7. BEARING AND SEAL TECHNOLOGY

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Space-power systems and advanced aircraft gas turbines require bearings ranging in size up to approximately 6 inches in bore diameter and seals to about 7 inches in diameter. The application of our bearing and seal technology to power generating equipment would, therefore, represent a considerable extrapolation of size. Nevertheless, some of the improvements in bearings and seals, which have their roots in aerospace technology, may very well influence a change in the design philosophy used in generating equipment. Some development work would be required, of course, since the actual hardware was designed to meet different requirements.

BEARINGS

Bearing Power Loss

As generating systems get larger and larger, the power loss in the bearing and seal systems becomes a factor of greater economic importance. Larger units mean higher sliding speeds in the bearings and seals. Sliding speeds in bearings are now high enough so that the flow in the bearing film is no longer laminar but is at least partially turbulent. This has resulted in a sharp rise in bearing power loss.

An illustration of turbulence effects on power loss is given in figure 7-1. Since the degree of turbulence depends primarily on the Reynolds number in the bearing film, total power loss has been plotted against film Reynolds number. The Reynolds number in the film is a function of the sliding velocity V, the clearance C, and the kinematic viscosity γ, of the oil. As the machine size is increased, both the velocity V and the clearance C increase, resulting in a sharp increase in Reynolds number, turbulence, and total power loss. Two units of 240 and 600 megawatts, now in service, and a projected unit of 936 megawatts are spotted on the plot. The 600-megawatt unit has a total bearing power loss of 4000 kilowatts, while the projected total bearing power loss for the 936-megawatt unit is some 12 000 to 13 000 kilowatts. In this unit, the flow in the bearing film is fully turbulent.

The power loss of a rolling bearing system is shown in figure 7-2, plotted against shaft surface velocity. At low speeds, the power loss in a sliding bearing is comparable to that in a rolling bearing, but at high speeds it is considerably greater, especially when the flow in the bearing film becomes turbulent. Rolling bearings are inherently low power-loss bearings, especially at high speeds. For this reason, in part, they are used in space-power systems and in aircraft turbine engines, where power loss is critical.

In addition, rolling bearings perform well with lubrication techniques designed to minimize power losses. To illustrate, a typical lubrication system concept for minimizing power loss is shown in figure 7-3. Since very little oil is actually required to lubricate a rolling bearing, only a small quantity of oil is allowed to pass through the bearing. This keeps the losses due to churning and viscous shearing low. The major portion of the oil flow is bypassed around the bearing to provide cooling. It may be made to flow under the inner ring as shown, through the housing, or both. In large turbine engine ball bearings the inner ring is often split and provided with slots in the bearing center plane. A portion of the oil flowing underneath the inner ring passes radially out through the slots and through the bearing to both lubricate and cool. The concept shown in this figure was developed under contract for NASA by the Pratt & Whitney Division of United Aircraft Corporation.

As was stated previously, bearing power losses in large generating units now in service range up to 4000 kilowatts. This represents a loss of some $250 000 per year, based on a 6-mil per kilowatt-hour rate, and projected bearing power losses in units now being designed are several times as high because of turbulence effects. The use of rolling bearings would significantly reduce these losses, but these gains would have to be weighed against development costs and predicted reliability.

Rolling Element Bearing Reliability

Application severity. - The use of rolling bearings in generating equipment raises questions regarding their speed limitations, life, and reliability. Much progress has been made in these areas in the past decade, however, as a result of aerospace research. Any comparisons of bearing performance and reliability must be made on the basis of the severity of the application. Table 7-I illustrates the relative severity of bearing applications in industrial gas turbines and in a projected 1000-megawatt gas-turbine generating system. Two parameters are used. The parameter DN is the product of bearing bore in millimeters and shaft speed in rpm. It is the most widely used severity parameter, and is analogous to a peripheral speed. In comparing the performance of very high-speed bearings, however, DN has distinct shortcomings. Limiting DN values do not remain constant for a range of bearing sizes. Data indicate that DN^2, which is analogous to a centrifugal force, may be a better parameter for gaging the severity of high-speed applications. This parameter provides an estimate of the loading between the rolling elements and the outer race. The ideal parameter may be DN^x where $1 < x < 2$, but a comprehensive program in this area has not been conducted.

As table 7-I indicates, the maximum continuous service DN value in industrial gas-turbine bearings now operating is about 1.32 million. This compares favorably with the 1.6 million value in a 1000-megawatt gas-turbine generating unit. On the basis of DN^2, the severity of bearing operation in industrial gas turbines greatly exceeds that in the 1000-megawatt projected gas-turbine generating unit. It, therefore, appears that bearings in present industrial gas turbines operate under conditions comparable to those in projected gas-turbine generating units.

Industrial gas turbines operate for periods of 16 000 hours without maintenance. The bearings and seals operate with high reliability for this time period, which is considerably greater than the longest overhaul time for aircraft turbine engines. However, aircraft engines must operate over a wide range of speeds, power ratings, and environmental pressures and temperatures, whereas industrial engines operate at a reasonably fixed set of conditions. Operation at a fixed condition improves the life of rolling bearings and especially of face seals.

Fatigue life. - The use of rolling bearings in applications where reliability is of paramount importance has usually been avoided because they are subject to fatigue failures. These failures can be predicted only on a statistical basis, so that failure by fatigue of an individual bearing cannot be predicted. It is highly improbable that rolling fatigue will ever be eliminated, but several developments of the past decade have significantly increased the fatigue life and reliability of rolling bearings. Three of these techniques are listed as follows:

(1) Consumable electrode vacuum melting

(2) Fiber control

(3) Hardness control

Since the origin of fatigue is in the material, attempts to improve fatigue life must deal with improvements in the raw material and its processing until the bearing is finished. Until recently, most quality bearing steels were melted in a ceramic crucible and in an air environment. Air-melted materials usually contain hard oxide inclusions that result from the reaction of the melt alloys and oxygen. In addition, ceramic contaminants from the crucible frequently go into the melt. These act as stress raisers which promote early fatigue cracking. The growing use of consumable-electrode vacuum-melted materials has resulted in more uniformly high-quality bearing materials and longer life bearings (fig. 7-4). In the consumable-electrode vacuum-melting process, an arc is struck between the ingot, which serves as an electrode, and a water-cooled copper crucible. A vacuum is drawn to remove gases as the energy from the arc gradually melts the ingot. The vacuum environment prevents the formation of oxides, and the copper crucible eliminates the source of ceramic contaminants.

The effects of vacuum melting on bearing life are dramatic, as indicated by the bar graph. Each remelt results in improved life with a fourfold increase in life after five remelts. The actual life values shown are not important because these data were obtained in highly accelerated life tests.

A second way in which bearing fatigue life can be improved is through better manufacturing. Whenever a material is shaped by rolling, forging, or extrusion, the metal grains take on a stringlike pattern, resembling fibers - thus, the term fiber flow. The fiber flow in a bearing race that has been cut from tubing is illustrated in figure 7-5. Research conducted here at Lewis has shown that metals are weaker in fatigue when the ends of the fibers are exposed to the stressed surface than when the fibers are parallel to the surface. Note the fiber ends, which are exposed to the stressed region in the race cut from tubing.

If a special forging technique is used, a race with essentially parallel fibers in the highly stressed ball groove can be produced. The Fafnir Bearing Company has developed specially forged races based on the concepts developed at this Center. These indicate that a tenfold life improvement can be expected.

A third way in which bearing fatigue life can be improved is through hardness control. Research at this Center has indicated that resistance to rolling fatigue generally improves with increasing hardness. It would seem to follow from this that the rolling elements and races of a bearing should be heat treated for maximum hardness. Further research at this Center has indicated, however, that maximum fatigue life is achieved if the balls are made two points harder (as measured on the Rockwell C scale) than the races. These results are shown in figure 7-6 in which relative bearing life is plotted as a function of the difference in ball and race hardness ΔH. Each point represents a life determination for a group of 30 bearings. It is apparent that fatigue life can be reduced as much as 80 percent if ΔH varies much from 1 to 2. Normal manufacturing variations may result in bearings being assembled with ΔH values several points away from the optimum, with a consequent significant reduction in fatigue life. This illustrates the importance of specifying not only the proper ΔH but, also, of closely controlling component hardnesses so that the desired ΔH is actually achieved.

These are but some of the techniques that have been found to improve fatigue life. Others, including improved nondestructive inspection techniques for the location of minute flaws, are under development. Improvements in life of greater than an order of magnitude have been demonstrated, and it is entirely possible that further improvements of a like order may be achieved.

SEALS

The performance and reliability of seals are also critical to the proper functioning of turbomachinery. Because of reliability considerations, buffer and labyrinth seals, rather than face seals, have generally been used throughout steam systems and, in some locations, in gas turbines. These noncontact seals are essentially reliable, but a significant loss in thermal efficiency is incurred because of seal leakage. In addition, these seals require a considerable axial length, and this has led to problems with critical speeds.

Face Seals

Leakage rates can be greatly reduced by using face seals (fig. 7-7), but there are problems that would have to be solved before their reliability could be considered satisfactory.

Figure 7-7 is a schematic of a conventional face seal used in many aircraft gas-turbine engines. It consists of a rotating seat attached to the shaft and a nonrotating nosepiece, which is held in sliding contact or close proximity to the rotating seal by a light spring force and by a force from the sealed gas pressure P_1 acting across the annular area behind the dam. For a typical 7-inch-diameter seal, sliding speeds to 300 feet per second are common.

A piston ring permits axial motion of the nose to accommodate for thermal expansions and to allow the nose to follow axial run-out movements of the seat face. The seal faces or dam (shown with an exaggerated leakage gap) restrict the leakage of the high-pressure hot gas into the sump for the oil lubricated roller bearing that supports the shaft. It is important to minimize leakage of hot gas through the seal because this degrades the lubricant and adds to the cooling requirements of the lube system.

Current pressure and speed limitations on the conventional face seal result from an inability to maintain a small net closing force on the nose, that is just large enough to make the nose follow the seat, yet not so large as to cause excessive wear. wear. To achieve long seal life, rubbing contact of the nose and seat must be prevented, but this must be accomplished with very small sealing gaps so that leakage rates are acceptable.

Deformation. - The problem is basically one of maintaining a proper balance between the closing and opening forces. The closing force is a function of the area over which the pressure P_1 acts and is readily controlled by the designer. The opening force is a function of the pressure in the gap. This is not easily controlled because it is greatly influenced by deformation of the sealing faces. This is shown in figure 7-8.

Portions of the nose and seat with greatly exaggerated sealing gap deformations are shown. Deformations of the seal faces can result in a sealing gap which is either convergent or divergent in the direction of gas leakage. If the deformation is convergent, seal operation is stable because, if a disturbance causes the nose to approach the seat, then a greater separating force is automatically generated in the gap. The equilibrium pressure profile is shown as the solid and the disturbed profile as the dashed line. Since the closing force (P_1 times its area) remains the same, the greater separating force opens the gap until equilibrium is again restored. Thus, a seal with a converging gap can run without rubbing contact.

In contrast, a divergent gap is unstable. A divergent deformation gap is more common and is usually caused by thermal gradients. If the nose is displaced to the dotted position, the leakage flow is effectively pinched off. This causes the average gap pressure (as shown by the dashed line) to decrease. Now the force tending to close the seal is greater than the separating force. This will cause the nose to slam down against the seat. At the high sliding speeds in advanced gas turbines, the resulting rubbing contact will generate more heat and make the divergence worse.

Face seals incorporating gas bearings. - In a development contract at Pratt & Whitney Aircraft, one of the approaches taken to solve the gap deformation problem was to add hydrodynamic or self-acting gas bearings to the sealing faces. These act to keep the faces separated even though divergence occurs. This seal is shown in figure 7-9.

The seal construction is similar to the conventional face seal shown previously. The only difference is that a self-acting gas bearing has been added under the sealing dam. This gas bearing consists of a series of shallow recesses arranged circumferentially around the seal face. Figure 7-10 is a view of the face of the nose, showing the sealing dam area and the gas bearing recesses. Each recess has a radial feed groove. An edge view shows the feed groove and recesses with the depth (typically 0.002 in.) greatly magnified.

The motion of the seat over these recesses drags or pumps the air from the feed groove into the recesses. This pumping action raises the gas pressure in the recesses creating a separating or lifting force. The closer the nose comes to the rotating seat, the greater is this force produced by the gas bearing (fig. 7-11).

The gas bearing lifting force for a typical seal is plotted as a function of the leakage gap height. When the gap is small (of the order of 0.2 mil), the recesses exert a large force to keep the faces separated. In this example, the lifting force generated is approximately 100 pounds. However, when the gap becomes large (1.0 mil) then only a small force of some 5 pounds is exerted. The self-acting gas bearing is inherently suited for seal operation because large forces are produced only when the faces are nearly touching. There is little tendency to produce a large gap that would result in high leakage rates.

Figure 7-12 shows a comparison of seal leakage in standard cubic feet per minute as a function of sealed pressure for labyrinth, conventional, and modified face seals. The leakage of a typical gas-turbine labyrinth seal is usually at least 10 times as great as that of a conventional face seal. The conventional seal works well at low pressure, but begins to fail at high pressures because divergent sealing faces cause heavy rubbing contact. The resulting vibration and wear lead to high-leakage rates. The leakage of a seal with a self-acting gas bearing is low over the entire range of pressures. Divergent surfaces have been created by thermal deformation, but the gas bearing generates enough force to keep the surfaces from rubbing. Inspection of this seal after test showed the surfaces to be in excellent condition.

The difference in leakage of this type seal as compared with a labyrinth seal, can be translated into efficiency gains, and for this reason, we are currently working on replacing certain compressor and turbine labyrinth seals with gas film seals of this type.

Visco Seals

In addition to noncontacting seals with a gas film between the sliding surfaces, studies have been made at Lewis and its contractors on noncontacting seals for liquids. The visco seal (fig. 7-13) is one such device and is used in the SNAP-8 space-power system to seal mercury. This seal consists of a rotor with a helical groove rotating within a close fitting housing containing the sealed liquid. The clearance between the shaft and the housing is dictated by mechanical considerations, vibration, and runout and is usually several mils. This type of seal is essentially a pump, which pumps the liquid back as fast as it tends to leak out, so that a liquid-to-gas interface is established within the seal length when the pumping effort of the grooves balances the pressure to be sealed. Thus, a pressure gradient is established from ambient at the interface, increasing across the seal, to the sealed pressure. For a given sealed pressure, the length of seal occupied by the liquid depends on the peripheral speed of the rotor outside diameter, the liquid viscosity, and the clearance between the rotor and housing. If sealing is required at zero shaft speed, then a positive contact seal must be added to the system.

Models of visco seals have been constructed for the purpose of conducting visual studies of their performance. The model illustrated in figure 7-14 consists of a steel shaft in a clear plastic housing. The right portion of the shaft has helical grooves cut in its surface, and rotates in a smooth bore housing. The left portion of the shaft is smooth and rotates in a helically grooved housing. As can be seen, the stationary and rotating grooves are equally effective; for each seal the same wetted length is required to seal against the small head of the fluid in the reservoir above the shaft. Although the pressures developed in this model are quite low, significant pressures can be developed in a practical seal.

Figure 7-15 illustrates the sealing capability of a visco seal for two fluids: oil and water. Pressure in psi per inch of axial length is plotted as a function of rotor peripheral speed. The pressure that can be sealed increases with speed and also with fluid viscosity. The oil chosen has a viscosity some 50 times that of water and pressures developed are in that ratio. As an example, a 6-inch-diameter seal with a clearance of 4 mils running at 3600 rpm would produce about 20 psi per inch of seal length when sealing water.

There are a number of factors that must be considered in applying the visco seal, and one of these is the fluid film power loss which is comparable to that of a journal bearing of the same size, speed, and clearance. In dissipating this power loss, advantage can be taken of the seal pumping action, which induces a constant exchange of liquid from the film to the cavity and back. The film temperature tends to be uniformly low if sufficient cooling flow is provided through the cavity.

Since the visco seal operates with liquid, in case of a seal malfunction, the possibility of liquid entering the turbine areas must be eliminated through the use of some type of mechanical trap or slinger.

Spiral Seal

Another seal that works on the same pumping principle, is the spiral seal which is shown in figure 7-16. This is very similar in construction to the conventional face seal shown earlier except that the nonrotating nose has spiral grooves cut in its face. The motion of the seat across the grooves produces a pumping effort to seal the liquid. The seal is designed so that the spiral groove pumping effort develops a pressure greater than the sealed pressure, thus assuring sealing face separation and the formation of a liquid-to-gas interface within the seal. Orifices permit recirculation of the sealed fluid to remove heat.

The spiral seal has two advantages over the helical-groove seal. It can be made smaller in size because smaller gap heights can be used. The gap magnitude is not dictated by shaft runout and vibration as for the helical groove seal. With smaller gaps, proportionally larger pumping pressures can be produced, and it will also seal at zero shaft speed because, when rotation stops, the seal closes.

CONCLUDING REMARKS

Although further development would be required to establish reliability and integration into present electric power generating systems, some bearing and sealing concepts discussed may have application in the system.

As pointed out, use of rolling bearings would significantly reduce the bearing power loss. The seal concepts could be used to achieve a marked reduction in leakage at the expense of some mechanical complexity. Incorporation of the seal types discussed would also make it possible to significantly reduce the axial length of machines. This would help to alleviate design problems associated with critical speeds.

The seal concepts discussed were derived from studies on seals in the 2- to 8-inch size range. However, the basic principles should be applicable to seal sizes found in large gas and steam turbine applications.

It has been shown that reliable bearing operation for periods of 2 years is being achieved at operating conditions at least as severe as those in projected industrial generating systems. Further improvements in bearing life and reliability can be expected over the next decade.

TABLE 7-I. - RELATIVE SEVERITY OF BEARING OPERATION

Application	Severity parameter	
	DN	DN^2
Industrial gas turbine	1.32×10^6	12.4×10^9
1000-mW open-cycle gas-turbine system	1.6×10^6	4.8×10^9

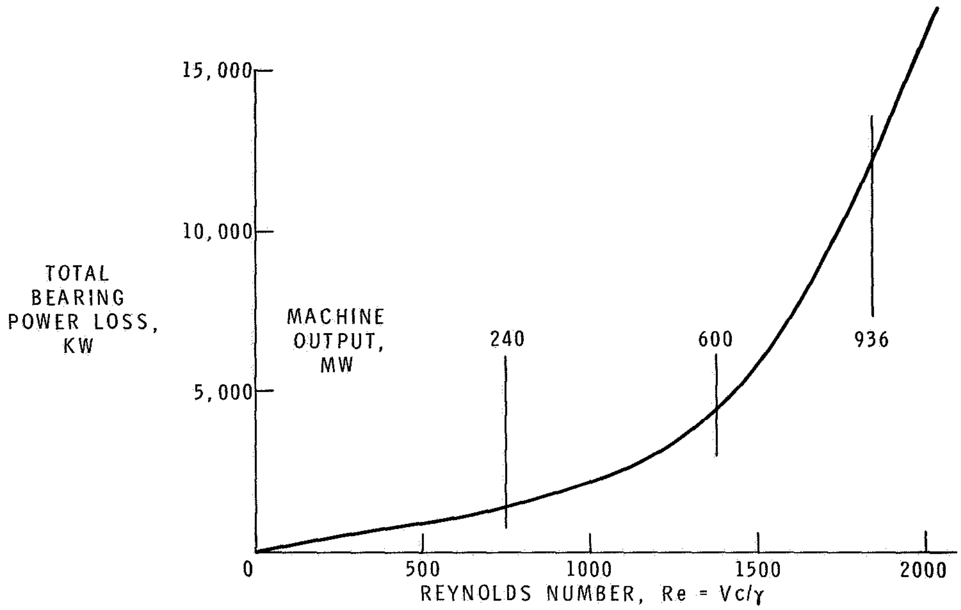

Figure 7-1. - Power loss in sliding bearings.

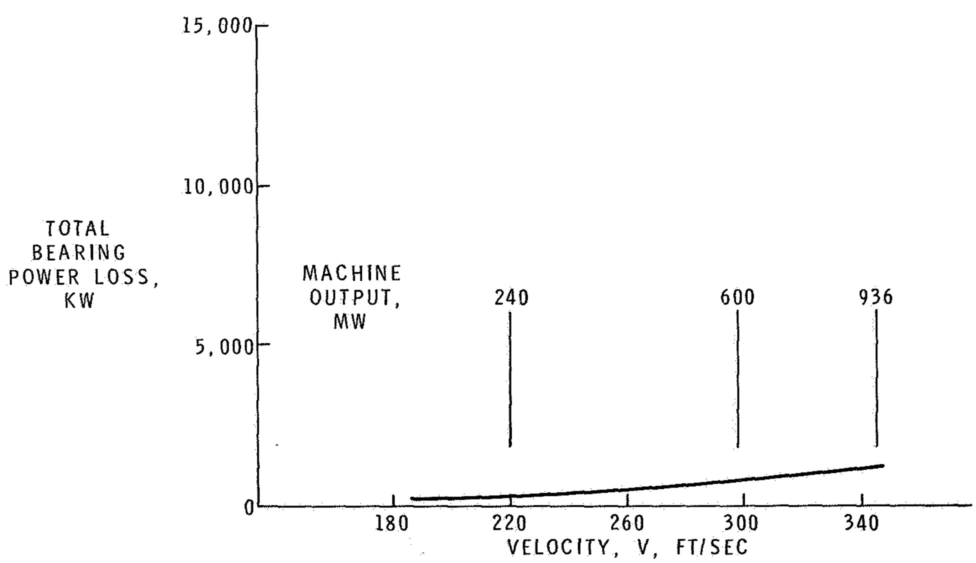

Figure 7-2. - Power loss in rolling bearings.

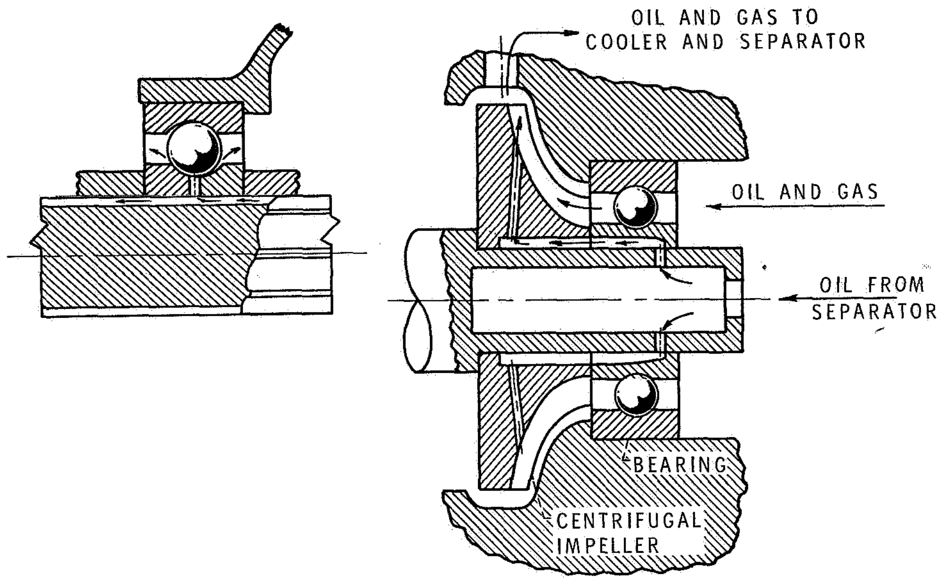

Figure 7-3. - Lubrication system concept.

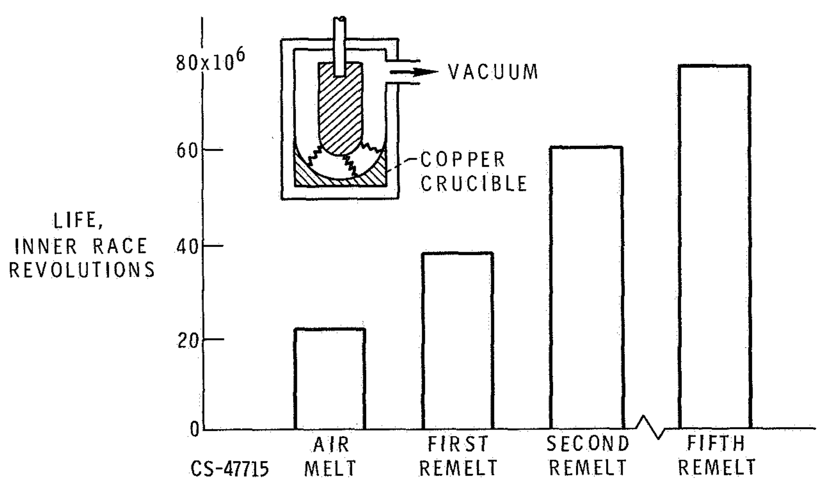

Figure 7-4. - Effect of successive consumable remelts on fatigue.

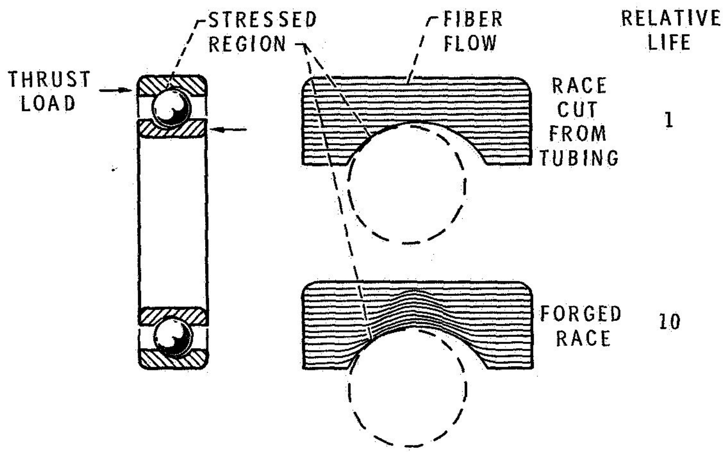

Figure 7-5. - Fiber flow in bearing races.

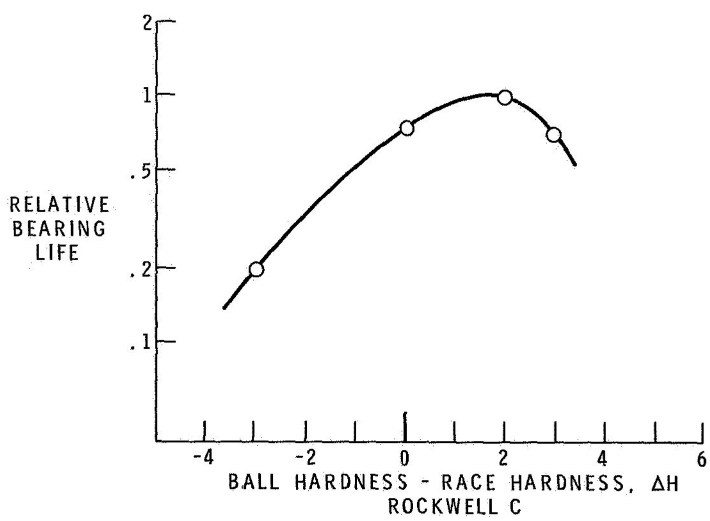

Figure 7-6. - Bearing life as function of component hardness.

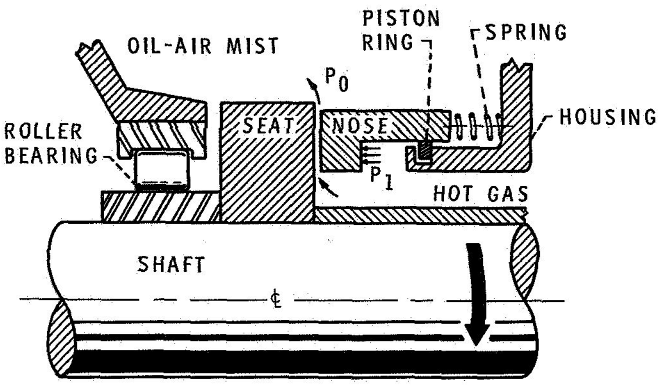

Figure 7-7. - Face seal.

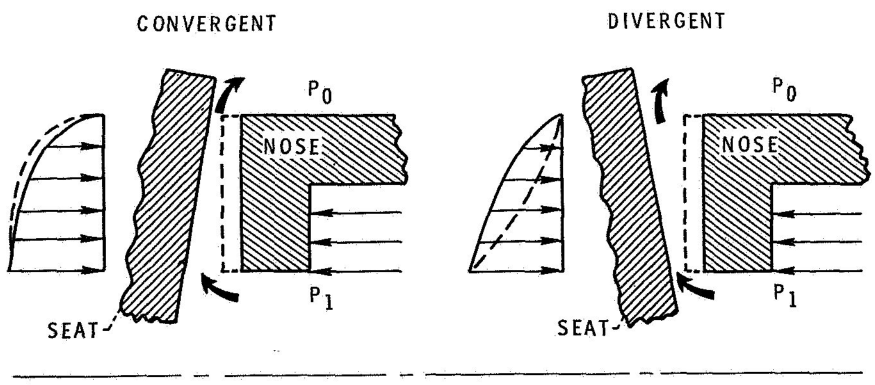

Figure 7-8. - Sealing face deformation.

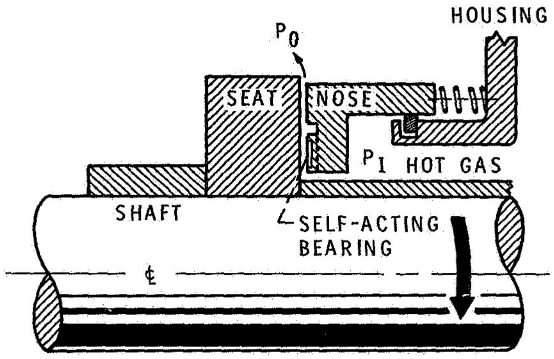

Figure 7-9. - Face seal with self-acting bearing.

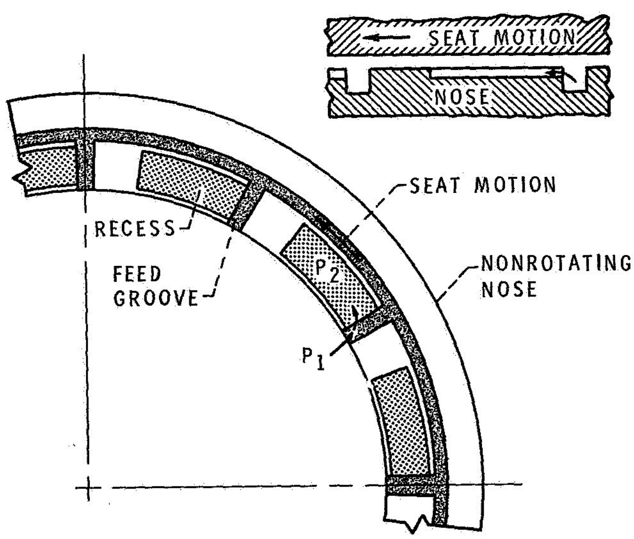

Figure 7-10. - Nose face with self-acting bearing.

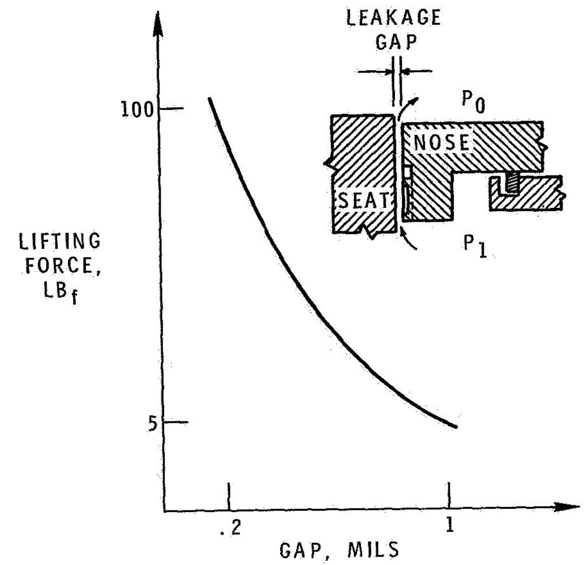

Figure 7-11. - Gas bearing for seal support.

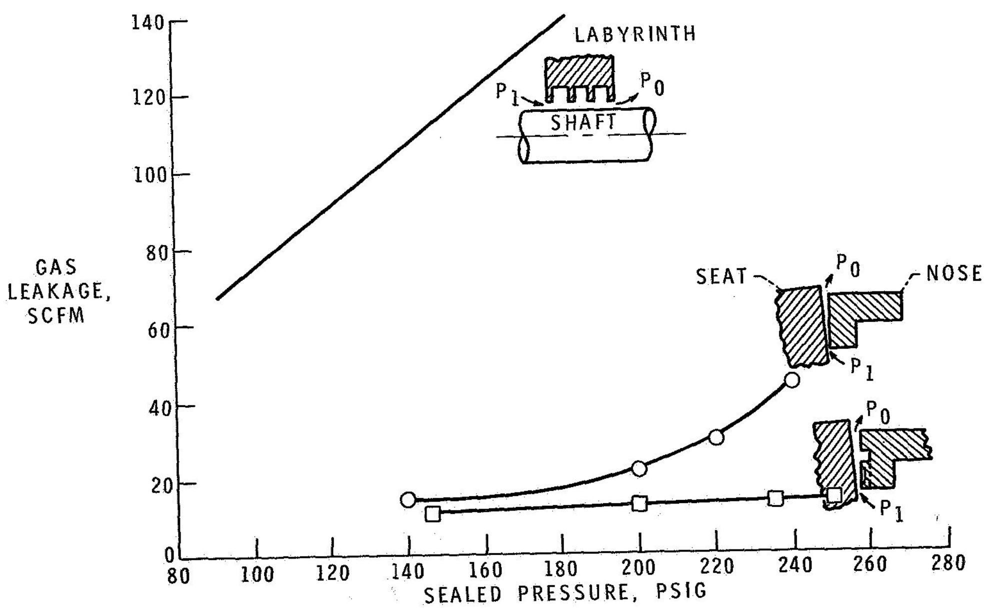

Figure 7-12. - Seal leakage.

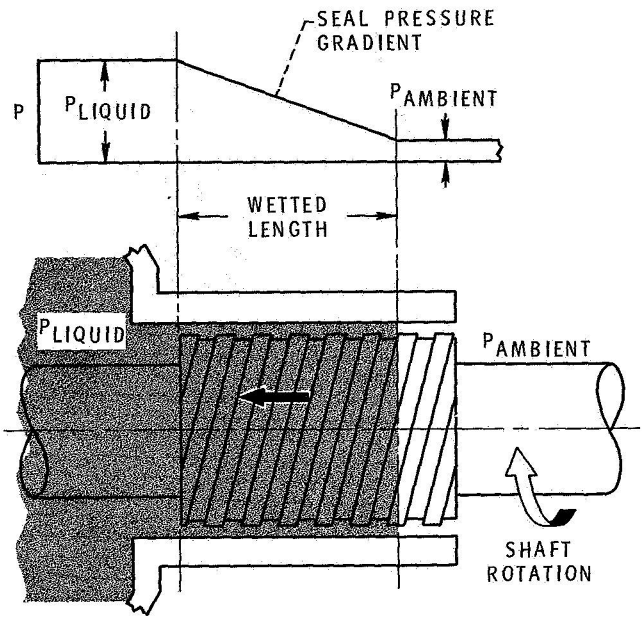

Figure 7-13. - Helical groove visco seal.

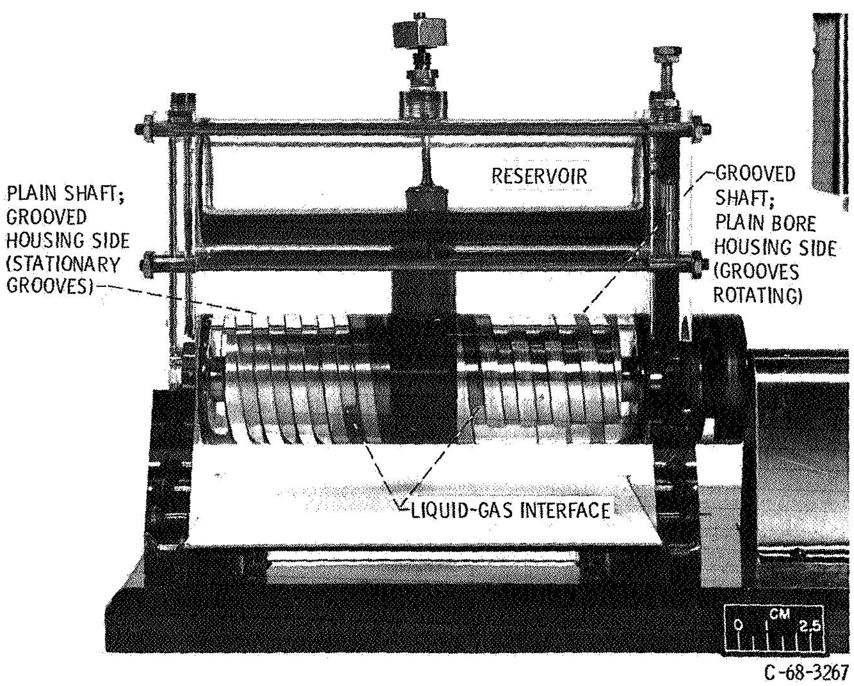

Figure 7-14. - Visco seal model.

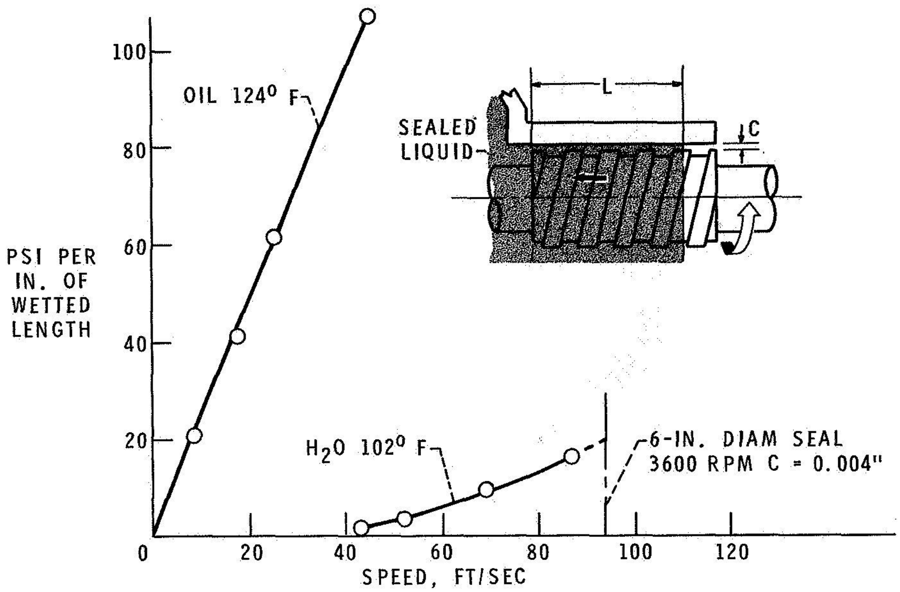

Figure 7-15. - Sealing pressures.

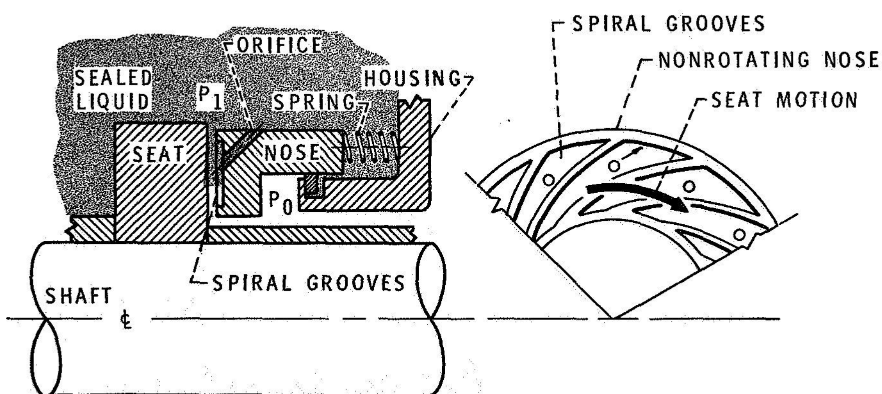

Figure 7-16. - Spiral seal.