Seals—Application Guide to Radial Lip—SAE J946d

SAE Recommended Practice
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SEALS—APPLICATION GUIDE TO RADIAL LIP—SAE J946d

SAE Recommended Practice

Report of Nonmetallic Materials Committee approved February 1966 and last revised by Transmission and Drivetrain Technical Committee April 1979.

1. Introduction—This manual is intended as a guide to the use of radial lip type seals. It has been prepared from existing literature, which includes standards, specifications, and catalog data of both oil seal producers and users. Incorporated is the work performed to date by the Seals Subcommittee T-13 of the SAE Transmission and Drivetrain Committee. Standard tolerances established by the Rubber Manufacturers Association (RMA) are included. One of the main reasons for the preparation of this manual is to make standard information available in one document to the users of oil seals.

2. Sealing Systems—There are two general classes of sealing systems:

(a) The standard lip seal operating on a conventional mating surface.

(b) Elastohydrodynamic sealing systems which incorporate supplemental sealing devices on the seal and/or the mating surface.

Seal manufacturers are generally standardized on the bonded construction, single or double lip, with or without springs and with or without inner cases. Coating or molded rubber outside diameters are a variation of each class.

Table 1 depicts the more standard seal types. Seals of an assembled construction are also used, although generally the bonded types are preferred.

2.1 Standard Sealing Systems—The lip seal used in this system prevents leakage in dynamic and static applications by controlling interference between the seal lip and the mating surface. No supplemental sealing mechanisms are used with this system. Consequently, it will function satisfactorily in an application only if the following conditions are met:

2.1.1 The seal lip material is chosen for its ability to maintain essential interference under the environmental conditions to be encountered.

2.1.2 The seal is manufactured to tolerances established herein.

2.1.3 The shaft surface is prepared as in paragraph 4.1 and the shaft's dynamic characteristics, that is, roundness, dynamic runout, shock loading, or deflection, are within limits that will establish and maintain a satisfactory oil film.

2.1.4 The seal is installed properly as outlined in section 4.

2.1.5 Maintaining the above four basic sealing requirements in production is not always economically feasible. In certain applications, a supplementary sealing mechanism may be necessary. However, the use of these devices cannot be substituted for lack of control of shaft and seal quality.

2.2 Elastohydrodynamic Sealing Systems—These systems utilize supplemental sealing devices designed to transfer lubricant in a predetermined direction. There are two general types of elastohydrodynamic sealing systems:

2.2.1 SUPPLEMENTAL SEALING DEVICE ON THE SEAL—This system utilizes protrusions or depressions to transfer lubricant in a predetermined direction.

2.2.1.1 *Unirotational*—A seal incorporating helical ribs located on the outside lip surface which may or may not terminate in a static lip. Fig. 1 illustrates schematics of some of the more popular designs which have reached the commercial market.

2.2.1.2 *Birotational*—A seal incorporating configurations located on the outside lip surface which function independent of direction of shaft rotation. Their fluid transfer capability is generally lower than that of the unirotational designs, which reduces their potential to tolerate seal and shaft defects. Figs. 2 and 3 are examples of this type of seal.

2.2.2 SUPPLEMENTAL SEALING DEVICE ON THE MATING SURFACE—This system utilizes protrusions or depressions on the mating surface in combination with a seal or packing to transfer lubricant in a predetermined direction.

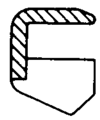
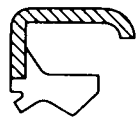
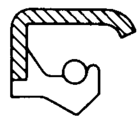
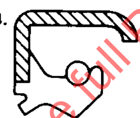
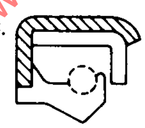
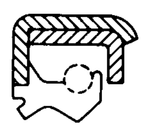
2.2.2.1 *Unirotational*—A mating surface incorporating a helical pattern. Static sealing is determined by groove configuration and seal material. Fig. 4 shows examples of this type of sealing system.

2.2.2.2 *Birotational*—A mating surface incorporating configurations which function independent of direction of shaft rotation. No known commercial application for this configuration exists.

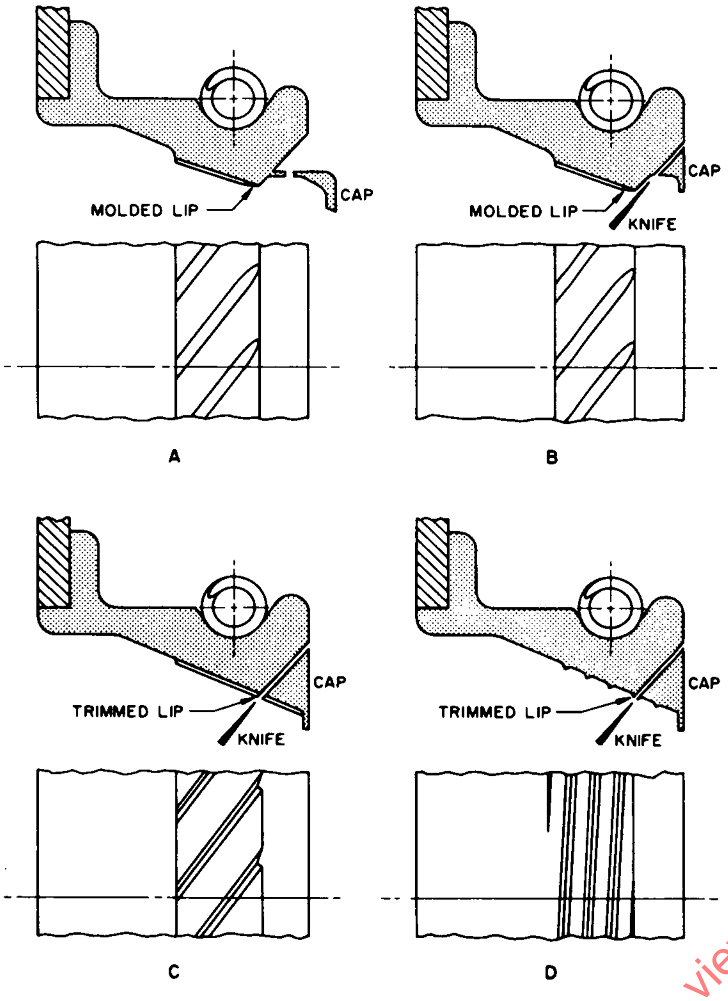
3. Seal Materials—Environmental conditions dictate the type of material which should be used in a specific application, so seal materials can be fully evaluated only in terms of specific operating conditions and performance requirements.

Seal material-lubricant compatibility is generally the governing factor. Because of the importance of seal material-lubricant compatibility, existing data and experience of both user and supplier should be fully considered. If adequate information is not available, it is recommended that candidate seal materials be evaluated for significant changes in properties in the actual application lubricant at normal operating temperatures. Particular importance should be given to those properties most directly affecting seal performance, such as volume swell (or shrinkage) and hardening (or softening). These evaluation tests should be of sufficient duration to establish the long-term effects.

TABLE 1—USES OF STANDARD SEAL TYPES

Cross Section	Type	General Application
1. 	Bonded single lip, non-spring loaded	Low cost design for viscous fluid and grease retention
2. 	Bonded double lip, non-spring loaded	Low cost design for viscous fluid or grease retention with dust, dirt, and moisture exclusion
3. 	Bonded single lip, spring loaded	Oil sealing applications and severe grease sealing applications
4. 	Bonded double lip, spring loaded	General oil sealing applications and severe grease sealing applications with dust, dirt, and moisture exclusion
5. 	Bonded single lip, with inner case	Provides all standard features plus additional inner case for greater structural rigidity
6. 	Bonded double lip, with inner case	Provides all standard features plus additional inner case for greater structural rigidity
7. Assembled construction	Single and multiple lip with and without springs	Special applications dependent upon the material selected

The ϕ symbol is for the convenience of the user in locating areas where technical revisions have been made to the previous issue of the report. If the symbol is next to the report title, it indicates a complete revision of the report.



SEALS SHOWN IN FIGS. 1A AND 1B HAVE A MOLDED LIP. THE EXCESS MATERIAL IS REMOVED BY TEARING THE CAP FROM THE MOLDED PART ON DESIGN 1A AND BY A KNIFE ON DESIGN 1B. THE HELICAL RIBS IN BOTH DESIGNS TERMINATE AT THE CONTACT POINT OF THE STATIC LIP.

SEALS SHOWN IN FIGS. 1C AND 1D ARE TRIMMED LIP SEALS; THAT IS, A KNIFE TRIMMING OPERATION FORMS THE CONTACT LIP AS THE EXCESS MATERIAL IS REMOVED. THE HELICAL RIBS PROTRUDE AT THE CONTACT POINT AND MUST BE COMPRESSED TO PREVENT THE SEAL FROM LEAKING WHEN THE SHAFT IS NOT ROTATING PRIOR TO INITIAL OPERATION.

FIG. 1—VARIOUS UNIROTATIONAL ELASTOHYDRODYNAMIC SEAL DESIGN CONCEPTS

It is also recommended that consideration be given to the effect of possible temperature extremes, particularly high-temperature compatibility and low-temperature flexibility as determined on both new and lubricant-aged seal materials.

Seal material-lubricant compatibility (or incompatibility) is influenced by four major variables which are interrelated in their effects.

3.1 Seal Material—The paragraphs at the end of this section give a general description of various seal materials.

3.2 Lubricant—Lubricants vary not only in their base composition, but especially in the additives used to achieve particular lubrication characteristics. It is recommended that material-lubricant compatibility be determined on a number of lubricants adequately representing those that might be used in the application. Lubricant decomposition as the result of heat, combustion products, etc., should also be considered, since decomposition products may themselves significantly affect seal materials.

3.3 Temperature—Seal material-lubricant compatibility over the normal operating temperature range is generally the governing factor, but adequate consideration must be given to the effects of the possible temperature extremes (low and high) on both seal material and the lubricant itself. High temperatures accelerate irreversible changes in both seal material and lubricant, as well as the interactions between them.

Low temperatures cause seal material to harden temporarily and cause lubricants to increase temporarily in viscosity, with seal fracture or abnormal wear possibly occurring as the result.

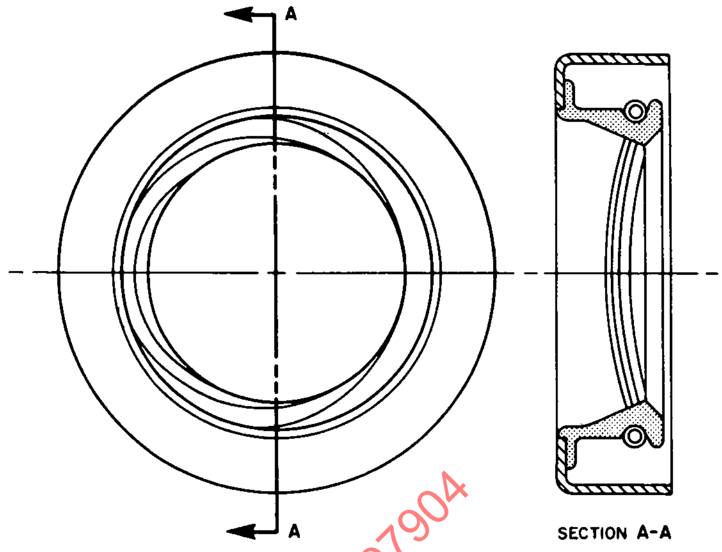


FIG. 2—BIROTATIONAL ELASTOHYDRODYNAMIC SEAL: RIBS PROTRUDING IN OPPOSITE DIRECTIONS PRODUCE SEALING IRRESPECTIVE OF DIRECTION OF SHAFT ROTATION

(The effect of shaft speed and/or pressure on seal material-lubricant compatibility is one of heat generation at the seal-shaft interface. Higher shaft speeds or pressures result in higher seal lip temperatures and in a greater differential between lip and bulk oil temperatures. These factors should also be considered in establishing material-lubricant compatibility.)

3.4 Time—Material-lubricant compatibility tests should be of sufficient duration to establish that compatibility is maintained over the required life of the seal.

3.5 The following paragraphs give general descriptions of the acceptable uses of elastomeric compounds, plastics, and leather, along with some of their advantages and disadvantages. The temperature ranges refer to normal lubricant bulk-oil operating temperatures. Acceptable upper and lower temperature limits may vary due to specific seal material and design, and are subject to substantial variations due to particular material-lubricant compatibility differences. When conditions approach extreme limits, the seal supplier should be consulted.

3.5.1 LEATHER—Leather is satisfactory for applications involving oil, grease, or foreign matter having temperatures ranging within limits of −65 to +180°F (−53.9 to +82°C).

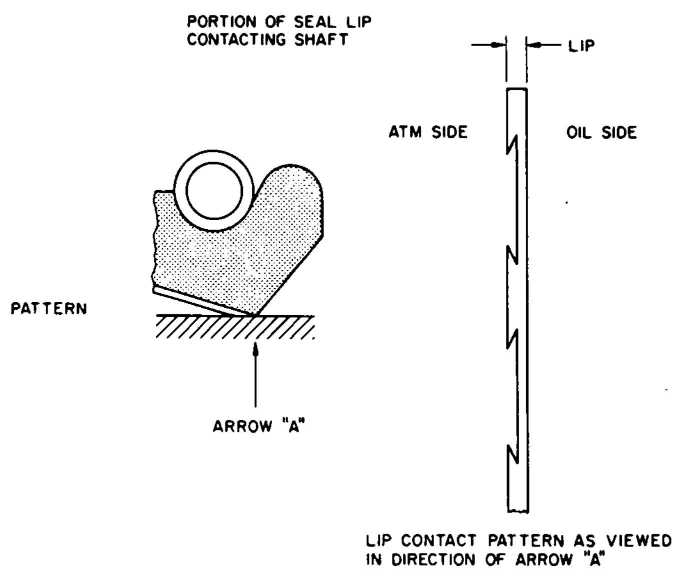
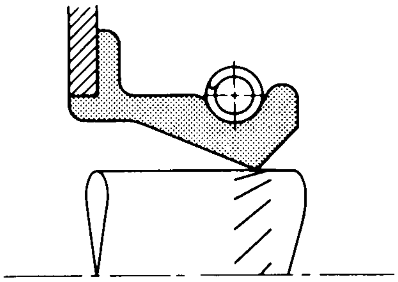
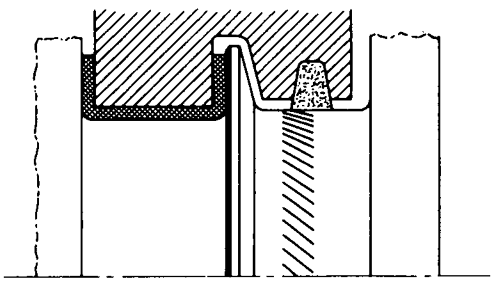


FIG. 3—BIROTATIONAL ELASTOHYDRODYNAMIC SEAL: TRIANGULAR DEPRESSIONS OR PROTRUSIONS IN CIRCUMFERENTIAL PATTERN IN OUTSIDE LIP SURFACE PRODUCE SEALING IRRESPECTIVE OF DIRECTION OF SHAFT ROTATION



DESIGN A



DESIGN B

DESIGN A ILLUSTRATES AN ELASTOMERIC, SPRING-LOADED SEAL OPERATING OVER GROOVES ETCHED OR EMBOSSED ON THE SHAFT SURFACE.

DESIGN B SHOWS AN ENGINE REAR CRANKSHAFT SEAL APPLICATION WHERE A ROPE PACKING IS USED TO OPERATE OVER A DEEPER KNURLED PATTERN ON THE SHAFT.

FIG. 4—UNIROTATIONAL HYDRODYNAMIC SEALING SYSTEM WITH SEALING DEVICE ON MATING SURFACE

3.5.1.1 *Advantages*
(a) Toughness—withstands difficult assembly.
(b) Accommodation of fairly rough shaft finishes.
(c) Good dry running characteristics.
(d) Good low temperature characteristics.
3.5.1.2 *Disadvantages*
(a) Poor heat resistance.
(b) Nonhomogeneous makeup makes consistent quality difficult.

3.5.2 Nitrile (Buna N) Compounds (NBR)[1]—Their operating range is −65 to +225°F (−53.9 to +107.2°C). When compounding a seal material for a low-temperature limit of −65°F (−53.9°C), the upper temperature limit of 225°F (107.2°C) must be lowered. Conversely, when compounding for the high-temperature limit, extreme low-temperature flexibility is sacrificed. Nitrile is recommended for general use in retaining lubricants and excluding mud, dirt, water, etc. These compounds have low volume swell in low aniline point oils. The nitriles are in the low cost range of oil seal compounds.
3.5.2.1 *Advantages*
(a) Fair dry running characteristics.
(b) Good processing.
(c) Good low-temperature and swell characteristics.
(d) Good oil resistance.
3.5.2.2 *Disadvantages*
(a) Lack of exceptional heat resistance.
(b) Tendency to harden during high temperature usage.

3.5.3 Polyacrylic Compounds (ACM or ANM)[1]—They are recommended for applications where temperatures are within 0 to +300°F (−17.8 to +149°C). If the shaft runout is low, these compounds may be used at lower temperatures. They are in the medium cost range of seal compounds.
3.5.3.1 *Advantages*
(a) Resistant to EP type additives.
(b) Good moderate temperature performance.
(c) Low swell characteristics.
(d) Good oil resistance.
3.5.3.2 *Disadvantages*
(a) Poor low-temperature properties with high shaft runout.
(b) Poor dry running characteristics.

[1]ASTM D 1418, Recommended Practice for Nomenclature for Synthetic Elastomers and Lattices (1976 Edition), should be used as a reference.

3.5.4 Silicon Compounds (V Si)[1]—They are recommended for applications where temperatures are within −65 to +350°F (−53.9 to +177°C). The maximum usable temperature is limited by the decomposition temperatures of the various lubricants. Silicon rubbers are in the high cost range of seal compounds.
3.5.4.1 *Advantages*
(a) Good heat resistance.
(b) Excellent low-temperature properties.
3.5.4.2 *Disadvantages*
(a) High swell characteristics in some oils.
(b) Poor chemical resistance to oxidized oils and some EP additives.
(c) Poor dry running characteristics.
(d) Easily damaged during assembly.

3.5.5 Fluorelastomer Compounds (FPM)[1]—They are recommended for applications where temperatures are within −65 to +400°F (−53.9 to +204°C). These compounds have good resistance to breakage at low temperatures. They are in the high cost range of seal compounds.
3.5.5.1 *Advantages*
(a) Excellent fluid resistance.
(b) Fair dry running characteristics.
(c) Excellent retention of original modulus and hardness in both dry heat and fluid service.
3.5.5.2 *Disadvantages*
(a) Poor low-temperature properties. Seals will leak at low temperatures with high shaft runout, but will not break down to −65°F (−53.9°C).
(b) Difficult to process.

3.5.6 Fluoroplastic Compounds (PTFE)[2]—They are recommended for applications which are chemically damaging to elastomers and for extreme temperatures within −400 to +500°F (−240 to +260°C). Some flexibility is maintained down to −110°F (−78.9°C). They are in the high cost range of seal compounds.
3.5.6.1 *Advantages*
(a) Superior fluid resistance.
(b) Excellent dry running characteristics.
(c) Low coefficient of friction.
(d) Withstands higher pressures than shown in Table 4.
3.5.6.2 *Disadvantages*
(a) Easily damaged during assembly.
(b) High thermal expansion and creep rates which cause seal lip instability.
(c) Limited ability to follow eccentric shafts due to high stiffness.

4. ***Applications Design Data***—Seal performance is greatly influenced by product design. Proper engineering of the components of the assembly which affect the seal is necessary for seal reliability. The following should be considered at the design stage of a new seal application or where existing applications are being updated.

4.1 Shaft Surface Texture—Shaft surface texture is a prime factor in the proper functioning of a lip seal. The surface roughness should be specified as 10–20 μin (0.25–0.50 μm) AA with no machine lead. The best known method to date for obtaining this texture is plunge grinding.

4.2 Shaft Diameter—The shaft diameter should be held within the tolerances shown in Table 2, although greater shaft tolerances may be used when agreed upon between user and supplier.

4.3 Shaft Hardness—Under normal conditions, the portion of the shaft contacted by the seal should be hardened to Rockwell C30 minimum. There is no conclusive evidence that hardening above this will increase the wear resistance of the shaft except under extreme abrasive conditions. Where the shaft is liable to be nicked in handling previous to assembly, it is recommended that it be hardened to Rockwell C45 minimum in order to protect against being permanently damaged during assembly.

4.4 Wear Sleeves—Where the use of wear sleeves is considered, hardened shafts generally are not required. Wear sleeves either soft or hardened can be made from mild steel rings and pressed over a soft shaft. They are recommended for shafts made of cast iron or other soft materials, and permit the replacement of wearing surfaces coincident with oil seal changes. New wearing surfaces generally are required with the replacement oil seal.

4.5 Offset—Offset is normally calculated from the tolerance stackup on the engineering drawings. Seal life can be shortened by excessive offset. Offset results in uneven wear. From a good practice standpoint, the offset should be kept under 0.010 in (0.254 mm).

4.6 Dynamic Runout—Generally, the shaft runout should be kept below 0.010 in (0.254 mm) TIR.

4.7 Bore and Seal Tolerances—The bore and seal tolerances shown in Table 3 apply only to ferrous materials. When a nonferrous material such as aluminum is used, the seal manufacturer should be consulted.

[2]ASTM D 1600, Tentative Abbreviations of Terms Relating to Plastics (1975 Edition), should be used as a reference.

TABLE 2—RMA OIL SEAL STANDARD TOLERANCES (SINGLE AND DUAL LIP SPRING LOADED BONDED SEALS)

Shaft Dia		Shaft Dia Tolerance		Seal Lip ID Range		Lip Opening Pressure Range		Radial Wall Variation, Max	
in	mm	in	mm	in	mm	psi	Pa	in	mm
Up to and including 4.000	Up to and including 101.6	±0.003	±0.08	0.040	1.02	Nominal pressure. 30% with a min range of 4 psi, and a min reading > 0	Nominal pressure. 30% with a min range of 27.6 kPa, and a min reading > 0	0.025	0.64
4.001- 6.000	101.63–152.4	±0.004	±0.10	0.050	1.27			0.030	0.76
6.001–10.000	152.43–254	±0.005	±0.13	0.060	1.52			0.040	1.02

4.8 Bore Surface Texture—On applications where a lubricant head is present against the outside diameter of the seal, if the bore surface roughness is approximately 100 μin (2.54 μm) AA or smoother, no outside diameter leakage problems should be encountered, if no tool removal defects are present.

If the surface is rougher than 100 μin (2.54 μm) AA, a case OD sealer should be used to insure that no outside diameter leakage occurs. Cements or sealers should be used with care to prevent contact with the sealing lip.

On grease sealing applications, a bore sealer generally is not required.

4.9 Pressure—Standard design radial lip type oil seals should not be used when the operating pressure exceeds the limits shown in Table 4.

When variable surge pressures exceeding these limits are present, a special condition exists and the seal manufacturer should be consulted.

Higher operating pressures may be feasible if a custom seal design is considered. However, when a pressure seal is used, features such as the ability to take greater offset and runout may be sacrificed. Whenever possible the design should be such that the system is vented. This will allow the lip seal to function more efficiently.

4.10 Shaft Lead Corners—To prevent damage to the seal lip and to facilitate installation, the lead edge of the shaft should have a chamfer or radius. If a chamfer is used, its dimensions should allow a seal lip to be assembled without damage. (See Fig. 5.)

4.11 Bore Lead Corners—The lead corner of the bore should be chamfered to facilitate efficient installation of the seal. (See Fig. 6.)

4.12 Bore and Seal Sizes—Whenever possible, shaft and bore size should be selected from the RMA standards. See Tables 2, 3, and 5.

4.13 Cocked Assembly—A factor in the functioning of a lip seal is the installed squareness of the outside seal face with respect to the normal shaft centerline. Keeping this within 0.010 in (0.254 mm) TIR is considered a good general practice. This squareness is obtained by pressing the seal flush with the front of the bore or bottoming against the back of the bore.

It is recommended that the seal case be designed so that a seal of maximum width is pressed approximately flush with the front of the bore. By doing this, various seal widths can be accommodated if the bore is sufficiently deep.

Installation tools should be used to press the seal into place. (See Fig. 7.)

Whether a seal is installed flush to the face of the bore or bottomed on the back of the bore (Figs. 8 and 9), the surface it is aligned with should always be a machined one. Unfinished surfaces should never be used for alignment purposes because of the danger of cocking the seal in the bore.

4.14 Seal Assembly—All surfaces which the seal lip must slide over during installation should be smooth and free from rough spots. To prevent damage to the seal lip, special installation tools should be used if the sealing element slides over splines, keyways, holes, or if the seal is assembled toe first. Assembly procedure should be carefully reviewed so that seal lips are not turned under assembly. A light film of grease or oil on the shaft or seal prior to the assembly will decrease the probability of damage during assembly.

*5. **Springs***—A spring is incorporated in the design of most oil seals to provide a uniform load at the seal contact line-shaft surface junction. There are two types of springs currently in use, the garter spring and the finger spring. (See Fig. 3 of SAE J111c (April, 1979).)

Since environmental conditions dictate the type of sealing element material used for a specific application, the spring design must take into account any physical, chemical, or mechanical property change experienced by the material, as well as provide the proper radial load requirement of the application. Seal manufacturers interrelate the design of the spring and sealing element in conjunction with the user's requirements to provide optimum performance.

5.1 Material—Material to be noted as carbon or stainless steel with SAE designation.

5.1.1 SAE 1065–1066—Hard drawn wire—general purpose material for temperatures up to 250°F (121°C) where no corrosive fluids or gases come in contact with the seal.

5.1.2 SAE 30302 Stainless Steel—Temperatures up to 450°F (232°C) in corrosive environments.

5.2 For garter spring dimensional and visual qualities and physical properties, refer to paragraphs 8.6 and 8.7.

*6. **Drawing Designation***—It is recommended that the standard SAE oil seal drawing be used. See Fig. 17.

*7. **Qualification Tests***—Qualification tests should be performed in accordance with SAE J110b (July, 1973).

*8. **Inspection and Quality Control Data***

8.1 Radial Wall Variation—See Fig. 10 for diagram of radial wall dimension. Radial wall variation is checked through use of an optical comparator. The seal outside diameter is placed on a base and rotated through 360 deg.

8.1.1 Recommended Tolerances—See Table 2.

8.2 Lip Opening Pressure—Lip opening pressure is used as a measure of consistency of manufacturer.

8.2.1 Recommended Tolerances—See Table 2.

8.2.2 Air Flow Method for Gaging Seal Lip Opening Pressure—The lip opening pressure of the seal assembly shall be gaged by means of an airflow occurring between the lip and a test mandrel when air pressure is applied. Satisfactory equipment is available commercially.

TABLE 3—RMA OIL SEAL STANDARD TOLERANCES (METAL OUTSIDE DIAMETER SEALS)

Bore Dia		Bore Tolerance		Nominal Press Fit		OD[a] Tolerance		Out of Round	
in	mm	in	mm	in	mm	in	mm	in	mm
Up to 1.000	Up to 25.4	±0.001	±0.025	0.004	0.102	±0.002	±0.051	0.005	0.127
1.001– 3.000	25.43– 76.2	±0.001	±0.025	0.004	0.102	±0.002	±0.051	0.006	0.152
3.001– 4.000	76.23–101.6	±0.0015	±0.038	0.005	0.127	±0.002	±0.051	0.007	0.178
4.001– 6.000	101.62–152.4	±0.0015	±0.038	0.005	0.127	+0.003/−0.002	+0.076/−0.051	0.009	0.229
6.001– 8.000	152.42–203.2	±0.002	±0.051	0.006	0.152	+0.003/−0.002	+0.076/−0.051	0.012	0.306
8.001– 9.000	203.23–228.6	±0.002	±0.051	0.007	0.178	+0.004/−0.002	+0.102/−0.051	0.015	0.381
9.001–10.000	228.63–254	±0.002	±0.051	0.008	0.203	+0.004/−0.002	+0.102/−0.051	0.015	0.381
10.001–20.000	254.03–508	+0.002/−0.004	+0.051/−0.102	0.008	0.203	+0.006/−0.002	+0.152/−0.051	0.002	0.051
20.001–40.000	508.03–1016	+0.002/−0.006	+0.051/−0.152	0.008	0.203	+0.008/−0.002	+0.203/−0.051	in/in of seal OD	mm/mm of seal OD
40.001–60.000	1018.03–1524	+0.002/−0.010	+0.051/−0.254	0.008	0.203	+0.010/−0.002	+0.254/−0.051		

[a]The average of a minimum of three measurements to be taken at equally spaced positions.

TABLE 4—OPERATING PRESSURE LIMITS

Shaft Speed		Max Pressure Permissible	
ft/min	m/min	psi	kPa
0–1000	0–304.80	7	48.2
1001–2000	305–609	5	34.5
2001 and up	610 and up	3	20.7

8.2.2.1 The seal case shall be mounted over the mandrel in a retainer fixture and be held concentric to the mandrel within 0.002 in (0.051 mm) TIR.

8.2.2.2 Air leakage around the seal case outside diameter and around the mandrel pilot shall be prevented by means of O-rings or other suitable gaskets.

8.2.2.3 The test mandrel diameter shall be equivalent to the mean of the shaft diameter limits specified on the seal drawing with a mandrel diameter tolerance of ±0.0005 in (0.013 mm).

8.2.2.4 The test mandrel shall have a surface roughness of 15 μin (0.38 μm) AA or less.

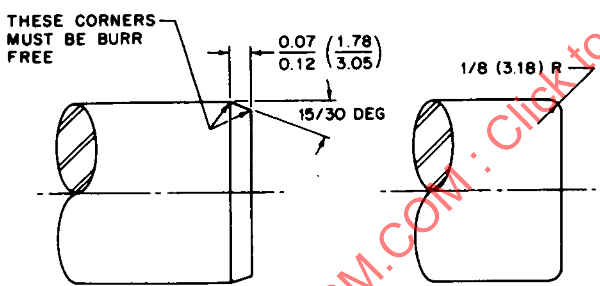
8.2.2.5 The seal shall be placed in the test fixture so that air pressure is applied to the outside face of a single lip seal or between the lips of a double lip seal, with care being taken to insure that the dust lip does not interfere with the reading.

8.2.2.6 The seal lip opening pressure shall be gaged at a flow of 10 000 cm^3/min.

8.2.2.7 To minimize material relaxation effect, the air pressure shall be increased from zero at a uniform rate such that the lip opening pressure shall be read within 3–6 s.

8.2.2.8 When both lip opening pressure and seal lip inside diameter measurements are to be taken consecutively, the seal lip inside diameter shall be measured first to avoid errors due to deformation of the material. Repeat measurements on the same seal shall not be taken within 16 h of a lip opening pressure measurement.

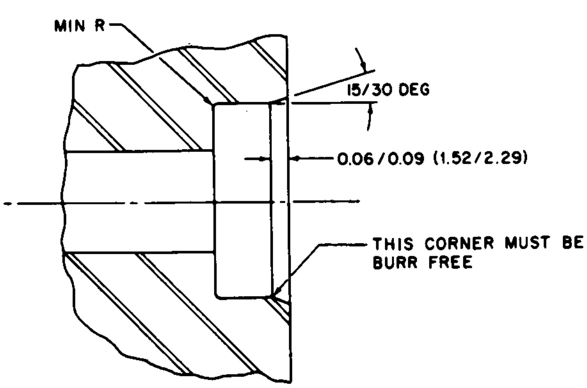
8.2.2.9 Measurements are to be taken at a room temperature above 60°F (15.6°C). The seal shall be exposed to room temperature for at least 1 h before measuring.



MUST BE SMOOTH AND FREE OF NICKS AND ROUGH SPOTS

DIMENSIONS ARE IN (mm)

FIG. 5—RECOMMENDED SHAFT LEAD CORNERS



DIMENSIONS ARE IN (mm)

FIG. 6—RECOMMENDED BORE LEAD CORNER

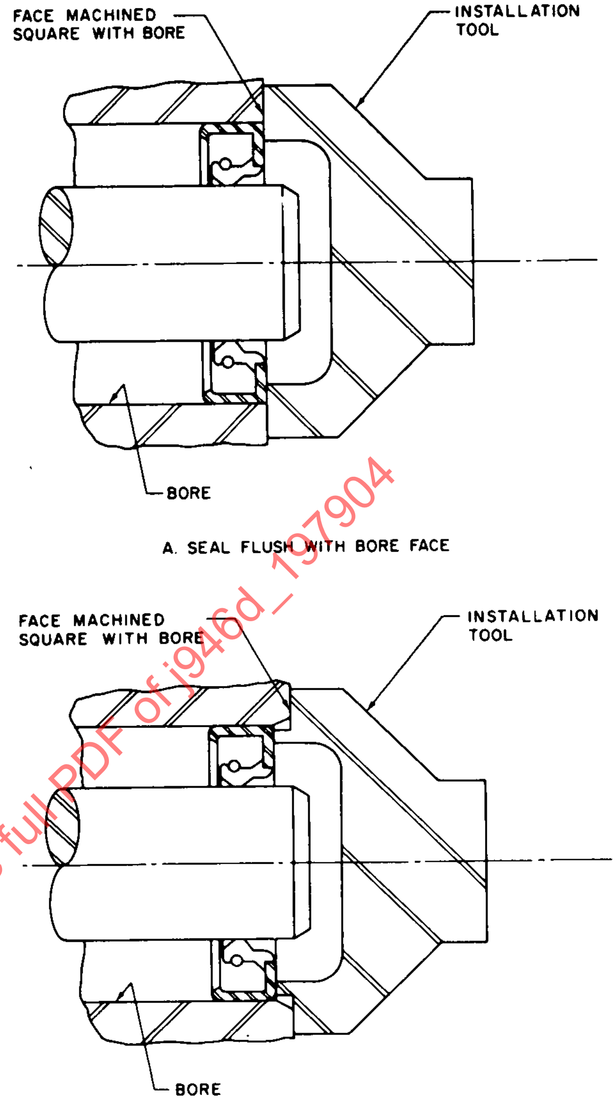


FIG. 7—THROUGH BORE: INSTALLATION TOOL BOTTOMS ON MACHINED BORE FACE

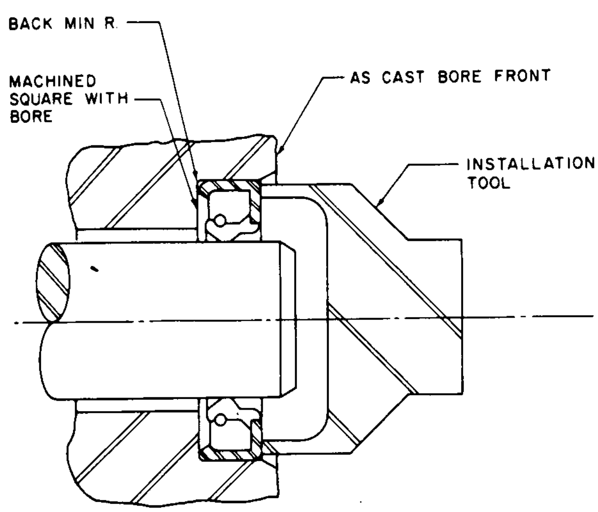


FIG. 8—BOTTOM BORE: SEAL BOTTOMS ON MACHINE BORE SHOULDER

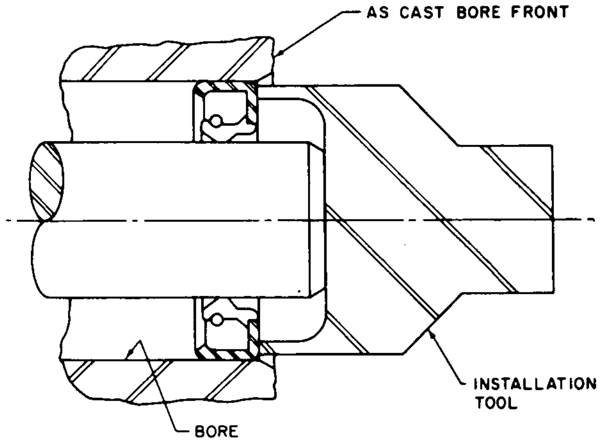


FIG. 9—THROUGH BORE: INSTALLATION TOOL BOTTOMS ON SHAFT

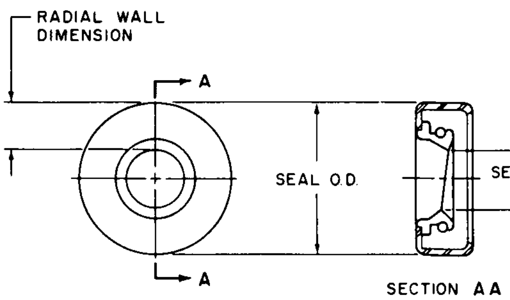


FIG. 10—RADIAL WALL DIMENSION

8.3 Inside Diameter—There are two types of inside diameters presently measured in the seal industry. They are seal lip diameter and functional lip diameter.

8.3.1 Seal Lip Diameter—See Table 2 and Fig. 10.

8.3.1.1 *Optical Comparator Method*—In one technique using an optical comparator, the seal lip diameter is measured in several positions and an average taken. The main disadvantage of this technique is lack of speed. A technique using the optical comparator which is more rapid involves the determination of the average radial wall dimension through the measurement of the minimum and maximum radial wall dimensions. The seal lip diameter is then equal to the seal outside diameter minus twice the average radial wall dimension. This method has been automated.

8.3.1.2 *Tapered Shaft by Light Method*—This is an acceptable method of approximating the seal lip diameter. While it does not measure diameter in the free state, it does give an acceptable approximation of the seal lip inside diameter.

In this method, a shaft with a taper of approximately 0.005 in per ⅛ in (0.127 mm per 3.18 mm) is used with a light source below. The seal is lowered until no light can be seen between the seal lip and the shaft. The seal lip diameter is read from markings on the shaft.

8.3.2 Functional Lip Diameter

8.3.2.1 *Airflow Method for Gaging Seal Functional Lip Diameter*—The functional lip diameter of the seal assembly can be measured by means of an airflow between the lip and a mandrel of known size when the air is applied at a standard pressure.

8.3.2.1.1 The seal case shall be mounted over the mandrel in a retainer fixture and be held concentric to the mandrel within 0.002 in (0.051 mm) TIR.

8.3.2.1.2 Air leakage around the seal case outside diameter and around the mandrel pilot shall be prevented by means of O-rings or other suitable gaskets.

8.3.2.1.3 The test mandrel diameters shall be the same as specified for maximum and minimum lip diameters with a mandrel diameter tolerance of ±0.0005 in (0.013 mm). The measurement using the minimum diameter mandrel shall be made first.

8.3.2.1.4 The test mandrel shall have a surface roughness of 15 μin (0.38 μm) AA or less.

8.3.2.1.5 The seal shall be placed in the test fixture so that air pressure is applied to the outside face.

8.3.2.1.6 An air pressure of 0.50 psi (3450 Pa) shall be used.

8.3.2.1.7 The airflow between the seal lip and the minimum diameter test mandrel shall be equal to or greater than 10 000 cm^3/min. The airflow between the seal lip and the maximum diameter test mandrel shall be equal to or less than 10 000 cm^3/min.

8.3.2.1.8 When both lip opening pressure and seal lip diameter measurements are to be taken consecutively, the seal lip diameter shall be measured first to avoid errors due to deformation of the material. Repeat measurements on the same seal shall not be taken within 16 h of a lip opening pressure measurement.

8.3.2.1.9 Measurements are to be taken at a room temperature about 60°F (15.6°C). The seal shall be exposed to room temperature for at least 1 h before measuring.

8.3.2.2 *Light Box Method*—In this method, the seal outside diameter is held concentric to a sizing mandrel in a similar fashion to that described above under the air gaging method. High and low limit functional inside diameter mandrels are arranged with light sources underneath them. Test seals are then positioned into the fixtures, noting the light passing between the seal inside diameter and the sizing mandrel. On the high limit mandrel, the seal lip must preclude all light to be acceptable (indicating it is smaller than the mandrel); while on the low limit mandrel, the seal lip must allow light to pass.

By using mandrels in increments of 0.005 in (0.127 mm) of diameter, the seal functional inside diameter can be determined in a fashion similar to that of the airflow method.

8.3.3 Recommended Tolerances—The Rubber Manufacturers Association has established tolerances for seal lip diameters as shown in Table 2.

Functional inside diameter tolerance ranges may frequently be the same as those stated above for seal lip diameter. However, under certain circumstances functional lip diameter tolerance ranges may be greater.

ϕ **8.4 Spring Axial Position—The spring axial position is the axial distance between the projected intersection of the inside and outside lip surfaces and the centerline of the spring coil diameter (center plane of the spring) with the spring in position and the seal located on the shaft.**

ϕ 8.4.1 Method of Measurement—The primary method to be used is designated, *On Shaft Casting and Sectioning Method.* An alternate, nondestructive method is described as the *Electrical Continuity Spring Location.*

ϕ *8.4.1.1 On-Shaft Casting and Sectioning*—The seal to be measured is placed in a fixture that simulates the shaft and housing bore assembly. This fixture can be constructed of various materials suitable for polishing and adhesion to a potting material. The shaft is to be concentric with the housing bore within 0.002 in (0.05 mm) TIR. The seal and fixture are then encapsulated in a potting material such as a dental casting plaster or an epoxy having a shrinkage of not more than 2%. The assembly is then cross sectioned through its center in a vertical plane and polished. Care must be taken not to distort the seal during the sectioning operation. The spring axial position is now measured by viewing the cross section using either a reflective optical comparator or a toolmaker's microscope. This method is the most accurate but is not practical for measuring large quantities of seals.

ϕ *8.4.1.2 Electrical Continuity Spring Location*—The seal to be measured is placed on a device equivalent to the one described in SAE Paper No. 740204, "Spring Position Measurement," presented at the 1974 SAE Automotive Engineering Congress and Exposition. The spring axial position is determined by adding the correction factor to the micrometer reading. This method is suitable for single case seals having sufficient clearance for a probe.

ϕ 8.4.2 Recommended Tolerance—The recommended tolerance for spring axial position from seal to seal shall be 0.020 in (0.51 mm) total for molded contact line seals, and 0.025 in (0.64 mm) total for trimmed contact line seals. The recommended tolerance for spring position within one seal shall be 0.010 in (0.25 mm) total for molded contact line seals, and 0.015 in (0.38 mm) total for trimmed line seals. These tolerances do not apply for seals employing variable spring axial position as a designed feature. The supplier should be consulted for spring axial position tolerances for these designs. The spring axial position tolerance should not allow the center line of the spring coil diameter (center plane of the spring) to shift from one side of the projected intersection of the inside and outside lip surfaces to the other.

ϕ **8.5 Contact Line Height Variation**—Contact line height variation is the difference between the maximum and minimum axial dimensions from the seal contact line to the outside face. See Fig. 11.

ϕ8.5.1 Measurement—The recommended method of measuring contact line height variation is the use of a microscope and prism arranged as shown in Fig. 12. In this method the contact line height variation is read directly as the seal is rotated above a fixed center. The method is easy to use and offers a good degree of repeatability. Necessary equipment is inexpensive and does not require tooling for each seal size. Since the seal contact line is viewed in its free position, handling should be minimized to avoid distortion. One method of minimizing distortion is to *shaft* the seal concentric to its OD using a nominal shaft size mandrel prior to measuring.

ϕ8.5.2 Recommended Tolerances—Recommended tolerances for limiting contact line height variation shall be within and shall not exceed the maximum allowable spring position tolerance as specified by the supplier.

ϕ **8.6 Sealing Edge Roughness**—Sealing edge roughness refers to the condition of the surfaces on the seal that form the seal to shaft interface. (See Fig. 13.)

ϕ8.6.1 Measurement—Optical examination using magnification is recommended to measure sealing edge roughness. A minimum of 7X magnification should be used. Conditions that should be noted are the following:

ϕ *8.6.1.1* Average contact line roughness.

ϕ *8.6.1.2* Surface characteristics immediately adjacent to the contact ϕ line, such as defects due to dirty mold surfaces, inclusions in compound, and voids.

ϕ *8.6.1.3* Angular marks on the outside lip surface adjacent to the contact line. This may be part of the design on elastohydrodynamic seals.

ϕ *8.6.1.4* Angled tears or edges that would give preferential pumping action. This may be part of the design on elastohydrodynamic seals.

ϕ *8.6.1.5* Any deviation that exists in a small increment of seal contact line length (approximately 1/4 in (6.35 mm)). This differentiates between *contact line height variation* and *contact line roughness*.(See Fig. 13.)

ϕ 8.6.2 Production Measurement—The method above will not lend itself to production seal examination. If such production seal examination of contact

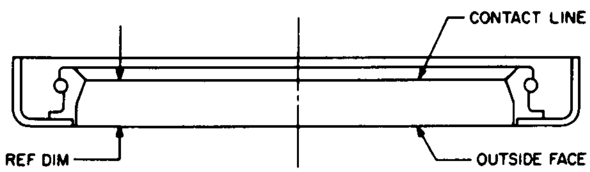


FIG. 11—CONTACT LINE HEIGHT VARIATION

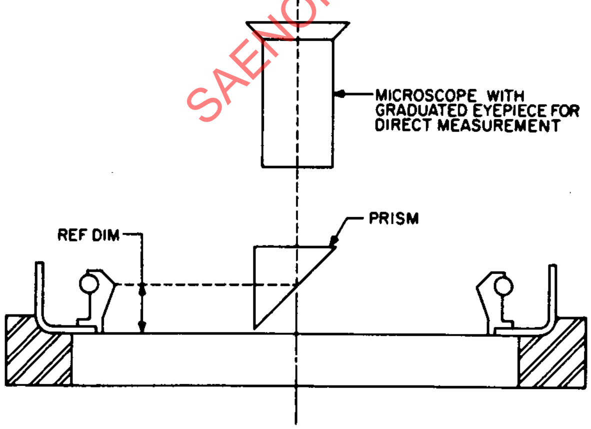


FIG. 12—MEASUREMENT OF CONTACT LINE HEIGHT VARIATION

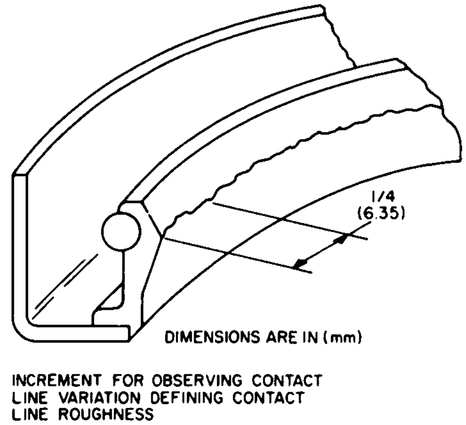


FIG. 13—SEALING EDGE ROUGHNESS

line roughness is required, it may be accomplished by the use of a suitable mechanical measuring device with a reasonable degree of confidence. When using these devices, they should be periodically referenced against some optical method.

ϕ **8.7 Garter Spring Dimensional and Visual Qualities**

ϕ8.7.1 Wire diameter is the total wire thickness measured prior to coiling, expressed in inches or millimetres. Wire diameter is used as a reference only. It is expected that wire diameter may vary to accomplish control on load at specified test length. (See Fig. 14.)

ϕ 8.7.2 Coil diameter is the total outside diameter of the coil expressed in inches or millimetres. Tolerance should not exceed ±0.005 in (±0.127 mm). A slight increase at the joint is permissible. However, the increase shall still be within the allowed tolerance. (See Fig. 14.)

ϕ 8.7.3 Free Length (excluding the nib)—Up to 4 in (101.60 mm) assembled inside diameter, the free length will be for manufacturing reference only. (See

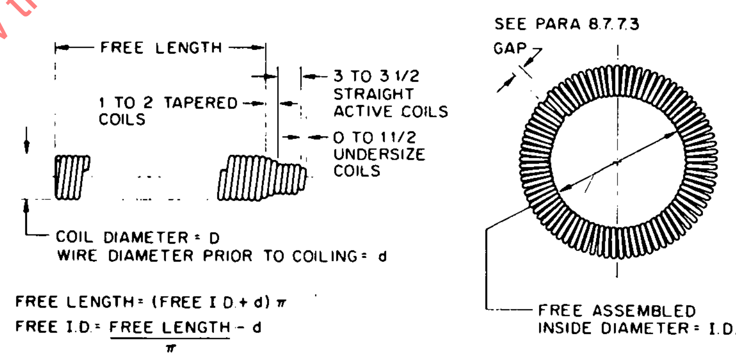


FIG. 14—SPRING NOMENCLATURE

ϕparagraph 8.7.4. Over 4 in (101.60 mm) assembled inside diameter, the free length tolerance will be ±2 wire diameters from the mean dimension. (See Fig. 14.)

ϕ 8.7.4 Assembled Free Inside Diameter

Inside Diameter		Tolerance	
in	mm	±in	±mm
0.000–2.000	0.00– 50.80	0.008	0.20
2.001–3.000	50.83– 76.20	0.012	0.30
3.001–4.000	76.23–101.60	0.015	0.38
4.001 and over	101.63 and over	Reference Only	

A spring inspection mandrel should be used to measure the assembled free inside diameter (Fig. 15). Each gage must include a minimum and maximum step. Slight out-of-roundness should be corrected by the shaft taper ahead of the NO-GO step. Springs are acceptable when:

(a) On the minimum limit step, with pressure applied, they slide smoothly without rolling.

(b) On the maximum limit step, with pressure applied, they just begin to roll.

ϕ 8.7.5 Spring windup is the tendency of a spring with its ends assembled together to deform from a flat surface. Excessive spring windup results in the spring forming a *figure 8* configuration.

Assembled garter springs that "figure 8" must snap back to their original circular form when dropped approximately 1 ft (0.3 m) onto a flat surface.

ϕ 8.8 Physical Properties of Garter Springs

ϕ 8.8.1 Initial tension is the *preload* that has been wound into the coils of a spring during the coiling operation. Initial tension can be determined by the following procedure:

(a) Cut the assembled spring opposite the point where the ends are joined.

(b) Install the free spring on test scale.

(c) Adjust test scale to straighten spring. Record position of scale.

(d) Adjust test scale to extend X inches (millimetres) and record the load "A" in ounces (grams). The X extension will be governed by the spring size. For springs less than 1 in (25.4 mm) assembled diameter, it is recommended that X be limited to 0.100 in (2.54 mm) maximum. For sizes over 1 in (25.4 mm), X should be limited to 0.200 in (5.08 mm) maximum.

(e) Extend test scale an additional X and record load "B" in ounces (grams). The total extension of the spring from its straightened position under step (c) is now $2X$.

(f) Measured initial tension is then determined by:

$$\text{I.T.} = A - (B - A) = 2A - B \text{ oz (g)}$$

ϕ 8.8.2 Calculated Initial Tension—The calculated value when added to the force developed by the spring rate equals the spring load at a specified test length.

$$T_t = T_c + R\Delta L \text{ or } T_c = T_t - R\Delta L$$

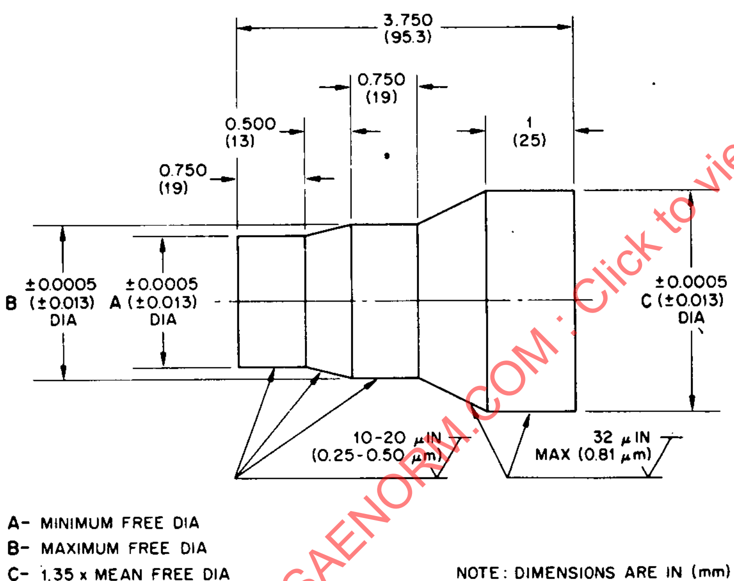


FIG. 15—SPRING INSPECTION MANDREL

where:

T_c = calculated initial tension
T_t = spring load at specified test length
R = spring rate
ΔL = change in spring length (test length − free length)

It is recommended that the calculated initial tension approach the test load as close as the design and manufacturing parameters will permit.

ϕ 8.8.3 Minimum Initial Tension—A value stated as a function of the calculated initial tension. This value establishes the lower limit of the tolerance range for the spring manufacturer in maintaining the calculated initial tension value. It is recommended that the minimum initial tension value be established as 80% of the calculated initial tension.

$$T_c \times 0.80 = \text{minimum initial tension}$$

ϕ The measured initial tension of the actual spring in paragraph 8.8.1(f) should always exceed the specified minimum initial tension value.

ϕ 8.8.4 Spring Rate—The force required to extend a spring a unit distance that is within the elastic limit of the spring. It is referred to for definition and reference only.

Refer to Fig. 16 for method of measurement.

$$\text{Spring rate oz/in (N/m)} = \frac{B - A}{\text{distance } (2X - X)}$$

ϕ 8.8.5 Spring load at specified test length is the total force (initial tension plus load created by spring rate) of the spring at a given extension. (See Fig. 16.)

Suggested tolerances are:

Up to 5 oz (142 g) = ±15% but no greater than 0.5 oz (14 g).
5 oz (142 g) and up = ±10%.

ϕ 8.8.6 Spring index is the ratio of the mean coil diameter to wire diameter. A spring should not be designed with an index less than 4 for satisfactory coiling of the garter spring.

$$\text{Spring index } C = \frac{D - d}{d} = \frac{Dm}{d}$$

where:

D = spring body diameter
Dm = mean coil diameter
d = wire diameter

ϕ 8.8.7 Nib Joint Section

ϕ *8.8.7.1 Strength*—The garter spring, when passed over a spring inspection mandrel whose large diameter is 1.35 times the mean assembled inside diameter of the spring and whose smallest diameter is less than the mean assembled inside diameter of the spring, should not separate at the nib joint. This is considered a semidestructive test. Springs used in this check should be discarded and not used in seals. (See Fig. 15.)

ϕ *8.8.7.2* Springs must remain assembled when given one-half turn in the direction of disassembly.

ϕ *8.8.7.3* Gap at the point of assembly must not exceed 2 wire diameters. (See Fig. 14.)

ϕ 8.8.8 Load Loss—The percent loss of load at a specified test length after aging the spring in its free disassembled condition in air at a temperature of 400 ± 5°F (204 ± 3°C) for a period of ½ h. The loss in load shall not exceed 0.25 oz (7 g) for springs under 5 oz (142 g) or 5% for springs over 5 oz (142 g) nominal check load. This test is intended as a check for adequate heat treatment of the spring during its manufacture.

TABLE 5—RMA OIL SEAL STANDARD SIZES, IN
(SINGLE AND DUAL LIP SPRING LOADED BONDED SEALS)

Standard Bore Sizes		Standard Shaft Sizes																		
		in																		
		0.500	0.625	0.750	0.875	1.000	1.125	1.250	1.375	1.500	1.625	1.750	1.875	2.000	2.125	2.250	2.375	2.500	2.625	2.750
		mm																		
in	mm	12.7	15.9	19.05	22.2	25.4	28.6	31.8	34.9	38.1	41.3	44.5	47.6	50.8	54.0	57.2	60.3	63.5	66.7	69.9
0.999	25.4	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.124	28.6	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.250	31.8	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.375	35.0	—	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.499	38.1	—	×	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.624	41.3	—	—	×	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—
1.752	44.5	—	—	—	×	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—
1.874	47.6	—	—	—	—	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—
2.000	50.8	—	—	—	—	—	×	×	×	—	—	—	—	—	—	—	—	—	—	—
2.125	54.0	—	—	—	—	—	—	×	×	×	—	—	—	—	—	—	—	—	—	—
2.250	57.2	—	—	—	—	—	—	—	×	×	×	—	—	—	—	—	—	—	—	—
2.374	60.3	—	—	—	—	—	—	—	×	×	×	×	—	—	—	—	—	—	—	—
2.502	63.6	—	—	—	—	—	—	—	—	×	×	×	—	—	—	—	—	—	—	—
2.623	66.6	—	—	—	—	—	—	—	—	—	×	×	×	×	—	—	—	—	—	—
2.750	69.9	—	—	—	—	—	—	—	—	—	—	×	×	×	×	—	—	—	—	—
2.875	73.0	—	—	—	—	—	—	—	—	—	—	—	×	×	×	—	—	—	—	—
3.000	76.2	—	—	—	—	—	—	—	—	—	—	—	×	×	×	×	—	—	—	—
3.125	79.4	—	—	—	—	—	—	—	—	—	—	—	×	×	×	×	×	—	—	—
3.251	82.6	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×	×	×	—	—
3.371	85.6	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×	×	×	—
3.500	88.9	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×	×	×
3.623	92.0	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×	×

Standard Bore Sizes		in																			
		2.625	2.750	2.875	3.000	3.125	3.250	3.375	3.500	3.625	3.750	3.875	4.000	4.250	4.500	4.750	5.000	5.250	5.500	5.750	6.000
		mm																			
in	mm	66.7	69.9	73.0	76.2	79.4	82.6	85.7	88.9	92.0	95.3	98.4	101.6	107.9	114.3	120.0	127.0	133.4	139.7	146.0	152.4
3.623	92.0	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
3.751	95.3	×	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
3.875	98.4	—	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
4.003	101.7	—	—	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
4.125	104.8	—	—	—	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
4.249	107.9	—	—	—	—	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—	—
4.376	111.2	—	—	—	—	×	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—
4.500	114.3	—	—	—	—	×	×	×	×	—	—	—	—	—	—	—	—	—	—	—	—
4.626	117.5	—	—	—	—	—	×	×	×	×	×	—	—	—	—	—	—	—	—	—	—
4.751	120.7	—	—	—	—	—	—	—	×	×	×	—	—	—	—	—	—	—	—	—	—
4.876	123.9	—	—	—	—	—	—	—	—	×	×	×	—	—	—	—	—	—	—	—	—
4.999	127.0	—	—	—	—	—	—	—	—	×	×	×	×	—	—	—	—	—	—	—	—
5.125	130.2	—	—	—	—	—	—	—	—	—	—	×	×	—	—	—	—	—	—	—	—
5.251	133.4	—	—	—	—	—	—	—	—	—	—	×	×	×	—	—	—	—	—	—	—
5.375	136.5	—	—	—	—	—	—	—	—	—	—	—	×	×	—	—	—	—	—	—	—
5.501	139.7	—	—	—	—	—	—	—	—	—	—	—	—	×	×	—	—	—	—	—	—
5.625	142.9	—	—	—	—	—	—	—	—	—	—	—	—	×	×	—	—	—	—	—	—
5.751	146.1		—	—	—	—		—			—	—	—	—	×	×	—	—	—	—	—
6.000	152.4	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×	—	—	—	—
6.250	158.8	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×	—	—	—
6.375	161.9	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×	—	—	—
6.500	165.1	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×	—	—
6.625	168.3	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×	—	—
6.750	171.5	—	—	—	—	—	—	—	—	—	—	—			—	—	—	—	×	×	
6.875	174.6	—	—	—	—	—	—		—	—	—	—	—	—	—	—		—	×	×	—
7.000	177.8	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	—
7.125	180.9	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×	×
7.500	190.5	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	×

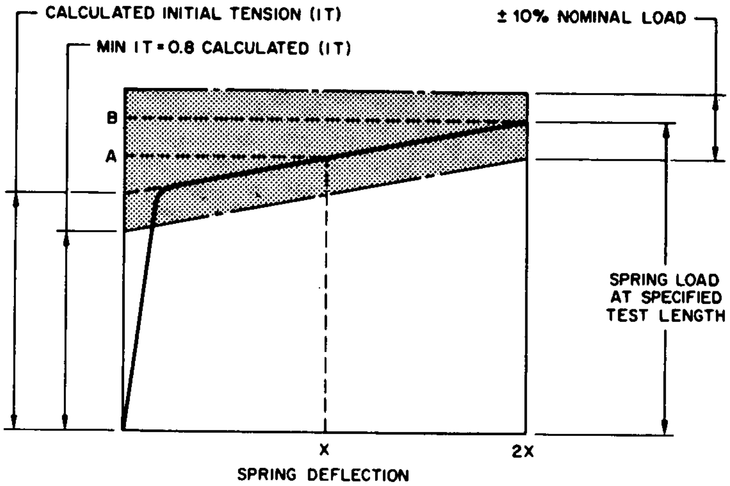


FIG. 16—SPRING FORCE DIAGRAM

ϕ 8.9 Visual Radial Lip Variations

ϕ 8.9.1 Identification of Visual Radial Lip Variations—The variations described and shown in this section are the result of manufacturing and handling variations and in no way are intended to describe variations as a result of service. Typical visual variations for elastomeric radial lip type seals are shown in Fig. 18. These variations are shown in the areas where they normally occur. They may occur at other areas on the seal and on other seal types. Reference radial seal nomenclature and glossary SAE J111c (April, 1979), for definition of terms.

ϕ 8.9.2 Location of Visual Elastomeric Variations—The location of an elastomeric variation is very significant. Fig. 18 shows the surface area on a single lip and a dual lip seal by letter. Each area is defined.

ϕ *8.9.2.1 Area A—Primary Lip Contact Area*—The lip of any single lip seal is the primary lip. If a multiple lip seal is used for fluid retention, the fluid retaining lip is the primary lip and contaminant excluding lips are secondary lips. If a multiple lip seal is not used to retain a fluid, the primary lip would be the most critical lip. A seal could have two primary lips if two fluids are separated by a multiple lip seal. The maximum wear anticipated on the primary lip establishes the boundaries of Area A.

ϕ *8.9.2.2 Area B—Secondary Lip Contact Area*—The secondary lip or lips are intended for dirt, dust, or other contaminant exclusion. The maximum wear anticipated on the secondary lip or lips establishes the boundaries of Area B.

ϕ *8.9.2.3 Area C—Flex Sections and Other Critical Areas*—The flex section, the remainder of the primary and secondary lips not included in Areas A and B, and any other flexing and critical areas constitute Area C.

ϕ *8.9.2.4 Area D—Seal O.D.*—The seal O.D. where contact is provided with the mating bore constitutes Area D.

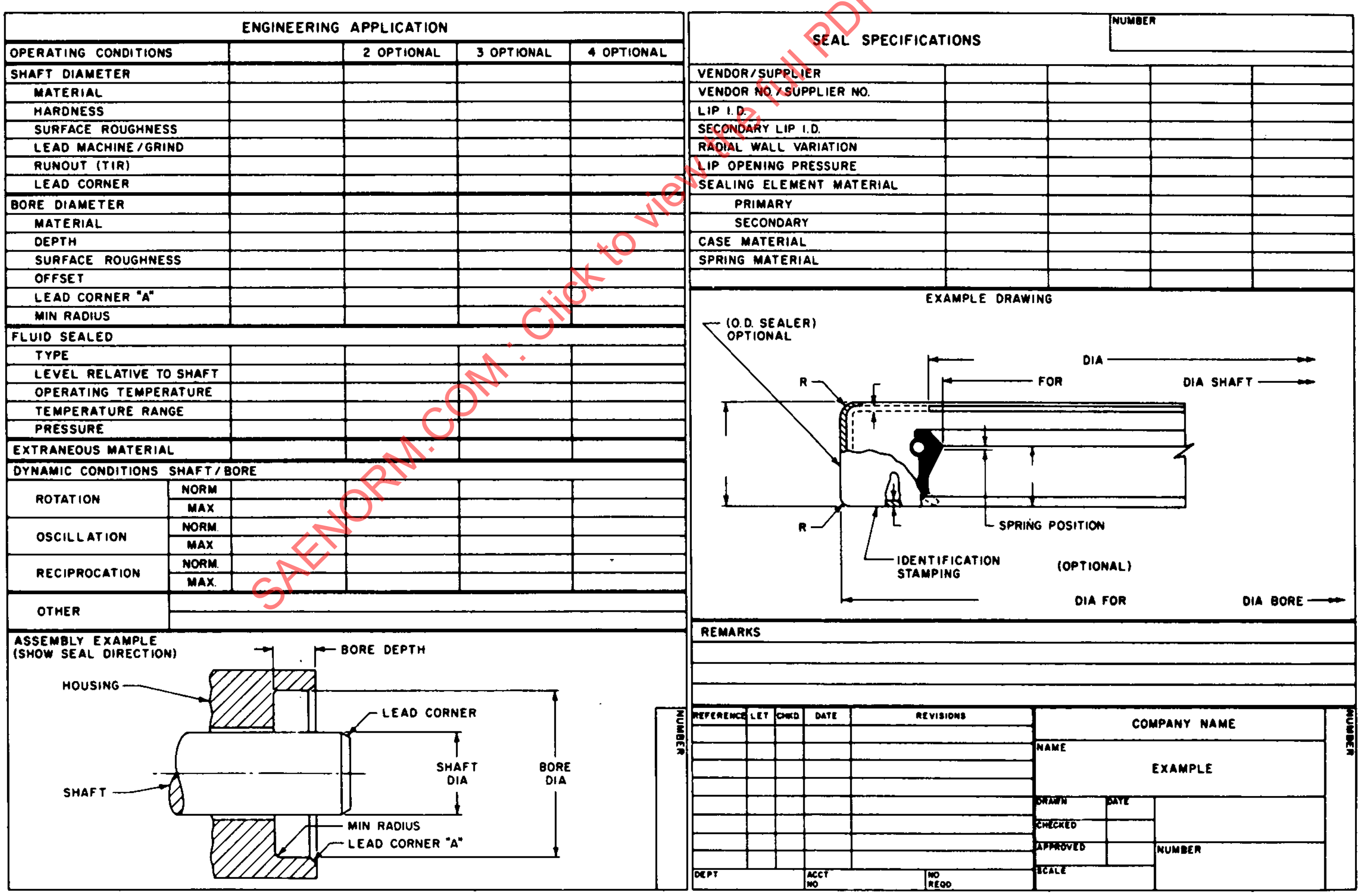


FIG. 17—OIL SEAL DRAWING FORMAT