pins mounted on orthogonal axes about which the gimbal assembly is free to pivot. In actual operation of the turbocompressor the ability of the gimbal and thrust bearing assembly to follow run-out and distortion of the shaft runner has been fair. Some cyclical variations in thrust bearing film thickness are observed in operating the unit. These have been shown to be fairly small compared to the average film thickness.

Some of the material coating and finishing specifications for both the journal and thrust bearings in the radial flow turbocompressor are listed in table III.

AXIAL FLOW TURBOCOMPRESSOR

Design

A second turbocompressor or gas generator which utilizes axial flow components is also under development. Figure 18 shows a schematic cross section of the axial flow turbocompressor. The compressor consists of 6 subsonic stages and a turbine of a single stage. The thrust bearings and the number one journal bearing are located within the compressor inlet to the left of the compressor stages. The number two journal bearing is located between the compressor and the turbine.

Bearing Design

Design data for both journal bearings and the thrust bearing are shown on table IV. Both journal bearings are four-pad, tilting pad bearings. The number one journal bearing is 1.5 inches in diameter and the number two journal bearing is 2.125 inches in diameter so that, in contrast to the radial flow machine, there is asymmetry in the journal bearings. As in the radial flow turbocompressor, the thrust bearing is a dual bearing fitted with orifices on both faces so that thrust load in both the forward and reverse direction can be supported hydrostatically. Both hydrostatic faces are inherently compensated for maximum stability. The self-acting face of the thrust bearing in the axial flow turbocompressor is an inward pumping spiral groove bearing with an outside radius of 1.625 inches and an inside radius of 0.40 inch. The rotor weight is about 11 pounds, and bearing loads are 3.5 pounds on the number one journal and 7.5 pounds on the number two journal. Thrust loads vary widely depending on the position of the unit. With the compressor up and the unit oriented vertically the thrust bearing is required to support the difference between the aerodynamic thrust and the rotor weight which is just about 5 pounds. With the compressor oriented down the maximum operating thrust load would be about 25 pounds. Transient thrust loads at start up vary from about 50 pounds in the reverse direction to 100 pounds in the forward direction. These loads are supported hydrostatically.

Because of aerodynamic considerations a design rotative speed of 50 000 rpm was selected for the axial flow turbocompressor. The unit is also required to operate at an overspeed condition of 60 000 rpm. The problems are quite similar to those in the radial flow turbocompressor, although the slightly higher design speed makes the bearing stability problem a bit more severe. This is the reason for again choosing tilting pad journal bearings for this lightly loaded, very high speed application. The inward pumping spiral groove design was chosen for the thrust bearing because it offers excellent load capacity. Alternatively to this type of bearing one might consider the Rayleigh step bearing, which is used in the radial flow turbocompressor, or a self-equalizing tilting pad thrust bearing. This latter type, however, has very limited load capacity so that it would be necessary to use a larger diameter bearing and runner. A tilting pad thrust bearing would have better alinement capability, but the achievement of satisfactory operation of the many linkages in the self-equalizing system at high temperatures may require a major development program.

Four pad bearings were chosen over three pad bearings because they offer a more uniform stiffness and also because they simplify the problem of keeping the pad mass low enough so that the various natural frequencies of the pad can be kept above the operating speed range of the unit. Journal bearing diameters are more or less dictated by the resulting system critical speeds. Since the speed is high care must be taken to keep the flexural critical speed well above the overspeed operating condition. Shaft stiffness considerations determine the minimum journal bearing diameters that can be employed. This is critical in the number two bearing because of heat generation. Attempts to reduce the number two journal bearing diameter to below $2\frac{1}{8}$ inches would lead to a flexural critical speed which is not sufficiently far above the overspeed operating condition. This would lead to a significant amplification of bearing loads at the overspeed operating condition.

In some of the early lectures design methods were presented for various types of gas bearings. These can be used with reliability provided that the bearings operate in an isothermal environment so that bearing geometries are not distorted. In addition bearing material technology is somewhat limited so that bearing temperatures must be kept within the limits of actual experience to avoid an extensive materials development program. To satisfy the maximum bearing temperature requirement, studies were conducted with the axial flow turbocompressor to determine the best possible cooling techniques to be used. A maximum journal bearing temperature of 400° F was the goal of the cooling system investigation. Table V outlines the various cooling techniques investigated and figure 19 illustrates schematically the final cooling system design and the flow path of both the liquid and gas coolants. Since the number two journal bearing is located adjacent to the turbine, its cooling requirements are the most severe and the bearing temperatures given on table V are those of this bearing. Attempts to cool this bearing by utilizing bleed argon gas at 440° F were not satisfactory and the resulting bearing temperature shown was about 915° F. Cooling argon at 350° F with the addition of a heavy silver shunt underneath the journals improved the situation considerably, but the bearing temperature was still 673° F. The addition of a liquid cooling jacket around the bearing pad with clearance space provided between the jacket and the pad to allow freedom of pad motion, together with bleed argon at 350° F produced a bearing temperature of 615° F. This again was unsatisfactory. The use of bleed argon at 350° F, the liquid jacket and also the silver shunt improved the situation further but the number two journal bearing temperature was still 472° F. The use of a cooling jacket around the bearing pad and also a cooling plug within the shaft cavity with clearance between the plug and the inside surface of the shaft resulted in a bearing temperature of 511° F. The use of argon at 350° F plus the introduction of liquid coolant into the pads resulted in a bearing temperature of 350° F. Temperature-wise this was quite satisfactory, but this type of cooling scheme has never been used in an actual machine. The connection of the liquid coolant lines to the pads would offer considerable restraint and would probably have a seriously detrimental effect on the stability of the tilting pad journal bearing. The final scheme considered was the use of argon precooled to 100° F with a liquid coolant in an external heat exchanger, with the addition of finned cylindrical heat exchangers on the inside of each journal within the shaft cavity. The circulation of argon coolant is shown on figure 19. With this scheme the resulting number two journal bearing temperature was 367° F, and this cooling technique was chosen for the final unit. As shown in figure 19 argon gas is bled from the compressor discharge and passed through a heat exchanger which cools it to 100° F. It is then passed into the shaft cavity area around the number one journal bearing. It passes through the finned tube inside the journal called the heat exchanger number 1, and then flows down through the shaft cavity to the region of journal bearing two where it flows through the second finned tube heat exchanger. It then passes through the small metering holes in the shaft and into the number two bearing cavity from where it is discharged through a tube back into the number one bearing cavity to warm the pads and the bearing carrier. From here it bleeds through a labyrinth to the first stage of the compressor. The thrust bearing is cooled directly by the liquid coolant. It is introduced through a coil around the shaft inside the compressor inlet, entering a cavity in the thrust bearing carrier from whence it circulates through the stationary thrust bearing parts and out again.

Journal Bearing Performance

Figure 20 illustrates the performance characteristics of the number one journal bearing with a clearance ratio of 0.001. The various curves illustrate the pivot film thickness, the friction horsepower, the film stiffness and the natural frequencies of the pad in both the pitch and the role modes, all plotted as a function of rotative speed. The requirements of the journal bearings are that (1) they have adequate load capacity, that is, that they maintain adequate film thicknesses at the pivot under all loading conditions, (2) their power absorption be reasonable, (3) their stiffnesses in the radial direction be such that the system critical speeds do not fall close to design speed, and (4) all of the natural frequencies of the bearing pads be above the operating speed. As indicated on figure 20, it is apparent that both the ratio of the pitch natural frequency to the rotative speed and the roll natural frequency to rotative speed remain greater than 1 over the entire speed range up to the 60 000 rpm overspeed condition. Note that the friction horsepower loss in the number 1 bearing is about 110 watts at the design speed.

Figure 21 indicates the performance characteristics of the number two journal bearing, also with a clearance ratio of 0.001. Again all of the characteristics required of this bearing design are satisfactory with the possible exception of power consumption. The friction horsepower loss of approximately 330 watts at design speed is undesirably high. Use of this particular bearing would result in a significant penalty to the overall system efficiency.

Figure 22 illustrates the effect of bearing stiffness on the rigid body criticals of the axial flow turbocompressor. The stiffnesses of the two journal bearings, as obtained from the curves of figures 20 and 21, place the first and second rigid body criticals at about 9000 and 12 000 rpm, which is quite satisfactory for the 50 000 rpm operating speed. The third or flexural critical is approximately 95 000 rpm, which is far enough above the overspeed condition so that no amplification of load takes place. Prototype bearings with clearance ratios of 0.001 were manufactured and evaluated experimentally in the bearing development program. A simulated rotor operated stably over the entire speed range to the overspeed condition on the journal bearings with a clearance ratio of 0.001.

In order to investigate the possibilities of reducing the friction power loss in the journal bearings, performance curves were generated for the number one and number two bearings with somewhat larger clearance ratios, since the power loss is a rather strong function of the clearance ratio. Since the power loss results from shearing a gas film the thinner the average film, of course, the higher the power loss. Figure 23 illustrates the characteristics of the number one journal bearing with a clearance ratio of 0.0018. These particular curves are for the 12 psia bearing cavity pressure condition. All of the design characteristics are still satisfactory for this bearing and the power loss at design speed has been reduced to 65 watts. The pitch and roll natural frequencies are still above the operating speed range of the unit. For the 6 psia bearing cavity condition, however, some problems are likely to result with this particular design. Figure 24 illustrates the predicted performance of this bearing at the 6 psia bearing cavity pressure condition. Note that the roll natural frequency of an unloaded pad is about 47 000 or 48 000 cpm. This would indicate that the unloaded pads may go into a roll resonance when the unit reaches a speed of 47 000 rpm. How serious such resonances are can only be determined experimentally.

The predicted performance characteristics of the number two journal bearing with a clearance ratio of 0.003 are given in figure 25 for the 12 psia condition and in figure 26 for the 6 psia condition. Since the power loss in this bearing is much greater than that in the number one journal bearing, the clearance ratio was increased more than in the number one bearing. Note that the friction horsepower loss has been drastically reduced to approximately 125 watts. Note also, however, that the roll natural frequency of the loaded pads is about 30 000 at the 12 psia condition and approximately the same at the 6 psia condition. This would indicate that these pads may go into a roll resonance when the operating speed of the unit reaches 30 000 rpm.

Prototype journal bearings with these large clearance ratios have been fabricated and evaluated experimentally. While pad resonances did not prove to be a problem, some shaft instabilities were apparent in the higher speed range when the bearings were run with a preload of 0.5. All of the performance curves of the various journal bearings shown up to now were calculated for a preload value of 0.5. In order to damp out the observed shaft whirl with the large C/R journal bearings, it was necessary to increase the preload. With this increased value of preload the simulated rotor could be run up to the overspeed condition with complete stability. The final choice of journal bearing design for the actual turbocompressor has not, of this date, been finalized. The probability is, however, that journal bearings with clearance ratios intermediate between 0.001 and 0.003 will be used with an initial value of preload sufficient to maintain stable rotor operation with some margin of safety.

Thrust Bearing Performance

The predicted performance characteristics of the self-acting spiral groove thrust bearing are shown in figure 27. At the maximum predicted steady state thrust load of 24 pounds, the operating film thickness of the self-acting bearing is about 0.0008 inch. At this operating film thickness the total power consumption of the bearing is 180 watts. This includes both the hydrostatic and hydrodynamic portions of the power loss. The principal problem with the thrust bearing is minimization of thermal distortions due to temperature gradients within the stationary thrust bearing parts. Liquid coolant is fed directly into flow passages within the stationary thrust bearing parts to maintain an approximately isothermal condition to minimize distortions.

Effect of Rotor Unbalance

It is of interest to examine the effect of rotor unbalance on the journal bearing loads. It is anticipated that the actual turbocompressor rotor will be balanced in place on the gas bearings until the unbalance orbits are on the order of 20 microinches. This will result in a very small initial unbalance but long time operation of the machine at high temperatures may cause mass shifts to occur in the turbine. The allowable magnitude of this mass shift has been set at 0.002 ounce inch. With this assumed unbalance, the total bearing load on each of the two journal bearings has been calculated as a function of speed. This is illustrated in figure 28. Shown is the static load and also the total bearing load which consists of the static load and the dynamic load due to the rotating unbalance. It can be seen that, at this degree of unbalance, the increase in bearing load is fairly nominal, being slightly greater than 1 pound on the number two bearing.

TURBOALTERNATOR

Design

In contrast to the relatively light weight, high speed turbocompressors the turboalternator is a somewhat heavier, lower speed machine. Since it is a two pole machine, its design speed of 12 000 rpm is dictated by the desire to have it supply 400 cycle per second alternating current. The predicted rotor weight is about 55 pounds with a load of 28 pounds on one journal bearing and 27 pounds on the other journal bearing. Again the gas in the bearing cavities is argon at a design pressure of approximately 12 psia, but with provision for operating the bearings in an ambient pressure as low as 6 psia. The steady-state thrust load is predicted to be 86 pounds, and the transient thrust load at startup is 250 pounds maximum in the forward direction to 100 pounds maximum in the reverse direction. The maximum unbalance for the rotor assembly is predicted to be 0.005 inch in the plane of the turbine wheel.

Bearing types chosen for the turboalternator are similar to those incorporated in the axial flow turbocompressor. One major change in the design concept is the inclusion of hydrostatic lift off provisions in the journal bearings as well as in the both faces of the thrust bearing. A schematic of the turboalternator is shown in figure 29. The two-stage axial flow turbine is overhung on the left side and the dual face thrust bearing is overhung on the right side. The two journal bearings are located between the overhung components and the alternator stator.

Bearing Design

A complete description of the journal and thrust bearings in the turboalternator is given in table VI. As in the axial flow turbocompressor the journal bearings are four tilting pad bearings, each pad with a span of 80°. Nominal diameters and lengths of the two journal bearings are 3.5 inches. The C/R ratio is 0.001 in both bearings and all of the pads pivot on semi-fitted ball and socket pivots with each pad flex-mounted. The stiffness of the two upper pad flexures when the unit is in the horizontal attitude is 50 000 pounds per inch and the stiffness of the two lower pad flexures in each journal bearing is 150 000 pounds per inch. The choice of these flexures is discussed in the section on critical speeds.

The thrust bearing, as in the axial flow turbocompressor, is an inward pumping spiral groove bearing with outside and inside radii of 3.5 inches and 0.5 inch, respectively. The reverse bearing is somewhat smaller and both faces of the thrust bearing are equipped with a ring of orifices which are pressurized to about 80 psi during startup. During startup, transient thrust loads are supported hydrostatically until the rotor is up to speed.

Material specifications are also given in table VI. Note that the bearing surfaces have a surface coating of chrome-oxide. Chrome-oxide surface coatings have been found to be extremely effective in resisting wear and surface damage during contact at startup and shutdown and also under conditions of momentary high speed rubs.

The predicted performance characteristics of the turboalternator journal bearings are shown in figure 30. All of these data were calculated for a preload of 0.5 and a radial load of 35 pounds on each bearing. The curves shown are for an ambient pressure of 12 psia. At the design speed the pivot film thickness at the 12 psia pressure condition is about 0.00058 inch. For operation at the 6 psia operating condition the predicted film thickness is 0.00045 inch. The pad natural frequencies in both pitch and roll at both pressure conditions is predicted to be well above the operating speed. The lowest ratio of N_R to N occurs at the 6 psia operating condition for the unloaded pad, and this is predicted to be 2.5; therefore, the natural frequencies of the journal bearing pads are, conservatively, far in excess of the operating speed range of the unit. The predicted power loss of each journal bearing at design speed is 105 watts.

All of the predicted performance characteristics of the journal bearings have been generated without including magnetic load effects. Magnetic loads may constitute a serious problem but, as of this writing, only a preliminary assessment of the problem has been made. The magnetic forces are quite complex. If they are expressed as two orthogonal forces, such as F_x and F_y, each force contains 18 terms. Each force contains components which are rotating at twice per revolution and components dependent on electrical angles. Attempts are being made to determine bearing response to magnetic loads, but the analysis is not complete.

Performance curves for the turboalternator self-acting spiral groove thrust bearing are shown in figure 31. As can be seen the design of the bearing is extremely conservative. At the design load of 85 pounds, the operating film thickness is about 0.0012 inch. The bearing could easily carry large overloads without being overloaded. The power absorption in the thrust bearing at design load is less than 90 watts because of the large operating film thickness.

The principal problems with the thrust bearing are to minimize thermal distortion so that serious losses in load capacity do not occur, and also to design a gimbal system for the stationary thrust faces so that they can follow runouts or any sort of wobble that may occur in the rotating thrust runner. The design of a satisfactory gimbal system is not a simple problem, and the evaluation of the existing design has not been completed. The design presents a problem simply because the film stiffness of a self-acting gas bearing is quite low so that, in order for the gimbal to work properly, it has to be extremely flexible. Yet it must be stiff enough to prevent sizable axial movements of the shaft. Great flexibility must be built into the gimbal mount to allow the bearings to accommodate angular misalinement and wobble.

Crowning of the thrust faces must also be prevented for this reduces load capacity. Liquid and gas cooling are employed to maintain a nearly isothermal structure.

Rotor Critical Speeds

Figure 32 illustrates the effect of journal bearing film stiffness and also the bearing mount stiffness on the turboalternator rigid body critical speeds. The dashed line curves indicate the condition that would prevail with all the journal bearing pads rigidly connected to the frame. The cross hatched area between the two bearing stiffness curves indicates the calculated range of journal bearing stiffness for ambient pressures of 6 to 12 psia in the bearing cavity. It would seem that these limits of bearing stiffness cross the rigid support critical speed curves quite close the the design speed. Therefore, rather than use rigid supports, the pads of both journal bearings are mounted on flexures. This would also provide somewhat better thermal accommodation in the journal bearings. The upper two pads of each journal bearing in the horizontal attitude are connected to flexures with a stiffness of 50 000 pounds per inch and the lower two pads to flexures with a stiffness of 150 000 pounds per inch. The critical speeds with these flexure stiffnesses are indicated by the solid curves which cross the curves of bearing film stiffness in the region of 8 to 9000 rpm. This is sufficiently below the design speed while the flexural critical speed of 29 000 rpm is sufficiently above the overspeed condition of 16 000 rpm.

SUMMARY

The gas bearings in Brayton cycle space power systems turbomachinery are required to operate at high speeds and light loads and may be subject to distortions resulting from thermal gradients. These operating conditions dictate the use of tilting pad journal bearings which have good stability characteristics and a high tolerance to distortions resulting from thermal and mechanical stresses. Rayleigh step and spiral groove thrust bearings, which have load capacities superior to self-equalizing tilting pad thrust bearings are being used in conjunction with gimbal mounts which allow them to accommodate wobble of the thrust runner.

Both pressurized and self-acting startup schemes are being evaluated in the journal bearings. A chrome-oxide coating has been developed for the bearing surfaces in sliding contact. This coating has shown extremely good resistance to wear and surface damage under start and stop conditions and under momentary high speed rubs. Because of the high magnitude and reversal of transient thrust loads during startup, dual faced thrust bearings are employed with provision for hydrostatic load support. Both gas and liquid cooling are employed to cool the bearings to minimize thermal distortion and materials problems.

In contrast to theoretical predictions of complete stability in tilting pad journal bearings, some rotor instabilities have been observed. These fractional frequency whirls can be damped out by increasing bearing preloads at the cost of slightly higher power consumption. The causes of these instabilities are currently being investigated.

In electrical machinery there is much concern over the behavior of gas lubricated journal bearings under the action of magnetic forces. These forces are quite complex and they appear in various frequencies and magnitudes. Theoretical and experimental studies of rotor behavior under the action of magnetic forces are presently being conducted.

REFERENCES

1. "Development of Gas Bearings for Brayton-Cycle Turbomachinery Applied to Space Systems" by L. A. Matsch, E. L. Wheeler, and R. B. Gildersleeve. Presented at the ASME Spring Lubrication Symposium, June, 1965.

2. "Gas Bearing Systems for NASA Solar Brayton Cycle Axial Flow Turbocompressor and Turboalternator" by P. W. Curwen, A. Frost, and E. B. Arwas. Presented at the ASME Spring Lubrication Symposium, June, 1965.

3. "The Requirements of Gas Bearings in Brayton Cycle Axial Flow Machinery" by P. Bolan, W. B. Spencer, and E. E. Striebel. Presented at the ASME Spring Lubrication Symposium, June, 1965.

4. Seventh Monthly Progress Report on Contract NAS 3-4179, Pratt & Whitney Aircraft, January 19, 1965.

5. Final Report on Contract NAS 3-2778, AiResearch Manufacturing Co., January 1966.

TABLE I. - 8.5 KW BRAYTON CYCLE POWER SYSTEM

Working fluid: Argon
Compressor inlet temperature: 80° F
Compressor inlet pressure: 6 psia
Compressor outlet pressure: 13.8 psia
Turbine inlet temperature: 1490° F
Bearing cavity pressure: 12.0 psia or 6 psia

TABLE II. - RADIAL FLOW TURBOCOMPRESSOR BEARINGS

Rotative speed: 38 500 rpm

Journal Bearing Data:

Type: 3 tilting pad
Diameter: 1.750 inches
Length: 1.312 inch ($L/D = 3/4$)
Pad angular span: 100°
Radial clearance: 0.0023 inch
Pivot location: 0.65
Pivot type: Fitted-ball socket
Mount system: One resiliently supported pad ($K_R \cong 2000$ lb/in.), 2 rigidly supported pads
$\Lambda \cong 1.7$

Thrust Bearing Data:

Type: Rayleigh step, 8 pads
Leading sector angle/total pad angle: 0.33
Step height: 0.00065
Outside radius: 1.62 inches
Inside radius: 0.95 inch
Total axial clearance: 0.0061 inch
Mount system: Gimbal utilizing cylindrical pin pivots

All bearings hydrostatic start up.

TABLE III. - RADIAL FLOW TURBOCOMPRESSOR

Bearing Materials and Specifications:

Pivot - Fitted ball in socket
Match lapped ≈ 0.0001 Δ dia.

Ball - 1/2" dia.
Kennametal 96 (tungsten carbide)
Sphericity within 25 microinches TIR

Socket (Integral with tilting pad)
AMS 5737 (A286) Flame plated LWI-N40
(Cobalt 15 to 17 percent, 1000 to 1150 VPH, 1000° F)
90 to 100 000 psi rupture
0.002 to 0.003 finish thickness

Shaft and integral thrust runner
A286 - Rc 60 min. Coated LWI-N40
0.002 to 0.003 finish thickness
Runner 3 He light bands (22 microinches)

Thrust Bearings - Stators
M-2 tool steel Rc 60 min. 3 He light bands flat

Gimbal Bearing pivots 1/4" dia.
Alpha Molykote X15 0.0002 to 0.0003
Both pins and sleeves of hardenable material
steel on steel with phosphate undercoating -
Clearance 1/2 to 1 mil with wipe on coating

TABLE IV. - AXIAL FLOW TURBOCOMPRESSOR BEARINGS

Rotative speed: 50 000 rpm

Journal Bearing Data:
Type: 4 tilting pad

	No. 1 (Compressor end)	No. 2 (Turbine end)
Diameter:	1.500	2.125
Length:	1.500	2.125
Pad angular span:	80°	80°
Radial clearances investigated:	0.75×10^{-3} to 1.35×10^{-3}	1.06×10^{-3} to 3.18×10^{-3}
C/R	$.1\times10^{-3}$ to 1.8×10^{-3}	1×10^{-3} to 3×10^{-3}
Pivot location:	0.65	0.65
Pivot type:	ball on cylinder	ball on cylinder

Mount system: Flexures with approximately the same stiffness as the bearing film.

Thrust Bearing Data:
Type: Whipple plate inward pumping spiral groove
Outside radius: 1.625 inches
Inside groove radius: 1.20 inch
Inside radius: 0.40 inch
Materials:
Rotor journals: Waspalloy
chrome oxide surface coating
Pivoted pads: M-1 tool steel
chrome oxide surface coating
Pivots: M-1 tool steel
Electrofilm surface treatment
Thrust runner: Titanium
chrome oxide surface coating
Stationary thrust bearing members:
Normal thrust face - Aluminum alloy
chrome oxide surface coating

Reverse thrust face - Low alloy carbon steel
chrome oxide surface coating

Journal bearings self-acting startup
Thrust bearing hydrostatic startup

TABLE V. - AXIAL FLOW TURBOCOMPRESSOR BEARING COOLING STUDY

Cooling gas	Gas temp. (°F)	Liquid temp. (°F)	Liquid location	Thermal shunt	Thermal shunt thickness (in.)	Max. bearing temp. (°F)
Argon	440	---	--------	---------	---	915
Argon	350	---	--------	Silver	0.3	673
Argon	350	100	Jacket	---------	---	615
Argon	350	100	Jacket	Silver	0.3	472
Argon	350	100	Jacket & plug	---------	---	511
Argon	350	311	Pad	---------	---	350
Argon	100	---	--------	Nickel finned heat exchanger	---	367

TABLE VI. - TURBOALTERNATOR BEARINGS

Rotative speed: 12 000 rpm

Journal Bearing Data:

Type: 4 tilting pad

	Turbine end	Alternator end
Diameter, in.	3.500	3.500
Length, in.	3.500	3.500
Pad angular span	80°	80°
Radial clearance	0.0018	0.0018
C/R	0.001	0.001
Pivot location	0.65	0.65
Pivot type	semi-fitted ball and socket	semi-fitted ball and socket
Mount system	flexure 50 000 lb/in. upper pads 150 000 lb/in. lower pads	flexure 50 000 lb/in. upper pads 150 000 lb/in. lower pads

Thrust Bearing Data:

Type: Whipple plate inward pumping spiral groove

	Thrust bearing	Forward bearing
Outside radius	3.500 inches	2.461 inches
Inside groove radius	1.960 inches	------------
Inside radius	0.500 inch	1.790 inches
Total axial clearance	0.004 inch	

Materials:
- Rotor journals: AMS 6294C (AISI 4620)
  chrome oxide surface coating
- Pivoted pads: M-1 tool steel
  chrome oxide surface coating
- Pivots: M-1 tool steel
  Electrofilm surface coating
- Thrust plate: AMS 4027C (6061 aluminum)
  chrome oxide surface coating
- Thrust runner: AMS 6415F (AISI 4340)
  chrome oxide surface coating

All bearings hydrostatic startup

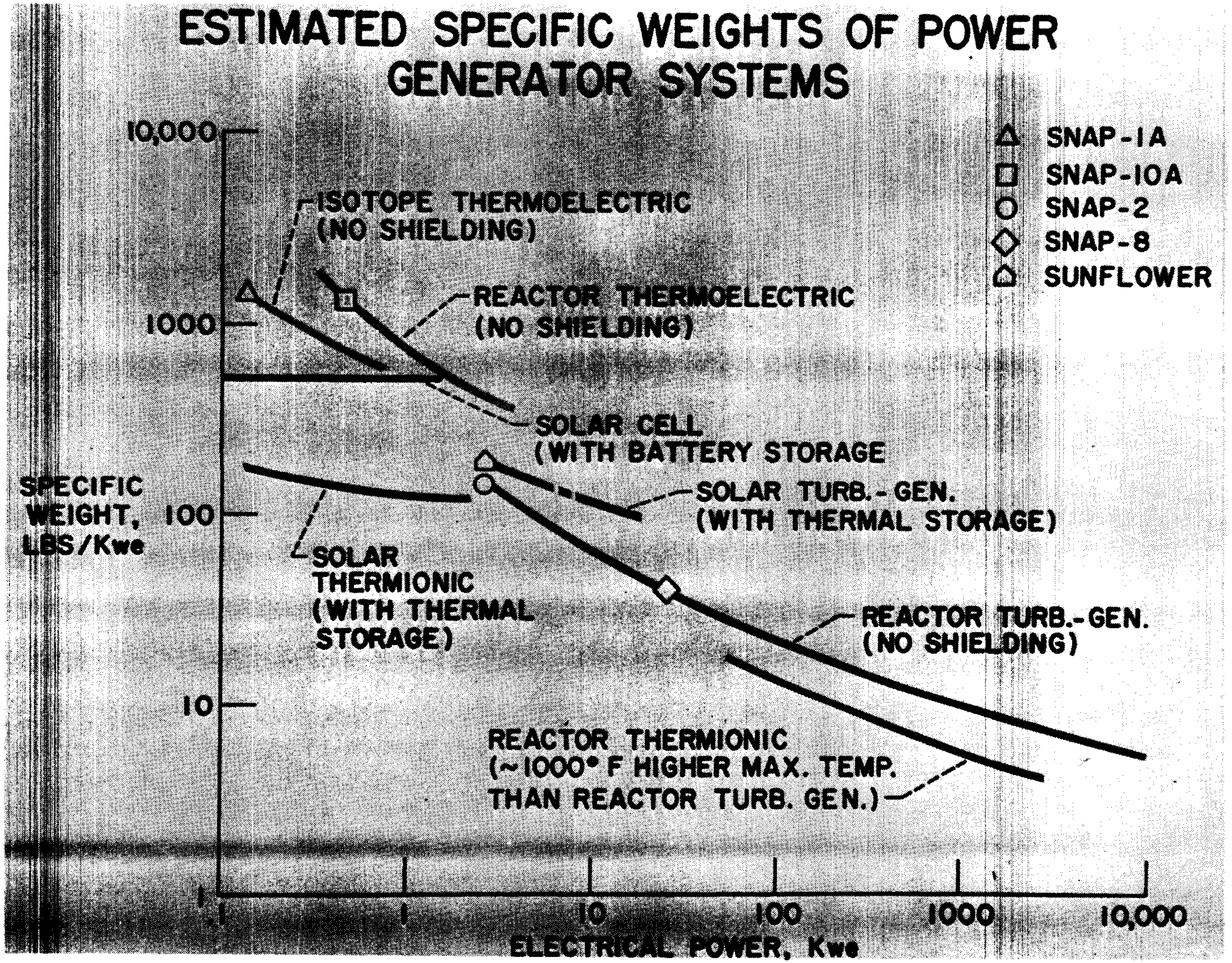

ESTIMATED SPECIFIC WEIGHTS OF POWER GENERATOR SYSTEMS
SNAP-1A
SNAP-10A
SNAP-2
SNAP-8
SUNFLOWER
ISOTOPE THERMOELECTRIC (NO SHIELDING)
REACTOR THERMOELECTRIC (NO SHIELDING)
SOLAR CELL (WITH BATTERY STORAGE
SOLAR TURB.-GEN. (WITH THERMAL STORAGE)
SOLAR THERMIONIC (WITH THERMAL STORAGE)
REACTOR TURB.-GEN. (NO SHIELDING)
REACTOR THERMIONIC (~1000° F HIGHER MAX. TEMP. THAN REACTOR TURB. GEN.)
SPECIFIC WEIGHT, LBS/Kwe
10,000
1000
100
10
1
.1
1
10
100
1000
10,000
ELECTRICAL POWER, Kwe

Figure 2

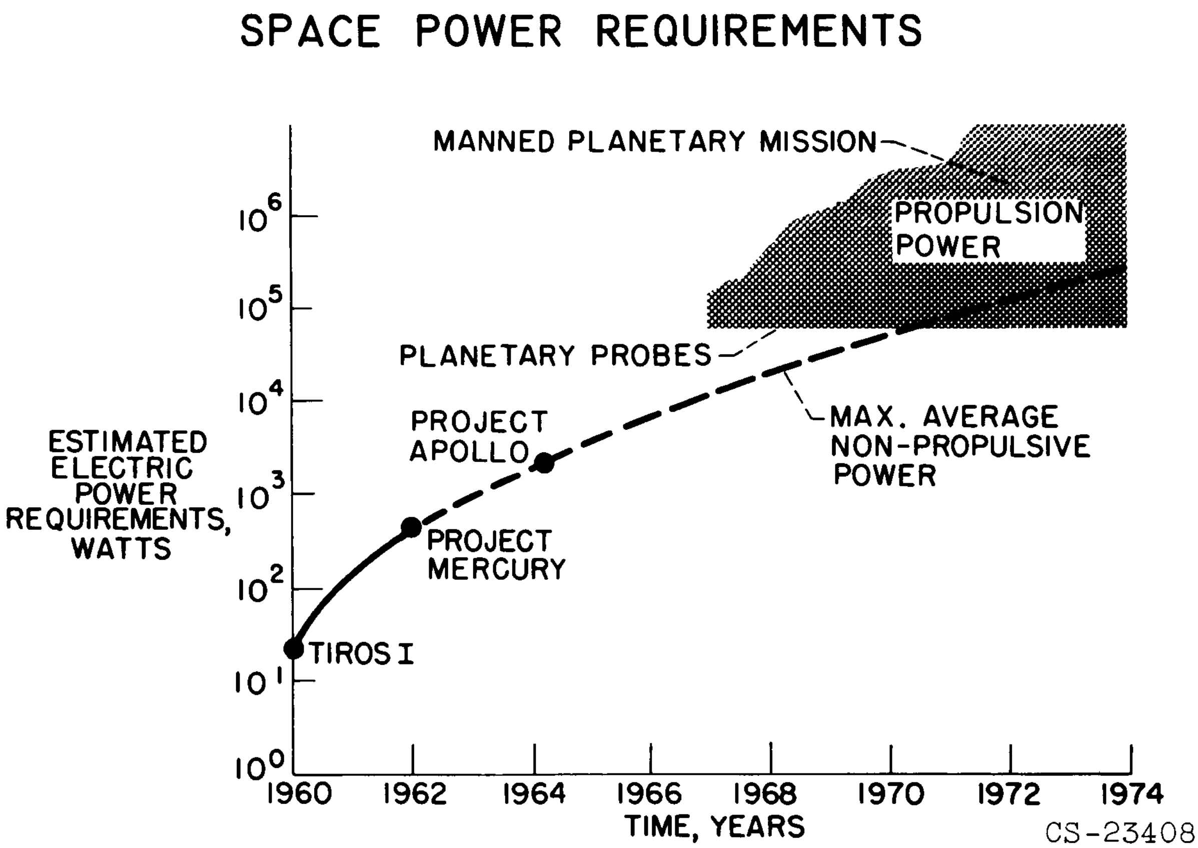

SPACE POWER REQUIREMENTS
MANNED PLANETARY MISSION
PROPULSION POWER
PLANETARY PROBES
MAX. AVERAGE NON-PROPULSIVE POWER
PROJECT APOLLO
PROJECT MERCURY
TIROS I
ESTIMATED ELECTRIC POWER REQUIREMENTS, WATTS
10^0
10^1
10^2
10^3
10^4
10^5
10^6
1960
1962
1964
1966
1968
1970
1972
1974
TIME, YEARS
CS-23408

Figure 1

SCHEMATIC OF BRAYTON CYCLE SPACE POWER SYSTEM

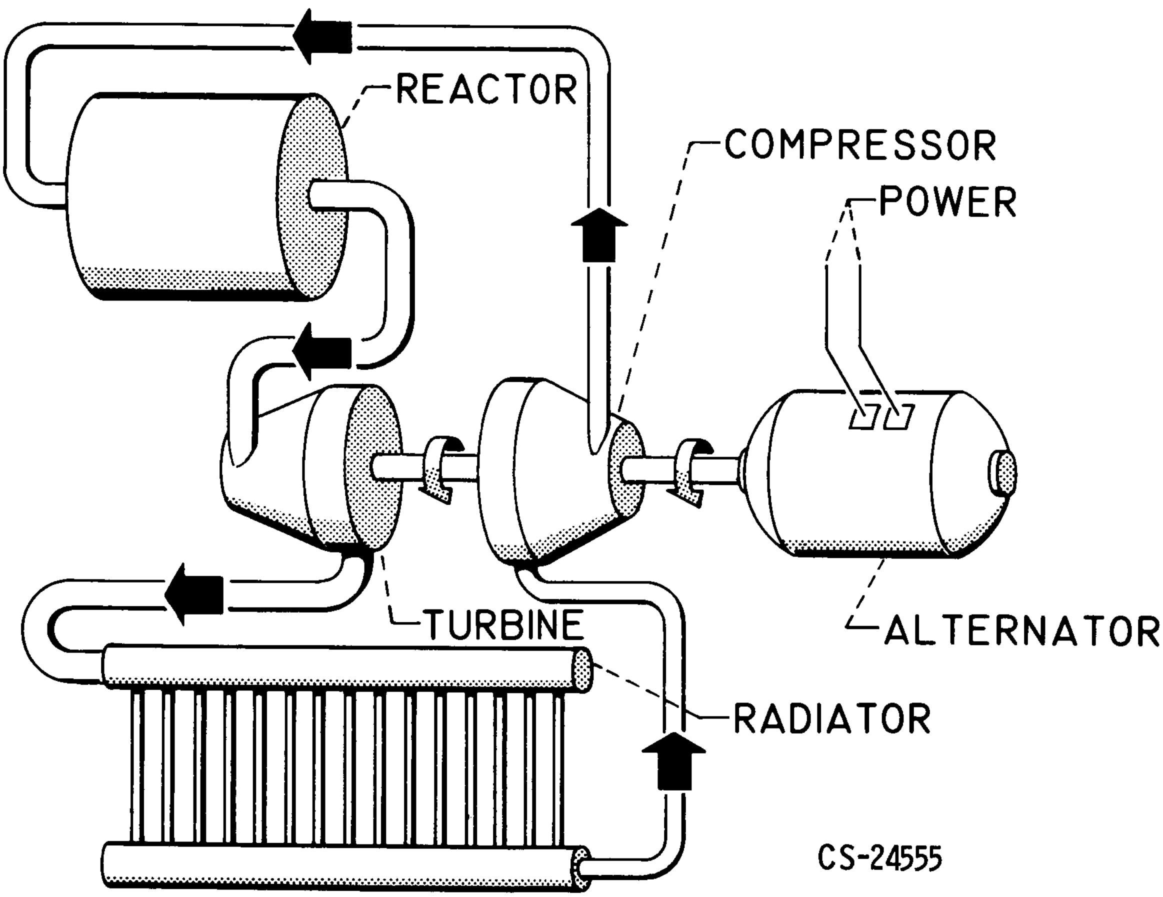


Figure 3

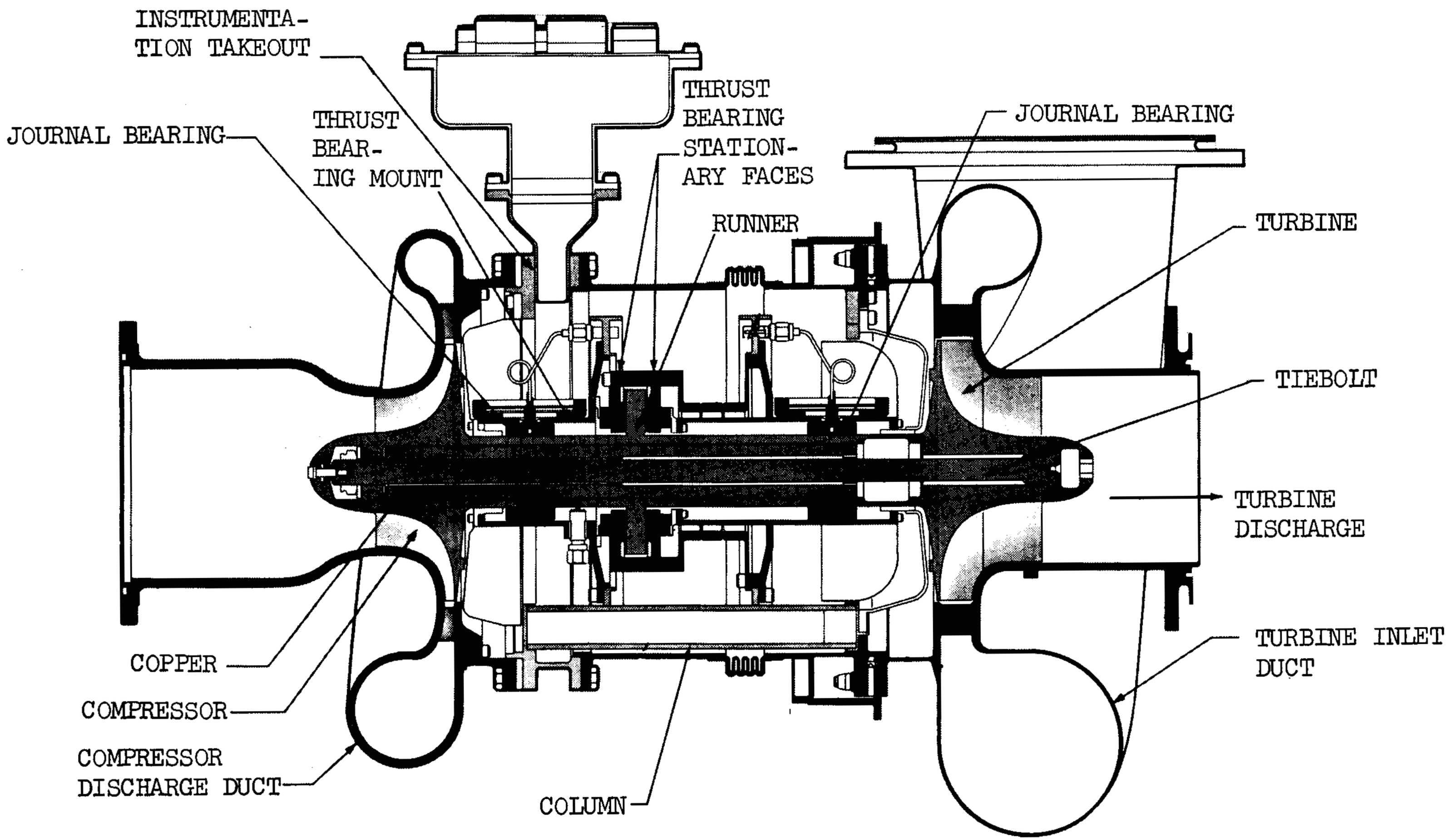


FIGURE 4. - RADIAL FLOW TURBOCOMPRESSOR.

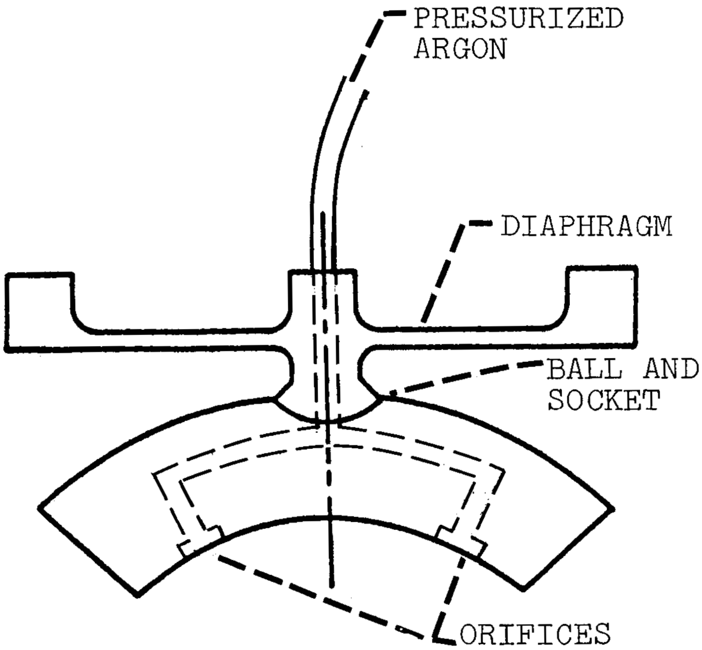


FIGURE 5. - PIVOT AND EXTERNAL PRESSURIZATION ARRANGEMENT. RADIAL FLOW TURBOCOMPRESSOR JOURNAL BEARINGS.

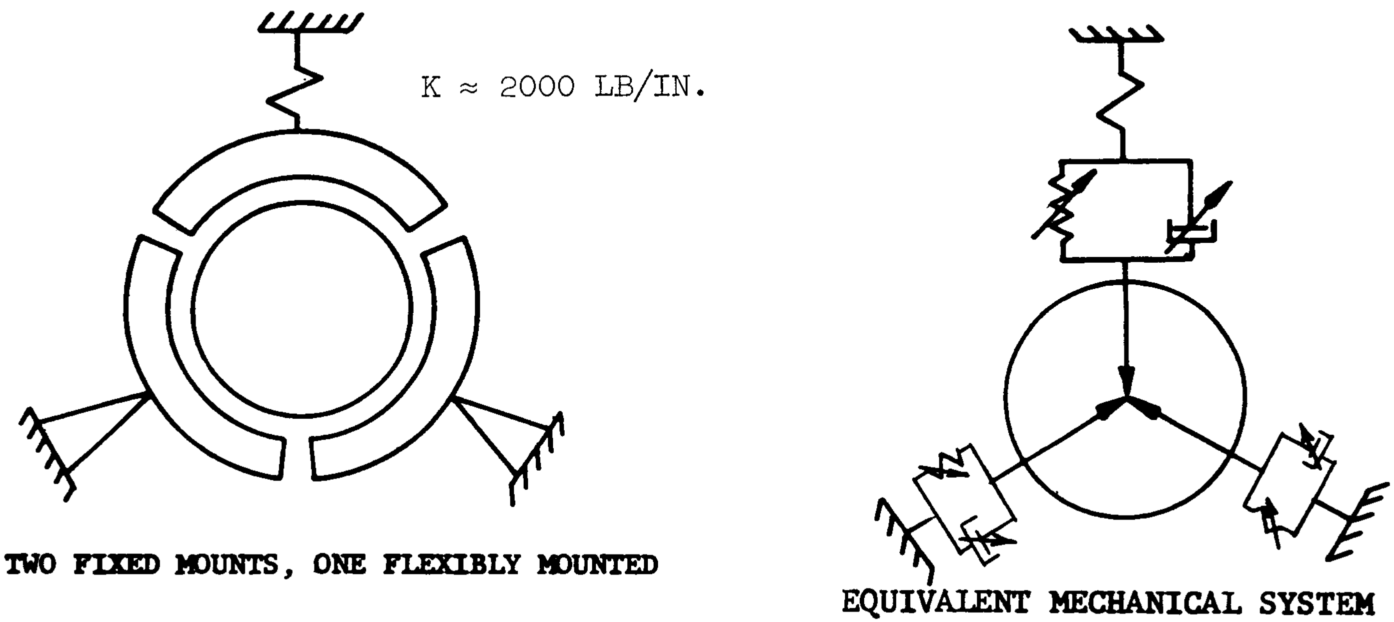


FIGURE 6. - JOURNAL BEARING MOUNTING METHOD. RADIAL FLOW TURBOCOMPRESSOR.

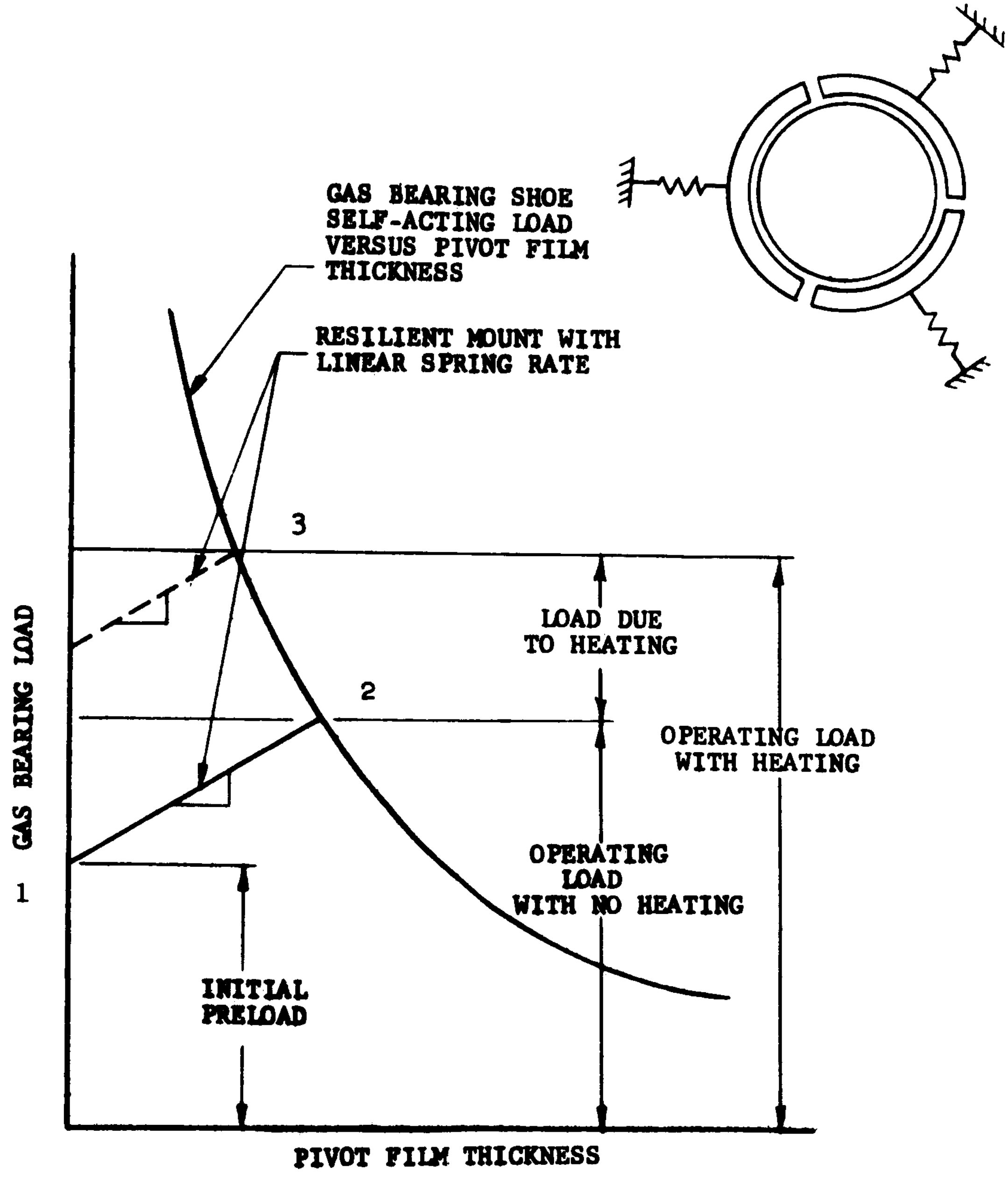


FIGURE 7. - BEARING LOADS AT VARIOUS OPERATING CONDITIONS WITH SHOE RESILIENT SUPPORTS. RADIAL FLOW TURBOCOMPRESSOR.

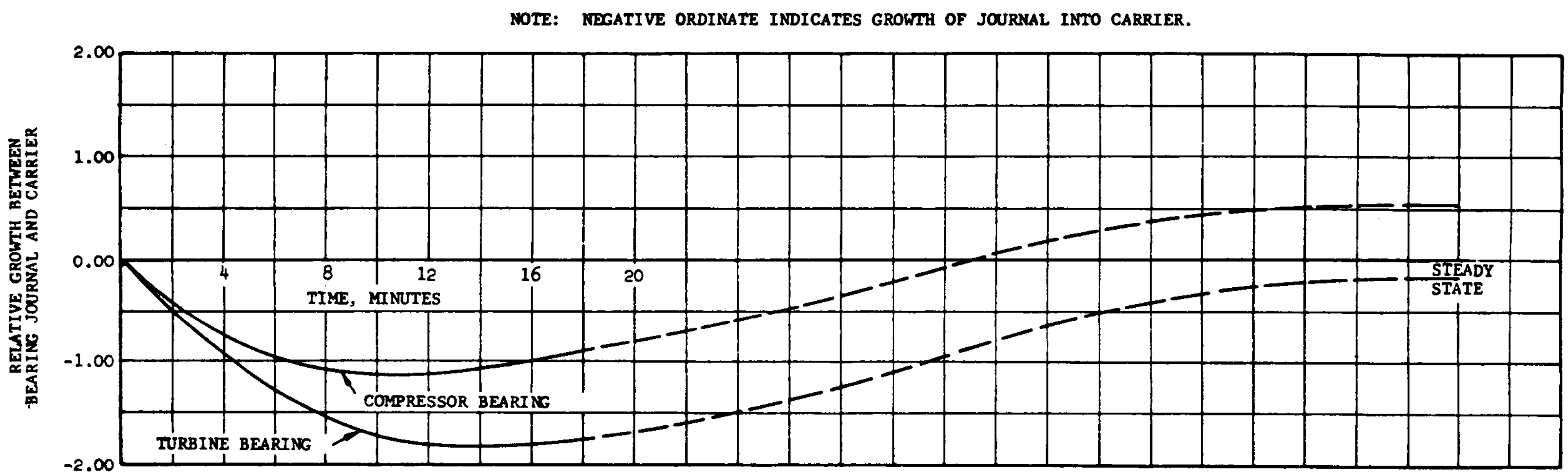


FIGURE 8. - TRANSIENT AND STEADY-STATE BEARING JOURNAL TO CARRIER RELATIVE EXPANSION. RADIAL FLOW TURBOCOMPRESSOR.

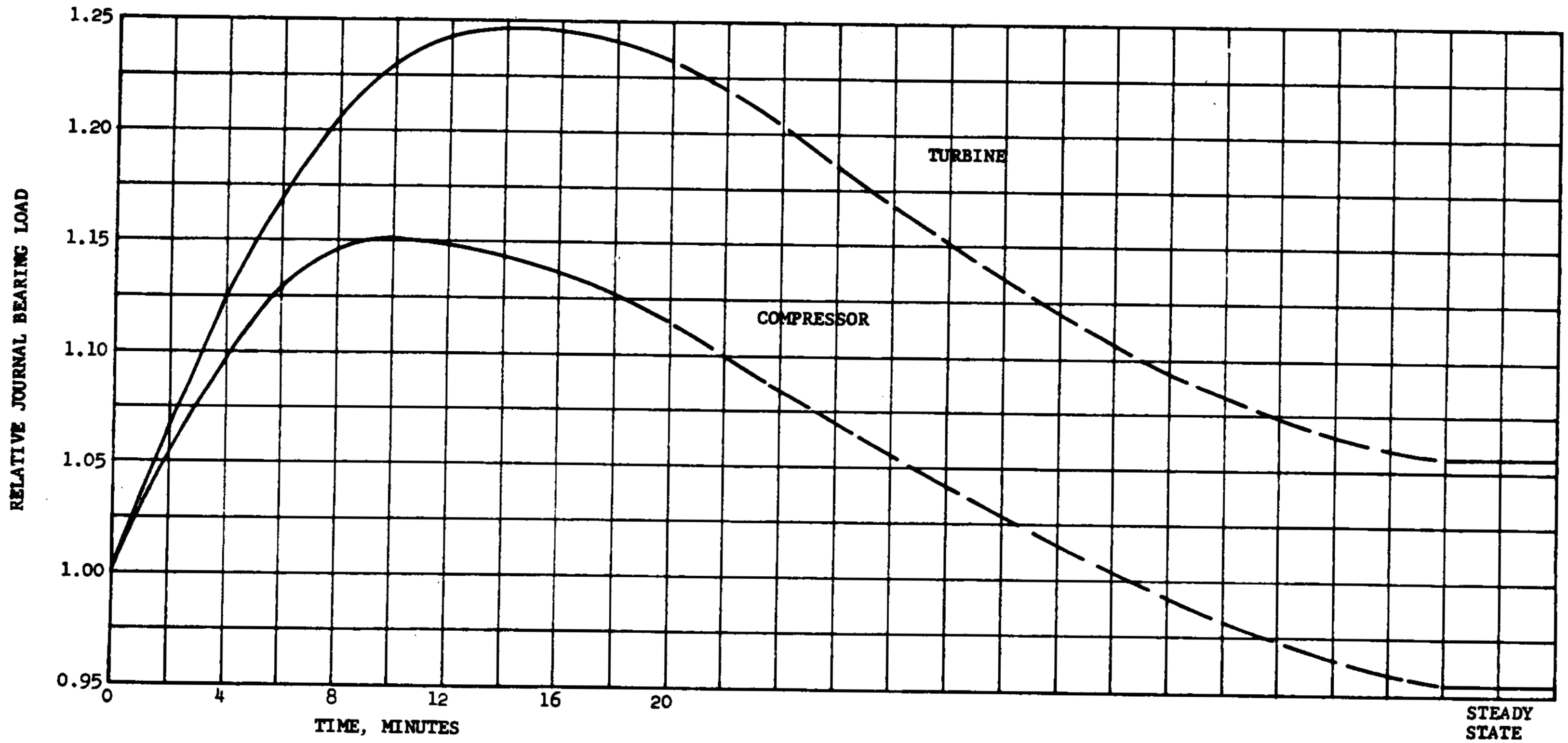


FIGURE 9. - TRANSIENT AND STEADY-STATE JOURNAL BEARING LOADS. RADIAL FLOW TURBOCOMPRESSOR.

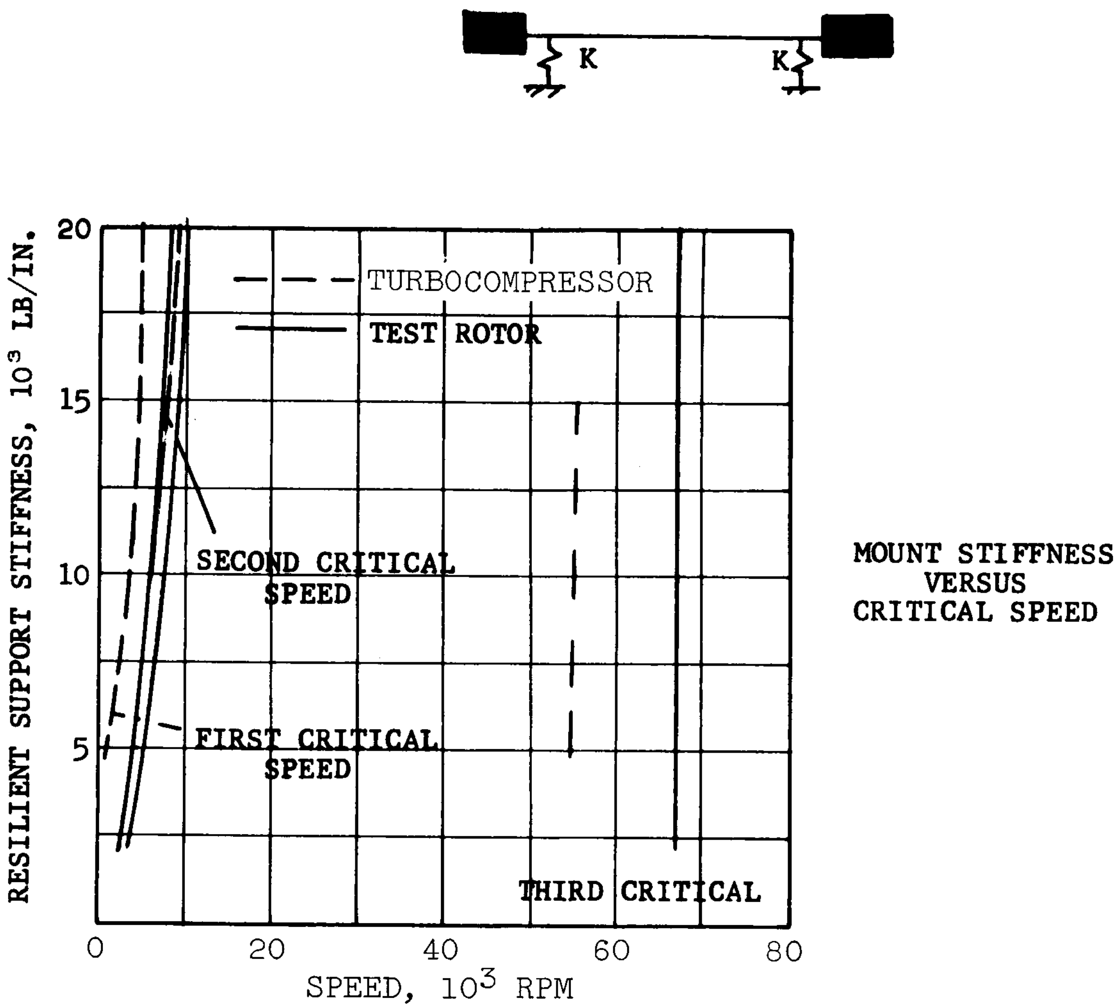


FIGURE 10. - CRITICAL SPEEDS AND BEARING LOADS FOR TEST ROTOR AND RADIAL FLOW TURBOCOMPRESSOR.

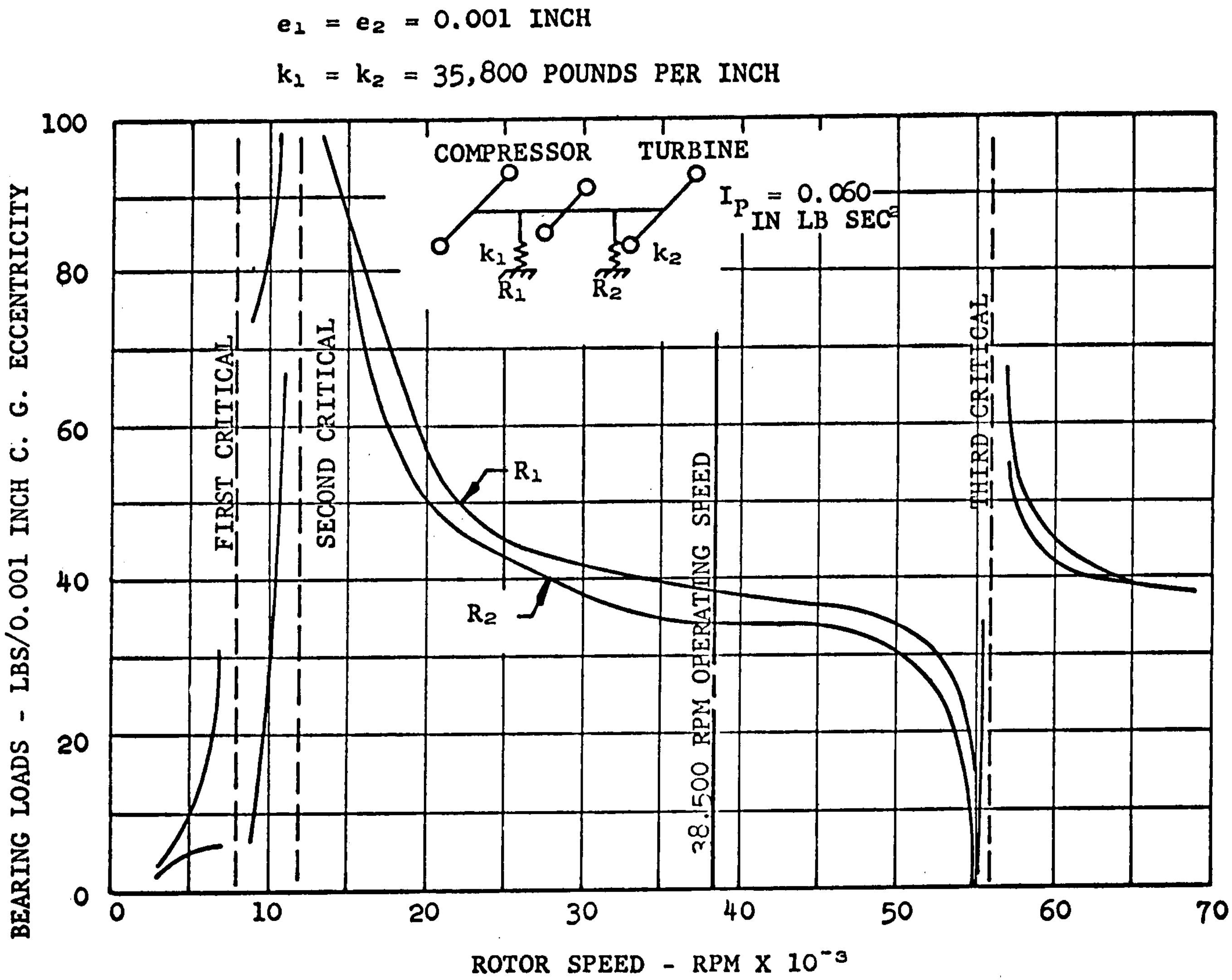


FIGURE 11. - BEARING LOAD VS ROTOR SPEED.

NOTE:

1. POSITIVE MOMENT ACTS TO LIFT LEADING EDGE AWAY FROM SHAFT

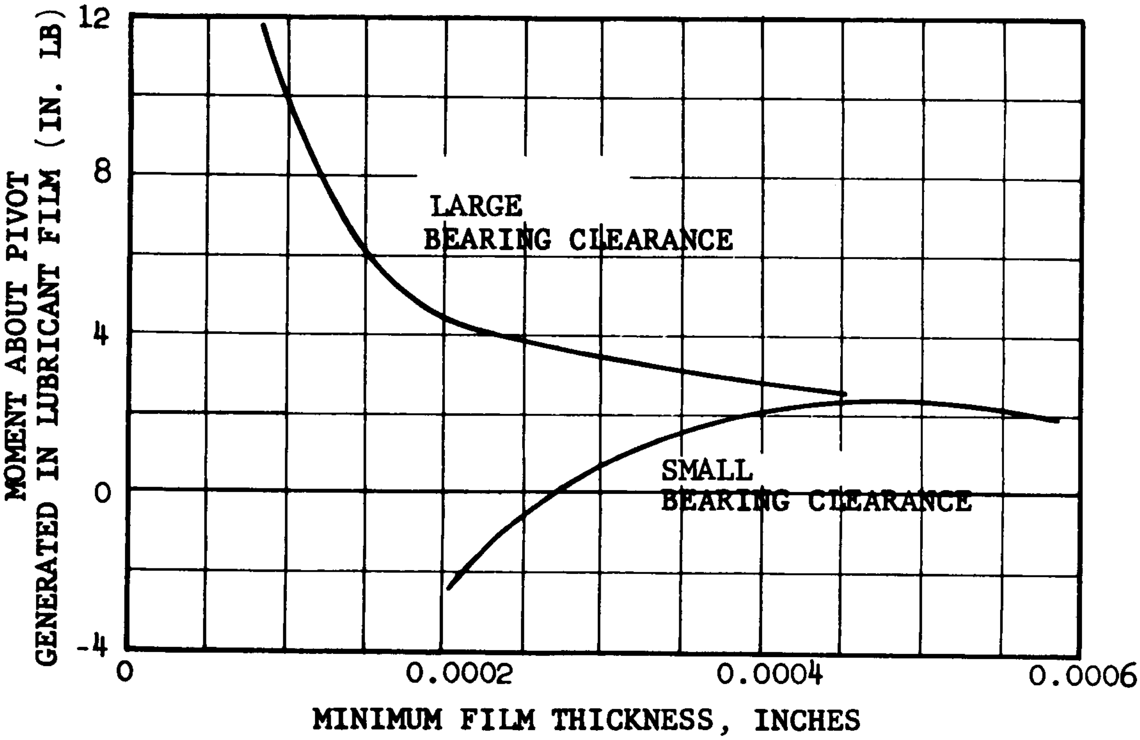


FIGURE 12. - HYBRID EFFECTS AT 20,000 RPM FOR TWO BEARING CLEARANCES. RADIAL FLOW TURBOCOMPRESSOR.

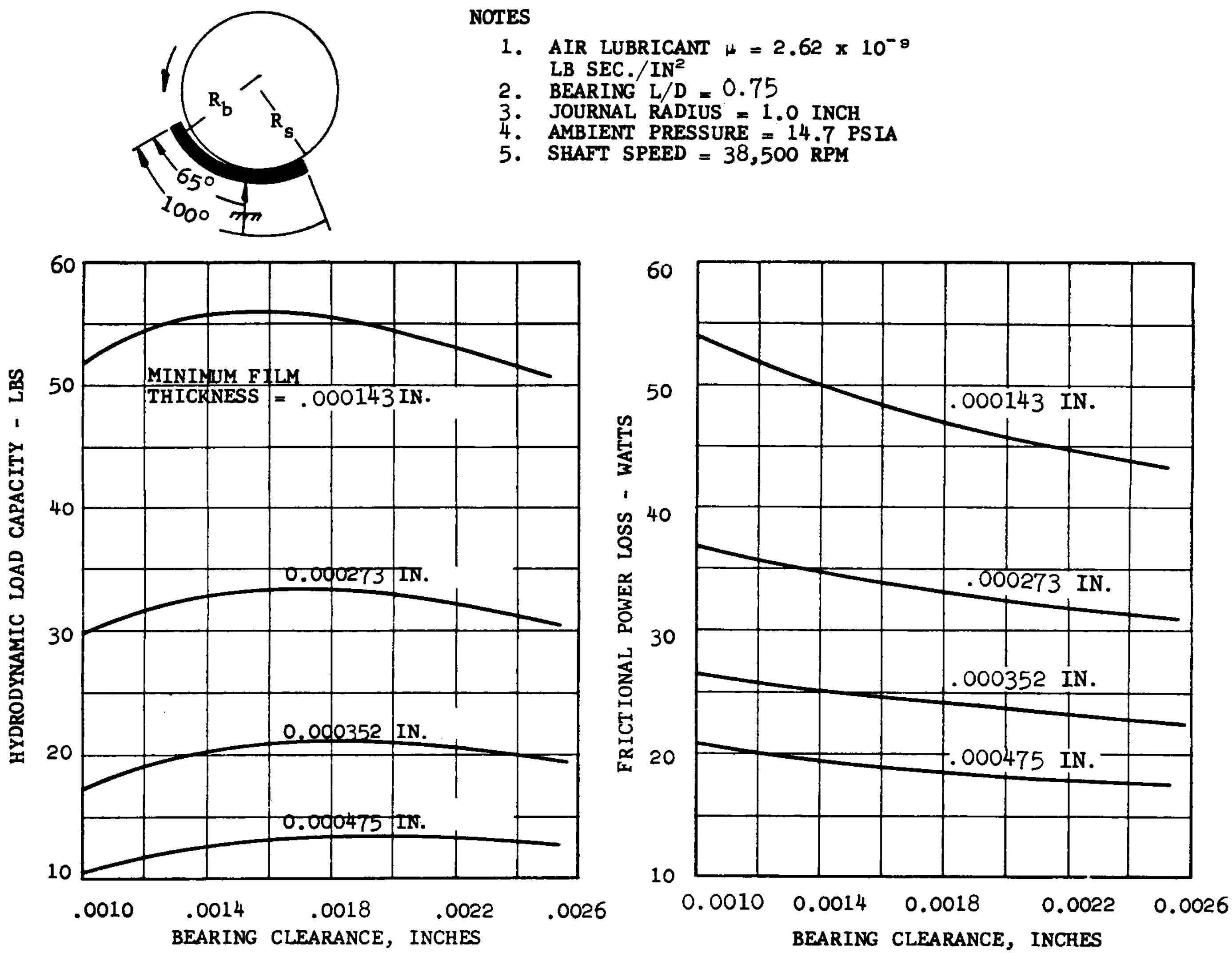


FIGURE 13. - LOAD CAPACITY AND FRICTIONAL POWER LOSS VERSUS BEARING CLEARANCE. RADIAL FLOW TURBO-COMPRESSOR.

BEARING CONFIGURATION

1. ANGULAR SPAN - 100 DEGREES
2. $L/R = 1.5$
3. AMBIENT PRESSURE = 12 PSIA
4. $R = 0.875$ INCHES
5. ARGON AT TWO TEMPERATURES

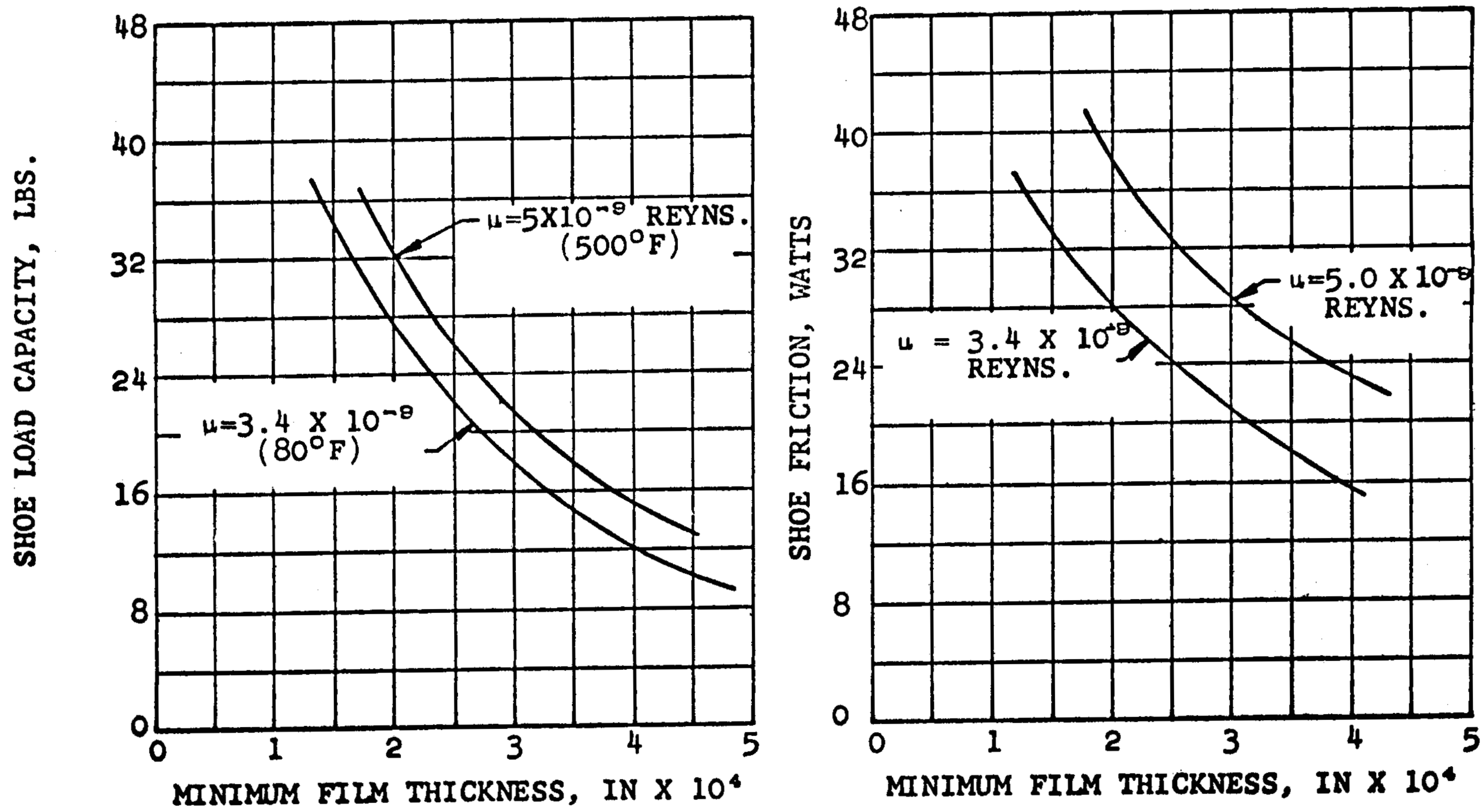
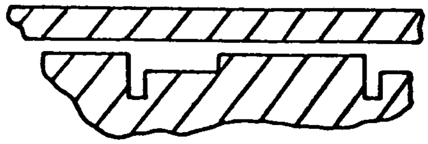


FIGURE 14. - PREDICTED JOURNAL BEARING PERFORMANCE AT DESIGN SPEED. RADIAL FLOW TURBOCOMPRESSOR.



HYDRODYNAMIC STATOR CROSS-SECTION

BEARING CONFIGURATION

1. ARGON LUBRICANT, 500°F, $\mu=5X10^{-9}$
2. SHAFT SPEED = 38,500 RPM
3. THRUST SURFACES ASSUMED PLANE, NO THERMAL DISTORTION CONSIDERED
4. OUTSIDE PAD RADIUS = 1.62 INCHES
   INSIDE PAD RADIUS = 0.95 INCH
5. FRICTION POWER GENERATED IS INSENSITIVE TO SMALL CHANGES IN AMBIENT PRESSURE

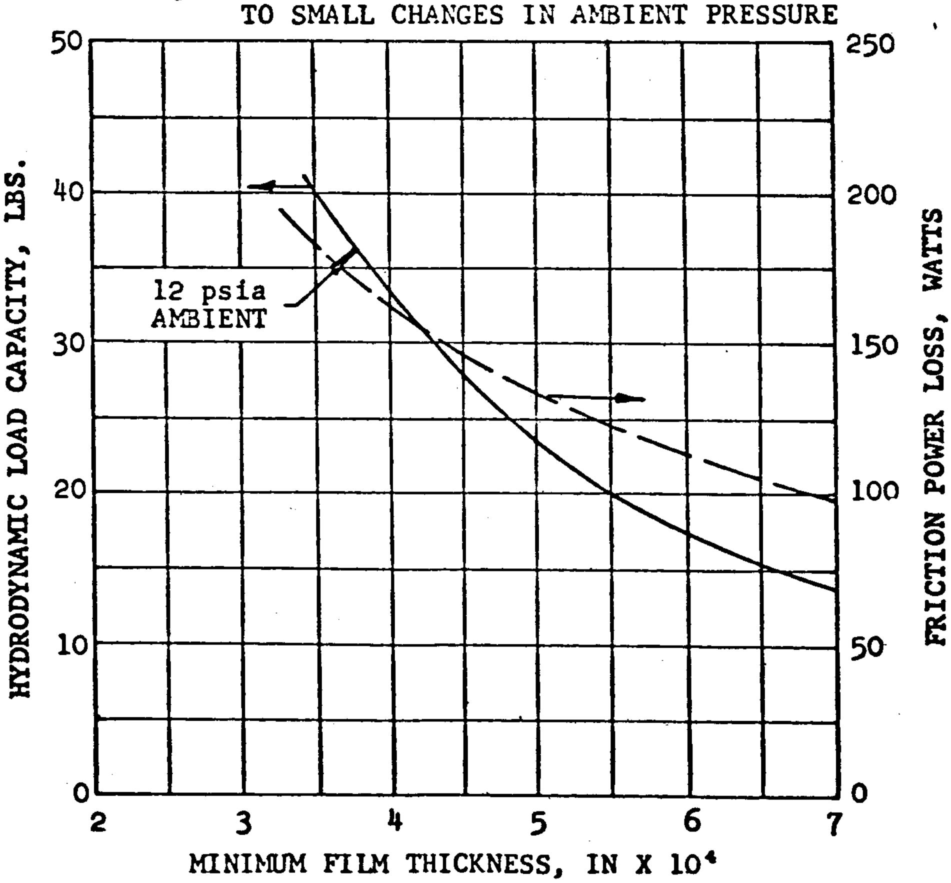


FIGURE 16. - HYDRODYNAMIC THRUST BEARING PERFORMANCE. RADIAL FLOW TURBOCOMPRESSOR.

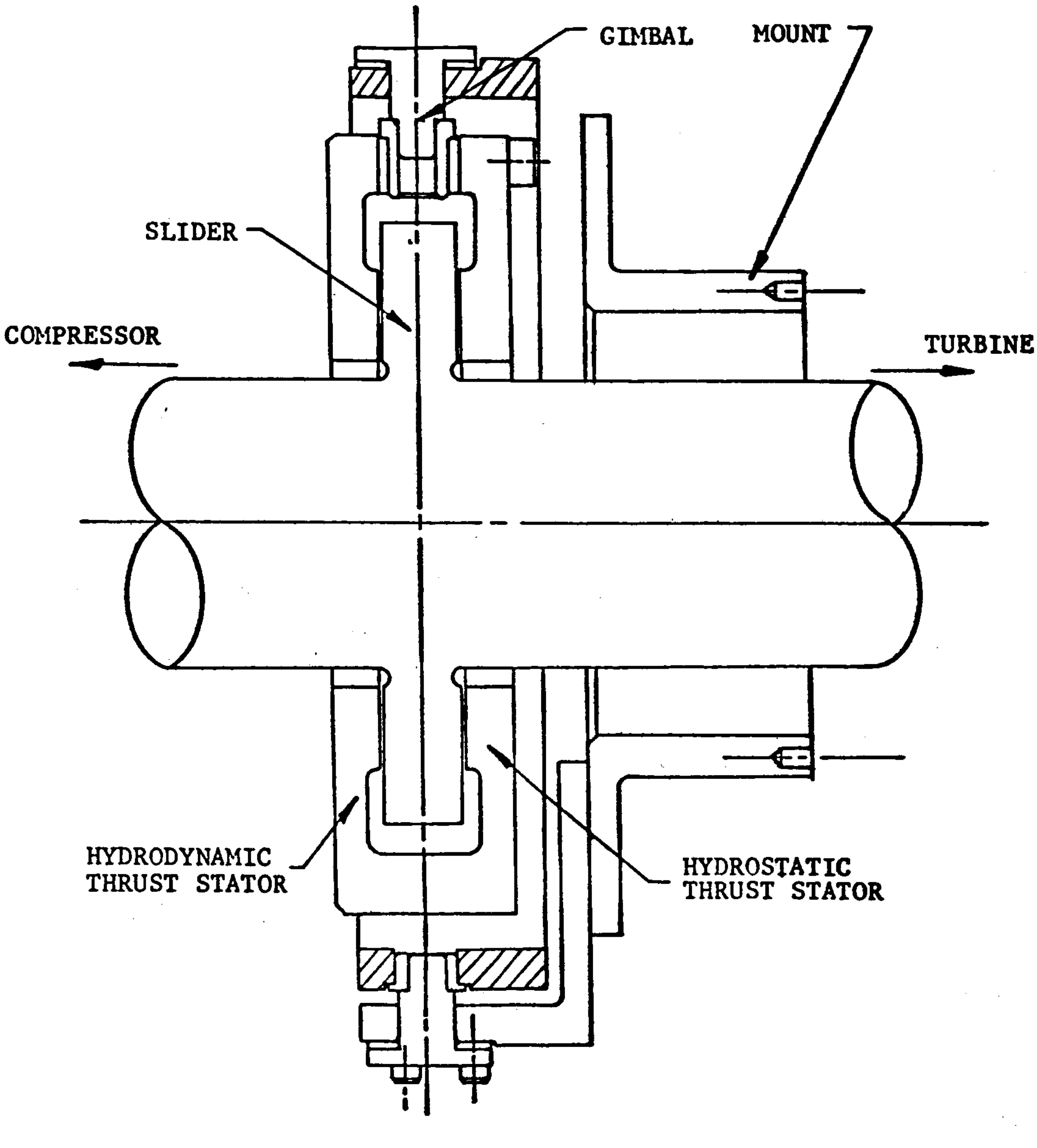


FIGURE 17. - THRUST BEARING AND GIMBAL ASSEMBLY. RADIAL FLOW TURBOCOMPRESSOR.

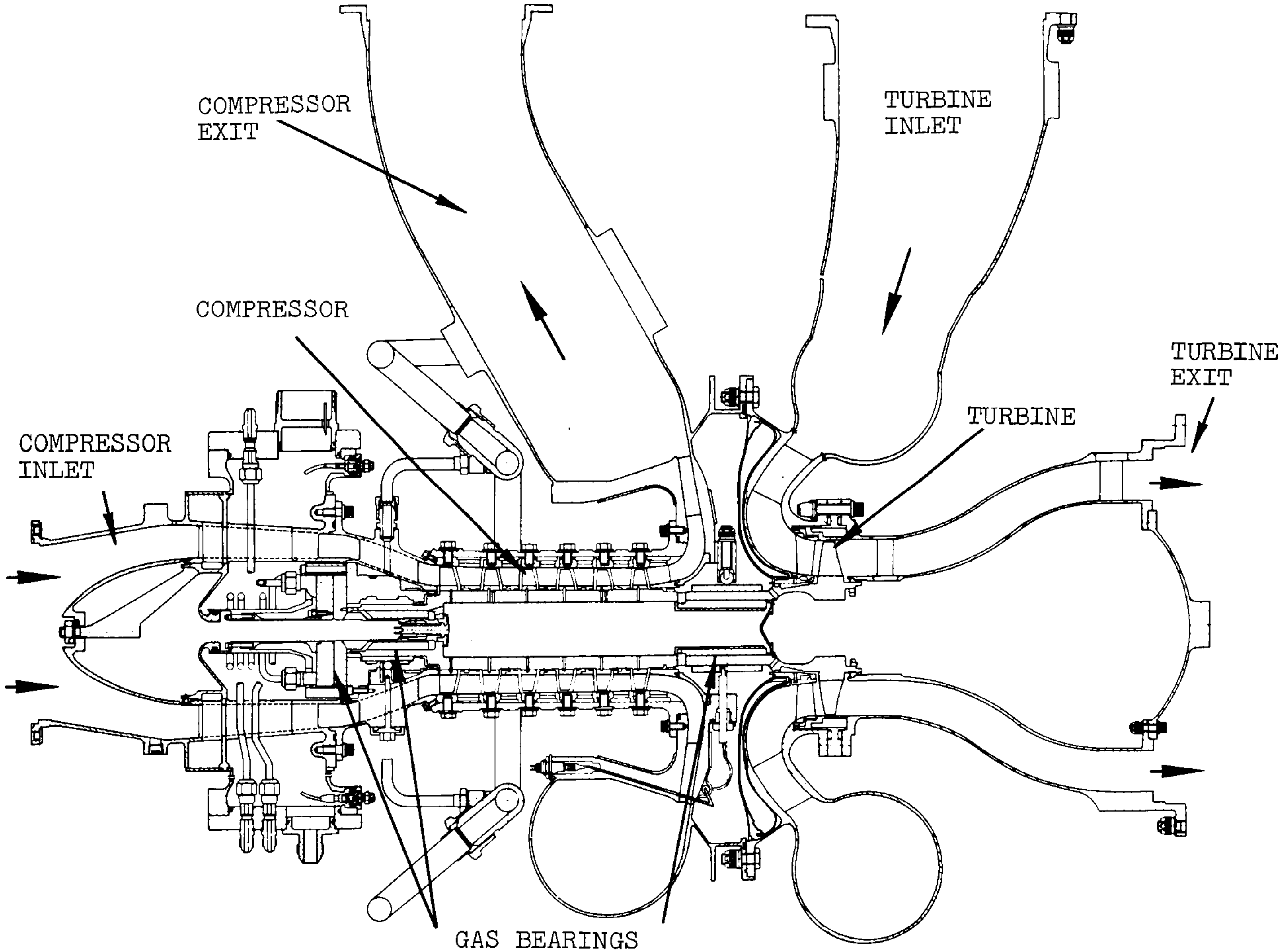


FIGURE 18. - AXIAL FLOW TURBOCOMPRESSOR.

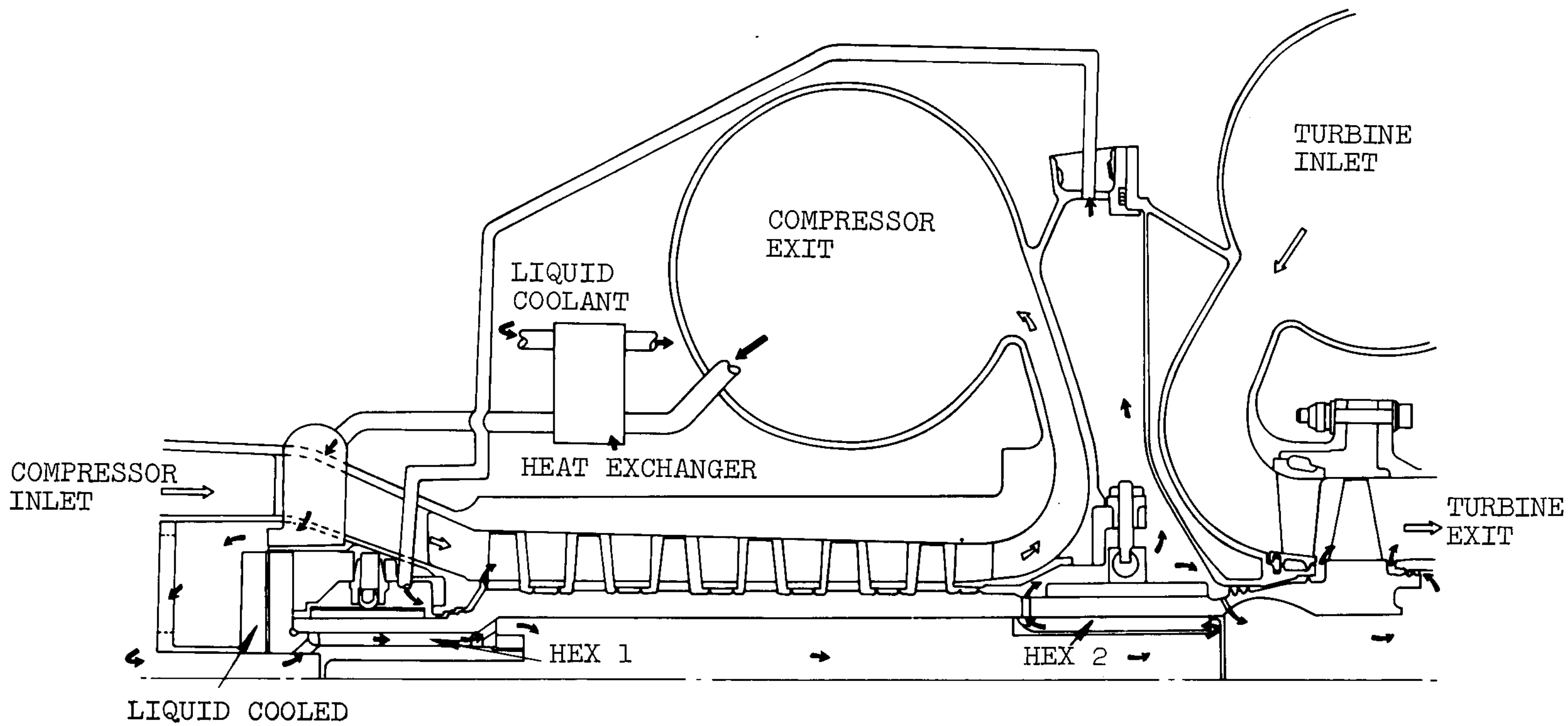
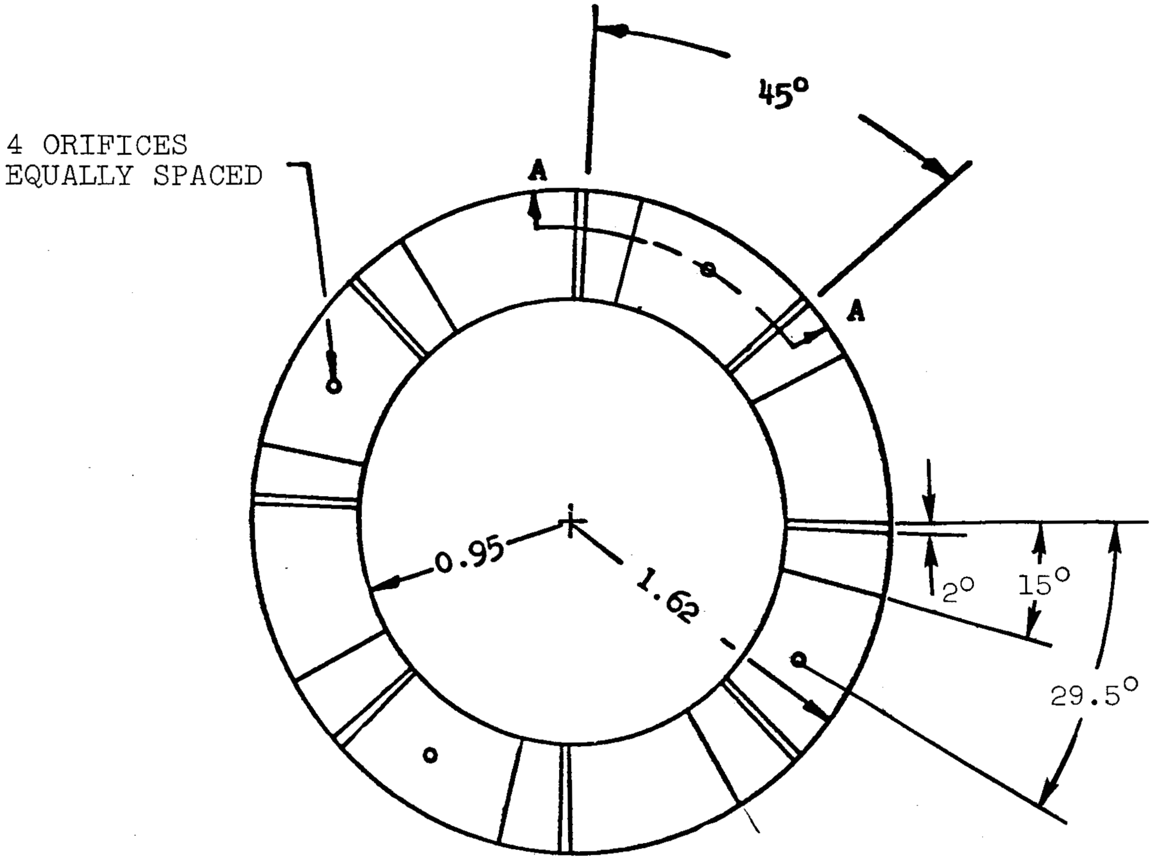


FIGURE 19. - GAS COOLING FLOW PATH. AXIAL FLOW TURBOCOMPRESSOR.



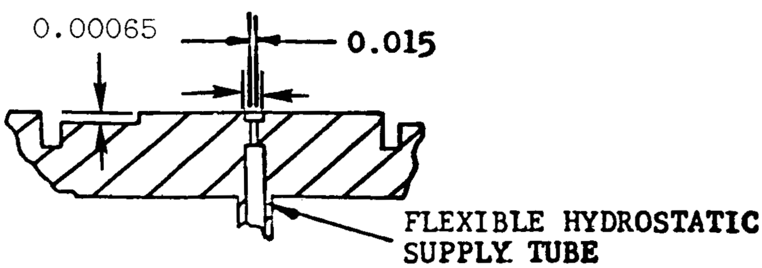


FIGURE 15. - HYDRODYNAMIC THRUST BEARING. RADIAL FLOW TURBO-
COMPRESSOR.

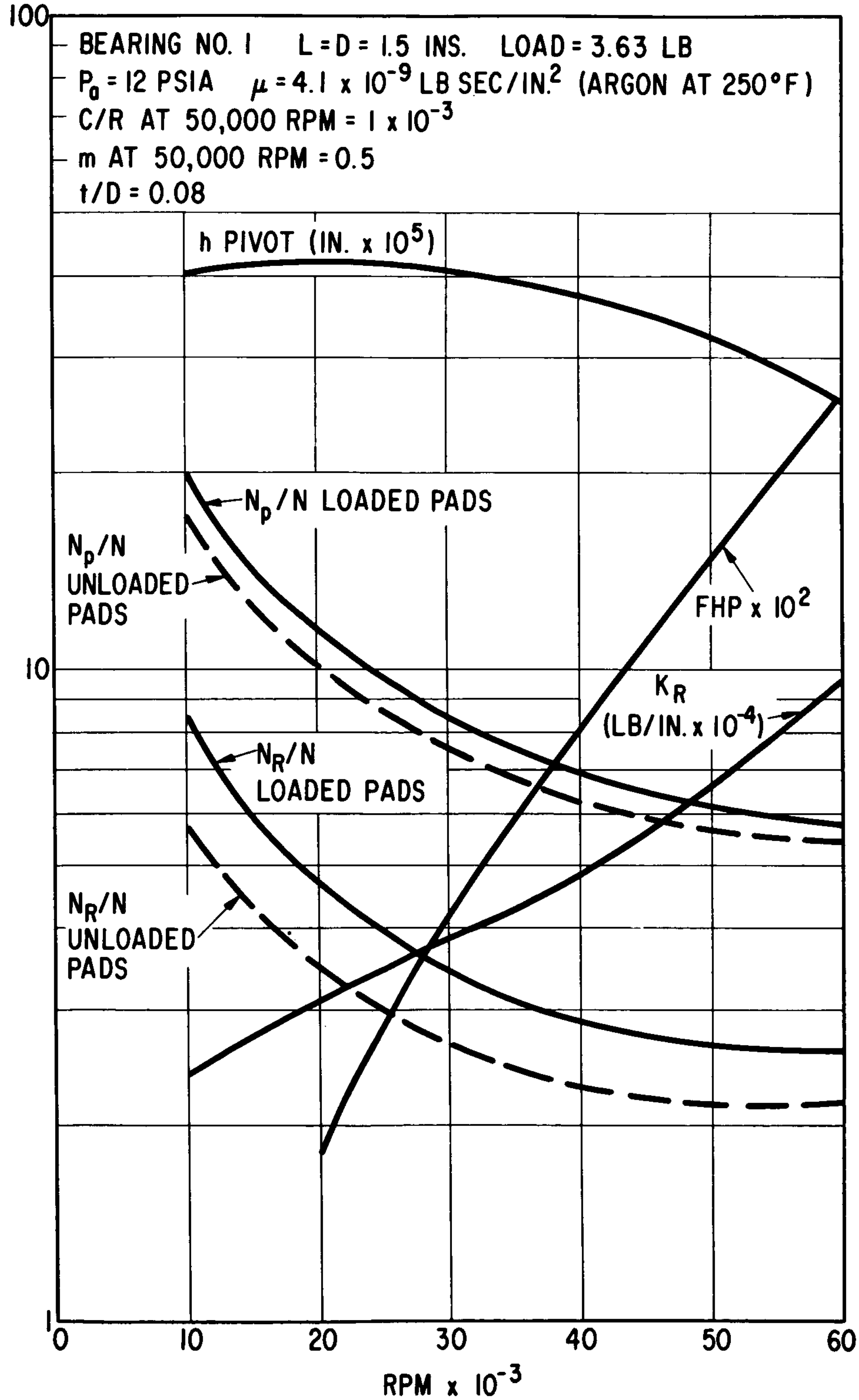


FIGURE 20. - NO. 1 JOURNAL BEARING CHARACTERISTICS WITH C/R = 0.001. AXIAL FLOW TURBOCOMPRESSOR.

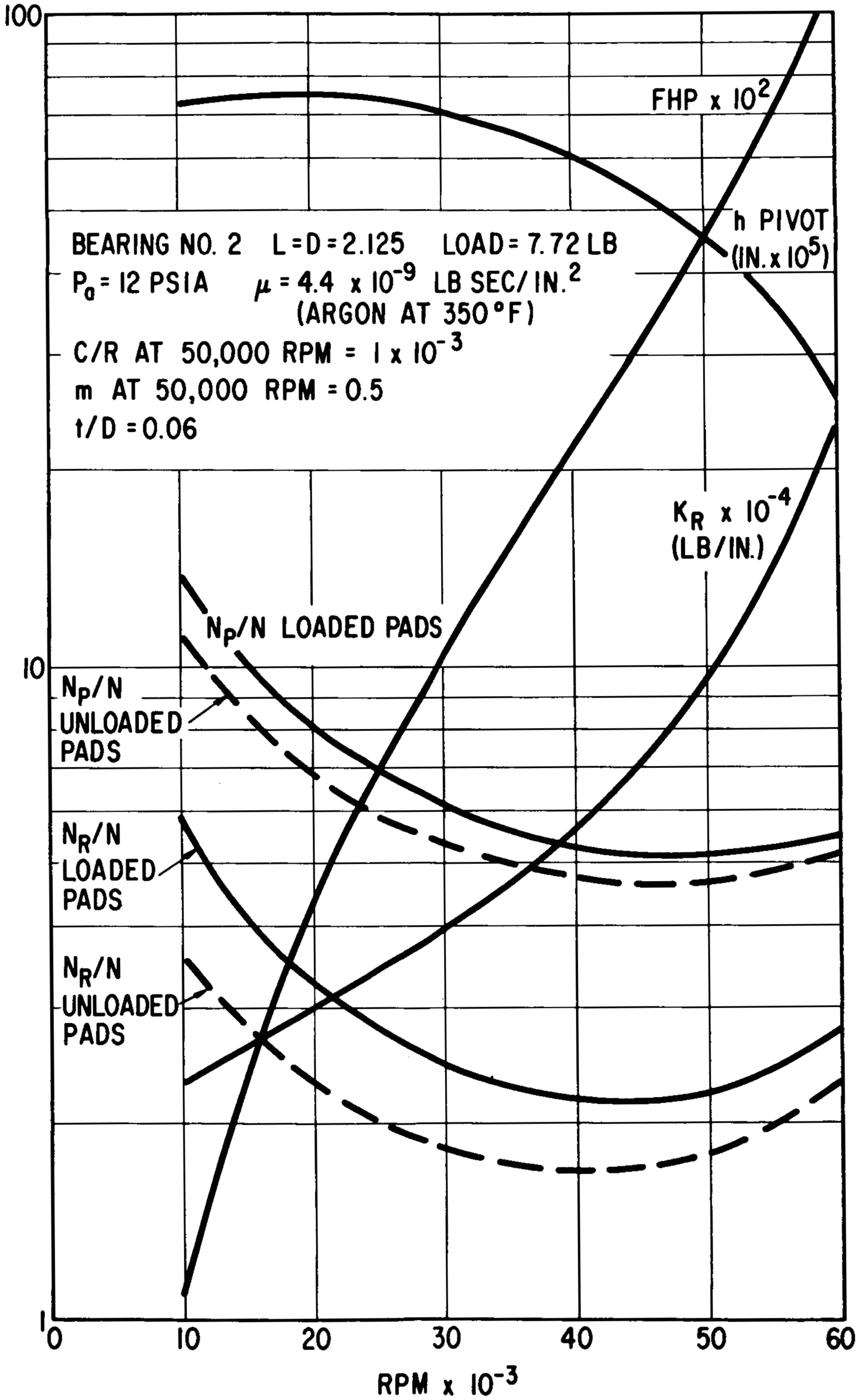


FIGURE 21. - NO. 2 JOURNAL BEARING CHARACTERISTICS WITH C/R = 0.001. AXIAL FLOW TURBOCOMPRESSOR.

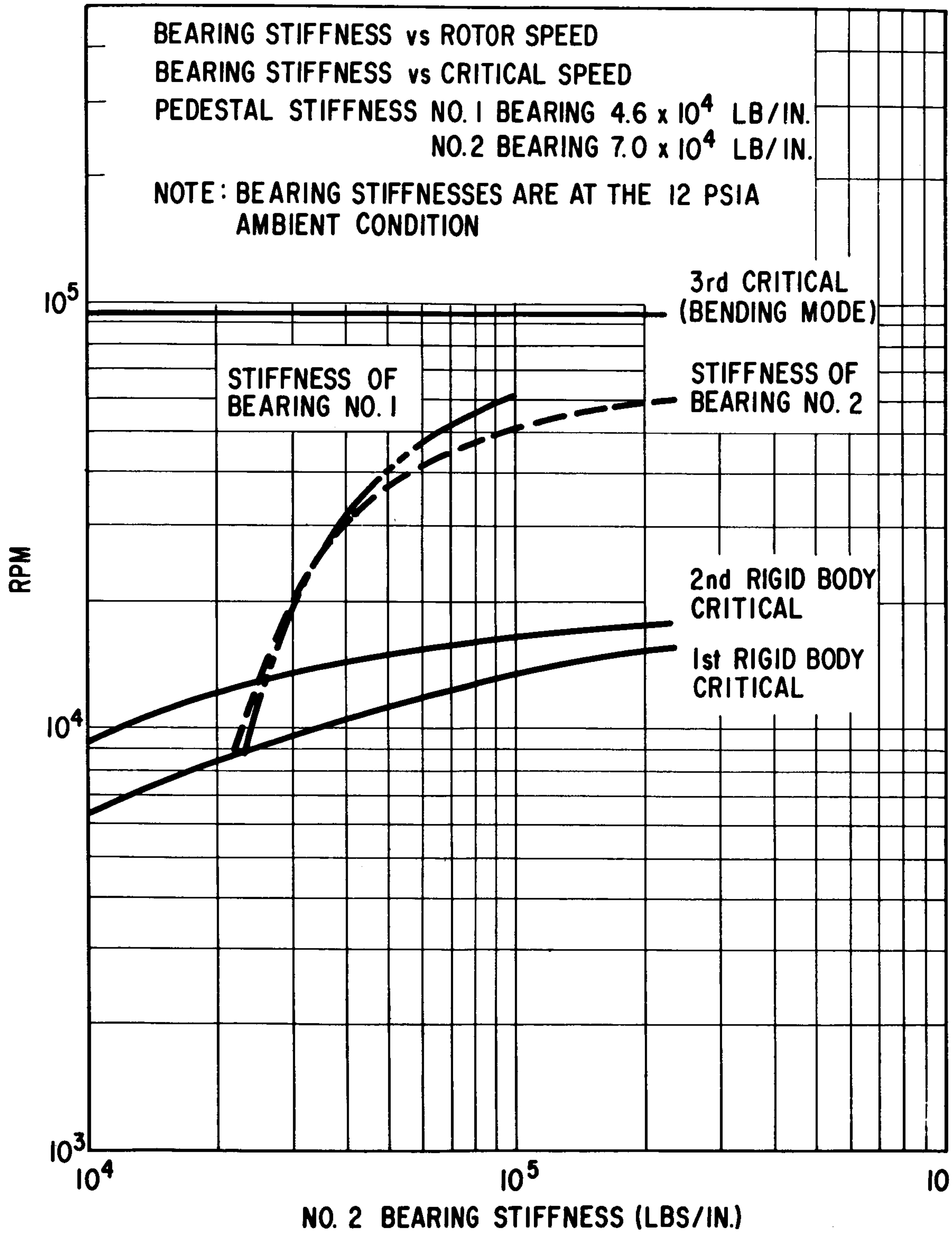


FIGURE 22. - CRITICAL SPEED VERSUS BEARING STIFFNESS. AXIAL FLOW TURBOCOMPRESSOR.

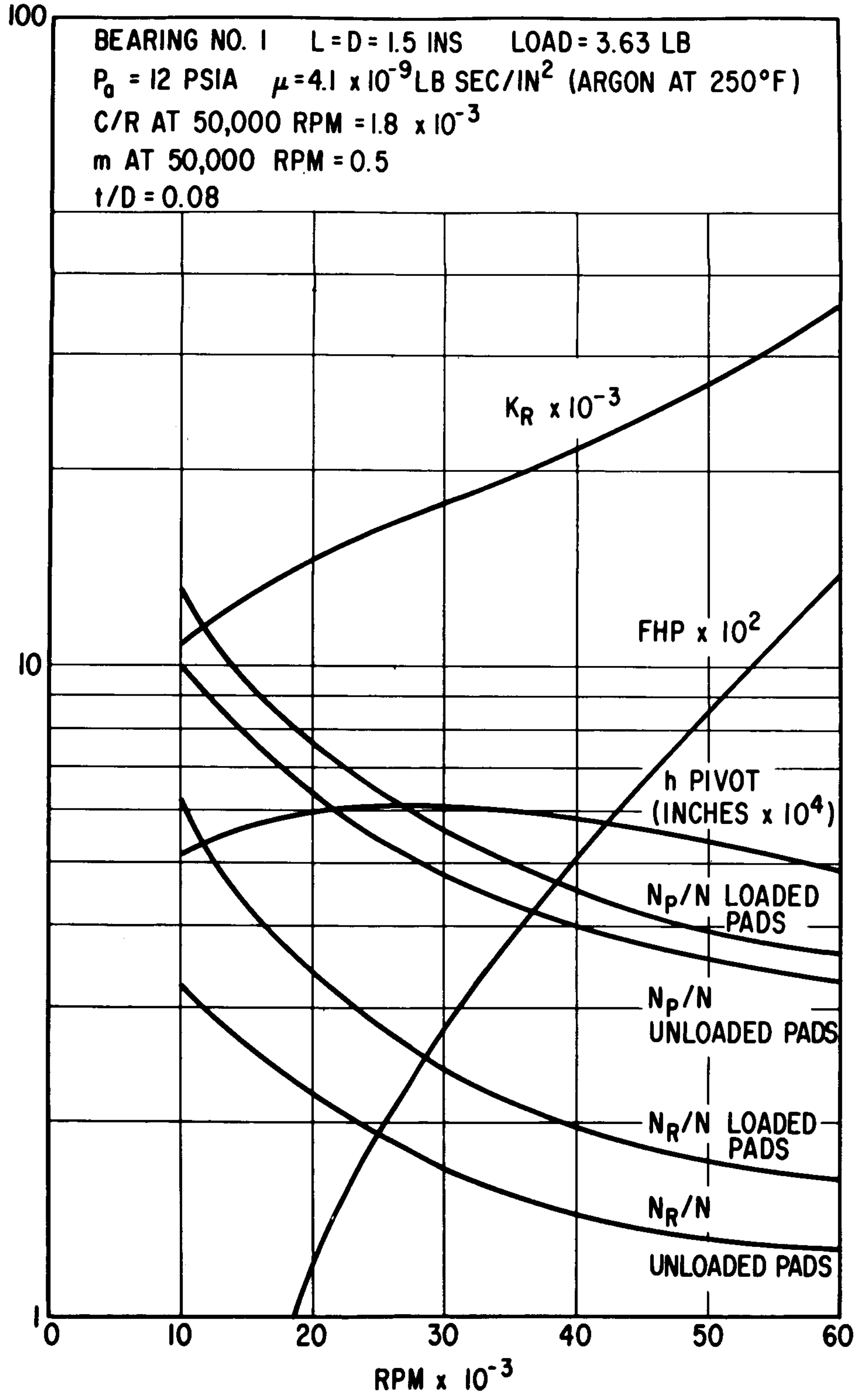


FIGURE 23. - NO. 1 JOURNAL BEARING CHARACTERISTICS AT 12 PSIA WITH C/R = 0.0018. AXIAL FLOW TURBOCOMPRESSOR.

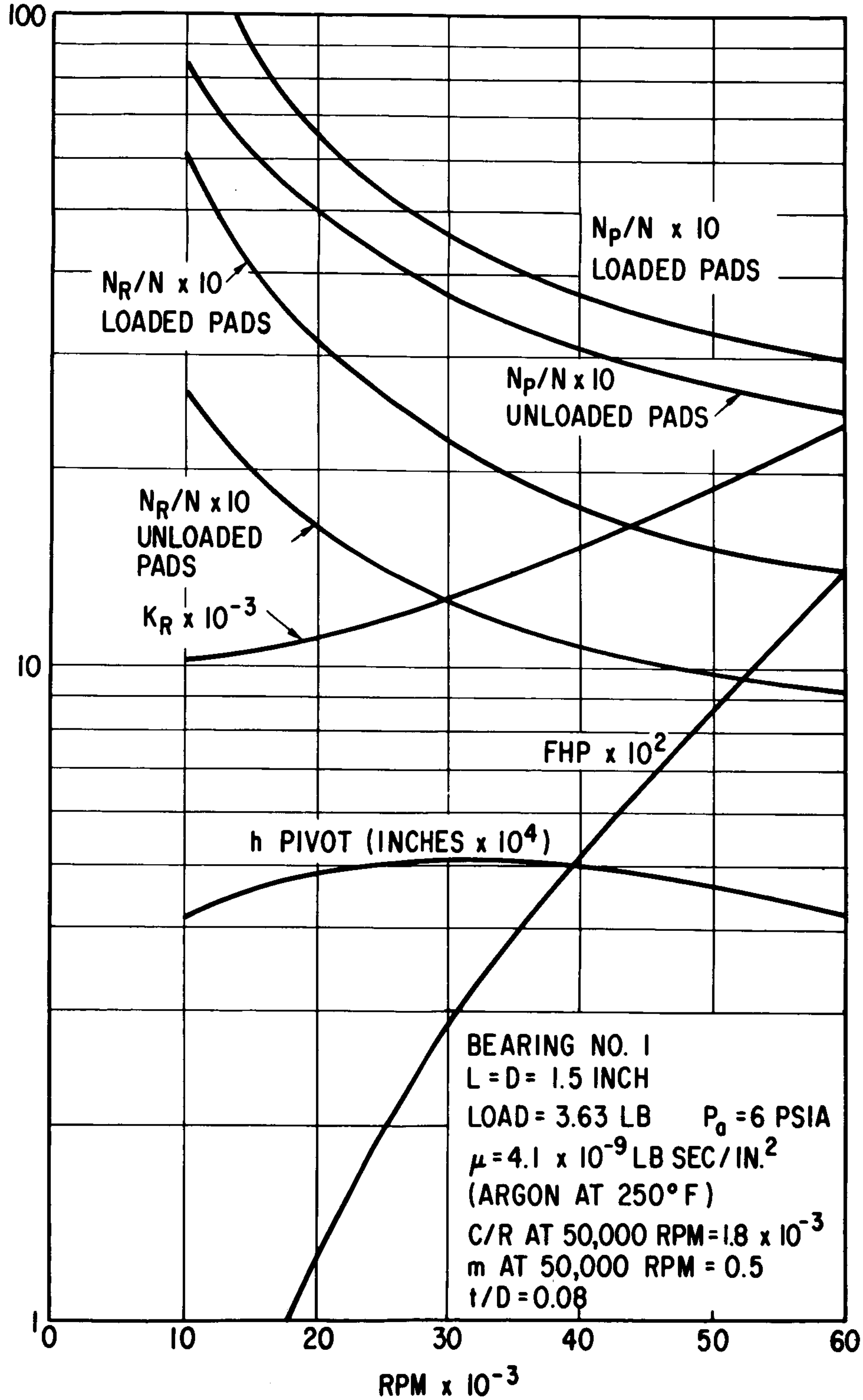


FIGURE 24. - NO. 1 JOURNAL BEARING CHARACTERISTICS AT 6 PSIA WITH C/R = 0.0018. AXIAL FLOW TURBOCOMPRESSOR.

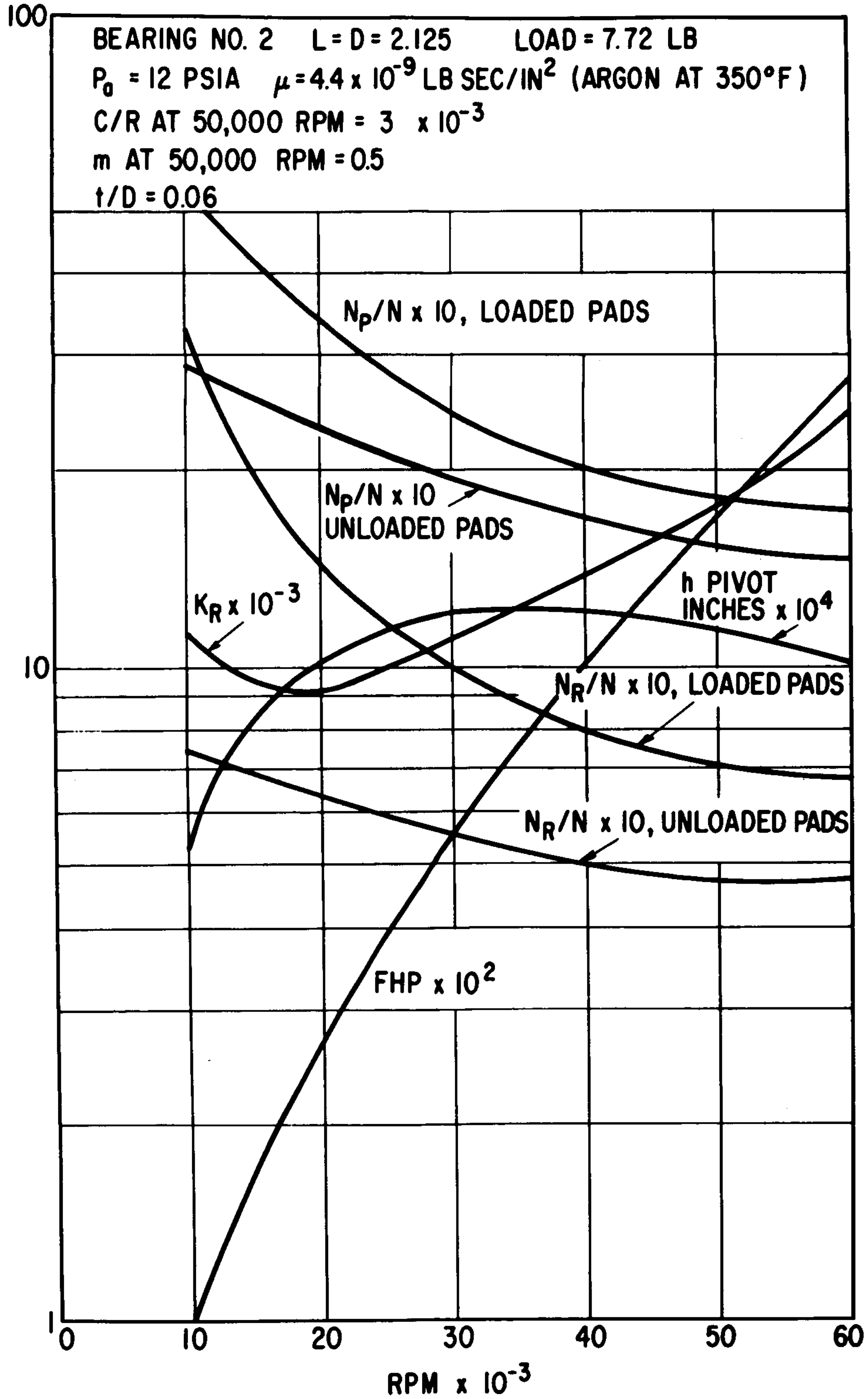


FIGURE 25. - NO. 2 JOURNAL BEARING CHARACTERISTICS AT 12 PSIA WITH C/R = 0.003. AXIAL FLOW TURBOCOMPRESSOR.

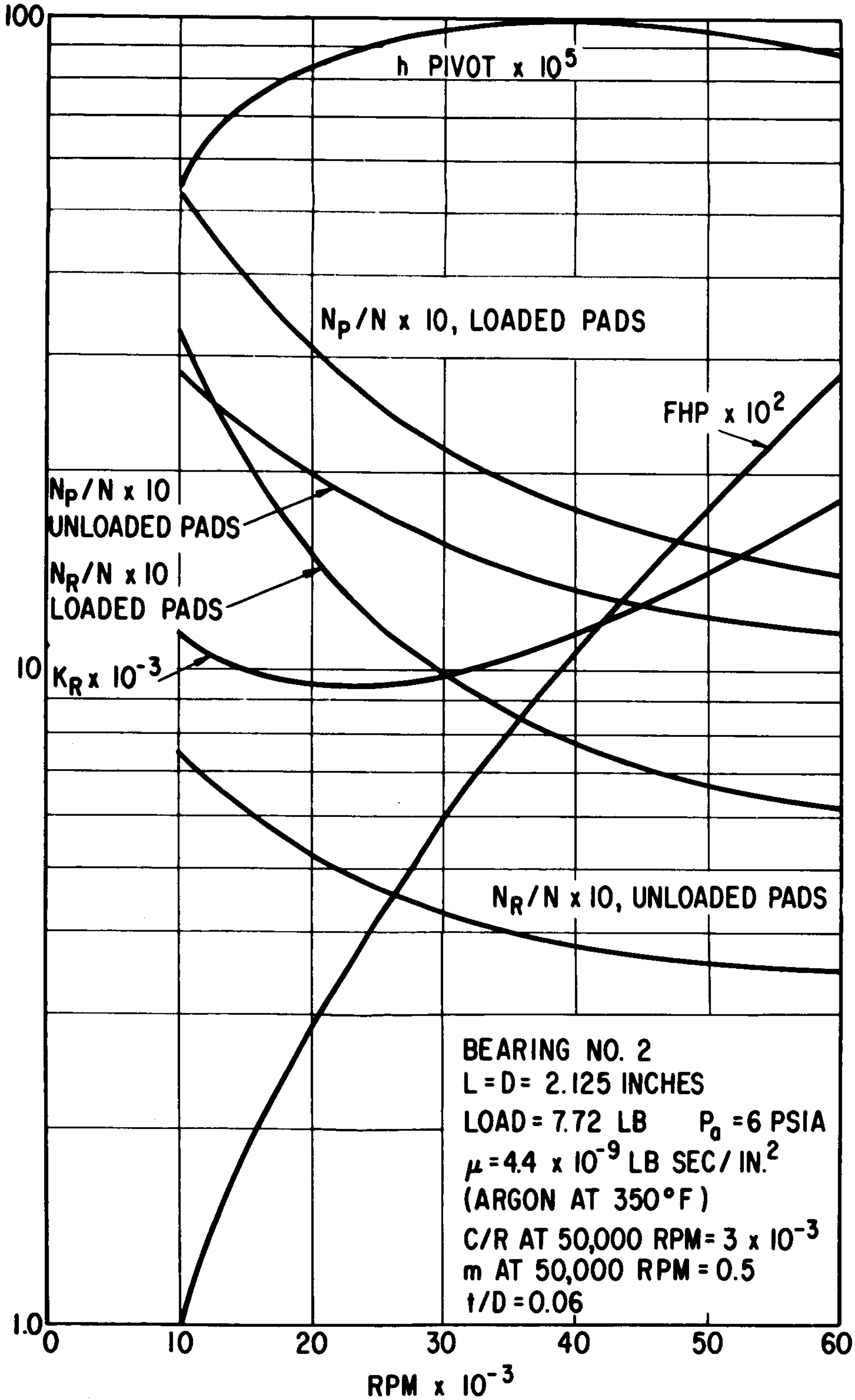


FIGURE 26. - NO. 2 JOURNAL BEARING CHARACTERISTICS AT 6 PSIA WITH C/R = 0.003. AXIAL FLOW TURBOCOMPRESSOR.

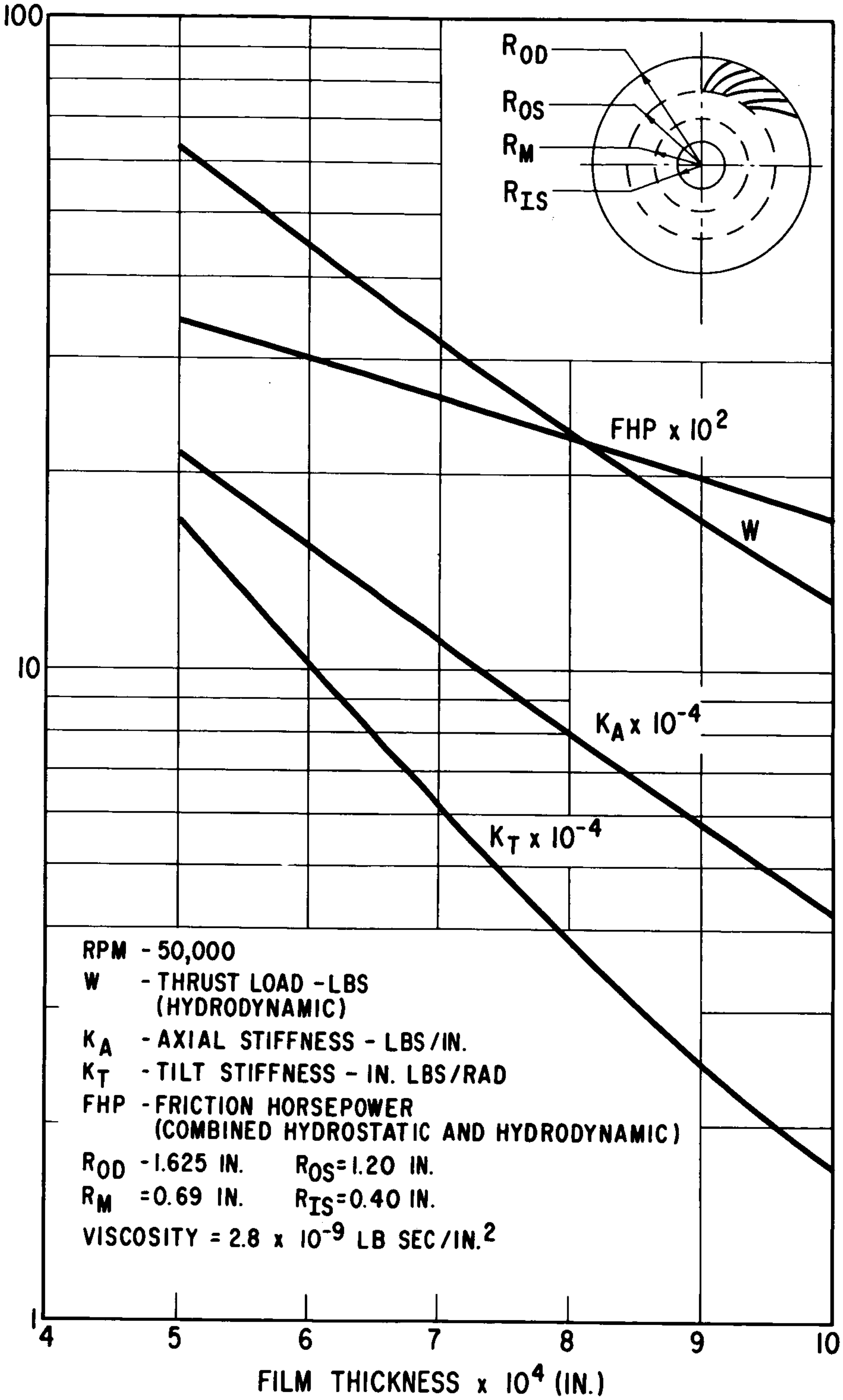


FIGURE 27. - PERFORMANCE CHARACTERISTICS OF THE SELF-ACTING SPIRAL GROOVE THRUST BEARING. AXIAL FLOW TURBO-COMPRESSOR.

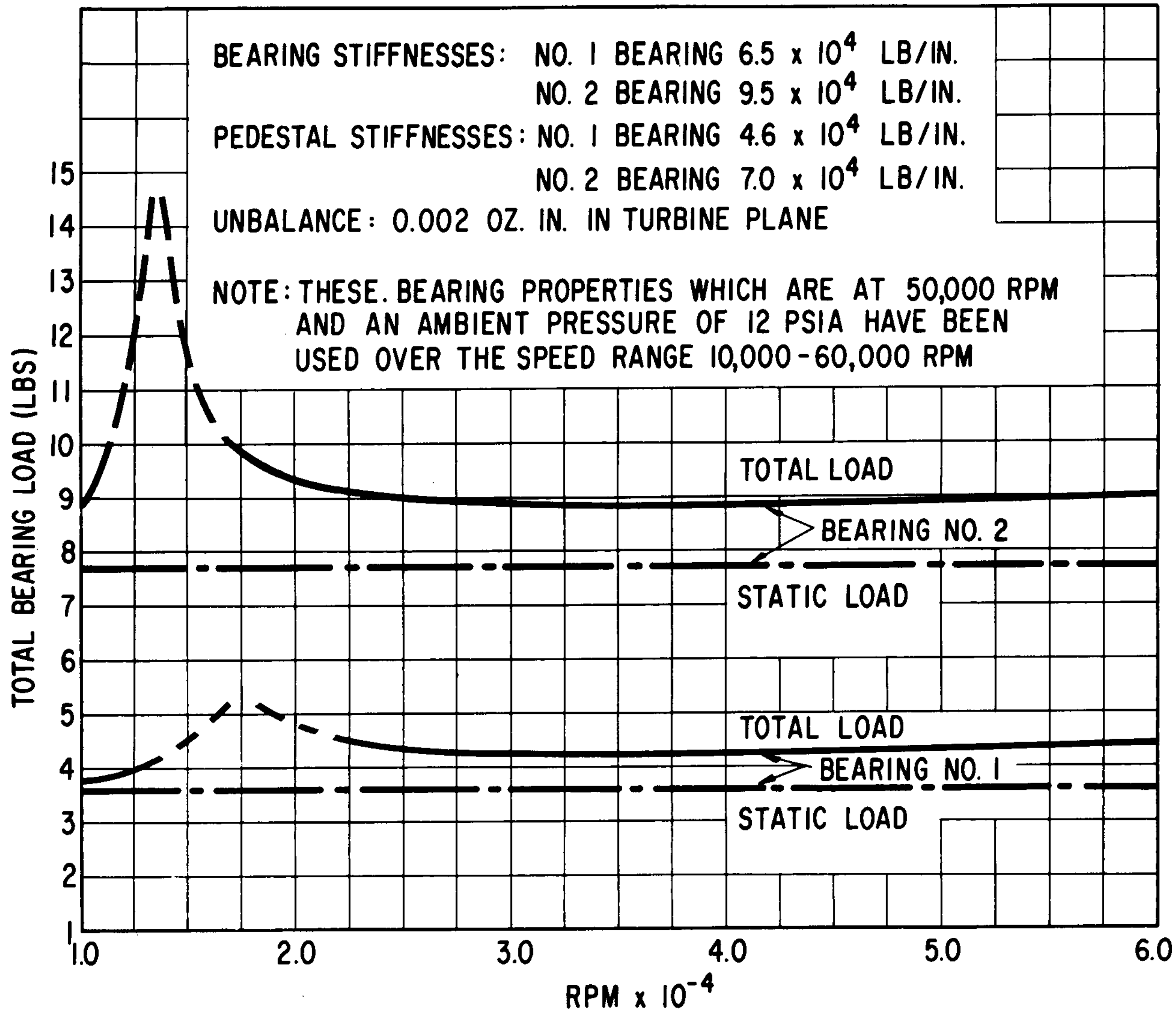


FIGURE 28. - UNBALANCE RESPONSE OF AXIAL FLOW TURBOCOMPRESSOR ROTOR.

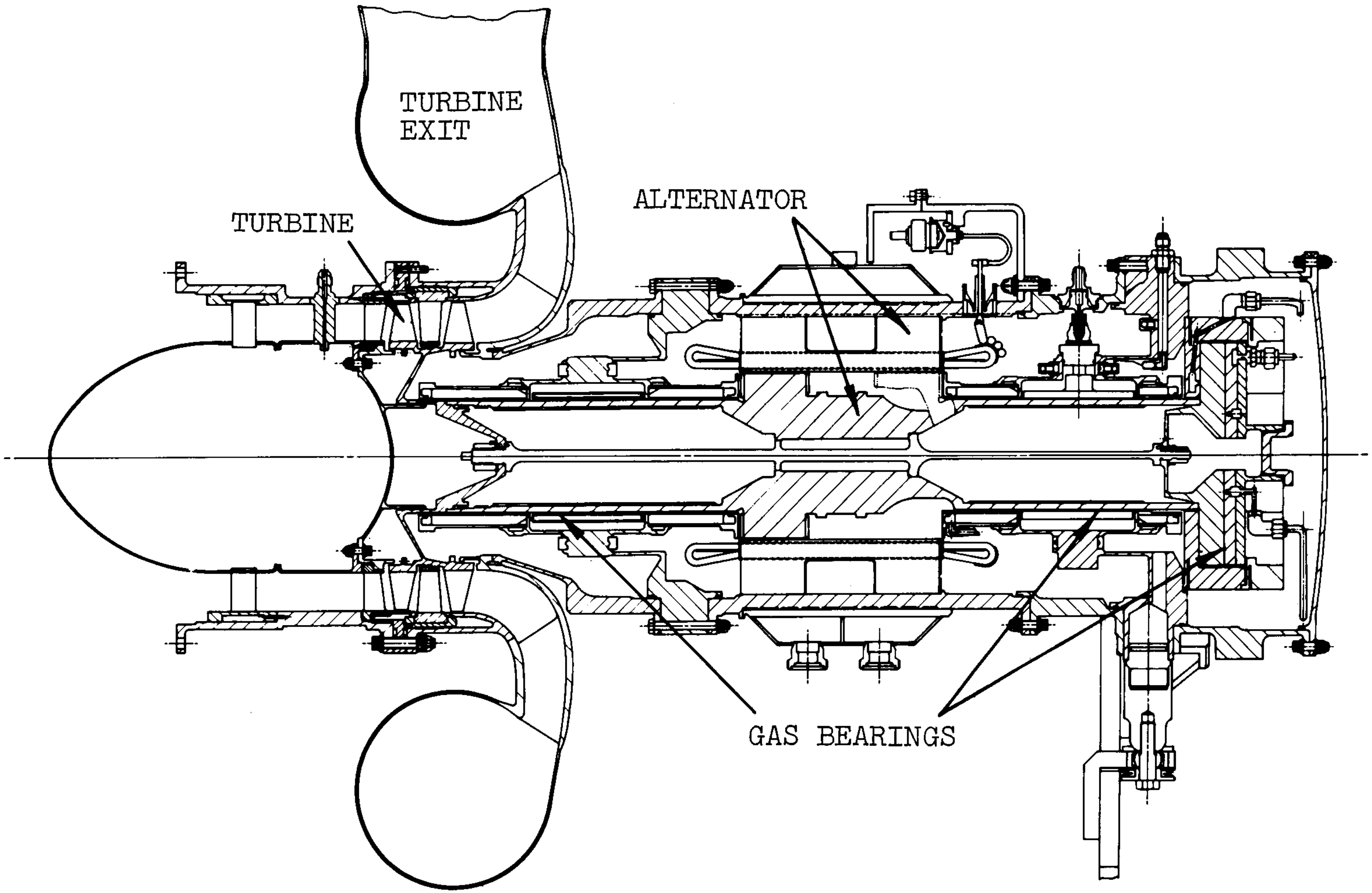


FIGURE 29. - TURBOALTERNATOR.

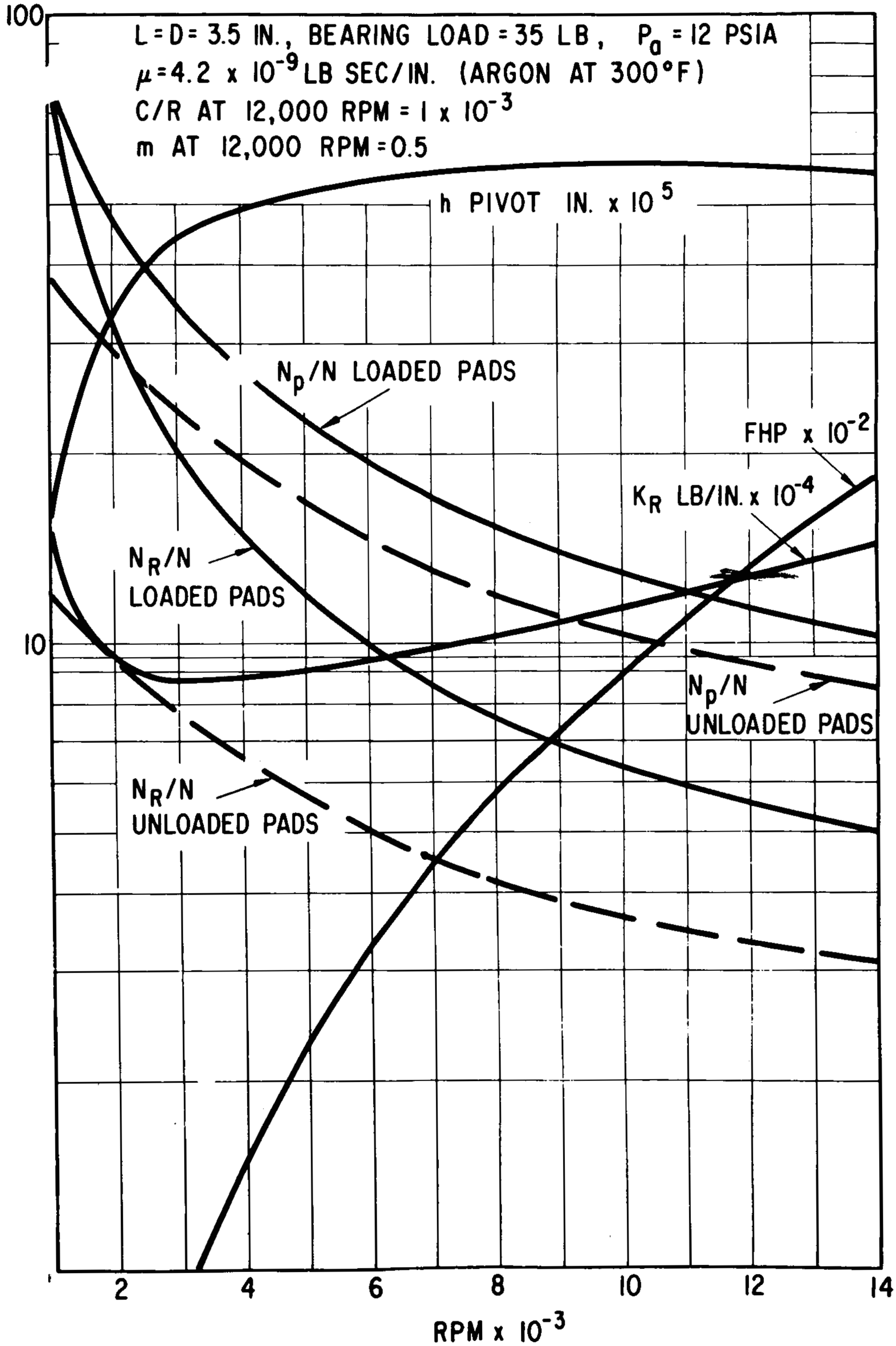


FIGURE 30. - TURBOALTERNATOR JOURNAL BEARING PERFORMANCE CHARACTERISTICS AT 12 PSIA.

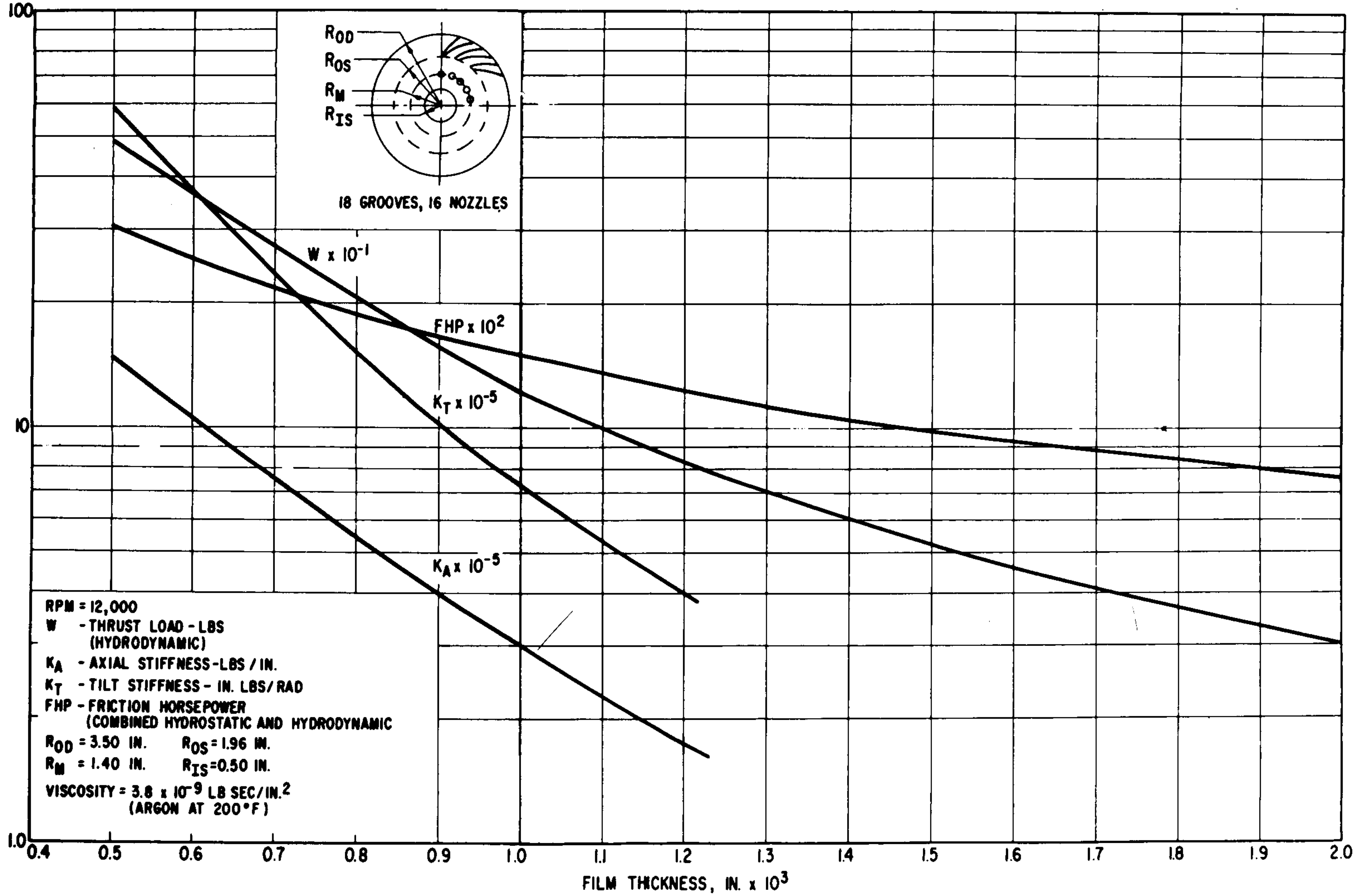


FIGURE 31. - PREDICTED PERFORMANCE OF TURBOALTERNATOR SELF-ACTING SPIRAL GROOVE THRUST bearing.

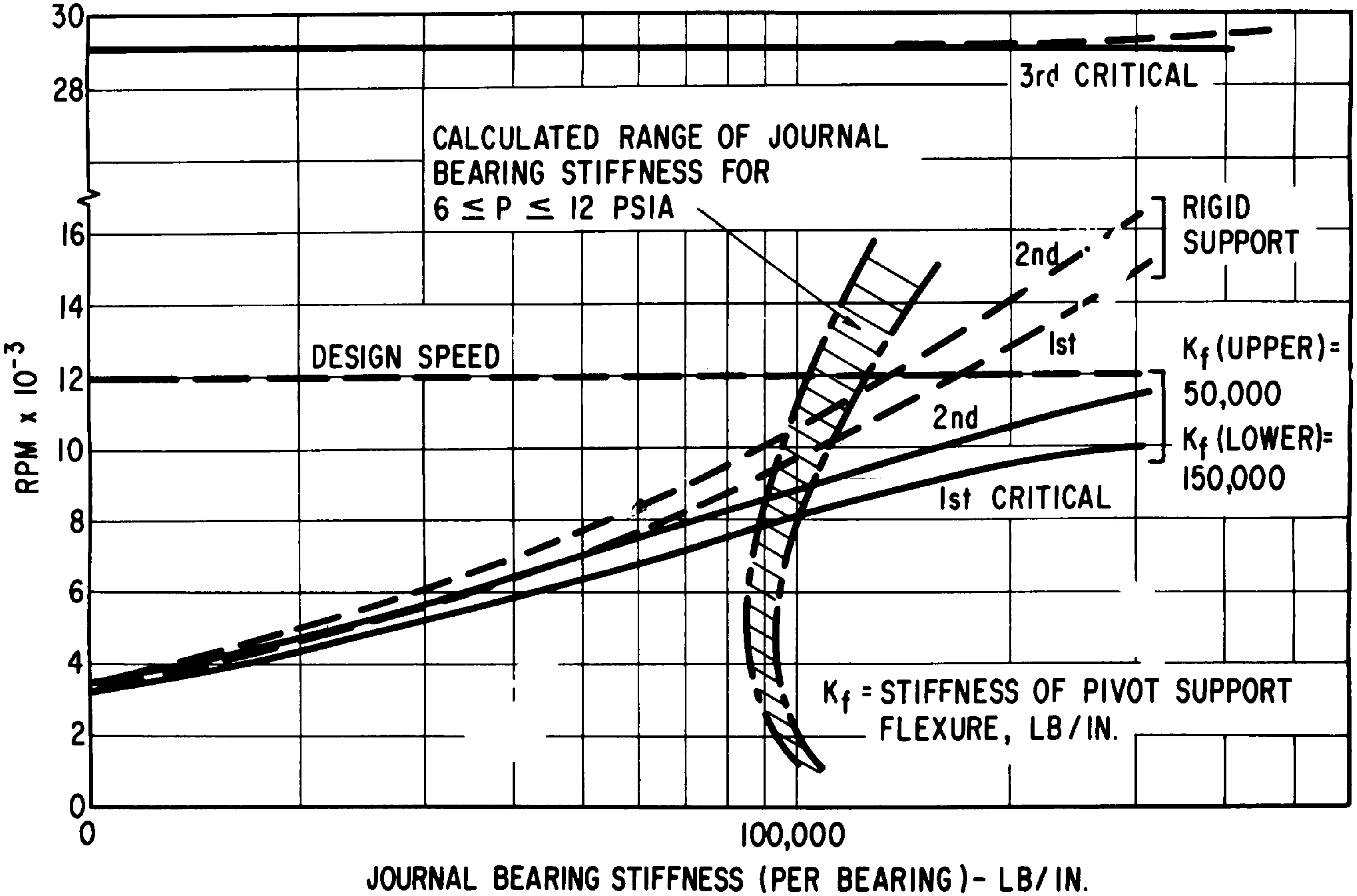


FIGURE 32. - CRITICAL SPEEDS VERSUS JOURNAL BEARING STIFFNESS - TURBOALTERNATOR ROTOR.