DESIGN AND EVALUATION OF AN AUTOMATIC-DISENGAGING DYNAMIC FLUID SEAL

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DESIGN AND EVALUATION OF AN AUTOMATIC-DISENGAGING

DYNAMIC FLUID SEAL

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FOREWORD

This investigation represents a portion of the duties of the author while assigned to the Fluid Seals Project as a graduate assistant. The Fluid Seals Project was conducted for the Service Engineering Division of the Oklahoma City Air Materiel Command under contract number AF 34(601)-5470, Project Authorization Nr 6.

CHAPTER I

INTRODUCTION

With the high speed requirements, expense of operation, and importance of mission of modern aircraft, failure of a relatively small part can be disastrous. One of the small parts that poses a large problem in today's aircraft is the fluid seal. These seals are used in various aircraft accessory units such as fuel pumps, hydraulic pumps, motors, and fluid transmissions. A small amount of leakage might not impair the function of the aircraft accessory unit; however, the presence of any leakage is normally classed as a failure for the particular accessory unit.

There are many types and varieties of fluid seals on the market. These can be classified as two general types with a number of different seals associated with each one. The two types are usually referred to as Static Seals and Dynamic Seals. Static Seals are largely gaskets or O-rings and will not be considered further.

Dynamic fluid seals are used to prevent migration of fluids across a joint or opening in an assembly where the mating parts have a relative motion. Normally, this means sealing the opening where a rotating shaft passes through the housing wall. The commonly used types of dynamic fluid seals, with their method of sealing, will be briefly discussed. This will provide background information for the dynamic seal design proposed in this report.

Radial Positive-Contact Seal

This type of seal is also known as a dynamic rubbing seal (1)[1]. It is a device which applies a radial contact pressure to a cylindrical surface to prevent passage of fluid or to exclude foreign matter.

The seal is composed of a stamped metal shell with a sealing lip bonded or fastened to the shell, as shown in Figure 1. The sealing lip is molded from a resilient material such as butyl rubber, neoprene rubber, or leather. The assembly is designed in such a manner that it can be pressed into the housing as a unit.

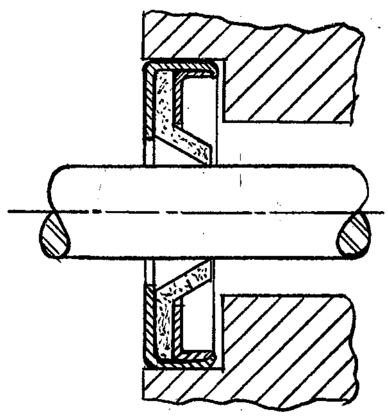

Figure 1. Radial Positive-Contact Seal

The sealing lip is held firmly against the shaft either by the resiliency of the lip material alone, or in conjunction with a spring. Garter springs and finger springs are the most common types used as spring-tension elements for the seal.

Leakage of fluid is prevented as long as the imposed load on the seal lip is greater than the fluid pressure tending to lift the lip. Sealing is accomplished at the knife edge of the seal where is it in

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[1]( )Refers to Selected Bibliography

contact with the shaft. As the pressure increases, this knife edge flattens out into a broad band. Obviously, this is not desirable because the increased friction force causes an increase in seal torque and wear of the seal material.

There are a number of materials available for the resilient part of the seal, depending on the application. Leather is used when it is anticipated that the seal may have to run "dry" for some time since porous leather tends to retain a portion of the lubricating fluid. Synthetic rubber that is specifically compounded to resist deterioration by the sealed fluid finds wide application, especially in the automotive field.

Labyrinth Seals

The labyrinth seal is probably the most common of the clearance seals (2). The term "clearance" indicates that there is no actual contact between the rotating and stationary sealing surfaces. A labyrinth seal, as shown in Figure 2, usually consists of a series of thin fins or knife edges attached to either the moving or stationary part. Design clearance between the fins and the shaft or between the fins and the housing is dependent on bearing tolerance, amplitude of vibration and expansion of the seal metals.

Sealing is accomplished by providing a large pressure drop in the leakage path. The pressure drop is obtained by forcing the fluid through an orifice created by the small clearance between each fin and the mating part. Therefore, the leakage fluid is exposed to the sum of the pressure drop of all these individual orifices. The amount of leakage will be a function of the number of fins placed in the leakage path, the seal clearance, and the total pressure drop.

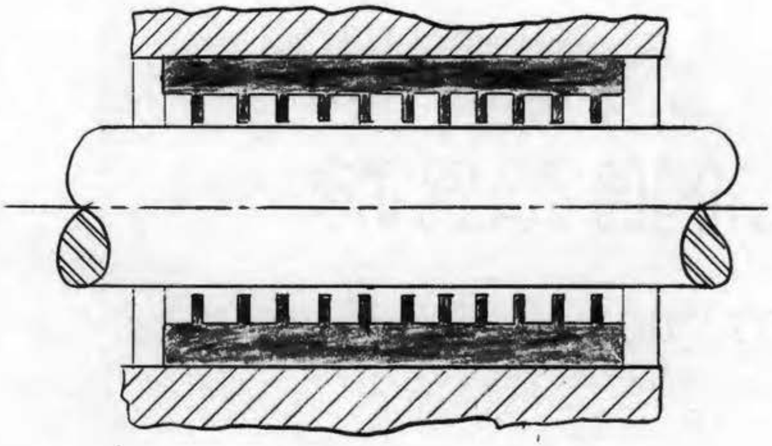


Figure 2. Labyrinth Seal

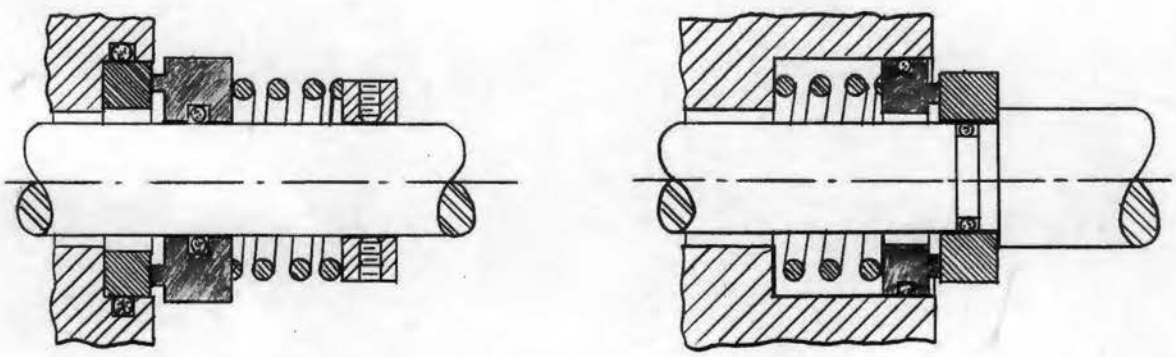
Since there is no rubbing contact between the parts, material used for the sealing parts is not critical. However, bronze is normally used for temperatures up to 600°F, and stainless steel for greater temperatures.

The primary disadvantage to using this type of seal for high-speed applications in the aircraft and missile field is its large size and weight. Consequently, this seal is normally used on large machinery handling hot fluids or gases, such as steam turbines or air compressors.

Axial Mechanical Seals

"End Face" is another term designating an axial mechanical seal (1). As the name implies, an axial mechanical seal is a device that forms a seal between precision finished surfaces having relative motion. The sealing surfaces are perpendicular to the shaft while the contact force is applied parallel to the shaft. This seal does not depend on deformation of some resilient material to form a seal, hence the contact surfaces are usually very hard.

The two general types of axial mechanical seals are designated as rotating and stationary. In the rotating seal, the sealing ring, spring and resilient packing all rotate with the shaft while the mating ring remains stationary. The other type (stationary) indicates that the sealing ring, spring and resilient packing remains stationary while the mating ring rotates with the shaft. The rotating and stationary types are pictured in Figures 3a and 3b, respectively.



(a) Rotating Seal (b) Stationary Seal

Figure 3. Axial Mechanical Seal Types

The axial mechanical seal includes a sealing ring, spring, O-ring or some other type of static seal, housing, and a mating ring with its associated static seal. Also, since the rotating part must be driven by the shaft, some means must be provided to hold this part firmly to the shaft. The rotating seal part could be held on the shaft by a key or some other positive device. In some cases, friction between the part and its static O-ring seal is used to provide the positive drive.

Sealing is ensured by the two precision finished mating surfaces preventing migration of fluid between them. The seal ring and its mating ring are initially held in contact by the action of springs. These may be multiple coil springs, a single coil spring, a wave spring, or the spring-action of a metallic bellows. After hydraulic pressure has built up in the accessory unit, the primary contact force is provided by a differential fluid pressure.

The mating seal faces are the most important parts of the seal mechanism. Carbon graphite against nitrided steel or stellite seems to be the most popular material for these faces. Many other combinations of hard materials would be acceptable depending on the application. A finish on these surfaces, giving a flatness of within three light bands, or 0.000035 inches, is desirable. The smoothness of the surfaces should be such that while there are no scratches of magnitude sufficient to promote excessive leakage of fluid, a very minute amount of fluid should be present on the faces to act as a lubricant.

Axial mechancial seals find wide application in the airplane and missile fields. Because of their ability to withstand extreme variations of temperature in combination with high rotative speeds and high pressures, these seals are used to a great extent on airplane accessory components.

Screw Thread Seal

As the name implies, this method of sealing incorporates a screw thread. It is basically a clearance type of seal with either the shaft or housing threaded. As the shaft turns, the fluid adheres to the moving and non-moving surfaces, and the thread causes axial displacement of the fluid. Figure 4 shows a screw thread seal with a threaded shaft.

Sealing pressure is directly proportional to viscosity of the fluid. Therefore, sealing depends also to a great extent on the temperature of the fluid. If heating became a problem it could be somewhat alleviated by cooling the housing.

Since the screw thread seal depends on rotation for its sealing, it would not work at all under static conditions. Therefore, some method of engaging and disengaging a static seal would be desirable. Such a method is discussed in this report.

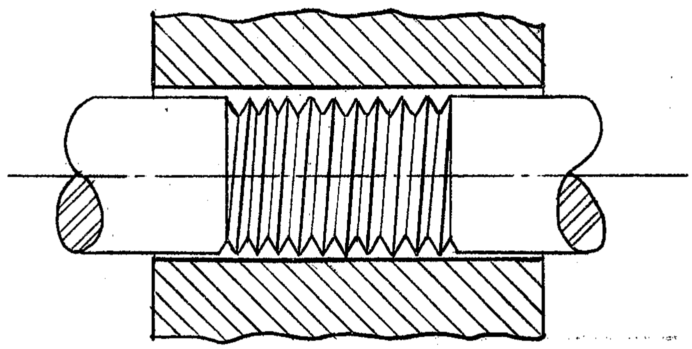


Figure 4. Screw Thread Seal

As a result of tests at Oklahoma State University on axial mechanical seals currently in use on aircraft, several factors contributing to seal failure have come to light. Perhaps the most important factor is that the precision with which these seals are manufactured makes it imperative that careful handling be exercised during the installation. Another factor contributing to seal failure is simply the wearing-out of the sealing surfaces. Contaminants in the sealed fluid, or those induced at the time of installation, greatly increase the wear of these surfaces. Also, the complexity of these high-speed fluid seals increases the likelihood that they will be assembled in an incorrect manner. This of course can be an immediate cause of failure. It might be added that the preciseness and complexity of the seals makes them a costly item to manufacture.

With these problems in mind, it was desired to investigate the possibility of designing a high-speed rotating fluid seal that would be simple in design and possess long-wearing qualities. Tolerance requirements and the number of parts were to be held to a minimum, so that precision machining operations and complexity of assembly could be minimized. A new seal design is presented which strives to meet these objectives.

CHAPTER II

DESIGN PARAMETERS OF AN AUTOMATIC-DISENGAGING DYNAMIC FLUID SEAL

The Automatic-Disengaging Dynamic Seal design incorporates two independent types of sealing devices. A dynamic seal of the clearance type is used in conjunction with a radial positive contact seal. The radial positive contact seal has an automatic disengaging feature to permit releasing of the seal from its mating surface at high rotative speeds. A production illustration of the seal is shown in Figure 5.

The specific requirements which dictated the design of the seal elements were as follows:

1. Seal speed range - 0 to 25,000 rpm.
2. Fluid pressure - 0 to 25 psi
3. Type of fluid - kerosene, to closely approximate turbojet engine fuel.

These requirements were specified in order to closely approximate the actual service requirements of a particular seal under test on the High-Speed Seal Test Facility in operation at Oklahoma State University.

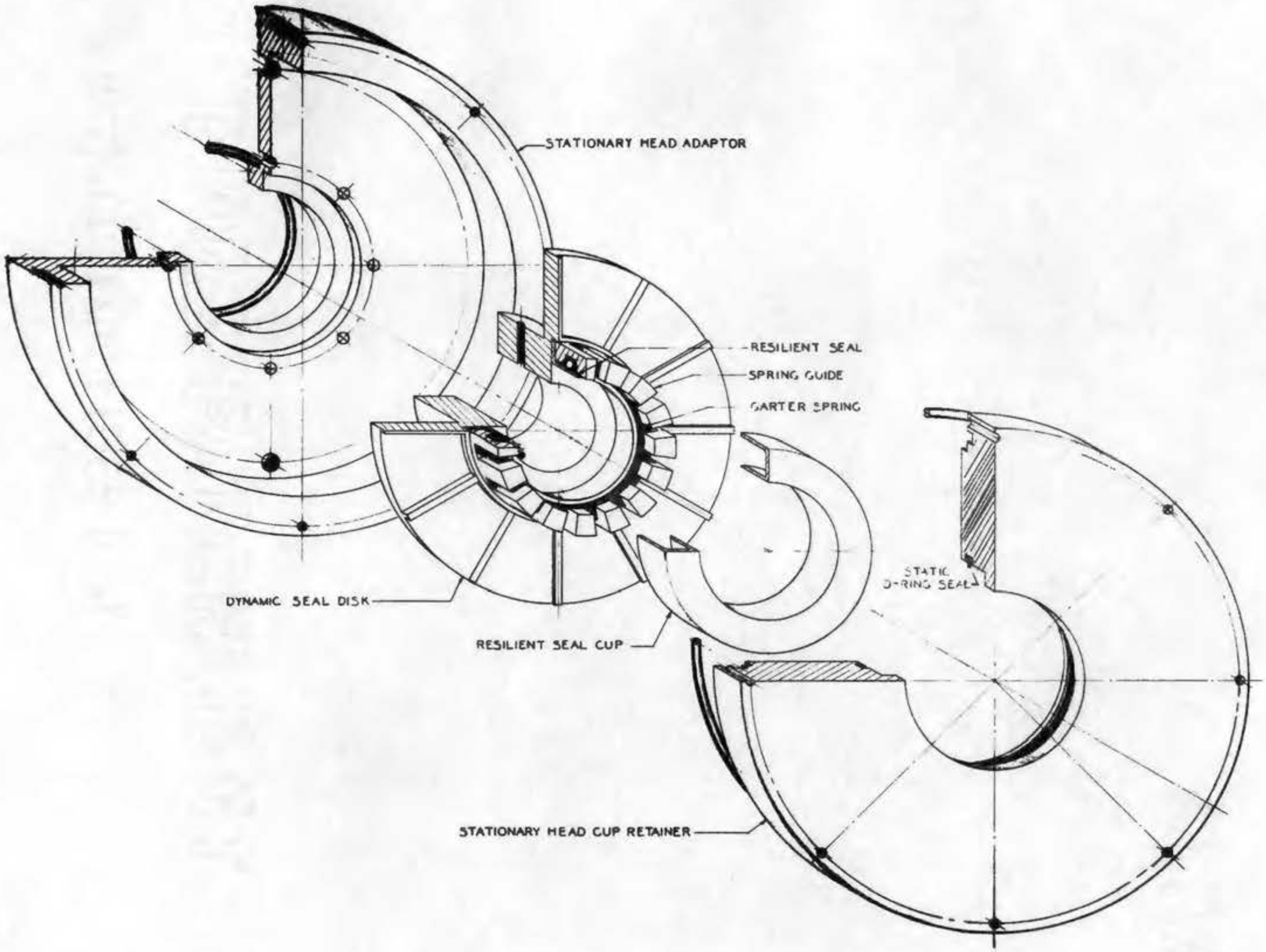


Figure 5. Production Illustration of the Automatic-Disengaging Dynamic Seal.

CHAPTER III

AUTOMATIC-DISENGAGING DYNAMIC FLUID SEAL DESIGN

The Automatic-Disengaging Dynamic sealing concept involves two components that must work together, yet possess separate sealing functions. The physical relationship of these components is best seen by referring to Figure 6. When the shaft is not rotating, fluid leakage is prevented by the resilient lip being pressed against its mating part. As the shaft quickly transcends from a state of non-rotation to high-speed rotation, the disengaging resilient seal releases its contact with the shaft, and the dynamic seal disk assumes the sealing function. Ideally, the sealing functions overlap sufficiently so that transition from one mode of sealing to the other occurs with no fluid leakage.

A detailed description of the design of the automatic-disengaging and dynamic disk sealing components will be presented separately.

Automatic-Disengaging Seal

The name for this seal gives a hint to its unique feature. Actually, it can be classified as a special type of radial positive contact seal. There are two pronounced differences. One is that the automatic-disengaging seal rotates with the shaft, while the normal radial seal does not rotate. The other is that this new seal disengages from contact with the sealing surface at high speeds, while the normal radial seal remains in contact with the shaft at all speeds. This releasing

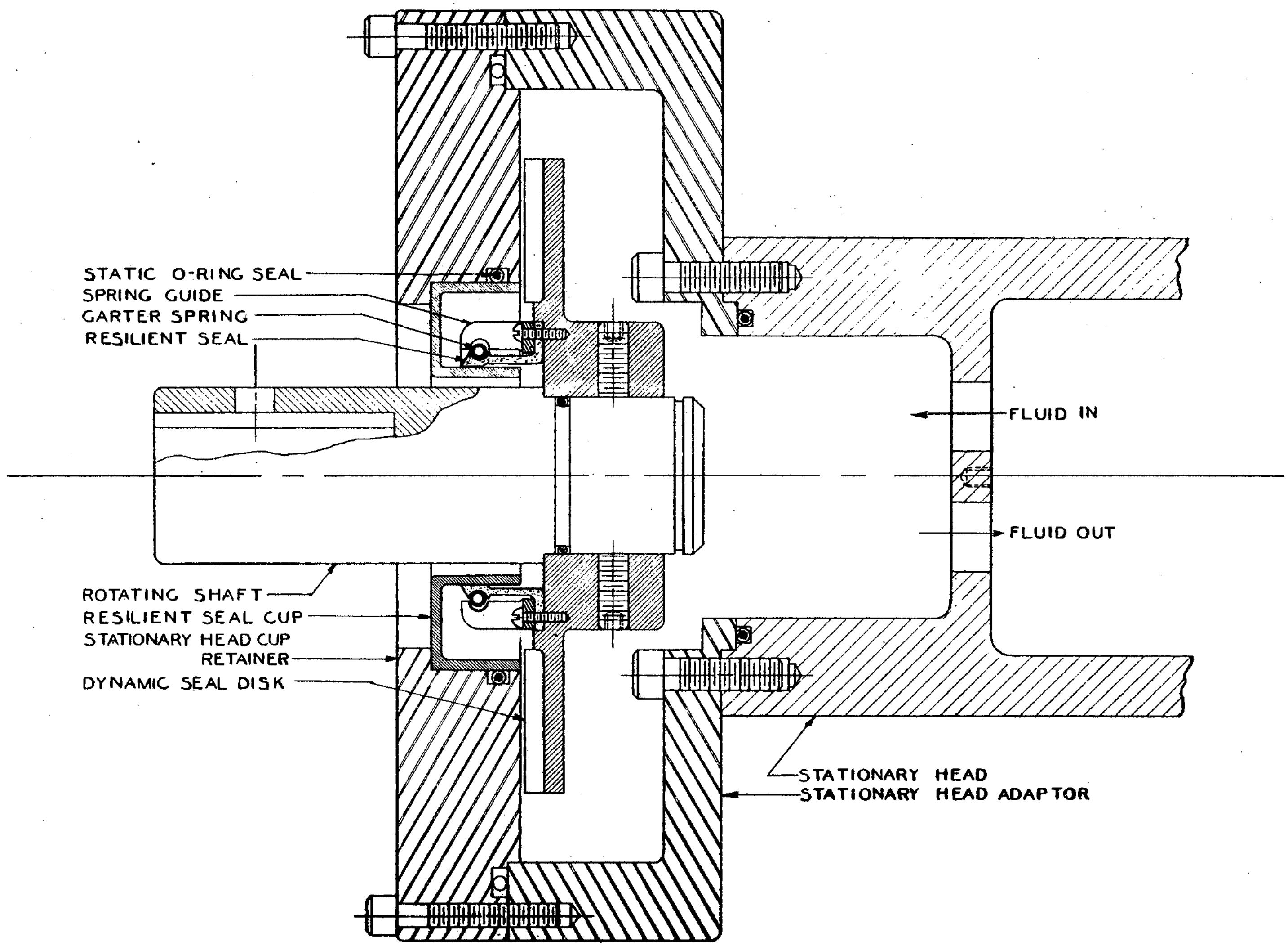


Figure 6. Cross-Sectional View of Automatic-Disengaging Dynamic Seal.

action is intended to prevent wear on the seal lip at high speeds, and thereby greatly prolong its life and reliability.

Sealing pressure on the lip in the ordinary seal is obtained by deformation of the resilient seal material by the shaft and the garter spring (if one is used at all). In the proposed new seal, there is no deformation of the sealing lip by the mating surface. The initial sealing pressure depends on a garter spring and the fluid pressure acting on the seal lip. With an increase in rotational speed, centrifugal force will displace the lead-filled garter spring outward releasing pressure on the seal lip. Simultaneously, the dynamic seal disk evacuates the cavity of fluid relieving hydrostatic pressure on the sealing lip. This then is the disengaging mechanism mentioned above. Observation of the seal in operation shows that the effect of centrifugal force on the resilient seal itself causes a slight increase in diameter which tends to aid the disengaging function of the seal.

RESILIENT MATERIAL SELECTION

Requirements dictating the selection of the resilient seal material included flexibility, porosity, compatibility with the fluid used (kerosene), and molding characteristics. Several materials were considered. Leather has been universally accepted as a good sealing medium, but its flexibility and molding characteristics prohibited its being used for this seal. Butyl rubber has excellent flexibility and molds quite easily, but kerosene has a rapid and extreme detrimental effect on butyl rubber.

The material finally selected was a neoprene compound.[2] Since

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[2]TR-121, 40-durometer hardness, Tulsa Rubber Goods Company, Tulsa, Oklahoma.

neoprene was developed to be used in conjunction with modern fuels, kerosene does not affect it. This material has the desired flexibility to permit proper operation of the seal disengaging mechanism and to allow for minor run-out of the shaft. When correctly molded, the neoprene is dense enough to prohibit fluid from leaking through its pores.

No lip seal was commercially available that had the desired dimensional characteristics. Therefore, a two-part steel mold was designed and fabricated to make the required lip seals. This mold is illustrated in Figure 7. This first seal was formed with a 150-ton steam heated press. The seal was cured at 350°F for 15 minutes. Subsequent modified versions of the seal were molded in an 8-ton laboratory press. These seals were cured in an electric oven at 350°F for 60 minutes.

Two methods were considered for attaching this molded seal to the dynamic disk. One method would have been to vulcanize the neoprene seal directly to the disk. However, correct axial alignment would have posed a problem. Therefore, the resilient seal was bolted to the dynamic disk, using the Spring Guide for a back-up ring and as an aid in alignment. This method allows the seal to be readily removed for inspection or replacement.

GARTER SPRING SELECTION

Selection of the garter spring is dictated by the forces acting on the resilient seal lip. A study of Figure 8 will indicate that these forces are the initial load imposed by the garter spring, the initial load imposed by the fluid pressure, and the centrifugal force on the garter spring and seal lip.

Thus, the seal will be in contact with its mating surface when the combined fluid and initial garter spring loads are greater than the

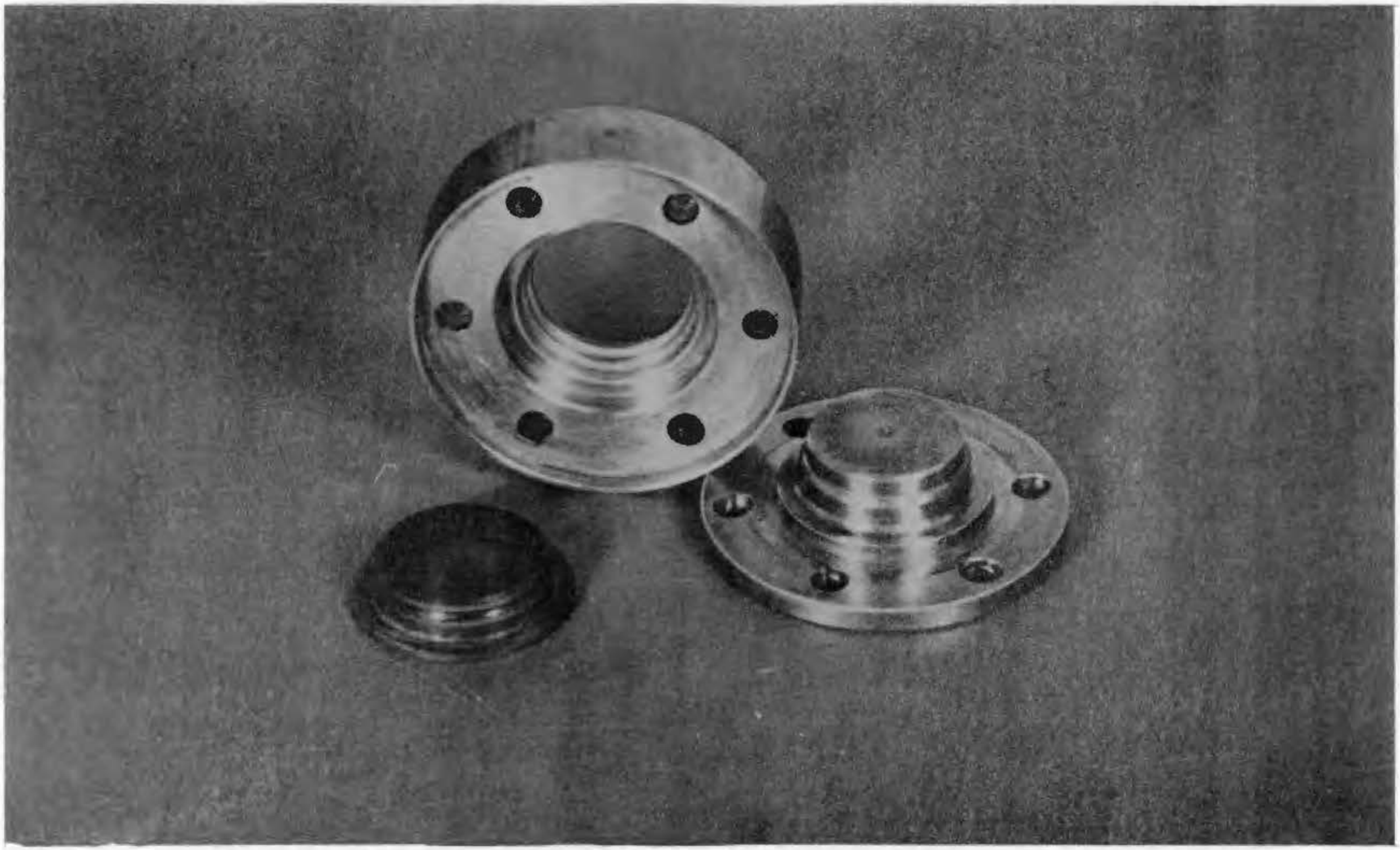
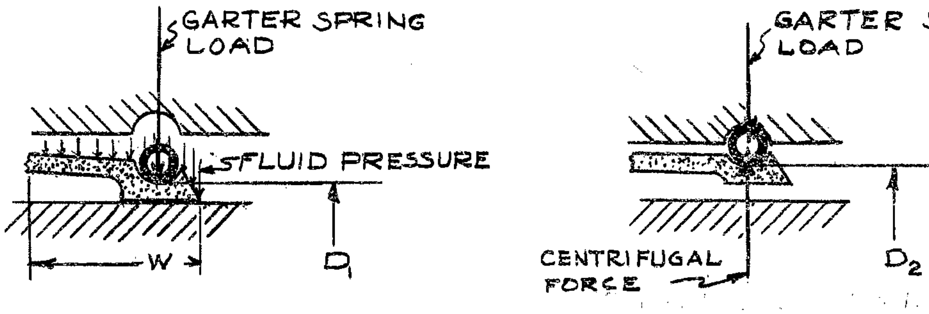


Figure 7. Mold Parts and Resilient Seal



(a) Static

(b) Rotating

Figure 8. Forces Acting on Resilient Seal Lip

centrifugal force. Conversely, the seal will be open when the centrifugal force on the garter spring and seal lip is greater than the load imposed by the extended garter spring.

Three modes of operation were considered:

1. mating surfaces in contact with no rotation,
2. breakaway of sealing surfaces,
3. totally apart.

With no centrifugal force acting, the static load on the circumference per inch of diameter is

$$L_s = P_i + p_f W \pi \tag{3-1}$$

where L_s = static load per inch of diameter, lbs/in.

P_i = compressive load per inch of diameter due to initial spring tension, lbs/in.

p_f = fluid pressure, lbs/in.^{2}

W = width of pressure area, in.

As the seal rotates, centrifugal action first relieves the compressive force due to the garter spring. Now only the fluid pressure keeps sealing pressure on the lip. Assuming that the seal lip material induces a negligible centrifugal force against the fluid pressure,[3] breakaway will occur only when the fluid pressure is relieved by the dynamic seal.

When a completely open condition exists, the dynamic seal has evacuated the automatic-disengaging seal area of all fluid; and centrifugal force has the garter spring resting against the spring guide. A summation of forces thus includes initial spring tension, plus the additional tension incurred in expanding the spring from diameter D_1 to D_2, versus the centrifugal force acting on the spring; or

$$F = P D_2 \tag{3-2}$$

where

F = centrifugal force, lbs.

P = total spring load (See Equation 3-5) lbs/in.

D_2 = fully open inside diameter, in.

The garter spring design evolves from consideration of the forces acting during this last mode of operation, plus the initial load discussed in the first mode of operation.

An equation for determining the initial load on the mating surface imposed by the garter spring can be derived as follows (3): Assume a spring of inside diameter D extended to diameter D_1 when stretched over

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[3]For instance, assuming that the seal lip weighs as much as the heavy garter spring-lead combination, and that it is rotating at the dynamic seal's operating speed of 3500 rpm, the force from Equation 3-7 is

$$F = (2.15)(10)^{-5}\left[\frac{(3500)(2\pi)}{(60)}\right]^2 = 0.355 \text{ lb}$$

Assume P_f = 1 psi, then the force due to the fluid pressure is

$$F_f = (1)(\pi D)(W) = (\pi)(1.5)(0.125) = 0.592 \text{ lb.}$$

Thus, it is easily shown that even though the garter spring is extended, a small fluid pressure will keep the seal lip in contact with its mating surface.

the combined seal lip and cup.

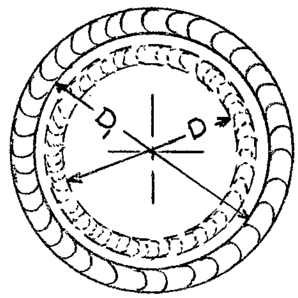


Figure 9. Garter Spring Extension

The total extension will then be, as illustrated in Figure 9

$$\pi(D_1 - D) .$$

If the spring rate is k lb/in., then the tension will be

$$T = \pi(D_1 - D)K . \qquad (3\text{-}3)$$

From a free body of the forces acting on half of the spring, the compressive load per inch of diameter is

$$P_i = \frac{2\pi K(D_1 - D)}{D_1} \qquad (3\text{-}4)$$

where

P_i = compressive load per inch of diameter due to initial spring tension, lbs/in.

K = spring rate, lbs/in.

D_1 = initial stretched diameter, in.

D = unstretched free diameter, in.

Numerical values corresponding to the actual spring are

K = 0.1473 lb/in. (See Appendix A.)

D = 1.390 in.

D_1 = 1.427 in.

Then

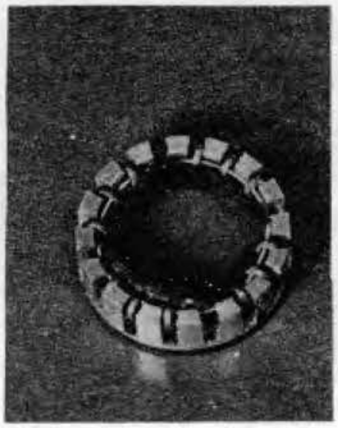
$$P_i = \frac{2\pi(0.1473)[1.427-1.390]}{1.427} = 0.023 \text{ lb/in.}$$

The corresponding initial spring tension would be

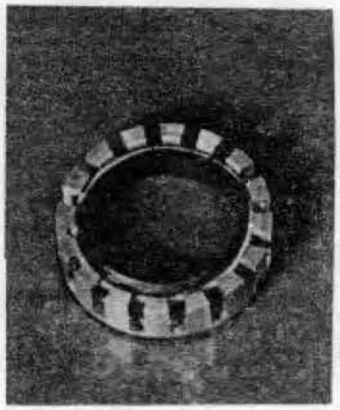
$$T_i = \frac{(0.023)(1.427)}{2} = 0.0164 \text{ lb.}$$

This tension is extremely light, however it was found to be sufficient to pull the sealing lip into contact with the mating cup.

For the mode of operation involving centrifugal force acting on the spring, two conditions were apparent. First, the spring could be designed of sufficient stiffness and mass so that it would not release until the dynamic seal became effective. Second, the spring could be designed so that it would release at a relatively low rpm. Since it was shown previously that even a slight fluid pressure would keep the sealing surfaces together until the dynamic seal was operational, the latter approach was chosen. This would allow relief of the initial garter spring load at a low rpm and thus mean less rubbing pressure on the resilient sealing surface.



(a)



(b)

Figure 10. Methods of Attaching Weights to Spring.

One way to increase the centrifugal force acting on the garter spring was to increase its mass by adding lead weights. Initially, the weights were spaced around the spring like beads. However, reference to Figure 10a on the previous page, will show that this tended to put uneven pressure on the sealing lip and was very difficult to assemble. So a larger cross-sectional diameter spring was procured, and the inside filled with short cylinders of lead. The results, as shown in Figure 10 b, was a weighted-spring combination that induced even pressure on the sealing lip, and was easily assembled.

The minimum speed at which the garter spring will be completely opened can be determined by considering a free body of the forces acting on one-half the spring when stretched to its maximum diameter;

$$2T = PD_2 \tag{3-5}$$

where

T = spring tension, lb.

D_2 = maximum inside diameter of the spring

P in this case includes the initial tension, given by Equation 3-4, plus the additional force gained by stretching the diameter D_1 to diameter D_2, or

$$P = \frac{2\pi K(D_2 - D_1)}{D_2} + \frac{2T_i}{D_2}$$

where T_i is the initial tension.

Since

$$\frac{2T_i}{D_2} = \frac{P_i D_1}{D_2}$$

then

$$P = \frac{2\pi K(D_2 - D_1) + P_i D_1}{D_2} \tag{3-6}$$

Numerical values for the actual spring are

$k = 0.1473$ lb/in.

$D_1 = 1.427$ in.

$D_2 = 1.496$ in.

$P_i = 0.023$ lb/in.

Therefore

$$P = \frac{(2\pi)(0.1473)(1.496-1.427)+(0.023)(1.427)}{(1.496)} = 0.064 \text{ lb/in.}$$

The centrifugal force, F, opposing the spring tension can be derived by reference to Figure 11.

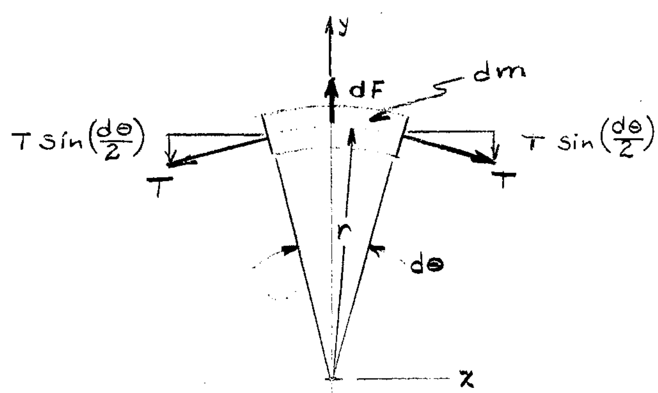


Figure 11. Centrifugal Force on Garter Spring.

$$dF = dm\, r\, \omega^2$$

where

$$dm = \gamma\, r\, d\theta$$

with

γ = mass/unit length of circumference

Therefore,

$$dF = \gamma r^2 \omega^2 d\theta \quad . \tag{3-7}$$

Summation of forces in the y-direction gives

$$2\left[T \sin\left(\frac{d\theta}{2}\right)\right] = dF$$

Assuming small angle approximations, and substituting for dF,

$$2T \frac{d\theta}{2} = \gamma r^2 \omega^2 d\theta$$

or

$$\omega^2 = \frac{T}{\gamma r^2}$$

but

$$2T = PD_2$$

Therefore

$$\omega^2 = \frac{PD_2}{2\gamma r^2} \tag{3-8}$$

The numerical values for the above terms are

$$P = 0.064 \text{ lb/in.}$$

$$D_2 = 1.496 \text{ in.}$$

$$\gamma = \frac{2.73 \times 10^{-5}}{2\pi r} \text{ lb sec}^2/\text{in}^2$$

$$r = (1/2)(1.496) + 1/2(0.111) = 0.803 \text{ in.}$$

Then

$$\omega^2 = \frac{(0.064)(1.496)}{2\left(\frac{2.73 \times 10^{-5}}{2\pi r}\right) r^2}$$

$$\omega = 117 \text{ radians/sec.}$$

$$N = 1112 \text{ rpm.}$$

This computation indicates that the selected garter spring will reach its maximum open diameter and be completely free of the lip seal at approximately 1100 rpm.

Any binding of the garter spring by catching on the guide, or the lead weights restricting coil movement, or any other binding action, would necessitate a higher speed in order to open completely. Therefore, to restrict coil movement as little as possible at the joint, the ends of the spring were joined by a method suggested by the Associated Spring Corporation (3). This involved spreading the coils out at each end slightly, forming a cone of one end, then twisting the ends together. The Associated Spring Corporation claims that this joint is sufficiently strong to withstand a strain equivalent to the elastic limit of the spring wire. Although proper coiling machinery and heat treatment were not available, the spring joint that was made by this method was adequately strong.

Garter Spring Guide

If the garter spring were allowed to expand freely with no means provided to arrest its movement, the spring would be permanently deformed. Also, there would be no assurance that the spring would return to rest in its proper groove as the sealing mechanism slowed down. Therefore, a guide is provided to contain the expanded spring and also insure its being guided back to its proper seat in the seal lip.

There is yet another aspect of this guide that bears mention. It can be easily visualized that as the sealing mechanism gains speed, the heavier spring will quickly expand beyond the sealing lip and hence be free of its driving mechanism. So, it is necessary that the upper limit of the guide be such that it will "catch" the spring as it is expanding and thus provide the driving force.

As secondary function of the spring guide concerns its use as a retainer ring for the resilient lip seal. As mentioned before, instead of vulcanizing the neoprene to the dynamic disk, it was bolted on through the spring guide. Figure 6 illustrates the physical relationship of the spring guide to the other components of the seal, while Figure 10 shows the actual part with the spring and resilient seal in place.

Aluminum was chosen as the guide material for two reasons. First, aluminum is very easily machined, and second it is a light weight material. The need for a light weight material that would not induce heavy unbalance forces is apparent when the 25,000 rpm speed requirement is considered. This critical balancing will be discussed further in a subsequent chapter.

Dynamic Seal Disk

After the resilient seal releases, there is nothing to prevent fluid leakage except the dynamic seal. Of primary consideration in the design of the dynamic seal was the fact that it should be a non-rubbing, clearance type seal.

There were two sealing principles brought out in the discussion of clearance seals in the introduction. The principle mentioned in connection with the labyrinth seal was to provide a small clearance and long path so as to create a high pressure drop in the leakage path. The screw thread seal worked on the fluid to create a flow in opposition to the leakage fluid.

It is this latter principle that is the basis for the design of the seal presented here. The sealing mechanism is an impeller disk similar to that of a centrifugal pump. As long as the disk is rotating at the proper speed, the centrifugal action on the leakage fluid prevents its escape across the sealed area.

Of the numerous vane configurations possible, two were considered, with the governing factor being simplicity of manufacture. A straight radial vane impeller, as shown in Figure 12, was one choice. The other was a spiral-groove impeller, also pictured in Figure 12.

STRAIGHT RADIAL VANE IMPELLER

The straight vane impeller has 12 vanes, each 1/8 inch wide by 1/16 inch deep by 1 1/8 inches long. The outside diameter of the disk is 4 1/2 inches, and it is bored to fit a 63/64 inch shaft.

Normally, a centrifugal pump impeller imparts a velocity to the fluid which could be expressed in terms of feet of head or pressure drop in the fluid. However, this depends on continuous fluid flow across

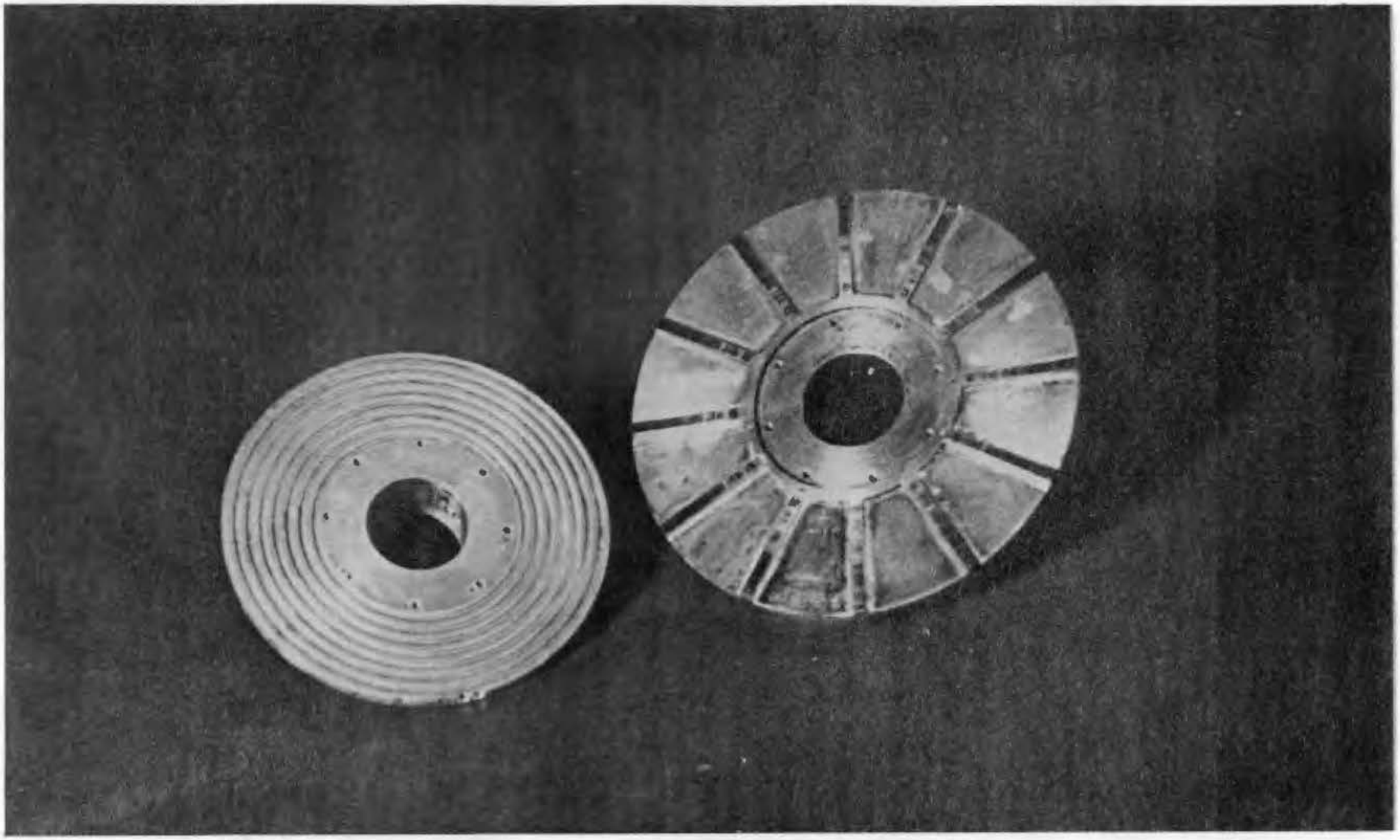


Figure 12. Dynamic Seal Disks

the pump vanes from the entrance to the exit. It is obvious that this phenomenon cannot exist in the sealing area of the dynamic fluid seal because there is no fluid introduction into this area to sustain flow. Two conditions can prevail then, assuming no leakage. Either the impeller disk pumps air into the system, or the fluid flow ceases across the impeller face and a certain amount of fluid rotates with the disk.

Since, for proper operation of the seal there can be neither air forced into the system or fluid leakage out of the system, it was assumed that the latter condition prevails. Reference to Figure 13 serves to illustrate this. The shaded area indicates a portion of fluid rotating with the disk in the small clearance space between vanes , and between the disk and the housing. Centrifugal action on the moving fluid creates a pressure acting against the system fluid. When equilibrium occurs at the proper speed and system pressure there is an interface between the fluid and air areas where no mixing occurs; and centrifugal force on the rotating fluid is sufficient to prevent leakage flow.

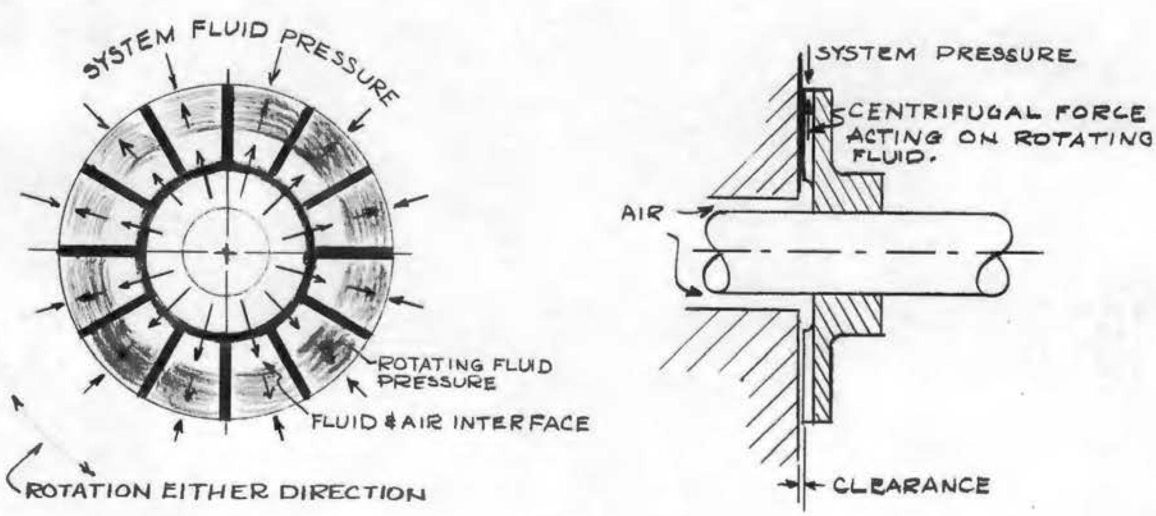


Figure 13. Forces Acting on Straight Vane Disk.

Ideally, it should be simple to formulate a theoretical analysis of this dynamic seal which would allow a prediction of rpm versus system pressure for a given disk size. However, the variables present are such that this problem becomes beyond the scope of this writing. Such variables include: inability to accurately measure clearance; prediction of fluid slip perpendicular to vanes, which would influence the fluid rpm and hence its centrifugal force; location of interface, which would influence mass of rotating fluid, and again, its centrifugal force.

SPIRAL-GROOVE DISK IMPELLER

As will be brought out later, one of the major problems connected with the straight vane impeller is that as the speed of rotation is increased, torque and fluid turbulence increases also. Therefore, the second impeller configuration was chosen to help alleviate this high-speed problem.

A smooth aluminum disk 3 3/4 inches in diameter was bored to fit the same 63/64 inch shaft as the straight vane disk. The impeller was completed by machining a shallow (0.021 inch) spiral groove in the sealing face. This groove is actually a coarse screw thread of 8 threads per inch. It was desired to leave a maximum amount of smooth disk face to create as little fluid turbulence as possible and yet have enough grooves to allow the disk to handle high pressures. Consequently, the groove spacing (pitch) was as large as could be machined with available lathes.

The action of the disk concerns three distinct methods of sealing. First, because of close clearance between the disk and its mating surface, the leakage fluid must undergo a large pressure drop in escaping. Second, the action of the spiral grooves tends to drive the leakage fluid from the center outward against the direction of leakage flow. Third, any fluid forced to spin with the disk would be thrown outward by centrifugal force, again tending to prevent any leakage flow.

As with the straight-vane impeller disk, a theoretical analysis of the fluid forces produced by this spinning disk was not attempted. Some of the uncontrollable variables present were: clearance between the disk and its mating surface; and percentage of leakage prevented by centrifugal force versus percentage of leakage prevented by the spiral-groove action.

An evaluation of these two dynamic seal disks is presented in a subsequent chapter.

CHAPTER IV

TEST PROGRAM

Simply stated, the testing procedure was to assemble the seal, run it, and determine whether or not it would seal properly. Therefore, a testing procedure was adopted that would indicate the performance of the seal in terms of leakage, contaminant introduction into the system fluid, and excessive heat generation in the system fluid.

Equipment and Instrumentation

It was desired to use the present Test Facility (5) with as little modification as possible. Only three new parts had to be fabricated in order to allow the seal to be tested using this facility. These were the Stationary Head Adaptor, Stationary Head Cup Retainer, and the Resilient Seal Cup. The first two parts were made of aluminum for ease of machining and light weight, while the Resilient Seal Cup was made of steel to provide a hard wearing surface.

The function of each of these parts can be noted by referring to Figure 6. It is seen that the Resilient Seal Cup provides a removable mating surface for the resilient seal lip. The Stationary Head Cup Retainer positions this cup properly with respect to the Stationary Head, and also acts as the mating surface for the dynamic disk. As the name implies, the Stationary Head Adapter allows these other two parts to be adapted to the present Stationary Head and provides a chamber for introducing system fluid into the test area.

System fluid provided three functions during the testing. First, it acted as the leakage fluid. Second, it was used as the sealing medium at high speeds. And third, fluid circulation helped to cool the seal parts.

The system fluid is circulated through the Stationary Head by means of two 3/8 inch plastic tubes. Flow and pressure capabilities were 1.5 gpm at 50 psi. The limiting factor on the pressure was the low bursting strength of the plastic tubing.

Leakage of fluid out of the system was determined by visual observation. A transparent plastic shield covered the seal apparatus to confine leakage fluid spray to a drip pan under the Stationary Head. Therefore, it was quite easy to maintain a constant observation of any leakage which occurred whether it was a slow drip or a gushing torrent.

System fluid contamination was in the form of air introduced into the system by the dynamic seal disk. Air in the fluid was detected by the presence of bubbles observed in the transparent plastic outlet tube leading from the rear of the Stationary Head. Also, since this fluid went directly to the rotometer, bubbles could be readily detected in the rotometer fluid. Any air entrained in the fluid was dissipated in the reservoir. Since clear fluid was going into the test section at all times, it was immediately apparent when the outlet fluid started receiving air.

The system fluid temperature and pressure were plotted by a Honeywell Recorder (4). A thermocouple was inserted into the outlet fluid stream at the point where the outlet tube is attached to the system-piping header, as pictured in Figure 14. A static pressure line led

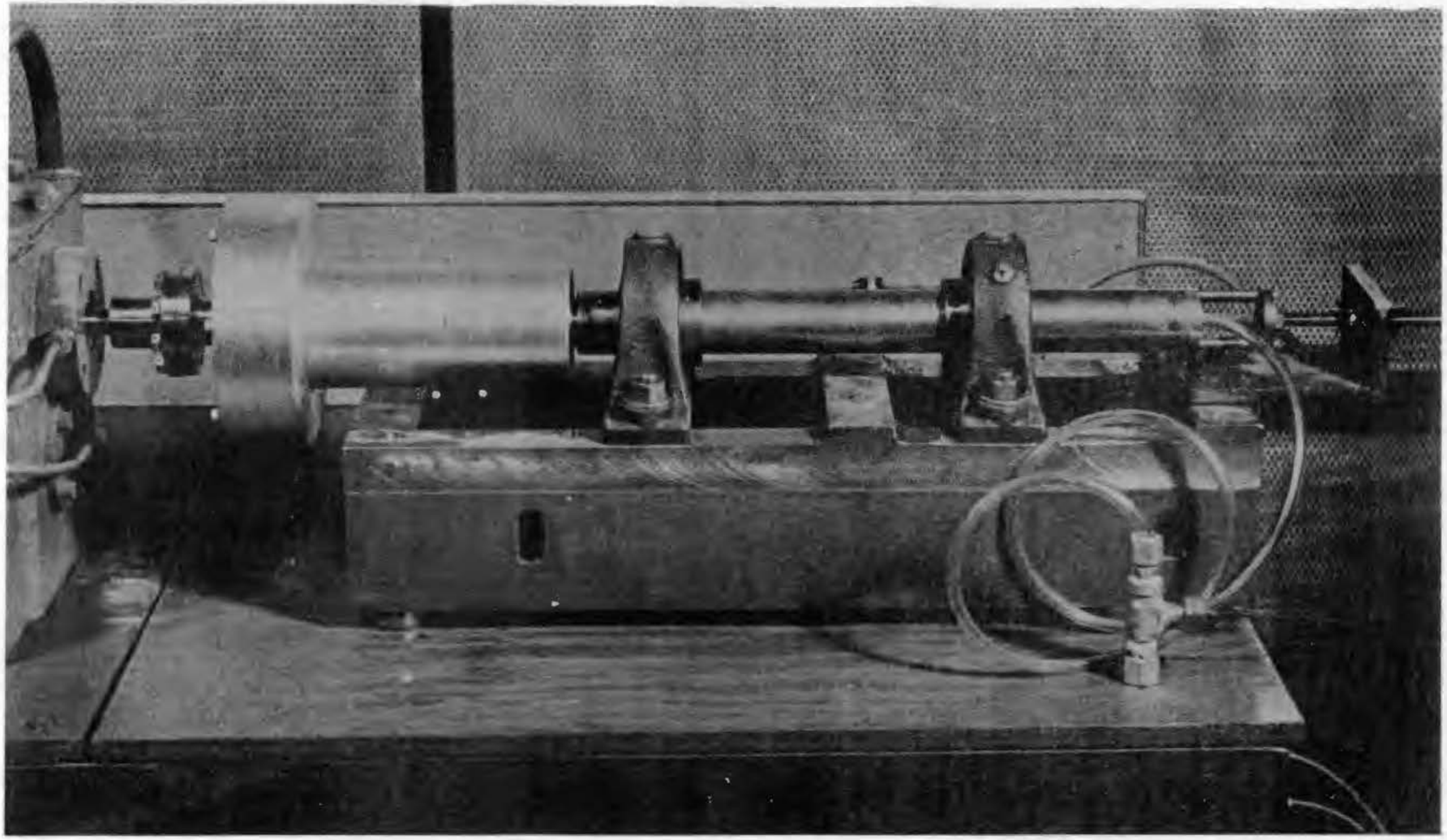


Figure 14. Testing Set-up Showing Fluid Lines and Thermocouple.

from the Test Facility on the other side. The thermocouple and the static pressure line were monitored by the recorder, and temperature and pressure indicated as lines on the recorder chart. Consequently, these two variables were monitored continuously, and changes were immediately reflected by the plotted points on the recorder chart. Since no torque measurement was provided for, the temperature of the fluid helped serve as a guide to the seal performance.

The seal driving force was provided by the facility's Varidrive (5). Speed capabilities were infinitely variable from 2500 rpm to 25,000 rpm. A direct reading electric tachometer was used to indicate the Varidrive output speed.

As mentioned before, clearance was one variable that was not fully controllable. The clearance was obtained and held by a screw-type loading device, shown schematically in Figure 15.

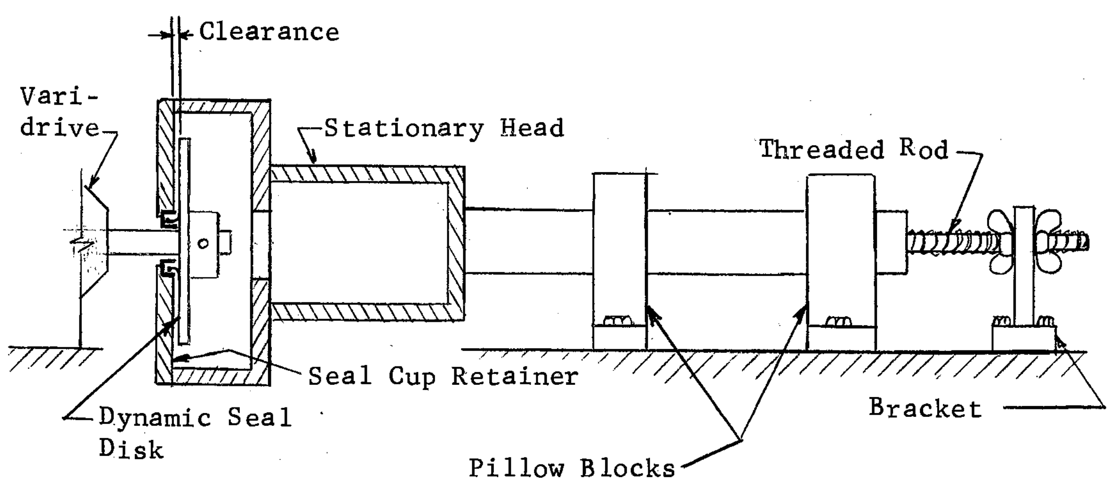


Figure 15. Schematic Diagram of Loading Device

As can be seen from this schematic, there was no means to accurately measure the amount of clearance when the seal was in operating position. To secure the approximate clearance, the disk was moved forward by the loading device until it jammed against the face of the Seal Cup Retainer, then the threaded rod was screwed back a specified length by turning the rear wing nut. The length the threaded rod was moved indicated the amount of clearance. Axial movement of the Varidrive shaft, and deflection in the loading device and seal parts, made it impossible to accurately maintain this clearance between the dynamic disk and its mating surface.

Since high speed operation was anticipated, it was deemed advisable to balance the seal disk with its resilient part, garter spring, and spring guide attached. The balancing machine used was an Annis Dynograph Balancer (6). Unbalance of the assembly was corrected by drilling holes in the heavy side along the periphery of the disk. The parts were then match-marked with paint so that they could be disassembled and reassembled without upsetting the balance of the unit.

Test Procedure

Before testing could begin, a logical sequence of operations had to be performed. The electronic instruments were energized first so that they might be stabilizing while the remainder of the preliminary operations were being accomplished.

Assembly of the seal and its associated components was begun next. The seal was assembled in correct order onto the Rotating Shaft. In the case of the Spiral-Groove Disk, it was necessary to heat the disk in order to be able to fit it on the shaft without destroying the O-ring.

The housing was fastened together, taking care to insure that all O-rings were properly installed.

After all fasteners were tightened, the next step was to adjust the loading screw for the correct clearance. It was found that about 1/16 inch clearance was sufficient to prevent seating and deflection of parts under pressure from causing the disk to rub on its mating member. The Varidrive output shaft was then rotated by hand to insure that all seal parts rotating freely.

The Stationary Head is free to rotate on precision bearings. However, in normal use, it is restrained from rotating more than a few degrees by a thin torque arm attached to the head by a holder. In order to handle the high torque load imposed by this seal, the thin torque arm was removed from its holder, and the holder itself was braced against a solid object. With the holder tightened securely, it adequately restrained the Stationary Head from rotating.

Actual operation of the test facility was exactly as prescribed in Fluid Seals Study, Report No. 2 (5).

A constant check was kept on the temperature as the speed and pressure were increased. When the temperature increased to a point where it was obvious that the flash point of the kerosene (minimum 115°F) would be exceeded, a cooling-down procedure was initiated. This included shutting off the Varidrive and decreasing the fluid pressure to about 10 psi. Circulation of the fluid through the system was continued at this pressure until the temperature dropped sufficiently to allow a normal shut-down procedure.

During this testing procedure, certain data was taken in order to allow an evaluation of the effectiveness of the seal. The data included:

1.) time

2.) temperature (as plotted by the Honeywell Recorder)

3.) pressure (as plotted by the Honeywell Recorder)

4.) speed (rpm)

5.) leakage of fluid (merely the occurance of this was noted)

It was considered that this was sufficient data for the preliminary design. However, it may be necessary to take more exacting data for subsequent work on the seal.

CHAPTER V

EVALUATION AND RECOMMENDATIONS

The primary criteria of a seal's performance is whether or not the leakage can be stopped entirely, or at least held to an acceptable rate. However, there are other factors, stemming from the application requirements of the seal, which play an important role in the overall evaluation of a seal's performance. Assuming that the seal design presented here would be handling inflammable jet engine fuel, and that fluid contamination could not be permitted, two other limiting factors would be high temperature and air-entrainment.

The overall seal performance can best be judged by referring to the graphs in Figures 16 and 17. Note that there are two graphs, one for each of the impeller designs. The plotted line in each represents pressure versus rotative speed. At any point on the line, a balanced condition exists of no leakage and no air-entrainment.

Since these curves represent the optimum sealing conditions, leakage and air-entrainment can be easily predicted from the graphs. If, for any point on the curves, the rpm is held constant and the pressure is decreased, air entrainment will result. Similarly, if the pressure is increased while holding the rpm constant, leakage will follow. Therefore, the area above the curves represents seal failure by leakage, and the area below the curves represents seal failure by air entrainment.

The lower limit of seal operation, 3450 rpm, was dictated by the

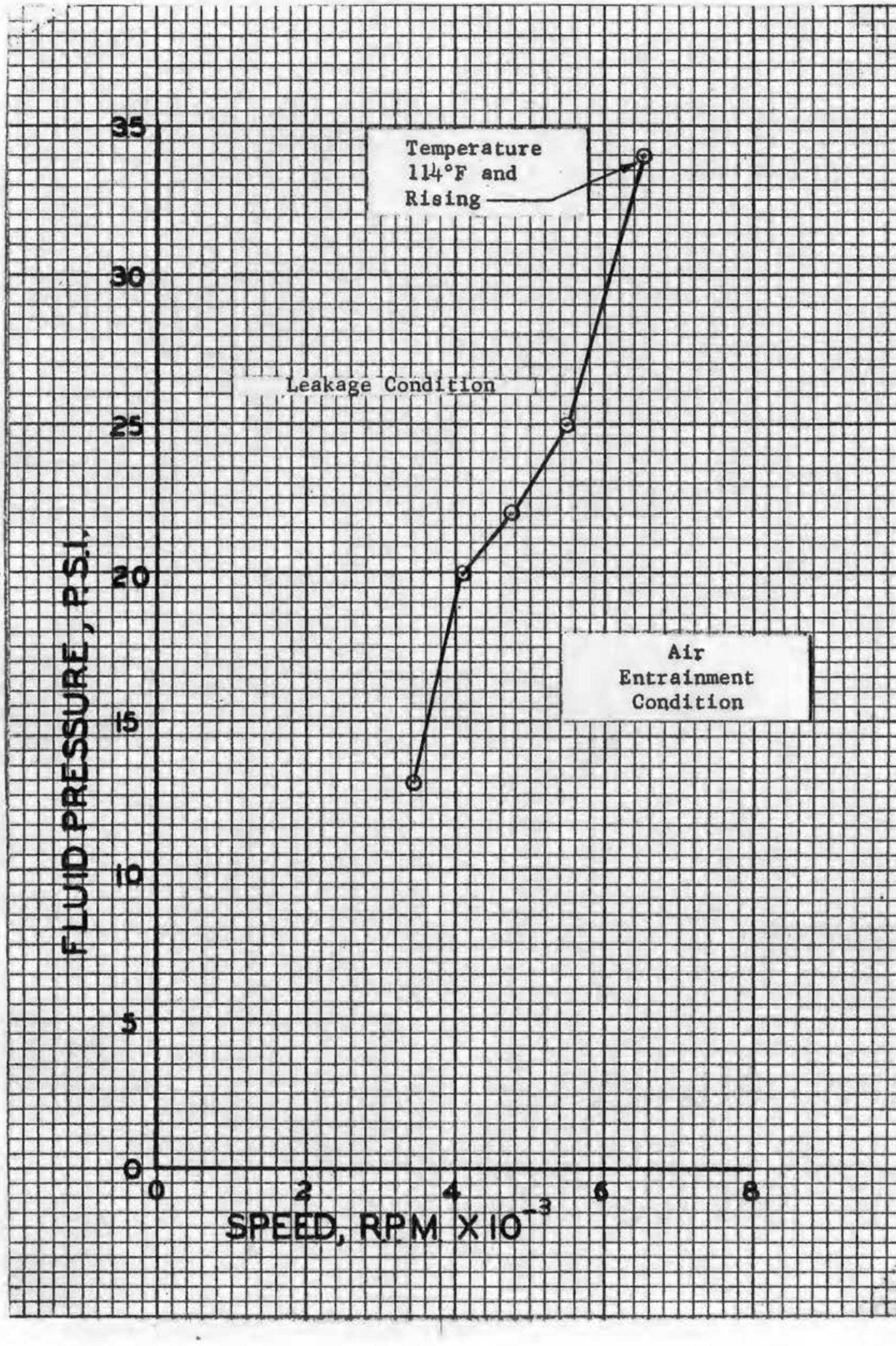


Figure 16. Steel Straight Vane Disk Performance Curve.

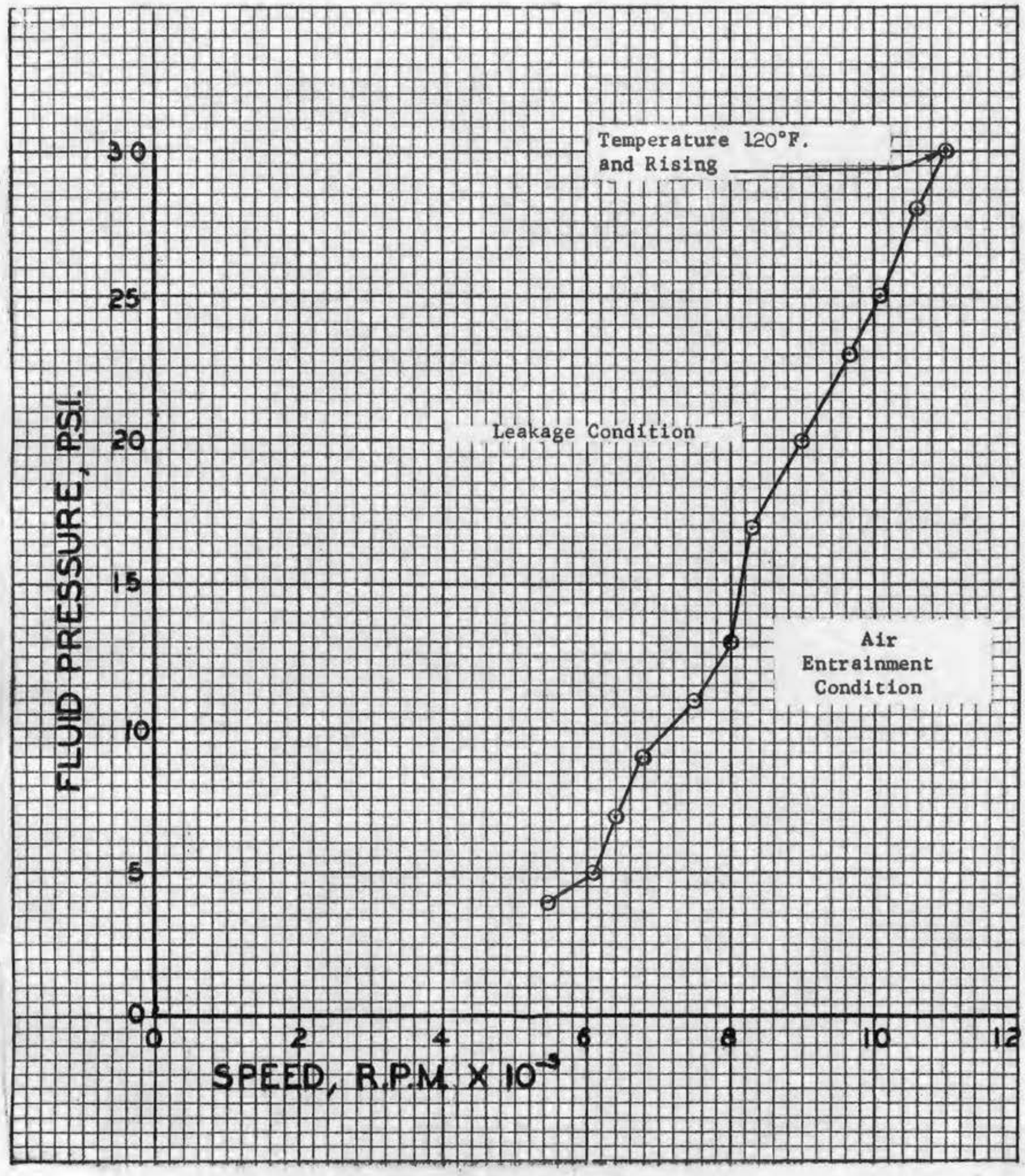


Figure 17. Aluminum Spiral-Groove Disk Performance Curve.

minimum Varidrive speed. The upper limit was set by an increasing temperature condition approaching the flash point of the kerosene. The graphs indicate that in neither case was the 25,000 rpm design speed possible because of excessive heat generation. The upper limit of operation of the Straight Vane Impeller was 6500 rpm, and the upper limit of operation of the Spiral-Groove Impeller was 11,000 rpm.

The 0-25 psi design pressure was obtainable by the use of either dynamic seal disk. However, the Straight Vane Impeller is better able to handle high pressure at low rpm than the Spiral-Groove Impeller, as indicated by 25 psi, versus 4 psi, at 5500 rpm.

The overall performance of the seal having been considered, it is desirable to discuss the evaluation of the two major components of the seal separately.

Automatic-Disengaging Seal

This part of the seal functioned as predicted. That is, no leakage occurred at either static conditions or when the seal was running at operating speed. However, leakage could be induced as a result of gradually stopping the seal while maintaining a high pressure. During the interval of time in which the dynamic seal is not effective, and the resilient seal lip has not yet seated, there is a small amount of leakage.

Rubbing of the seal lip was held to a minimum by the proper action of the disengaging feature. However, if the seal were started rotating while under high static pressure, wear was experienced because of "bellying" of the seal on the cup. As a solution to this problem, a support could be put under the long non-contacting portion of the seal to prevent the fluid from forcing it down into contact with the Stationary Cup. Or, a minor re-design could be made that would use the fluid pressure to force only the seal lip into contact with the mating part. Another improvement that could be made which would reduce the contact area, and hence initial drag, would be to design the lip form as a knife edge.

Dynamic Seal Disks

Even though testing of the two impeller designs proved that they were effective as a sealing device, the excessive fluid turbulence and heat generation were not desirable. It was noted by reference to the performance graphs that these factors severely limited the disks' effectiveness. However, it is believed that this heating problem could be minimized if a higher fluid flow rate could be obtained.

When the disks are sealing properly, there is an unbalance of fluid force on the seal parts. Thus, the full force of the fluid will be acting on one side of the disk and only atmospheric pressure on the other. As shown in Appendix B, the deflection of the seal disk is slight. However, the combined deformation of the loading members and the other seal parts was excessive. Not only can interference between the disk and its mating surface occur, but the excessive forces are transmitted to the seal housing and other members.

Recommendations for Further Study

It is suggested that future study of this seal be directed along the lines of improving the dynamic seal disk design. If a number of designs were evaluated, it is conceivable that a chart could be devised which would allow the selection of particular impeller sizes and configurations for a number of given applications.

Development of this Automatic-Disengaging Dynamic Seal was done entirely by experimental means. This has been generally the case with other types of seals developed by commercial concerns. Therefore, it is recommended that an endeavor be initiated to make a theoretical analysis to determine an optimum design for the dynamic seal disk.

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APPENDIX A

CALCULATION OF GARTER SPRING CONSTANT

The spring constant, k, was calculated by the use of the following equation (7):

$$\omega_n = \sqrt{\frac{K}{m}}$$

where

ω_n = natural frequency, radians/second

K = spring constant, lbs/inch

m = mass, lb. sec²/ft.

By attaching a one pound weight to the spring, displacing it downward, and timing the period of oscillation, it was determined that the natural frequency, f_n, was equal to 36 cycles/30 seconds.

Therefore, it f_n = 36/30 cps, then

$$\omega_n = 2\pi(36/30)$$

$$= 7.54 \text{ radians/sec.}$$

Since $\omega_n = \sqrt{\frac{K}{m}}$, then

$$\omega_n = \sqrt{\frac{K}{1/386}} = 7.54 \quad ,$$

or

$$K = 0.1473 \text{ lb/in.}$$

APPENDIX B

DEFLECTION OF DYNAMIC SEAL DISK

Consider the aluminum disk to be approximated by the configuration shown in Figure 18.

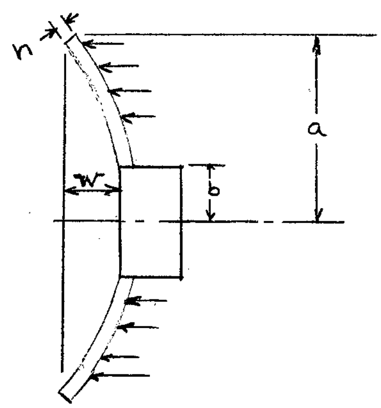

Figure 18. Fluid Load on Dynamic Disk.

The maximum pressure can be computed by the following formula (8);

$$w = \frac{K_1 q a^4}{E h^3}$$

where

h = thickness

b = 1/2 hub diameter, in.

a = 1/2 O.D., in.

q = fluid pressure, psi.

E = modulus of elasticity of aluminum, psi.

Using numerical values as follows,

$h = 1.25$ in.

$a = 1.875$ in.

$b = 0.97$ in.

$E = 10 \times 10^6$ psi.

$h = 0.125$ in.

$K_1 = 0.0837$

the pressure, q, necessary to produce a maximum deflection of 0.06 in., is

$$0.06 = \frac{(0.0837)(q)(1.875)^4}{(10)(10)^6(0.125)^3}$$

$$q = 1135 \text{ psi.}$$

VITA

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