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End Face Seals for Air-Conditioning Compressors

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INTRODUCTION

Proper mechanical seal selection for air conditioning compressors is critical if satisfactory seal performance and reliability is to be obtained. Selection of seal designs, arrangements and materials of construction are indirectly governed by compressor design, type of refrigerant, and compressor operating conditions. The following are the primary variables that directly govern mechanical seal selection:

- shaft speed
- shaft vibration
- seal pressure
- seal lubricant
  - type oil
  - oil temperature
  - oil flow rate to seal
  - type refrigerant
- seal leakage

The above variables are evaluated for all compressor types and designs, whether centrifugal or positive displacement (rotary or reciprocating).

SEAL DESIGN SELECTION

Seal design selection is always made from the following classifications[1]:

- balanced
- unbalanced
- rotating head
- rotating seat
- pusher
- nonpusher

Balance vs. Unbalance

Selection of balanced vs. unbalanced is made on the basis of maximum PV factor and subsequently maximum frictional power given by equations (1) and (2) respectively.

$$PV = [Ps\ (b-k) + Psp]\ V \quad (1)$$

Where:

- P = Seal Face Pressure, psi
- V = Mean Peripheral Speed, ft/min.
- Ps = Seal Pressure, psig
- b = Seal Balance
- k = Seal Face Pressure Drop = .5
- Psp = Seal Face Pressure Due To Spring Load, psi

$$Q = PVua \quad (2)$$

Where:

- Q = Frictional Power Developed At Seal Faces
- u = Coefficient Of Friction
- a = Seal Face Area, in^2

Maximum PV values for given seal face material combinations are established by imperical testing and are based on wear life (seal face wear rates) and maximum heat flows tolerate by the seal before thermal degradation takes place at the primary seal faces[2, 3], or secondary seal members. PV limits based on 16,000 hours wear life for common seals are given in Table I.

Unbalanced and balanced seals normally have balances in the range of 1.10 to 1.30, and .65 to 1.00 respectively. A change in seal balance from 1.30 to .65 has a very large effect on the quantity (b-k) of equation (1) and subsequently the PV value for a given compressor shaft size, speed and seal pressure. One can determine from equation (1) what seal balance is required to stay within wear life PV limits. Some adjustment can be made to the face loading caused by spring load, (Psp).

However, sufficient spring load must be provided to overcome secondary seal resistance thus assuring the seal head is capable of movement to accomodate seal face wear and/or axial shaft movement either/or mechanical or thermal. Vacuum sealing requirements may also dictate minimum spring load requirements where negative seal pressures result in large seal face opening forces[1]. Seal face loading due to spring load will vary from 30 to 100 PSI depending on the application.

With regard to frictional power generation, one can quickly recognize from equation (2) that in addition to control of PV by seal balance and spring load selection, seal lubrication and its effect on coefficient of friction is critical. Even more important is the removal of frictional heat by adequate flow of oil to and from the seal. Typical minimum oil flow rates for compressor designs which provide flooded seal cavities are given in Figure 1. These minimum flow rates are that required to remove all of the friction heat obtained from equation (1) while maintaining a maximum lubricant seal cavity entrance and exit ΔT of $15^{\circ}F$. Thus, seal balance selection must be made so that PV limitations given in both Table I and Figure I are not exceeded for compressor designs which provide flooded seal cavities.

Seals operating in compressors which incorporate oil splash or spray lubrication for the mechanical seal have greatly reduced maximum PV capabilities owing to the less efficient seal frictional heat removal. The maximum PV level of seal operation will range from 8×10^4 to 1.20×10^5, depending on seal face operating temperatures and frictional heat flow to the secondary seal elements within a given seal design. Details of this relationship will be discussed later.

Independent of the lubrication system used, it should be noted that in many cases, the maximum PV level will occur during compressor start-up when seal pressure is maximum. If slugging occurs during start-up, the seals are vulnerable to poor lubrication and high coefficient of friction. Consequently precaution should be taken to avoid slugging in the seal area of the compressor in as much as irreversible damage could take place to the seal faces and/or secondary seals.

<u>Rotating Head v. Rotating Seat</u>

In most cases, stationary seal designs are employed up to 3,500 and 5,000 feet per minute peripheral speed for single and multiple spring assemblies respectively. Above these speeds rotating seats are employed to eliminate centrifical force on springs, secondary seals and other seal head components. There are installations where compressor design may dictate the use of rotating seat independent of peripheral speed. The use of cartridge seals employing rotating seats can be very practical for high volume production. Such an installation is illustrated in Figure 2. Stationary seat designs are illustrated in Figures 4 - 7.

<u>Pusher v. Nonpusher</u>

The selection of nonpusher or pusher seals is based primarily on seal environment considerations, both mechanical and thermal. For definition purposes, pusher type seals are "mechanical seals employing secondary sealing elements (such as O rings, V rings, U-cups, wedges, etc.) that are pushed along the respective sealing surfaces while the primary sealing function at the faces is being performed."[1] Nonpusher seals employ bellows or diaphrams as secondary sealing elements.

Axial movement of the seal components caused by wear at the seal faces and/or shaft vibrations cause sliding movement of secondary seal in the pusher type seals. Movements in nonpusher seals is taken up by the bellows.

Nonpusher seals are preferred for air conditioning compressors where one or more of the following conditions exist:

1 - axial shaft vibrations greater than .002" in amplitude and frequencies greater than 60 Hertz.

2 - deposition of carbonized oil, foreign material, and/or corrosion products on the compressor shaft is sufficient magnitude to restrict the movement of pusher type secondary seals. This restriction prevents the seal from following shaft dynamics or prevents the primary seal faces from advancing to accomodate seal face wear. The end result is complete seal failure.

3 - Excess volumetic swell of secondary seal elastomers which may cause distortion of the primary seal faces in case of pusher type seals (see Figure 3).

4 - Minimum balance is required and standard balance seals cannot be accomodated. Unbalanced-nonpusher seals although classified as unbalanced can be designed to have a balance less than 1.0 at low seal pressure without any compressor geometry changes normally required in changing from unbalanced to balanced seals. It should be noted that seal balance on non-pusher seals will vary with seal pressure. The amount of balance variance is a function of bellows design and material of construction.

Nonpusher seals normally employ elastomer bellows where temperatures and other environmental conditions permit their use. Under extreme temperature conditions, more expensive teflon and metal bellows can be employed. Figure 4 illustrates two typical elastomer bellows, nonpusher type seals employed in air conditioner compressors. Figure 4B depicts a bellows having a one-half convolution flex section;4A depicts a elastomeric bellow seal having a full convolution flex section. The significant differences between one-half and full convolution bellows are as follows: Full convolution bellows are capable of withstanding axial shaft vibration up to .032 in. in apptitude. One-half convolution bellows are capable of withstanding axial shaft vibration up to .015 in. One-half convolution bellows require less axial space. Nonpusher elastomer seals generally have their secondary seal elements in close proximity to the seal faces and the frictional heat being generated the same. The particularly critical in a case of one-half convolution bellows, where the flex section of the bellows is located in close contact with the washer. Thus seals having one-half convolution bellows operating in non-flooded seal cavities will have lower PV limits than seal designs having full convolution bellows or pusher type seals. The secondary seal in pusher type seals can generally be put in a remote location from the seal faces.

Heat transfer capabilities in the primary seal faces is an important design and material selection consideration. Solid end plate designs shown in Figure 4 offer better heat transfer characteristics over alternative two-piece seat and gland plate arrangements as depicted in Figure 5. The gap between the seat and gland plate in a two-piece construction acts as an insulator and restricts frictional heat removal from the seal faces. In addition, the static secondary seal (O ring or gasket) is located closer to the seal face frictional heat source in a two-piece seat and gland plate construction, than in a solid end plate construction. "O" ring or similar gaskets that allow metal to metal contact between end plate and compressor housing as shown in Figure 4A are preferred over flat full face gaskets shown in Figure 4B. The latter restrict heat from flowing to the compressor housing.

Thermal conductivity of the carbon graphite materials typically used in seals range from 2.5 to 12 BUT-ft/hr^{o}F. Under marginal cooling conditions, it is desirable to employ low conductivity washer materials which will minimize heat transfer to dynamic secondary seals located in the seal head which are more sensitive to the effects of heat aging than static secondary seals located in the end plate or seat elements. Low conductivity washer materials are of particular importance in a case of one-half convolution bellows designs as discussed above. Heat shields as shown in Figure 4 are often incorporated between the carbon washer and the elastomeric bellows. The purpose of this heat shield is two-fold: first, to insulate the bellows from heat, and secondly, to minimize exposure of the elastomer to air; in that elastomers age and degrade at a greater rate in the presence of air.

The design given in Figure 4A having a low conductivity washer represents maximum PV capabilities for non-pusher elastomeric bellows seal operating in non-flooded seal chambers.

SEAL ARRANGEMENT SELECTION

There are several seal arrangements[1], but the two most commonly used for air conditioning compressor service are single inside mounted and double seals. Single inside seals shown in Figures 2, 4, 5 and 6 are the most popular, owing to their simplicity and low cost. Double seals as shown in Figure 7 are generally employed on large industrial compressors where cooled lubrication systems are available to supply fluid to the double seal chamber.

Double seals properly lubricated offer good seal reliability and maximum protection of refrigerant leakage to atmosphere or entry of air and moisture into the compressor. Seal fluid pressure should be maintained at 10 to 20 PSIG above compressor pressure. If lubricant contains refrigerant seal cavity should be vented before compressor start-up and lubricant pressure in the seal cavity should be maintained above refrigerant vapor pressure during operation.

Lubricant flow rate requirements for double seals are obtained from Figure 1 by totaling flow rate requirements for inboard and outboard seal as determined by the operating PV of each seal.

MATERIALS OF CONSTRUCTION

Face Materials

The non-galling-low friction characteristics of impregnated-fine grade carbon graphite materials are the main reason that these materials have become standard in mechanical seals supplied on air conditioning compressors. Maintenance of surface flatness and low roughness profiles even though wear is occuring are of utmost importance if the low leakage levels of refrigerant and lubricating oil are to be maintained. Additional prerequisites of carbon graphite washer materials for the application in question are:

- imperviousness
- flatness stability
- capability of showing polished to a low surface finish and free of scratches
- low frictional and wear properties
- blister resistance[4]
- thermal conductivity.

Without exception, imperical test methods have been established to evaluate, select and control carbon graphite materials with respect to these six prerequisites.

Blister resistance is of major importance on marginal lubrication applications where seal face temperatures may become high and cyclic. Occurance of blistering in carbon graphite washers will produce insipient seal failure. Carbon graphite grade vary in blister resistance, consequently, in addition to thermal conductivity grade selection is also made on the basis of blister resistance where marginal lubrication conditions are coupled with either high oil temperatures (greater than 200^{o}F) and/or high PV levels and seal cooling cannot be improved.

Cast iron in the form of end plates has been a popular economical seal face material in combination with carbon graphite. Separate seats are also machined from continuous cast iron bar stock. As in the case of carbon graphite materials, there are specific requirements for cast iron employed in air conditioning compressor seals.

These are as follows:

1 - fine grain structure
2 - high Youngs Nodulus (14 x 10^6 PSI)
3 - microstructure to be pearlitic matrix with predominant Type A graphites sizes 4 to 8
4 - 190 to 255 Brinell 3000 kg hardness

Six to eight percent cobalt or nickel cemented tungsten carbides are employed on large compressors where PV requirements given in Table I dictate their use. Tungsten carbide is the ultimate seat material for many seal applications air conditioning compressors included [2,3,5]. In addition to possessing excellent wear resistance, tungsten carbide has very high thermal stress resistance as well as high thermal conductivity and modulus of elasticity.

In addition to cast iron, 85% to 99% alumina ceramics have been employed along with medium carbon pearlitic steels as mating faces for carbon graphite washers.

Secondary Seal Materials

Neoprene elastomers and teflon offer the most universal resistance to common refrigerant from refrigerant oils. Teflon, for all practical purposes, has complete resistance with no restrictions. Neoprene, however, has a continuous service temperature limitation of 225°F and a maximum intermitant temperature limitation of 275°F. Close attention must be given to Neoprene and other elastomers swell characteristics in the presence of refrigerant and refrigerant oils. This is particularly true in the case of dynamic secondary seals employed on pusher type seals, as discussed previously. Excessive shrinkage can also cause obvious secondary seal problems in both pusher and nonpusher type seals. As a general rule, mineral base refrigerant oils having analine points of 180 to 228°F provide acceptable volume changes of Neoprene elastomers secondary seals.

Specific compound and molding practices are employed to minimize poor dispersion, air entrapment, and poor knit, all of which can lead to blistering failures of elastomeric secondary seals.

Nitrile elastomers have same temperature limitations as Neoprene but have poor imperviousness to Freon 22.

Viton elastomers have higher temperature resistance than neoprene (up to 375°F) but are restricted to amonia refrigerant service because of relatively poor imperviousness to Freon 12 and 22.

Current development work is being carried out on chlorinated polyethylene, epichcorohydrine, and chlorosulfonated polyethylene elastomer systems for air conditioning compressor seal service.

Hardware

Seal head hardware items such as springs, retainers, drive bands, discs, and snap rings are fabricated from wrought or sintered steel with black oxide coating or sinc plating for most Freon applications. Austinetic stainless steels or equivalent are acceptable hardware materials for all refrigerant applications.

SEAL LEAKAGE

Seal leakage for the most part is governed by equation [3].

$$W = \frac{Ps\ DH^3}{12NF} \qquad [3]$$

Where:

W = Leakage
D = Mean Seal Face Diameter
H = Channel Height (total seal face separation)
F = Face Width
N = Fluid Viscosity

Seal face separation or channel height is the most predominant variable affecting seal leakage in as much as leakage varies to the cube of channel height. Channel height is comprised of surface roughness, waviness and seal fluid film thickness. For this reason, it is important to control the surface finishes and distortion of both seal faces through proper design, materials selection and manufacturing procedures. Carbon washer and tungsten carbide seats surface finishes are held to within 5 CLA maximum. Cast iron surface finishes are held to within 10 CLA maximum. As manufactured seal face flatness is generally held to within two helium light bands. Stability of the seal face materials must be sufficient to maintain these flatness specifications until such time the seals are placed into compressors.

Carbon graphite seal faces normally distorted into a two node saddle shape configuration when subjected to hydraulic pressure and thermal gradients during compressor operation. The resulting waviness increased the channel height and subsequently, leakage as related in Figure 7. One means of correcting this waviness condition is deflecting .ame by increasing seal face loading either through larger spring loads and/or higher seal balance. Slight changes in seal face loading can cause significant reduction in waviness in as much as carbon graphite materials have low elastomeric modulic of 2 to 3 x 10^6 PSI.

Figure 8 relates deflection of seal face waviness as a function of seal face load. However, increased in seal face loading to reduce seal leakage must not cause seal operation PV values to exceed that limitation given in Table I and/or seal lubricant frictional power relationships are given in Figure 1 for flooded seal chambers and finally the PV limitations given for oil spray or splash lubricated seals.

The two-piece clamp plate and seat assembly although less desirable from a frictional heat removal standpoint, are more desirable in terms of flatness control. The separate seat in general has a more rigid section nodulus and is free of bolting or clamping stress. The higher modulus of cast iron (14×10^6 PSI) and the extremely high modulus of tungsten carbide (90×10^6 PSI) make them far less susceptible for distortion caused by hydraulic pressure, thermal gradients and volume changes in the secondary seals, than carbon graphite materials. Solid end plates should have sufficient rigidly and mounting support to maintain face flatness stability under bolting stress and hydraulic pressure. Here again, Figure 7 can be employed to determine what flatness is required in terms of desired maximum leakage.

Angular presentation of seal faces caused by deflection of same should be minimized to prevent excessive O.D. or I.D. seal face edge contact. Excessive edge contact will not only result in erratic leakage but thermal stress damaged faces materials and ultimate seal failure.

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TABLE I

MAXIMUM PV LIMITATIONS FOR SEAL FACE MATERIAL COMBINATIONS

Seal Face Material Combination	Maximum PV Limit Ft lbs/Min. in^2
Carbon Graphite vs. Cast Iron	2.56×10^5
Carbon Graphite vs. Tungsten Carbide	9.40×10^5
Bronze vs. Tungsten Carbide	1.00×10^6
Carbon Graphite vs. Alumina	1.52×10^5

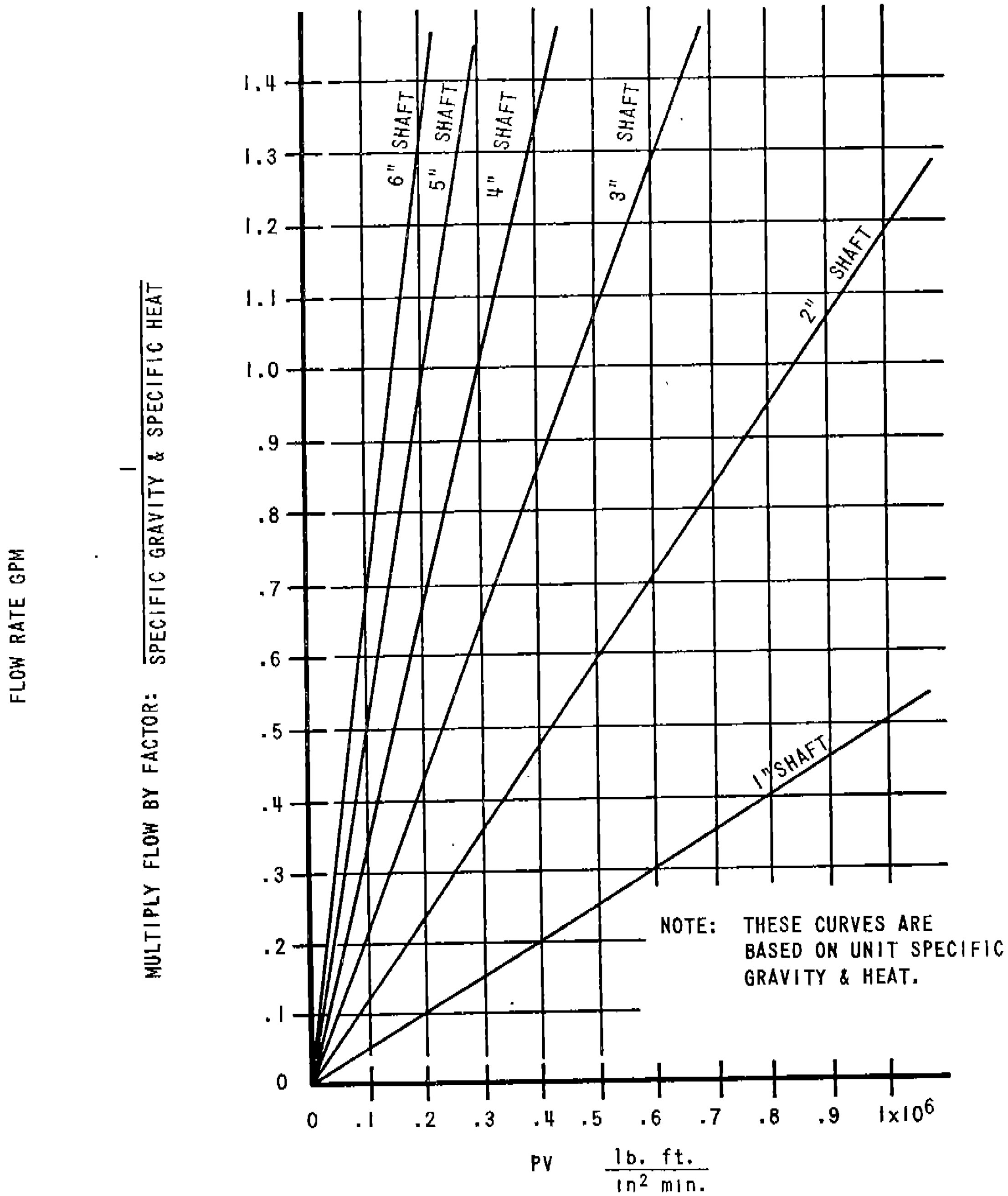

FIG. 1 SEAL LUBRICANT FLOW RATE VS PV FOR FLOODED SEAL CAVITIES

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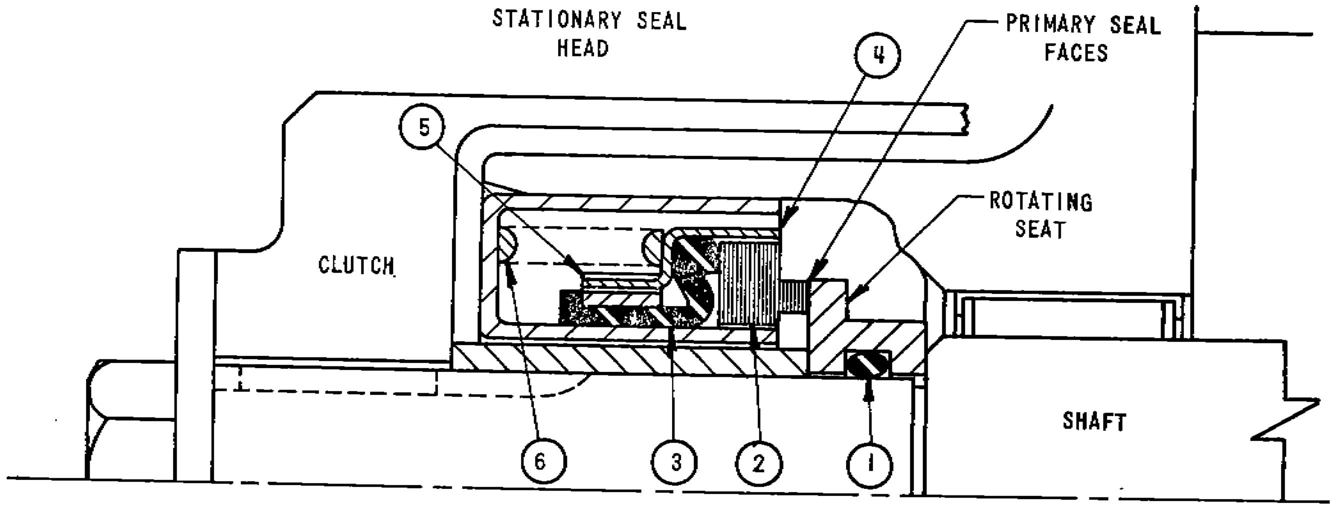


1. "O" RING SECONDARY SEAL
2. WASHER
3. BELLOWS SECONDARY SEAL
4. RETAINER
5. DRIVE BAND
6. SPRING

FIG. 2 CARTRIDGE SEAL WITH ROTATING SEAT

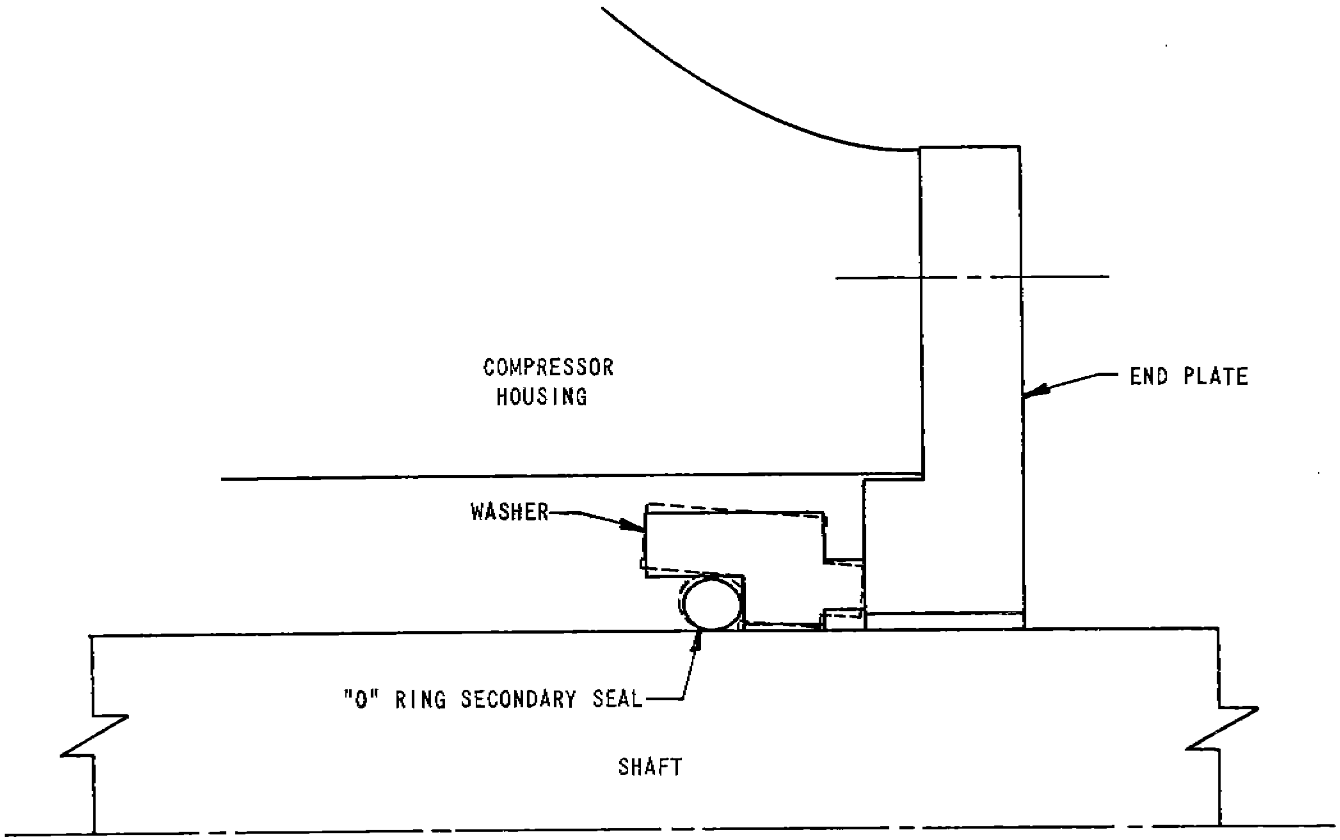
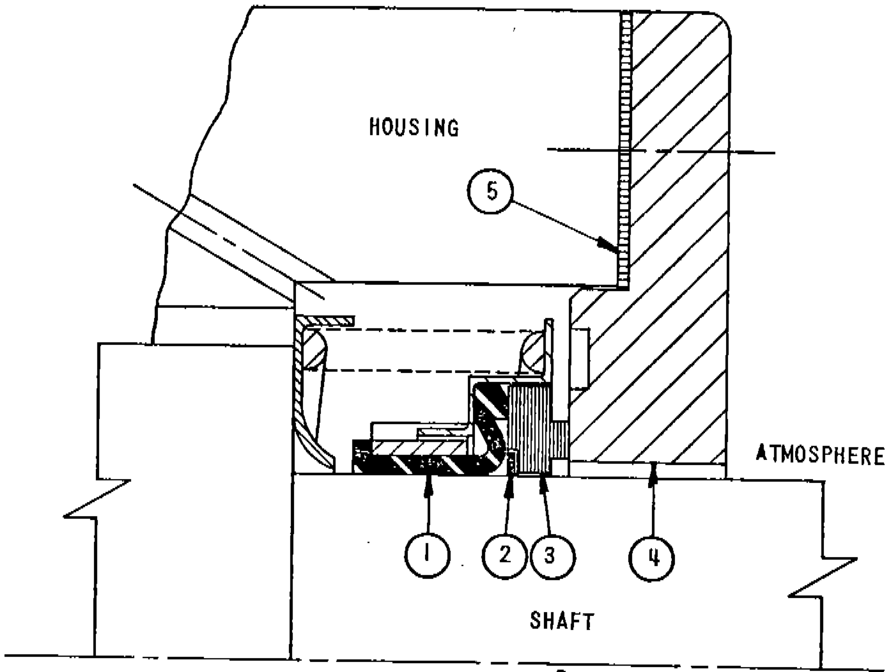
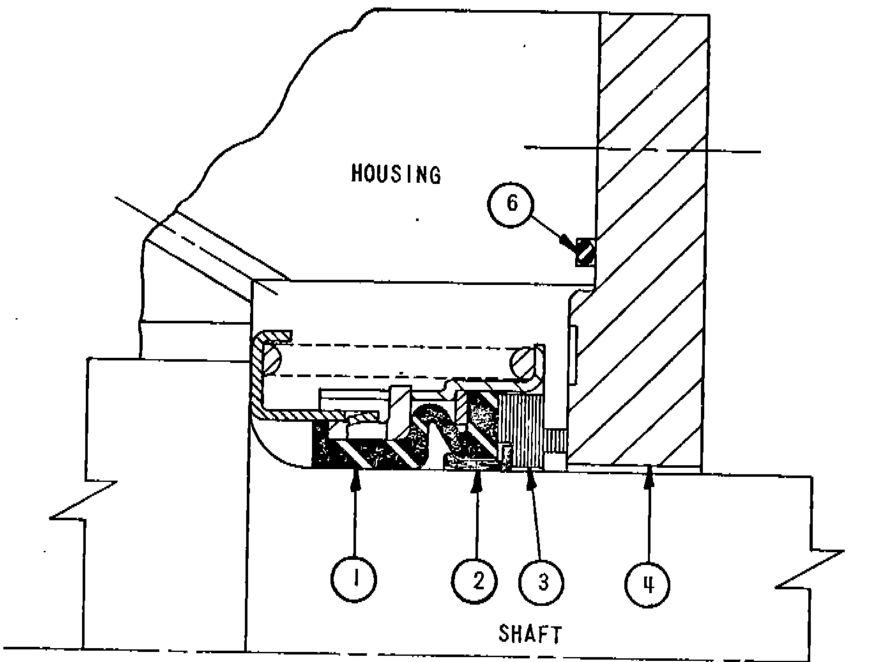


FIG. 3 DISTORTION OF PUSHER SEAL WASHER DUE TO SWELL OF SECONDARY SEAL



B

1. BELLOWS
   A. FULL CONVOLUTION
   B. ONE HALF CONVOLUTION
2. TEFLON HEAT SHIELD
3. WASHER
4. END PLATE (SEAT)
5. "O" RING
6. GASKET



A

FIG. 4 NONPUSHER ELASTOMERIC BELLOWS SEALS J.W. Abar

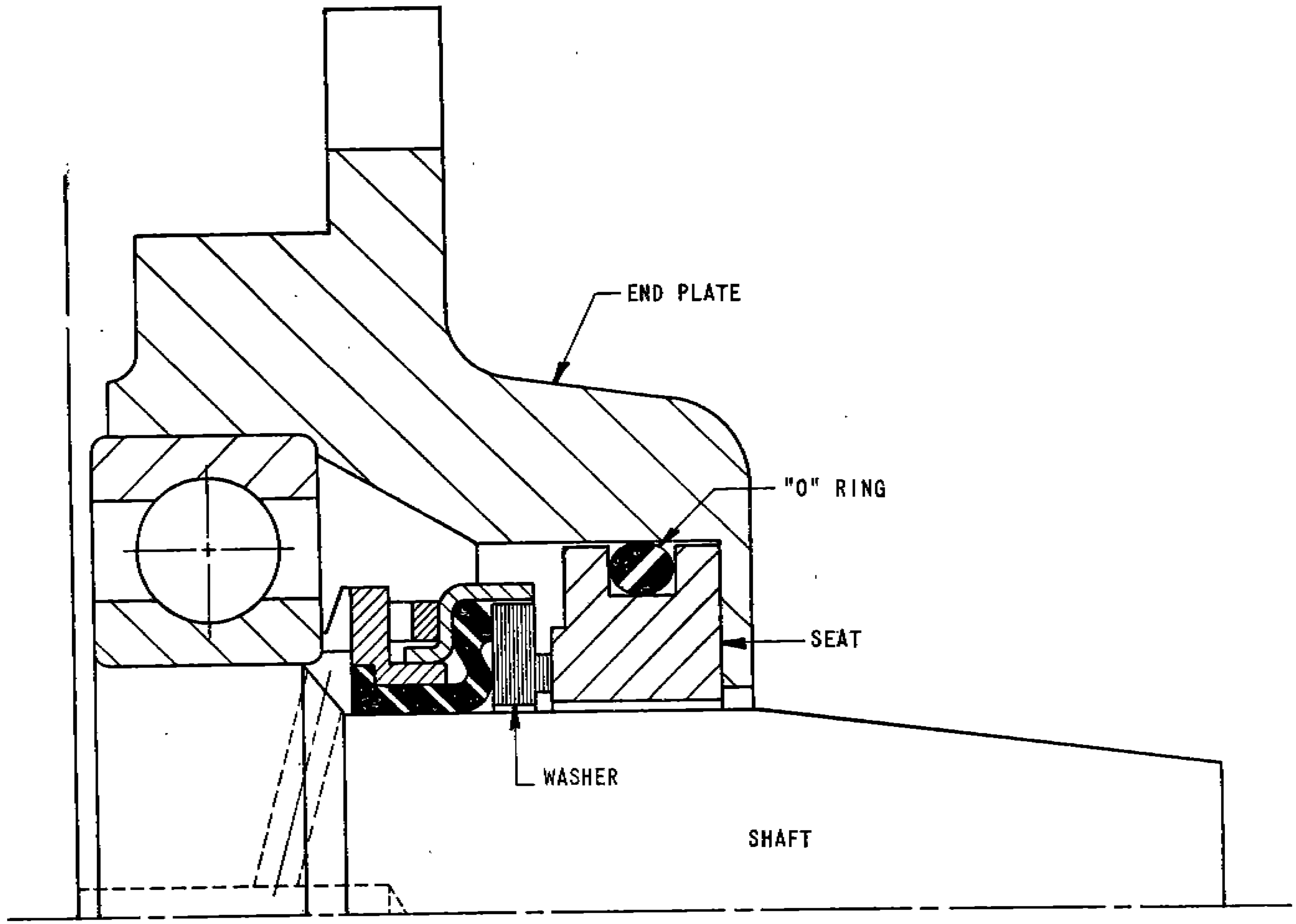


FIG. 5 TWO PIECE SEAT AND END PLATE

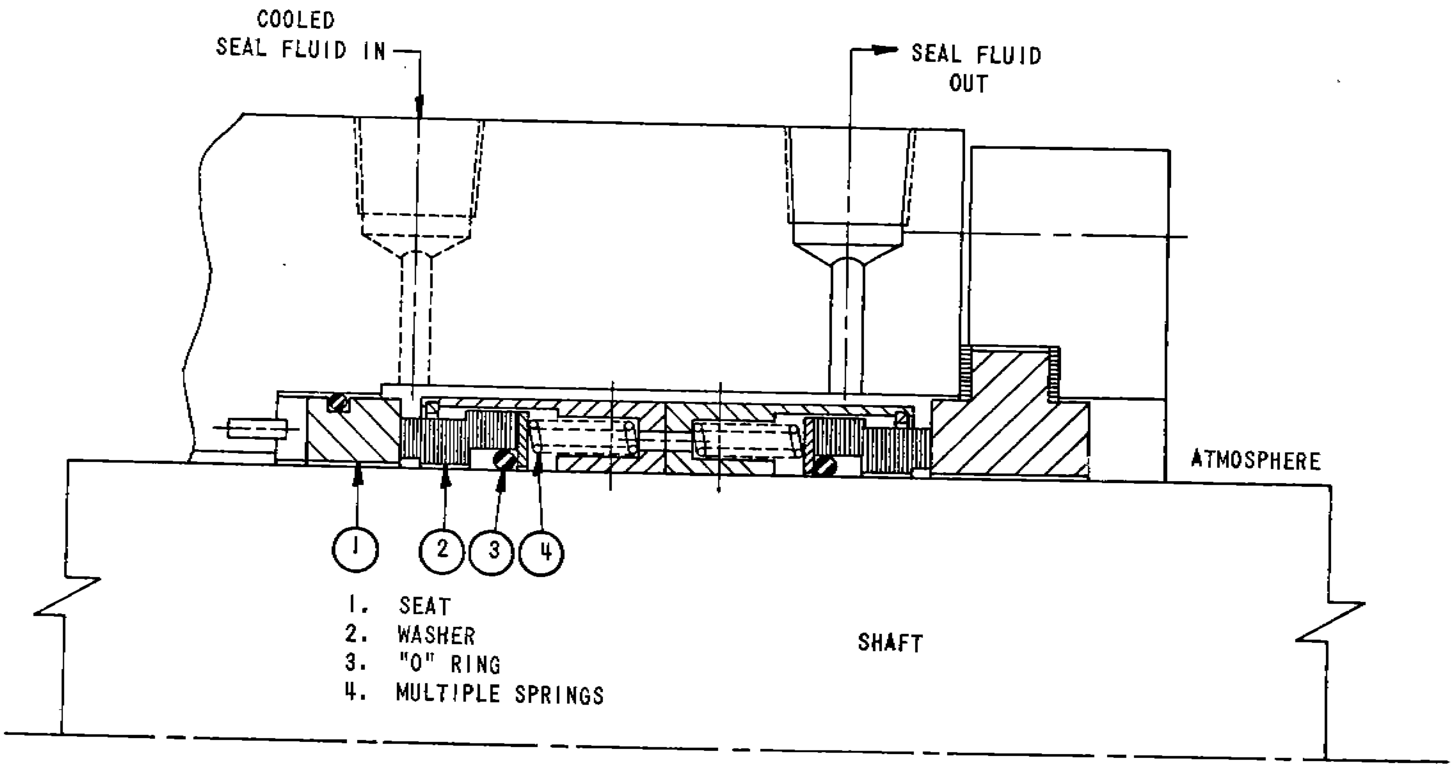


FIG. 6 DOUBLE-PUSHER SEAL ARRANGEMENT

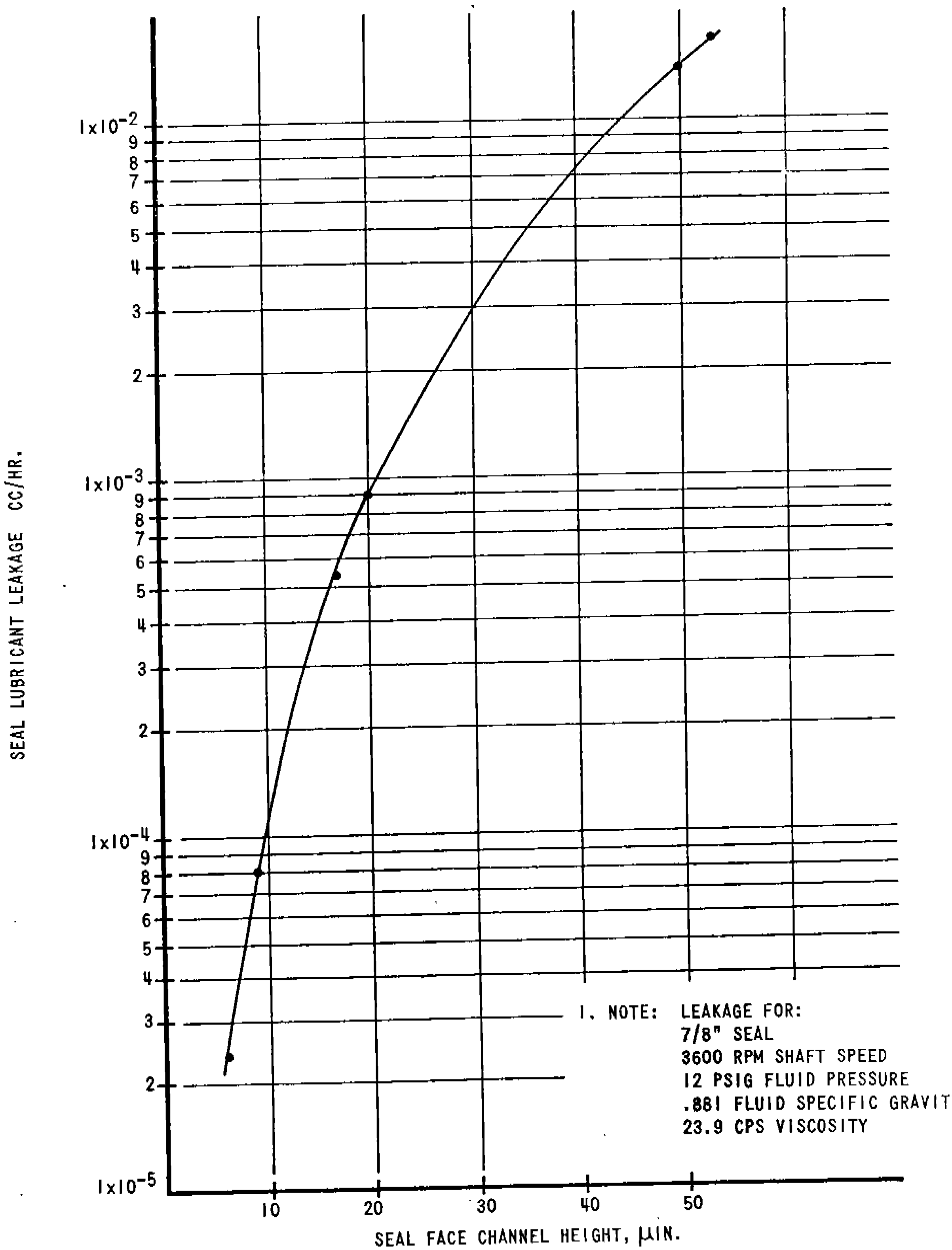


FIG. 7 SEAL LUBRICANT LEAKAGE VS SEAL FACE CHANNEL HEIGHT

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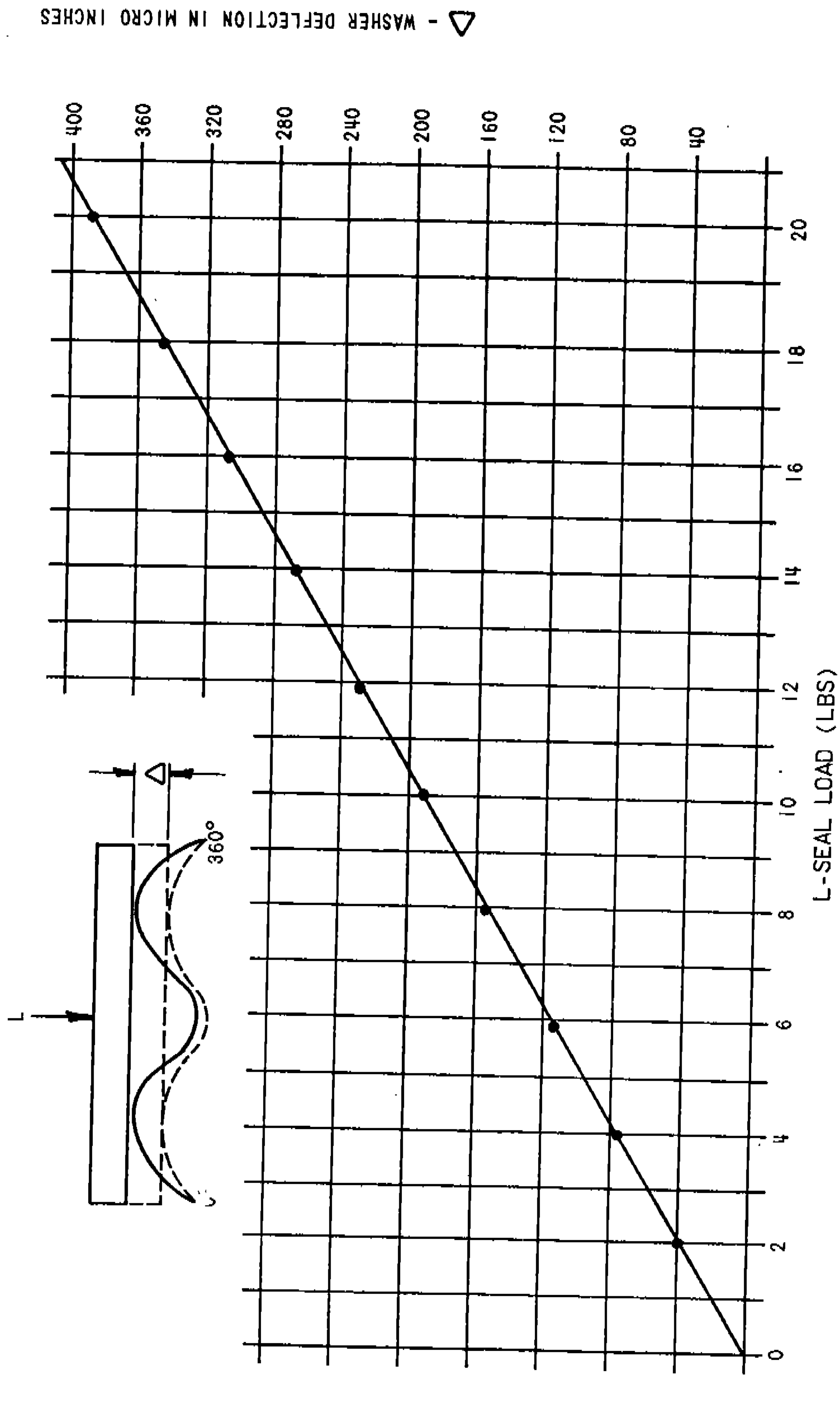

FIG. 8 THEORETICAL WASHER WAVINESS DEFLECTION FOR 7/8" SEAL