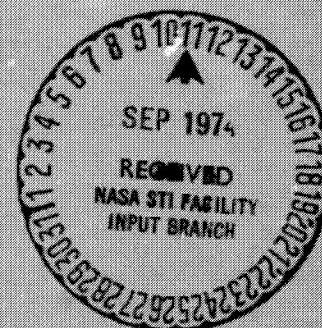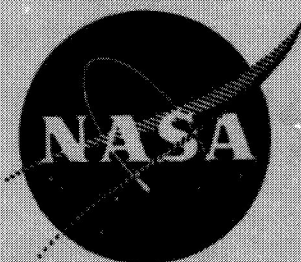NASA CR 120997
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DESIGN GUIDE FOR HELICOPTER TRANSMISSION SEALS

BY

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SUMMARY

Four seal types commonly used in helicopter transmissions are the conventional elastomeric lip seal, the hydrodynamic lip seal, the circumferential shaft seal, and the face seal.

Sealed pressures are usually less than 3.44 N/cm^2 (5 psi) and seal lubrication and cooling is usually provided by splash of the lubricant against the seal.

For splash lubrication and cooling it is recommended that the conventional lip seal be limited to sliding speeds of 1219 meters per minute (4000 ft/min) for sealed pressures less than 0.69 N/cm^2 (1 psi). For higher sealed pressures, the recommended sliding speeds are lower and are listed under the section titled Recommended Practices. The maximum recommended sliding speed for the hydrodynamic lip seal is 2133 meters per minute (7000 ft/min) when sealed pressures are 0.69 N/cm^2 1 psi. As with the conventional lip seal, the recommended maximum speed is reduced as the pressure is increased; these limits are stated in the Recommended Practices section. Higher speeds and pressures are feasible when special provisions have been made to cool the runner material that is under the seal lip. Lip seals have advantages of easy assembly, compact size and low cost.

The circumferential seal has high-speed capability (probably over 4572 meters per minute (15,000 ft/min)). However, the pressure capability is low (about 0.69 N/cm^2 (1 psi)) unless the seal is well drained and protected by lubricant windbacks and slingers. Circumferential seals are attractive because of easy assembly and compact size; however, they cost much more than lip seals.

The face seal has high-speed capability probably over 6096 meters per min (20,000 ft/min), and much higher pressure capability than the maximum pressures found in helicopter transmissions. Face seals have disadvantages of higher costs than lip seals, more difficult assembly, and restricted axial movement.

FOREWORD

This design guide on helicopter transmission seals was funded by the U. S. Army Air Mobility Research and Development Laboratory. It is to be regarded as a guide in seal application and not ac U. S. Army requirements, except as may be specified in formal project specifications. It is expected, however, that this document, revised as experience indicates to be desirable, eventually will provide a basis for a uniform practice.

This design guide was prepared under the direction of R. L. Johnson, Chief Lubrication Branch, NASA Lewis Research Center; project management was by L. P. Ludwig, NASA-Lewis Research Center. The design guide was written by T. S. Hayden and C. H. Keller of the Sikorsky Division of United Aircraft Corporation. To assure technical accuracy of this document, scientists and engineers throughout the technical community participated in critical review of the text. In particular L. A. Horve, Manager, Research and Development, Chicago Rawhide Manufacturing Co.; W. C. Davidson, Product Engineer, Gould Inc., Clevite Engine Parts Division; D. Lukes, Crane Packing Co.; J. D. Symons, Senior Research Engineer, General Motors Research Laboratory, E. Taschenberg and J. Heck, Koppers Co.; A. McKibbin and R. Thomas, Pratt and Whitney Aircraft Co.; and W. Hady, Dr. R. C. Bill and Dr. J. Zuk, NASA-Lewis Research Center reviewed the monograph in detail.

Comments concerning the technical content of this design guide will be welcomed by Mr. L. P. Ludwig, NASA-Lewis Research Center, MS 23-2, 21000 Brookpark Road, Cleveland, Ohio 44135.

GUIDE TO THE USE OF THIS MONOGRAPH

The purpose of the design guide is to organize and present, for effective use in design, the significant seal experience and knowledge accumulated in helicopter transmission development and operational programs to date. It reviews and assesses current design practices, and from them establishes firm guidance for achieving greater consistency in design, increased reliability in the end product, and greater efficiency in the design effort. The document has two major sections (sections 2 and 3); these are preceded by section 1 which consists of a brief introduction and some background data. A set of references and a glossary of terms compliment the two major sections.

The state of the Art, Section 2, reviews and discusses the total design problem and identifies which design elements are involved in successful design. It describes the current technology pertaining to these elements and points out those areas in which data is lacking. When detailed information is required, the best available references are cited. This section serves as a survey of the subject, provides background material, and prepares a technological base for the Design Criteria and Recommended Practices (Section 3).

The Design Criteria, Section 3, state briefly in CAPITALS what rule, guide, limitation, or standard must be imposed on each essential design element to assure successful design. The Design Criteria can serve as a checklist for the project manager to use in guiding a seal design or in assessing its adequacy.

The Recommended Practices, also in Section 3, state how to satisfy each of the criteria. Whenever possible, the best procedure is described; when this cannot be done concisely, appropriate references are provided. The Recommended Practices, in conjunction with the Design Criteria, provide positive guidance to the practicing designer on how to achieve successful design.

Sections 2 and 3 have been organized into decimally numbered subsections so that the subjects within similarly numbered subsections correspond from section to section. The format for the Contents displays this continuity of subject in such a way that a particular aspect of design can be followed through both sections as a discrete subject.

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1.0 INTRODUCTION

The purpose of a helicopter transmission is to transmit power from the engines to the rotor system and to various accessory components. This is done by a multistage gear system, the complexity of which can be realized by considering that the present production helicopter can have up to six gearboxes with a total of 30 seal positions. The function of the seals in these transmissions is to prevent lubricant escape and to prevent ingestion of water and debris that may affect operation of gears and bearings.

The minimum life of these seals should be at least equal to the scheduled overhaul time of the transmission; this is about 1500 hours. Lubricant leakage limits are sometimes specified as being less than 5cc/hr, with zero leakage and zero ingestion of water and debris being a design goal.

Seal problems appear in the form of seal leakage. This usually is not a flight-safety problem. However, seal replacements necessatated by excessive leakage causes maintenance costs to rise and aircraft downtime to increase; and the total costs associated with seal replacement are appreciable and deserve attention.

The following steps were followed in the construction of the design guide:

   (1) Identifying the successful transmission seal design practice and applications

   (2) Documenting the existing state of the art and its shortcomings

   (3) Listing what design criteria are necessary for successful design

   (4) Recommending practices to be used to achieve successful design

The end objective of this design guide is to reduce seal leakage problems through the establishment of a uniform seal design and selection practice.

Since the purpose to this design guide is to establish a uniform practice, the guide is based on experience obtained in a wide variety of applications using lip, circumferential and face seals. Particular attention is given to the capabilities and lubrication of the various seal types. Drainage considerations, as they affect the seal operation, are discussed. Also discussed are special limitations as a result of storage requirements, quality control, installation, operation and removal.

Not all seal locations in the transmission present the same degree of difficulty, but the following list of environmental extremes provides some insight into severity of operation:

A.   External environment (air side)

   (1) Air entrained debris (dust'
   (2) Water (such as rain encountered in forward flight and salt water spray)

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B. Internal environment (oil side)

   (1) Lubricant (compatibility characteristics)
   (2) Temperature to 394 K (250°F)
   (3) Start-up; dry starts and cold starts (219K(- 65°F))
   (4) Changes in axial location of seal parts due to thermal growth
        and tolerance variation from one assembly to another (as high
        as ± 1.26 mm (±0.050 inch) in one transmission)
   (5) Shaft dynamics
        -    runout (as high as 0.762 mm (0.030 TIR) on the
             O.D. of a shaft in one transmission)
        -    axial and radial vibration (no data on this
             item available)
   (6) Speed - (4572 meters per minute (15,000 ft/min)
        maximum speed now planned)
   (7) Pressure range - some locations specified as high as 3.44 N/cm^2
        (5 psi) maximum
   (8) Lubricant flooding of the seal possible in some applications

Tables 1.1 and 1.2 summarize the current successful helicopter transmission
seal practices for lip type and carbon element seals respectively. The
seal types, component materials, operating conditions, design parameters
and leakage data are listed for each seal. The data from these table
represent a focal point of the current state of the art and recommended
practices that will be discussed in the following sections.

Many types of seals have been used by various transmission manufacturers.
At Sikorsky Aircraft, however, as noted in Tables 1.1 and 1.2, the lip and
face seals are the most commonly used. Circumferential seals have been
used in well-drained positions where surface speeds are too high for lip
seals and axial tolerance variations in location or axial displacements are
excessive for face seals. The following descriptions introduce each seal
type; a more detailed analysis is presented later in the design guide.

Conventional Lip Seal

Figure 1.1 is a typical lip seal accessory application in a main transmis-
sion (item 7 in Table 1.1). The main element is the elastomeric lip that
is stretched over the shaft. Radial loads acting normal to the shaft
surface are a result of the stretched lip and the loading spring. The bore
of the lip and the runner (shaft) surface form the primary seal.

In operation, the surfaces of the primary seal are usually separated by a
thin oil film about 1.27 to 2.54 microns (50 to 100 microinches) thick.
Since this film must be maintained an important design feature is lubricant
transport to the primary seal. For the seal depicted in Figure 1.1, an
oil-splash type lubricating system is used. The splash results from oil
being forced out of the gear and support bearing area, which is jet-lubricat-
ed. An oil drain located at the bottom of the housing between the support
bearing and the seal relieves the seal area from a pressure buildup.

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TABLE 1.1 - SUMMARY OF HELICOPTER SEAL CURRENT PRACTICES - LIP TYPE SEALS

APPLICATION	TYPE OF LIP SEAL	SEALED MEDIA	MATERIALS			RUNNER OR SHAFT DIAMETER, mm(in)	SHAFT SPEED RPM	RUBBING SPEED, m/min (ft/min)	MAXIMUM STATIC ECCENTRICITY(TIR) mm (in.)	MAXIMUM RADIAL RUNOUT (TIR), mm (in.)	MAXIMUM FLUID PRESSURE, N/cm² (psi)	AVERAGE LEAKAGE cc/hr
			LIP	RUNNER	CASE							
Accessory (1)	Conventional	MIL-L-7808 MIL-L-23699	Fluroelastomer	Case Carburized	300 Series Stainless	19.1 (.750)	6600	396 (1300)	.127 (.005)	.076 (.003)	.689 (1)	1/2
Accessory (2)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	12.7 (.500)	6600	264 (865)	.102 (.004)	.089 (.0035)	1.38 (2)	1/2
Main Rotor Shaft (3)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	254 (10.000)	185	147 (484)	.48 (.019)	.254 (.010)	0 (0)	0
Main Rotor Shaft (4)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	152 (6.000)	185	88 (290)	.406 (.016)	.19 (.0075)	0 (0)	0
Accessory (5)	Conventional	MIL-L-7808 MIL-L-23699	Fluroelastomer	Case Carburized	300 Series Stainless	27 (1.0625)	7150	610 (2000)	.241 (.0095)	.125 (.005)	2.07 (3)	1
Accessory (6)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	15.9 (.625)	4270	213 (700)	.178 (.007)	.0635 (.0025)	.689 (1)	0
Accessory (7)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	36.5 (1.4375)	6023	690 (2265)	.357 (.014)	.152 (.006)	1.38 (2)	1
Tail Take-off (8)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	76.2 (3.000)	3060	732 (2400)	.14 (.0055)	.114 (.0045)	2.07 (3)	2
Input (9)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	58.7 (2.3125)	3060	564 (1850)	.165 (.0065)	.089 (.0035)	1.38 (2)	2
Output (10)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	58.7 (2.3125)	2310	425 (1400)	.178 (.007)	.114 (.0045)	.689 (1)	1
Input (11)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	92 (3.625)	2310	671 (2200)	.152 (.006)	.102 (.004)	2.07 (3)	2
Output (12)	Conventional	MIL-L-7808 MIL-L-23699	Fluoroelastomer	Case Carburized	300 Series Stainless	112.5 (4.437)	793	280 (920)	.38 (.015)	.216 (.0085)	0 (0)	0
Input (13)	Hydrodynamic	MIL-L-21260	Fluoroelastomer	Chrome Plated	300 Series Stainless	91.4 (3.600)	6600	1900 (6230)	.203 (.008)	.127 (.005)	1.38 (2)	2
Output (14)	Hydrodynamic	MIL-L-21260	Fluoroelastomer	Case Carburized	300 Series Stainless	127 (5.000)	2800	1120 (3670)	.254 (.010)	.127 (.005)	2.75 (4)	2
Clutch (15)	Hydrodynamic	MIL-L-7808 MIL-L-23699	Fluorelastomer	Flame Plated	300 Series Stainless	60.4 (2.375)	8200	1550 (5100)	.203 (.008)	.076 (.003)	.689 (1)	1/2

TABLE 1.2 - SUMMARY OF HELICOPTER TRANSMISSION SEAL CURRENT PRACTICES-CARBON ELEMENT SEALS

APPLICATION	SEAL TYPE	SEALED MEDIA	MATERIALS PRIMARY ELEMENT	MATING SURFACE	SECONDARY SEAL	CASE	SEAL DIAMETER (A), mm (in.)	SHAFT SPEED (rpm)	RUBBING SPEED m/min (ft/min)	SPRING LOAD (B) N (lbf)	PRESSURE BALANCE (C)	FACE WIDTH mm(in.)	AXIAL OPERATING RANGE mm(in.)	MAXIMUM FLUID PRESSURE N/cm²(psi)	AVERAGE LEAKAGE cc/hr
Input Shaft (1)	Circum-ferential	MIL-L-7808 MIL-L-23699 MIL-L-21260	USG 84	4340 Steel Chrome Plated	-----	416 Stainless	42.7 (1.6875)	20,000	2680 (8800)	13.35(3) Axial 2.224(.5) Radial	----	2.35 (.05) Ring	----	.38 (2)	1/2
Input Shaft (2)	Face	MIL-L-7808 MIL-L-23699	P658RC, CCA 72 or USG 39	440C Stainless	Fluoro-elastomer	300 Series Stainless	105 (4.115)	13600	4460 (14650)	80 (18)	d.	2.38 (.094)	±.785 (±.031)	2.76 (4)	5
Output Shaft (3)	Face	MIL-L-7808 MIL-L-23699	P658 RC, CCA 72 or USG 39	440C Stainless	Fluoro-elastomer	300 Series Stainless	77 (3.037)	6023	1460 (4800)	62.2 (14)	d.	1.98 (.078)	±.785 (±.031)	3.44 (5)	2
Input Shaft (4)	Face	MIL-L-7808 MIL-L-23699	P658, RC, CCA 72 or USG 39	440C Stainless	Fluoro-elastomer	300 Series Stainless	89.5 (3.537)	6023	1700 (5590)	71 (16)	d.	2.38 (.094)	±.785 (±.031)	.38 (2)	5
Input Shaft (5)	Face	MIL-L-7808 MIL-L-23699	658 RC, CA 72 or USG 39	440C Stainless	Fluoro-elastomer	300 Series Stainless	55.7 (2.200)	6023	1060 (3480)	57.8 (13)	d.	2.38 (.094)	±.785 (±.031)	.38 (2)	1
Input Shaft (6)	Face	SATO 35	P658 RC, CCA 72 or USG 39	440C Stainless	Fluoro-elastomer	300 Series Stainless	101 (3.984)	9000	2890 (9500)	75.5 (17)	d.	2.38 (.094)	±1.19 (±.047)	3.44 (5)	3
Accessory (7)	Face	SATO 35	P658 RC, CCA 72 or USG 39	440C Stainless	Fluoro-elastomer	300 Series Stainless	111.5 (4.392)	4600	1615 (5300)	80 (18)	d.	2.38 (.094)	±1.19 (±.047)	.38 (2)	3
Accessory (8)	Face	SATO 35	P658 RC, CCA 72 or USG 39	440C Stainless	Fluoro-elastomer	300 Series Stainless	53 (2.093)	8000	1330 (4380)	57.8 (13)	d.	1.57 (.062)	±.785 (±.031)	2.07 (3)	1
Clutch APP (9)	Face	MIL-L-7808 MIL-L-21260	USG 39 or P658 RC	416 Chrome Plated	Fluoro-elastomer	416 Stainless	68.1 (2.690)	8200	1760 (5780)	26.7 (6)	1.05	1.775 (.070)	±.76 (±.030)	2.07 (3)	3
Clutch APP (10)	Face	MIL-L-7808 MIL-L-23699	P658 RC USG 39	440C Stainless	Fluoro-elastomer	416 Stainless	95.5 (3.770)	8200	2460 (8080)	31.2 (7)	1.	1.65 (.065)	±.76 (±.030)	.38 (2)	1

NOTES:

(A) The seal diameter is circumferential seal bore or a face seal primary ring nose inside diameter.
(B) The circumferential seal radial load is a tension spring load.
(C) See section 2.1.2 for a definition of pressure balance.

SPLASH
LUBRICATION

AIR SIDE

LIP SEAL

LUBRICANT DRAIN
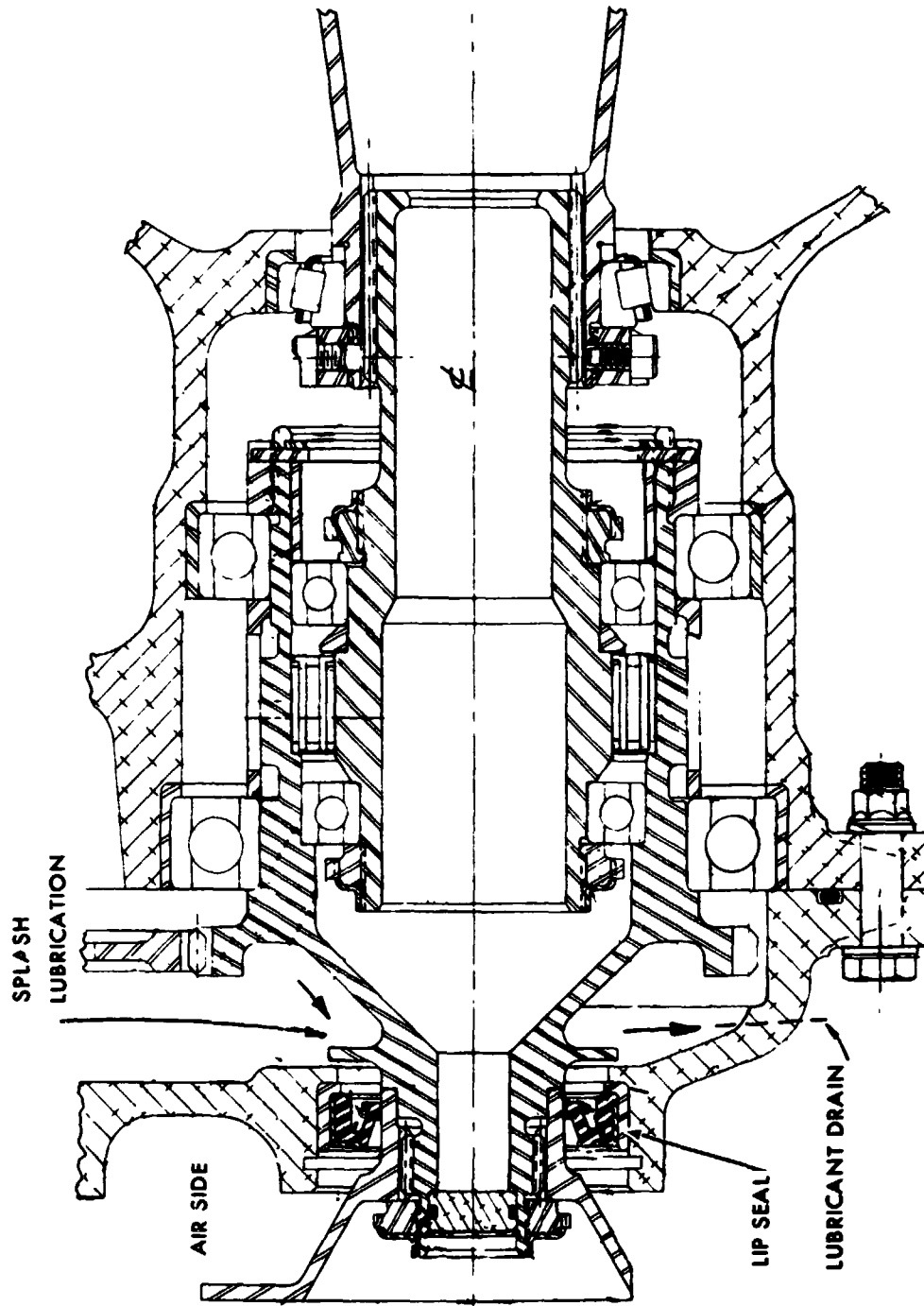
FIGURE 1.1 TYPICAL LIP SEAL APPLICATION

As can be seen from Table 1.1, conventional lip seals have been used successfully in applications where rubbing speeds are relatively slow (under 732 meters per minute (2400 ft/min)).

These lip seals in Sikorsky Aircraft accessory, main rotor shaft and tail drive shaft applications usually constitute over 80 percent of the seal applications, but less than 5 percent of the leakage problems in transmissions.

Hydrodynamic Lip Seals

Figure 1.2 is a sketch of a high-speed input shaft where a hydrodynamic lip seal is used (item 13, Table 1.1).

The hydrodynamic lip seal is essentially a conventional lip seal modified by the addition of molded ribs (helical grooves) on the air side of the seal. These molded ribs pump back any lubricant that leaks out.

The lip seal in this application has replaced a face seal in a retrofit program. As with the conventional lip seal, lubricant is supplied in splash form from the jet-lubricated bearing and is removed by a drain in the bottom of the housing.

Generally the lubricating film is similar to that of a conventional lip seal except that the helical groove pumping action allows the use of lighter lip loads, resulting in lower frictional torque and under lip temperatures and inhibits leakage. Thus, higher speeds than a conventional lip seal are obtainable.

Circumferential Seals

Figure 1.3 (item 1, Table 1.2) shows a circumferential seal used in a high-speed input application of a production main gearbox.

It is a three-ring design (segmented carbon rings) that is radially loaded by springs to the shaft surface. Relative rotation occurs between the shaft and the bore of the carbon rings.

As can be seen from the figure, the seal envelope is larger than the seal itself. The original seal used in the application was a face seal, but numerous failures showed that the axial shaft movement with respect to the housing was excessive for the face seal. The seal is splash lubricated, and the oil is then drained back into the sump by external plumbing.

As with the lip seals, a lubricating fluid film is probably established between the shaft and the carbon rings. Circumferential seals have higher speed capability than lip seals and can operate under dry or partial lubrication conditions.

Face Seals

Figure 1.4 shows a conventional face seal in the high-speed input shaft
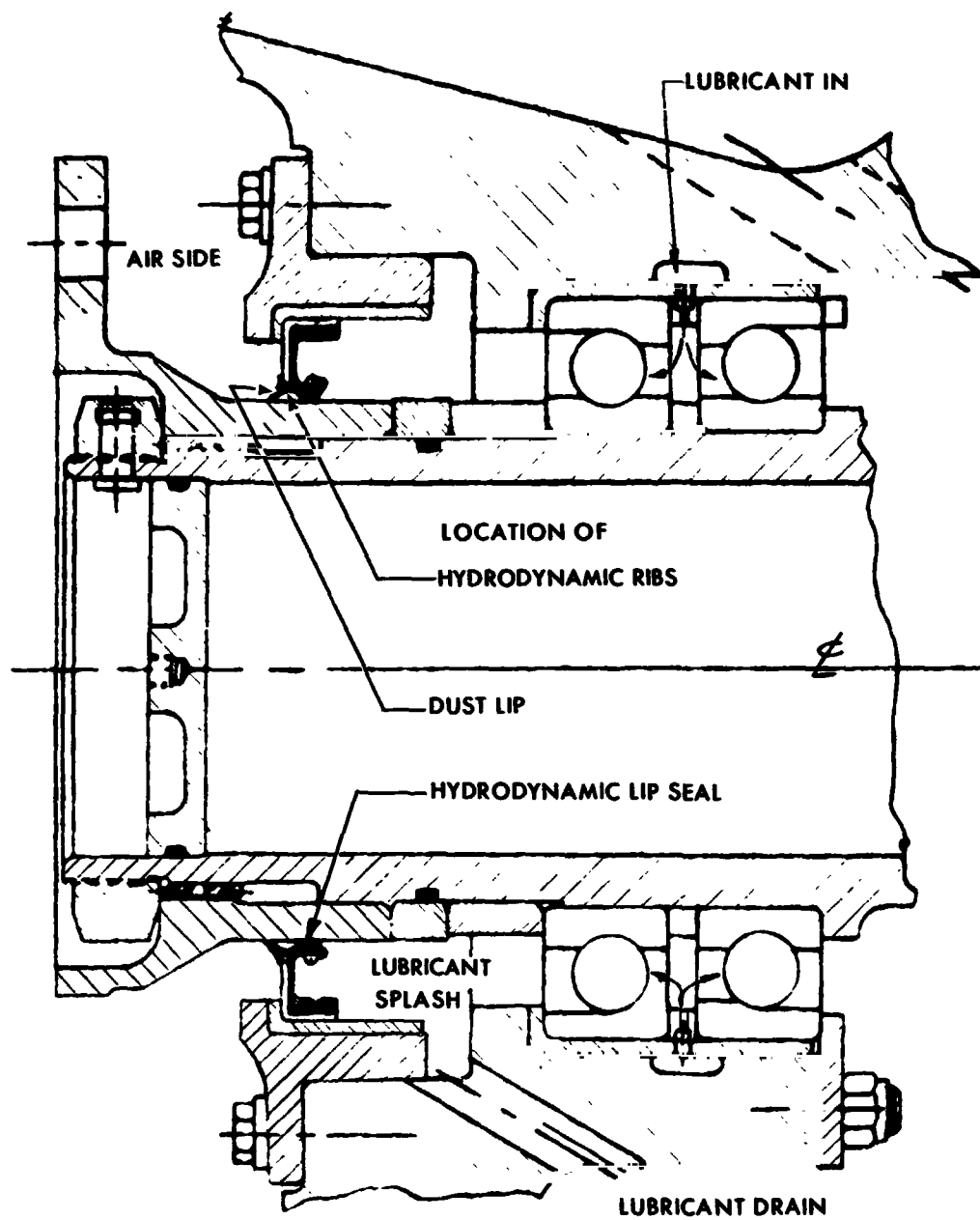
**FIGURE 1.2 TYPICAL HYDRODYNAMIC LIP
SEAL APPLICATION**

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CIRCUMFERENTIAL SEAL

AIR SIDE

LUBRICANT
SPLASH

BEARING

OIL SLINGER

LUBRICANT
DRAIN
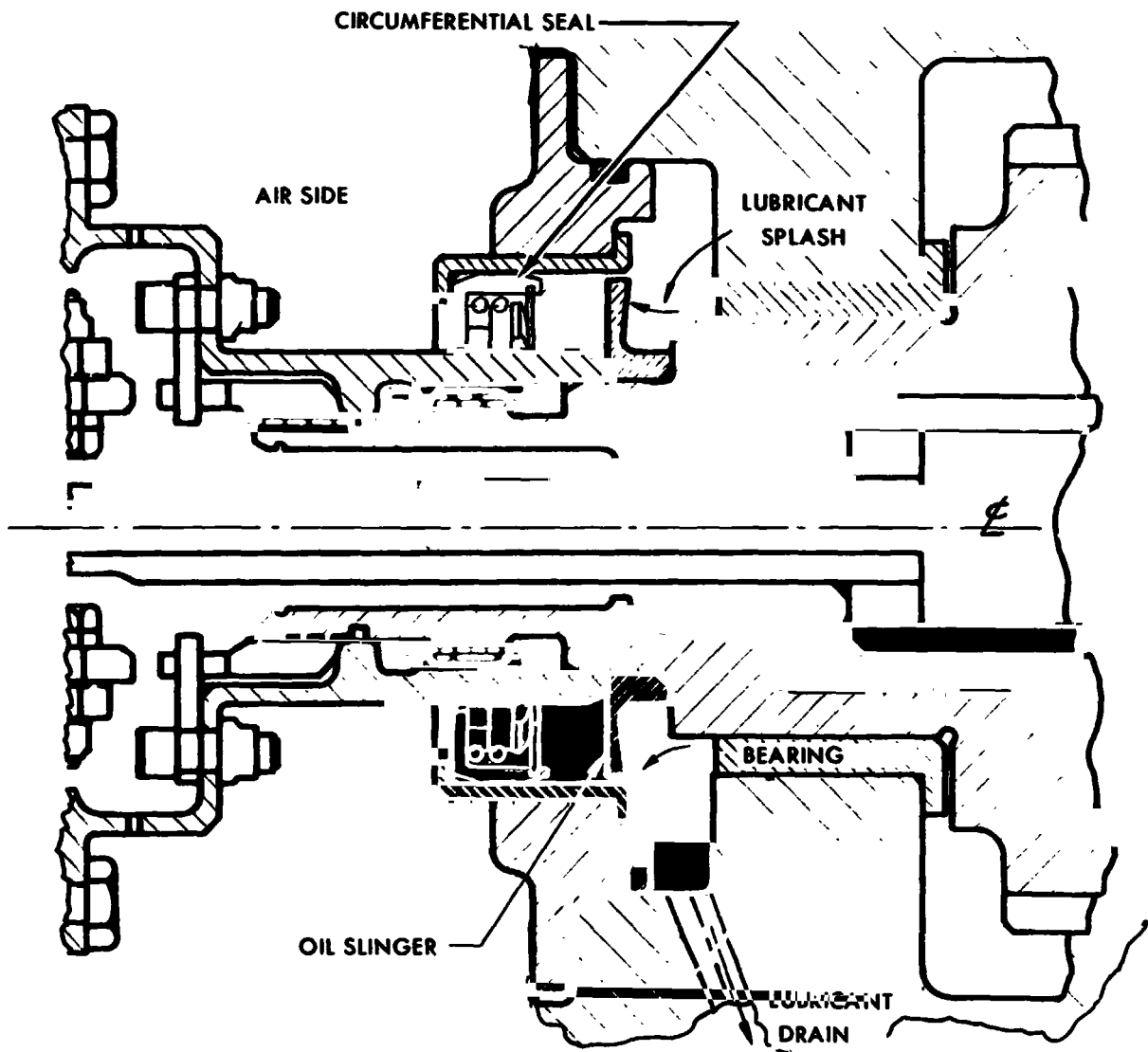
FIGURE 1.3 TYPICAL CIRCUMFERENTIAL SEAL APPLICATION
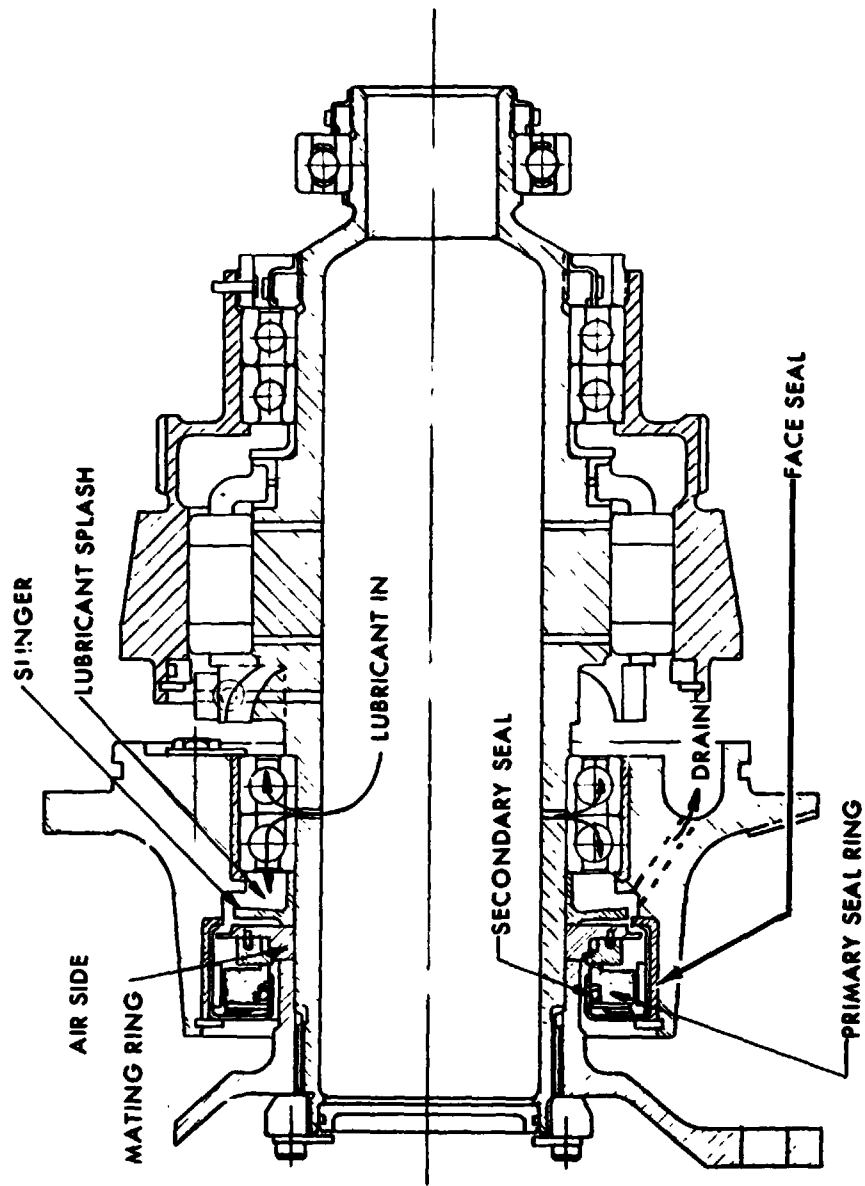
FIGURE 1.4 TYPICAL FACE SEAL APPLICATION

SLINGER

LUBRICANT SPLASH

AIR SIDE

MATING RING

LUBRICANT IN

SECONDARY SEAL

DRAIN

FACE SEAL

PRIMARY SEAL RING

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application on a Sikorsky Helicopter (item 4, Table 1.2)

This seal consists of a rotating mating ring and a non-rotating carbon-graphite seal ring flexibly attached to the seal case. A secondary seal between the primary seal ring and case permits axial motion of the primary seal ring. This axial motion is due to thermal expansion, runout of the mating ring face, and assembly tolerance stackup.

The seal is lubricated by an oil splash from the support bearing. A large drainage hole at the bottom of the housing grains oil from the seal area to an internal cored drainage line.

In transmission applications, experience suggests that an oil film exists between the mating ring and the primary seal ring. The mechanism responsible for this fluid film separation of the primary sealing surface is probably a hydrodynamic force that develops as a result of surface geometry. Face seals generally have lower leakage rates than circumferential seals and are better able to withstand oil flooding without leakage, because it is easier to accurately mate two flat surfaces than two cylindrical surfaces.

2.0 STATE OF THE ART

2.1 Seal Assembly Design

Successful seal design requires a complete definition of operating parameters. These parameters include speed, pressure, internal environment, external environment, life requirements and leakage limitations. The list shown in Table 2.1 is a format for stating the operating conditions and performance criteria for a seal application.

In general, helicopter transmission seals have similar operating conditions with large differences only in shaft rotational speed, which can vary from 20,000 rpm at the input to under 200 rpm at the main rotor shaft. Sliding velocity, which is dependent on shaft speed and diameter, currently varies from 4560 meter per minute (15,000 ft/min) to less than 152 meters per minute (500 ft/min).

In most applications, the internal environments are similar. A lubricant splash of almost zero gage pressure (a maximum of 3.44 N/cm^2 (5 psi)) at a temperature varying from 330 to 394 K (150 to 250°F) is sealed from the external environment. The lubricants used in helicopter transmissions are usually synthetic engine oils or mineral oils. Some common types are MIL-L-7808, MIL-L-23699, and MIL-L-21260. Usually the oil enters the seal area by a splash from the gears and support bearings and is drained to the sump by holes in the housing which lead to internal cored lines or external plumbing.

The external environment of the seal depends on the aircraft mission and operating area. Current operating extremes include (1) hot weather and dust, (2) low temperature(frigid) and (3) salt water mist.

Leakage limitations are dependent on the seal location and the effect of leakage on aircraft performance, but in general are less than 5cc/hr. Seal life requirements are based on the scheduled overhaul interval of the transmission components. Current gearbox TBO's range from 200 hours to 2000 hours.

As described previously, four seal types are generally used in helicopter transmissions. Additional comments on their relative advantages and capabilities are given in Table 2.2.

2.1.1 Speed Capabilities

The primary factor in the selection of the seal type to be used is sliding speed-capability. This sliding speed at the seal interface is

$$S = \pi \times D \times N \qquad (2.1)$$

where
S = sliding speed, meters per minute
D = mean diameter of the primary seal, m
N = revolutions per minute

TABLE 2.1

APPLICATION DATA

Shaft Outside Diameter
    Shaft Diameter
    Surface Finish
    Lead (RH, LH, None)
    Material
    Hardness
    Radial Eccentricity (Static)
    Radial Runout
    Out-of-Roundness
    Axial Displacement
    Axial Runout
    Shaft Speed - Range
                - Normal Operating
    Direction of Rotation (Viewed From Air Side)

Housing Bore
    Bore Diameter
    Depth
    Surface Finish
    Material

Performance Criteria
    Leakage Limitation
    Life Requirements

Operating Fluid
    Type of Fluid Being Sealed
    Temperature - Range
                - Normal Operaring
    Pressure - Range
             - Normal Operaring

External Environment

TABLE 2.2 - COMPARISON OF HELICOPTER TRANSMISSION SEALS

Seal Type	Advantage	Disadvantage	Primary Use
Conventional Lip	Low cost. Easy to install. Small space. Will tolerate eccentricity, misalignment and vibration. With proper materials, will handle most lubricant types now used.	Limited speed and pressure capability. Lip contact pressure must be controlled precisely. Will not operate dry.	Slow-speed applications including main rotor, accessory and tail rotor drive.
Hydrodynamic Lip	Capable of higher speeds than conventional lip seal. Helical grooves inhibit leakage by pumping oil back	Limited speed and pressure capability. More expensive than conventional lip seal.	For higher speeds than conventional lip seals such as an input shaft.
Circumferential	Small space. Can tolerate high temperature. Can run dry. Capable of high-speed operation.	More expensive than a lip seal. Limited pressure capability	For speeds higher than lip seal capability such as on input assemblies.
Face	Can operate at high speed, temperature and pressure. Can tolerate some misalignment and eccentricity. Can run dry.	Requires larger space envelope than lip or circumferential seals. Assembly more difficult. Axial displacement limited. More expensive than a lip seal.	For high-speed, high-pressure applications.

As previously stated, the sliding speed ranges from 4560 meters per minute (15,000 ft/min) to under 152 meters per minutes(500 ft/min). The speed capabilities of the various seal types cannot be given an absolute limit since they are dependent on other factors such as shaft runout, temperature, material, pressure and shaft vibration.

Conventional lip seals have been successfully applied at speeds to 732 meters per minute (2400 ft/min) in helicopter transmissions (Table 1.1 item 8). It is possible to run at higher speeds with lip seals, but this is dependent on controlled lubrication, cooling and dynamic factors.

Shaft outside diameter runout and heat generation in the lubricating film are major considerations in the operation of lip seals at high speeds. As the speed increases, the ability of the lip to follow the shaft runout decreases. Thus at high speeds, leakage gaps tend to form. This is termed by some as lip "camming", and others call the onset of "camming" the followability limit. Reference (1) presents data and a method of determining the followability limit. This data can be used as a guide. In general, as speed increases, greater attention must be given to seal cooling and the allowable magnitude of shaft imperfections and vibration. (The effects of imperfections are discussed in paragraph 2.1.4.) For conventional shaft tolerances and magnitude of shaft vibration, the speed limit of 1218 meters per minute (4,000 ft/min) is recommended by some manufacturers. But it is possible to operate faster with proper control of runout, temperature and vibration. Not enough published data exists to provide exact methods of determining the speed limit for a particular application.

One recent development, the hydrodynamic lip seal, has alleviated some of the problems associated with high-speed operation of lip seals. The hydrodynamic lip seal differs from a conventional lip seal by:

1. Grooves, ribs or flutes molded on the air side of the lip surface which pumps back the lubricant that leaks past the primary seal.

2. A lighter radial lip load.

With the addition of the grooves, lighter lip loads can be used and therefore, higher sliding velocities can be considered for hydrodynamic seals. Rubbing speeds of up to 3045 meters per minute (10,000 ft/min) are reported by some helicopter manufacturers, although the use of hydrodynamic lip seals in these applications is questionable from a life standpoint.

The documented applications reveal that hydrodynamic lip seals are used at higher speeds than conventional lip seals. For example, long-term successful operation has been achieved with hydrodynamic lip seals at sliding speeds of 1900 meters per minute (6,230 ft/min) (Table 1.1, item 13). The upper speed limit depends on the same factors mentioned in the preceding lip seal discussion.

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Circumferential seals have been successfully operated at 2700 meters per minute (8800 ft/min) in a production H-3 transmission application. (Experimentally, this type of transmission seal has been operated at 4270 meters per minute (14000ft/min).) Generally the carbon-graphite elements of a circumferential seal have a much higher speed capability than elastomeric lip seals because of the inherent better high-temperature properties of the material. Periods of dry starts and inadequate lubrication can be tolerated with carbon-graphite, whereas it is detrimental to lip seals. With proper attention to the elimination of lubricant flooding and with proper seal cooling, the speed limit is probably over 4560 meters per minute (15,000 ft/min).

Face seals have been successfully operated at 4460 meters per minute (14,650 ft/min) (see Table 1.2, item 2). This is not to be regarded as an upper limit, as face seals have been successfully operated at 6096 meters per minute (20,000 ft/min) in non-aviation applications. As with circumferential seals, the carbon-graphite material allows for a higher temperature in the lubricating film than elastomeric lip seals.

2.1.2 Pressure Capability

Pressure, although low, is an important parameter in the design of some seal types for helicopter transmissions. Venting of the gearbox to the atmosphere accounts for the low pressures experienced in the seal area. Pressure in the seal area is usually in the range of 0 to 3.44 N/cm^2 (0 psi to 5 psi). Only in a lubricant flooding condition where drainage is inadequate will higher pressures be recorded. Prevention of flooding and elimination of high pressures will be discussed in a later section.

Lip type seals (conventional and hydrodynamic) are limited in pressure potential. The allowable pressure of a lip seal is inversely dependent on rubbing velocity. Increased pressure tends to bellmouth the lip, causing the rubbing area of the lip to increase. This increase in lip area, in conjunction with an increased radial lip load, causes the heat generation and seal interface temperature to rise. Thus, with increasing pressure, failure eventually occurs. As shown in Table 1.1, lip type seals have been used where pressure was as high as 2.76 N/cm^2 (4 psi) when operating at 1120 meters per minute (3670 ft/min) sliding speed.

Special design features can be added to a lip seal to help compensate for increased pressure. Figure 2.1 illustrates a lip seal with the TFE stiffener ring to prevent bellmouthing. With this type of seal, lip flexibility is much less than that of a conventional lip seal, and allowable speeds are reduced.

17

STIFFENER RING

SEAL CASE

WEAR SLEEVE

GARTER SPRING

PRIMARY LIP
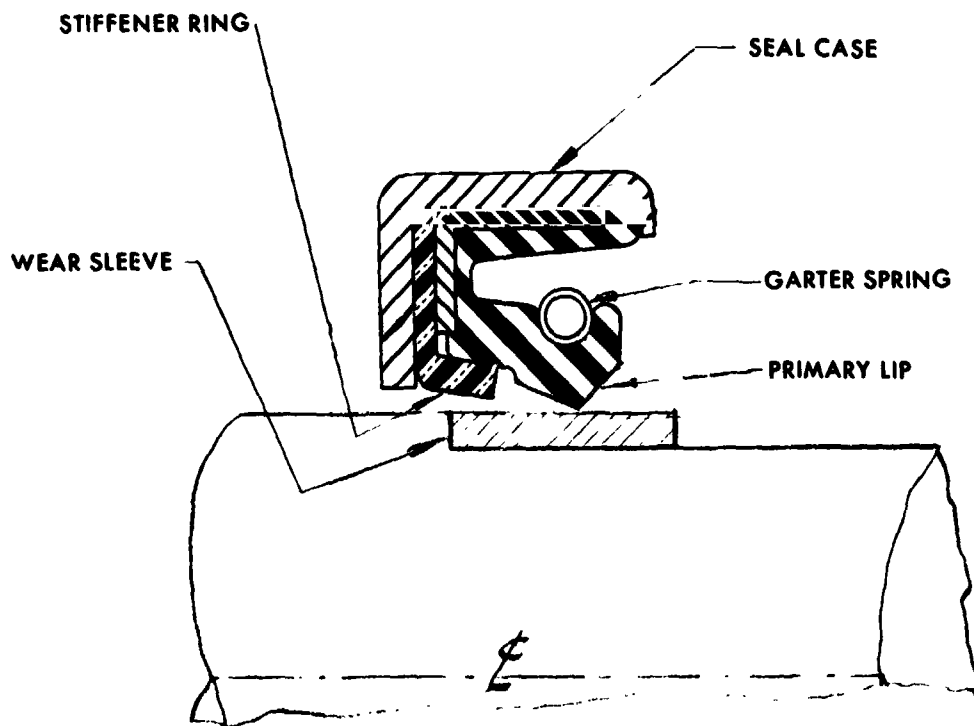
FIGURE 2.1 HIGH-PRESSURE LIP SEAL (*1)

Circumferential seals as used in helicopter transmissions have lower
pressure capability (from a leakage standpoint) than lip or face
seals. A circumferential seal generally leaks if the pressure on the
lubricant side is slightly higher than it is on the air side. How-
ever, leakage can be inhibited by the use of windbacks and slingers
to keep lubricant away from the seal, and as indicated in Table 1.2
circumferential seals nave been successfully applied at pressures up
to 1.38 N/cm^2 (2 psi).

Face seals can accommodate much higher pressure than a lip or cir-
cumferential seal. Since the transmission pressures are low, face
seals for transmissions are not pressure balanced. Although pressure
balancing is not necessary in transmission applications, the term is
introduced here to illustrate the difference between a pressure-bal-
anced seal and a seal that is not pressure balanced.

18

Seal balance is defined as

$$b = \frac{a_c}{a_o} \qquad\qquad (2.2)$$

where b = seal balance
  a_c = net area over which the sealed pressure acts
    to close the seal, m^2
  a_o = area of primary seal, m^2

The purpose of seal balancing is to reduce the hydraulic-pressure-induced contact force on the primary seal elements by decreasing the hydraulic closing force in relation to the opening force. Figure 2.2 illustrates pressure balancing. A more extensive description of pressure balancing can be found in Reference 2.

As compared to lip and circumferential seals, face seals can better adapt to pressure effects of lubricant flooding and are capable of operating at pressures much higher than those encountered in helicopter transmissions. For example, in industrial practice, unbalanced seals are used to 68.9 N/cm^2 (100 psi) and balanced seals are used for pressures up to 689 N/cm^2 (1000 psi).

2.1.3 Temperature Capability

The temperature ranges from 219 K (-65° F) in a cold start-up to 394 K (250° F) at maximum operating temperature. However, the primary seal temperature is higher than the ambient temperature because of shear of the lubricating film. Lubricating film temperatures equal to the lubricant boiling point have been observed. Even higher temperatures should be expected if the lubricant film is discontinuous and the solid surface rubbing contact occurs.

In lip type seals (conventional and hydrodynamic), the elastomeric material establishes the temperature limit Elastomers are capable of operating to temperatures up to 478 K, (400° F) and as low as 213 K (-75° F), although no one elastomer is suitable over the entire range. The temperature limits are time dependent and are also affected by compatibility with the lubricant. It should be noted that temperature, time and compatibility effects change in elastomer mechanical properties. For example, gradual hardening of the elastomer can decrease lip followability, which in turn could lead to seal leakage or failure.

Carbon-graphite rubbing elements in circumferential and face seals have a higher temperature limit than the elastomer in lip seals. Also, carbon-graphites do not have the compatibility problem associated with some elastomers; thus, mechanical properties do not change with time. Oxidation of the carbon-graphite limits the useful temperature, and this is around 533 K (500° F) for some grades; special high-temperature grades are good to 810 K (1000° F).

UNBALANCED

$$b = \frac{a_c}{a_o} \geq 1$$

BALANCED

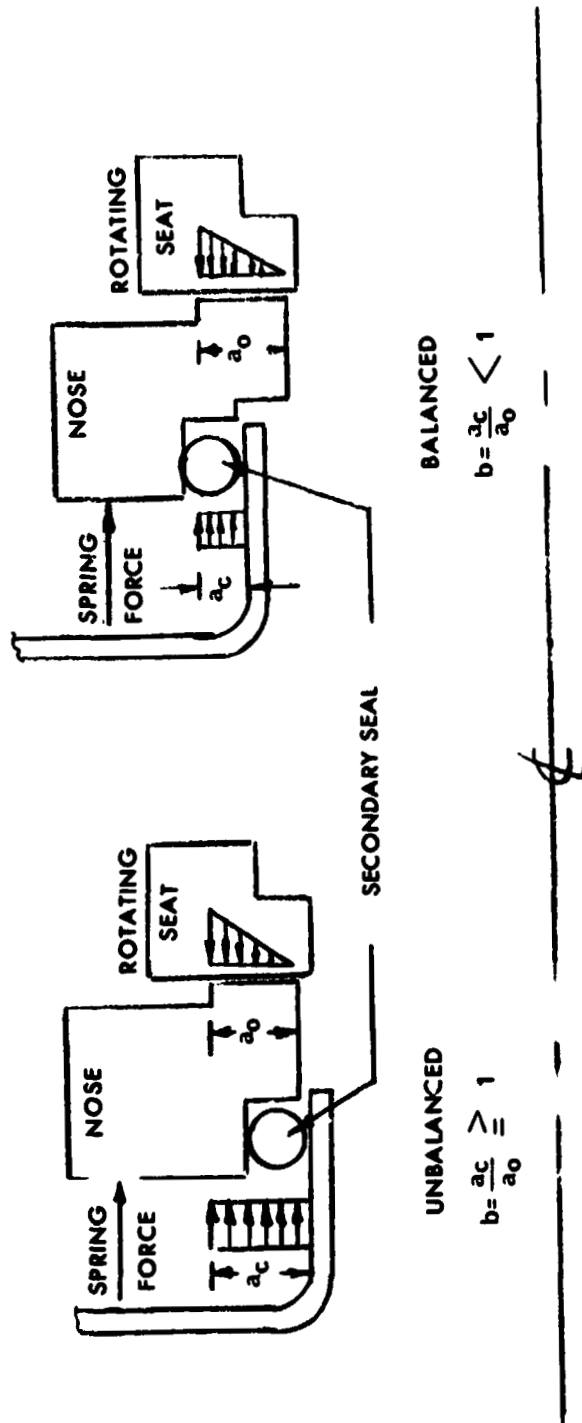$$b = \frac{a_c}{a_o} < 1$$

FIGURE 2.2 PRESSURE BALANCE
OF FACE SEAL

2.1.4 Misalignment, Shaft Runout and Shaft Roundness

Misalignment can be subdivided into the following specific types:

1. Parallel misalignment between housing bore centerline and shaft axis of rotation

2. Angular misalignment between housing and shaft centerlines

In general, parallel and angular misalignment between the housing and shaft centerlines arises from machining tolerances and elastic displacement that occur under operational loads. (In many cases angular misalignment is mainly due to gear forces.) The total misalignment (parallel and angular) that occurs under operational load is usually not known; however, the parallel misalignment under the no-load condition can be determined by inspection of assembled parts; this is referred to as static eccentricity. And an inspection of Table 1.1 reveals, a maximum static eccentricity of 0.48 mm (0.019 in.) can occur in one application.

Runout of the shaft surface, which would be detected by an indicator attached to the housing, is composed of the following:

1. Eccentricity of the shaft surface with respect to the axis of rotation

2. Shaft shirl (orbit of axis of rotation)

As with parallel and angular misalignment, the actual total runout under operational load is usually unknown. But eccentricity of the shaft surface with respect to the axis of rotation under no-load conditions is usually known (see Table 1.1) and this is referred to as radial runout.

In addition to misalignment and runout of the shaft and housing assembly, the seal assembly itself can have misalignment in the primary sealing faces due to thermally and mechanically induced displacement. Since the lubricating film is thin, these small-scale misalignments can have an effect on the lubrication of the primary seal. For a more detailed examination of seal mechanical and thermal distortion, see Reference 3.

Another factor which is extremely important with shaft riding seals is shaft roundness (lobing). Out-of-roundness is defined as the deviation of the shaft from a true circle. On a polar chart recording, it is measured as the radial distance between concentric circumscribed and inscribed circles centering the trace. This is illustrated in Figure 2.3 which is a trace of a shaft surface of a high-speed input assembly in a currently used helicopter. The out-of-roundness is approximately 17.8 microns (723 microinches). Out-of-roundness is caused by machining or assembly. Machine-oriented out-of-roundness originates from resonant frequency of the machine tool and workpiece

1.21 MICRONS
(0.00005 INCH)

SHAFT SURFACE

REFERENCE SURFACE
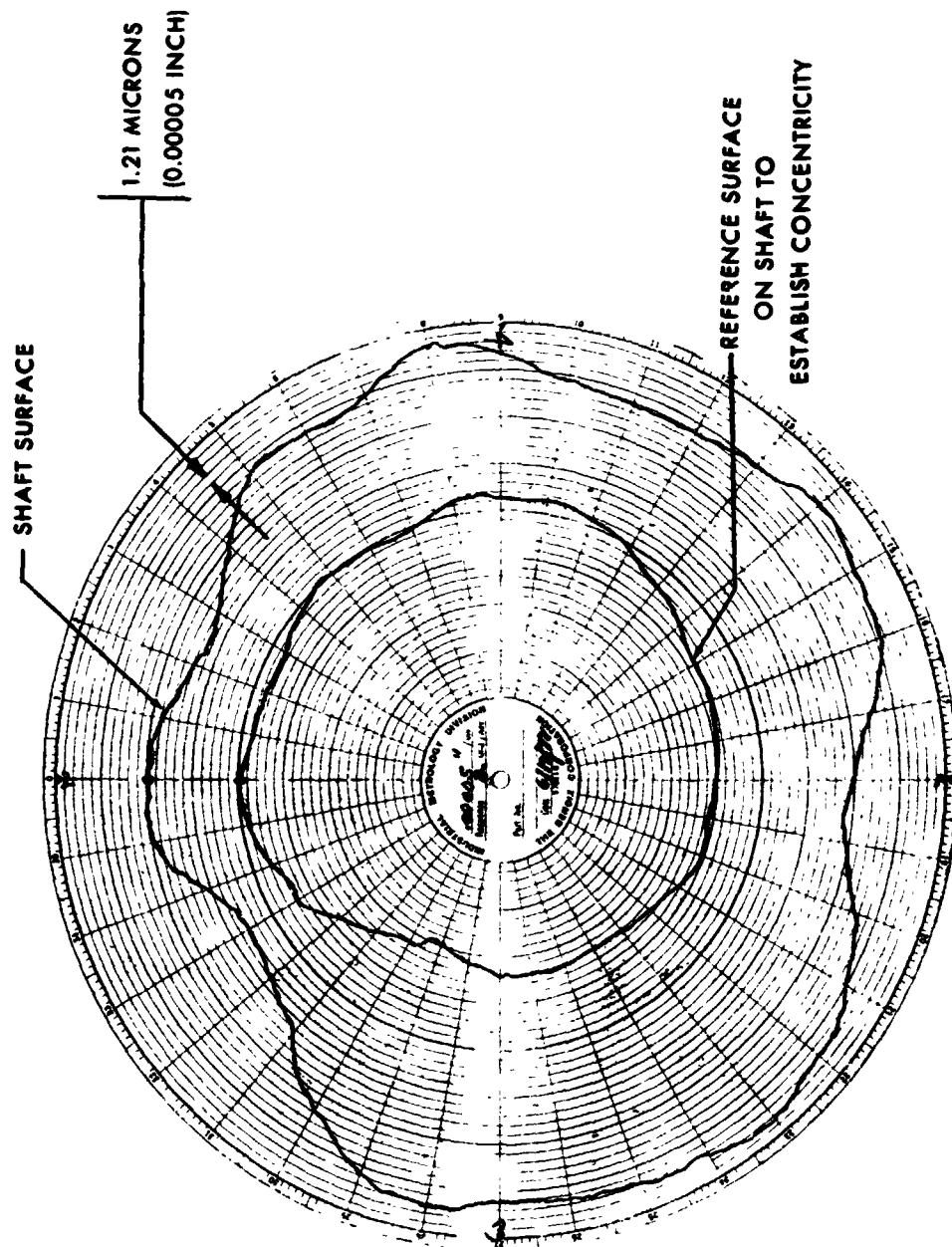ON SHAFT TO
ESTABLISH CONCENTRICITY

FIGURE 2.3 POLAR OUT - OF - ROUND TRACING OF HELICOPTER SHAFT

causing chatter, or from fixturing the workpiece. Out-of-roundness caused by chatter usually results in an uneven number of lobes. Assembly-oriented out-of-roundness can be the result of assembly of the runner with a press fit over a keyway or spline.

In lip type seals, static eccentricity tends to increase the radial lip load on one side of the seal while reducing load on the other side. This change in radial lip load is dependent, of course, on the amount of eccentricity and on the seal geometry and material properties. If the lip loading is not properly controlled (minimized), then design life will not be obtained because excessive lip loading affects lip lubrication and produces excessive underlip temperature. As noted in Table 1.1, lip seals have operated successfully at 1900 meter per minute (6230 ft/min) sliding speed with 0.203 mm (0.008 in.) static eccentricity. The allowable eccentricities are not well defined and are a function of speed in addition to the previously mentioned seal parameters. The data in the literature is not sufficient to establish maximum eccentricities.

As mentioned previously, runout of the shaft outside diameter can cause a "camming" action of the seal lip. This may increase the lubricating film thickness to a point where excessive leakage occurs. This ability of the lip to follow the shaft depends in part on the seal design and viscoelastic character of the elastomer, as well as on the shaft speed. Reference to Table 1.1 reveals that successful operation has been achieved at sliding speeds of 1900 meters per minute (6230 ft/min) with runouts of 0.127 mm (0.005 in.) TIR, and this is not the maximum limit. Reference 1 contains additional data on allowable runouts.

The out-of-roundness of the shaft can have significant effects on lip seal leakage and life. The effect of out-of-roundness is similar to that of radial runout, in that for proper operation the lip must "follow" the variation in shaft outside diameter. A general practice recommended by some users is to hold out-of-roundness within 5.08 microns (200 microinches) and to hold the number of lobes to a practical minimum.

Circumferential seals can adjust to static eccentricity since the segmented carbon rings have no radial constraint. The amount of static eccentricity that can be tolerated is determined by the radial clearance between antirotation pins, the carbon-graphite segments, and the garter springs. Normally, a circumferential seal is designed for 0.76 mm (0.030 in.) static eccentricity. However, runout promotes hydrodynamic lift-off of the carbon segments, and this increases the leakage. (Lubricant flooding of the seal will aggravate the hydrodynamic lift-off.) A literature search did not reveal any data on limiting magnitude of runout or misalignment for a circumferential seal. However, experimental data on one type of circumferential seal revealed excessive leakage at runouts of 0.076 mm (0.003 in.) when running at 2740 meters per minute (9000 ft/min). This data agrees with the general observation that circumererential seals tend to leak

if runouts are not minimized.

Eccentricity and radial runout by themselves have no noticeable effect on face seals since the seal interface is parallel to the direction of eccentricity. With poor lubrication, some fretting damage may appear. However, angular misalignment can have an influence on seal performance. The combination of eccentricity and angular misalignment in face seals has been shown by Findlay (in Reference 4) to produce a pumping action. This pumping action can be either inward or outward depending on the relationship of eccentricity and misalignment and the direction of rotation. These experiments show that orienting the radial eccentricity for inward pumping reduced hydrostatic leakage at low speed and caused reversed leakage at high speed.

Sufficient data is not available to establish limits on eccentricity and misalignment magnitude. As with lip seals, the allowable runout and misalignment decrease as speed increases. From a practical standpoint, there are some face seals that operate at misalignments (face runouts) of .0127 mm/mm of diameter (.0005 in/in of diameter). Also, a two-piece mating ring, Figure 2.30, eliminates clamping distortion and thus helps to minimize misalignment problems associated with face seals.

2.1.5 Axial Location and Axial Runout

The possible operating axial locations can be determined from a tolerance stackup of parts of the assembly. Consideration should also be given to axial thermal growth.

For shaft riding seals (lip type and circumferential seals), the only limitation imposed by axial travel is to insure that the shaft geometry and surface conditions are suitable for seal operation over the axial distance traveled.

Face seals are designed to accommodate a finite amount of axial travel. To compensate for large axial motions and variations in location, the allowable axial travel can be increased by using low spring rates; this tends to minimize axial load variations. Another method of reducing the effect of spring load variation is to shim out the tolerance stackup at each assembly. This method is not recommended, since human error is introduced as a variable. As shown from Table 1.2 current face seals have axial operating ranges up to ±1.19 mm (±0.047 in.).

Side runout of the shaft support rolling contact bearings causes an axial vibratory motion. Since the side runout of bearings is small (expecially in high-speed operation where close tolerance bearings are used), the axial runout has not been found to be detrimental to seal performance, although its effect in regard to heat generation could become significant in shaft contacting seals.

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2.1.6 Vibration Considerations

Helicopter transmissions are complex systems combining many different vibrating frequencies. These include shaft rotational frequencies and gear mesh frequencies. Shaft rotational frequencies are low, in the range of 3 cps to 350 cps. The vibrations caused from these frequencies are usually due to eccentric running or unbalance of the shaft. A maximum 10g vibration has occurred in this frequency range with large displacements.

Gear mesh frequencies as high as 5400 cps can excite vibrations up to 45 g with small displacements. These vibrations are audible and can excite vibrations in housings and other hardware such as seals in both the axial and radial direction of a seal.

Marginal lubrication of the seal interface can excite a circumferential vibration due to the stick-slip effects. This is not a common occurrence in transmission seals, since lubrication is usually adequate.

No special provisions for damping are made in lip type seals. However, the elastic properties of the lip does provide damping, and the damping of the seal from the housing can be provided by rubber bonded on the outside diameter of the seal case.

The construction of the circumferential seal is such that considerable damping is obtained by the axial loading of the carbon-graphite elements against the case flange (frictional damping). Excessive torsional vibration (or stick-slip condition) can cause the anti-rotation pins to wear into the seal segments. Rectangular pin blocks or metal shields on the segments have been used effectively in eliminating this "pin fretting".

The primary carbon-graphite element in face seals is damped by the use of elastomeric secondary seals (O-rings, wedges, V-cups). The elastomer positioned between the seal case and primary element damps out housing vibration in both the axial and radial direction.

2.1.7 Environmental Limitations

Although helicopters operate in a fairly dirty and corrosion-producing environment, the transmission and drive shafts are usually covered with a sheet metal cowling that provides some protection. However, because of the need for surface cooling of the transmission housing, some seals are exposed in this dirty and corrosion-producing environment. Environmental contamination and temperature extremes can have a severe effect on seal performance.

Lip seals are especially vulnerable to unfavorable environmental conditions. Dirt and dust accumulating at the lip/shaft interface cause abrasion of the rubber lip and possibly scoring of the runner surface. This abrasion increases seal leakage and leads to premature

failure. The pumping action of a hydrodynamic lip seal adds to the induction of harmful contaminants into the seal contact area. Moisture in the atmosphere can also be detrimental to lip seal performance if the exposed portion of the shaft or runner is not a corrosion-resistant steel or protected by a plating.

To control the entry of foreign material, external barriers should be used. The barrier could be a separate wiper seal, (Figure 2.4) or a dirt exclusion seal molded integral with the primary seal (Figure 2.5). The exclusion seal has little lubrication and can cause considerable abrasion in a dirty environment. These conditions can often lead to rapid wear of the lip, causing the exclusion lip to be nonfunctional. Exclusion seals normally have short, sturdy lips with enough flexibility to compensate for shaft radial runout. Radial load between lip and shaft is usually maintained by lip interference, although in extremely dirty environments, springs are sometimes used. The material of these seals must have good abrasion resistance and low friction. Common exclusion seal materials are leather, carboxylated nitrile, polyurethane and TFE.

Low-temperature operation of lip type seals affects both the elastomer and the lubricant. The lip has reduced flexibility and physical property changes; while the lubricant's viscosity increases, causing increased viscous shear forces at the seal lip/shaft interface. In severe cold-weather helicopter operations (military requirements are 219 K (-65°F)), seal leakage sometimes occurs during startup; but after the gearbox warms up, the seals regain their flexibility and generally perform satisfactorily.

At low temperatures, radial runout of the shaft is more likely to cause leakage by reducing the flexibility of the seal lip. From a flexibility standpoint, silicone rubber has been established as the best low-temperature elastomer, with fluoroelastomers only slightly better than nitriles and polyacrylates.

The effect of environment on circumferential and face seals with regard to both contamination and low temperature has been shown to be negligible when operating under protective cowlings. If solid contaminates enter the seal interface in the form of dust, dirt and salt crystals (from the evaporation of salt water), severe abrasive wear will occur. The use of stainless steel or chrome-plated steel mating rings and shaft runners prevents corrosion of vital sealing elements. In a dirty environment, relatively soft runners and mating rings will wear due to abrasion; therefore, chrome-plated surfaces or hardened surfaces (>55 R_c) should be used. Generally, the harder the surface, the longer will be its life under abrasive conditions.

2.1.8 Seal Envelope Requirements

The available spaces for the seal and assembly requirements will affect the choice of the type of seal. The method of holding or locking the seal or seal components for location purposes is also an

AIR SIDE

FIGURE 2.4 SEPARATE WIPER EXCLUSION SEAL (*1)

SEAL CASE

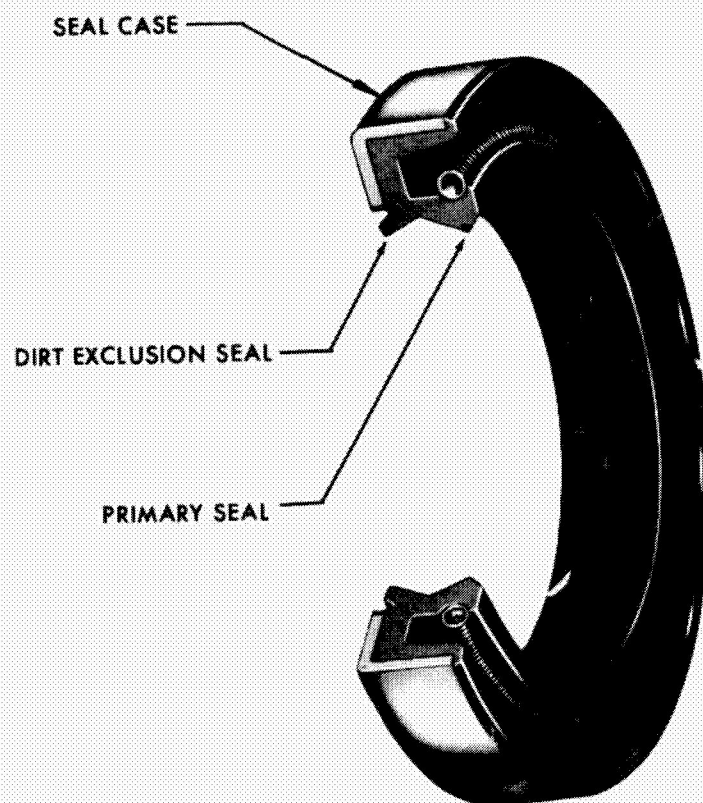DIRT EXCLUSION SEAL

PRIMARY SEAL

FIGURE 2.5 EXCLUSION SEAL MOLDED
INTEGRAL WITH PRIMARY SEAL (*1)

influencing factor. The rotating seal component must be securely held in place, with little distortion of the sealing surface. This can be accomplished in a variety of ways, including press fits, O-rings, and clamping. But all of these methods will produce some distortion and increased out-of-roundness or waviness. Current practice is to push the runner or mating ring over an O-ring. The O-ring serves two functions; it drives the mating ring and is a static seal. In some applications, a drive flange also serves as a seal runner, using splines as the driving mode. In other applications, such as a main rotor shaft, stress considerations will not allow the use of threads and O-ring grooves on the shaft. In these cases, runners have either been press fit or bonded on the shaft with excellent results.

Not all lip and circumferential seals operate on shaft runners; many operate on the shaft itself (see Figures 1.2 and 1.3). The advantage of operating on the shaft is less accumulation of tolerances (runout and out-of-roundness). The advantage of a sleeve (wear ring) is that it can be readily and inexpensively replaced in case of abrasive wear.

Usually transmission seal housings are comprised of a light-weight material (magnesium or aluminum), a press-fitted hardened steel liner in a lightweight housing, or a hardened steel seal carrier mechanically attached to the main housing. Most seal cases are made from either steel or stainless steel, although lightweight materials such as aluminum, are sometimes used. Stainless steel seal cases are preferred due to their corrosion resistance.

The seal housing is required to locate and hold the nonrotating component of the seal, which is usually the lip, circumferential, or face seal itself. To accomplish this task, various methods are currently being used. The most often used method is to press fit the seal case in the housing; this prevents axial movement and serves as an antirotation lock.

Since the thermal growth of the seal housing is usually greater than that of the seal case due to a difference in coefficients of expansion, press-fit requirements must include calculations of change in fit due to temperature. Depending on the housing configuration, two-ring or three-ring calculations are used in the analysis. The interference fit between the seal case and housing must be maintained at elevated operating temperatures to prevent static leakage and spinning in the housing. Seal distortion problems caused by housing press fits can be relieved by using O-rings on the seal outside diameter. The amount of O-ring squeeze will determine the allowable thermal growth, while still maintaining a static seal. Lip seals with rubber bonded to the case outside diameter can perform the same function as an O-ring. Figure 2.6 shows typical housing requirements for a lip, circumferential and face seal. For a press-fit design, the surface finish should be 1.52 microns (60 microinches) or less to prevent leakage. Bore sealers can be used if the surface finish is rougher.

15/30 DEG

1.5~2.25 mm
(0.06/0.09 in)

THIS CORNER MUST
BE BURR FREE

DIAMETER CONCENTRIC
WITH SUPPORT BEARING
BORE WITHIN .0254 mm
(.001 in.) TIR

PERPENDICULAR TO SEAL
BORE WITHIN .0508mm
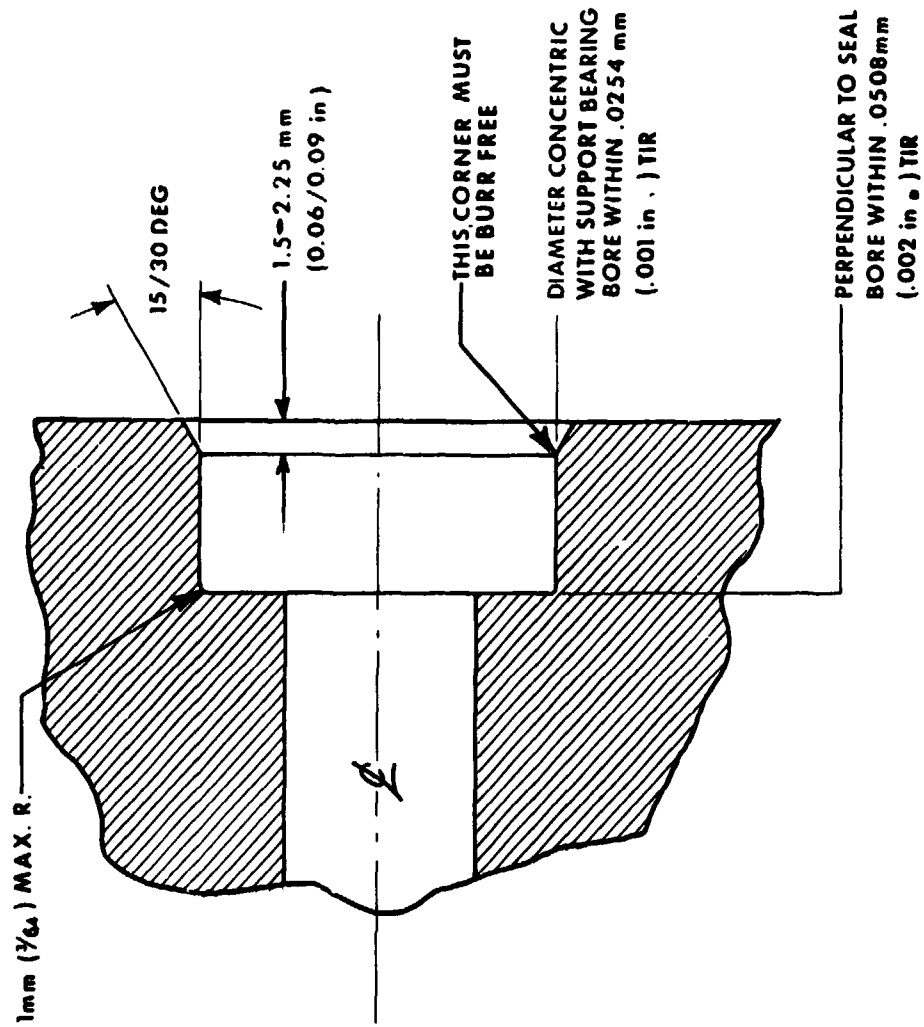(.002 in.) TIR

1mm (³/₆₄) MAX. R.

℄

FIGURE 2.6 RECOMMENDED TRANSMISSION HOUSING REQUIREMENTS (*4)

As a fail-safe measure to prevent the seal from moving axially due to an oversized bore or undersized seal outside diameter, a ctain- ing or snap ring is normally used.

There is no standard cross-sectional space requirement for any one seal type. The application design requirements will influence not only the type of seal but the space requirement of the seal. Application requirements of a lip seal might require increased axial space for a dirt exclusion seal for operation in a very dirty environ- ment or an extended beam length of the primary lip to compensate for a large radial runout. A face seal might require a longer spring to increase its operating range. Generally, sections are as small as possible to minimize weight.

Some general guidelines are as follows:

(1) Lip seal envelopes generally have a square cross section. This cross section increases with increased shaft diameter. Some typical helicopter lip seal sections are listed in Table 2.3.

TABLE 2.3

Lip Seal Envelopes

Shaft Diameter - mm	Cross Section Radial x Axial - mm
15.9 (0.625 in.)	9.5 x 9.5 (.375 in. x .375 in.)
25.4 (1.0 in.)	9.5 x 9.5 (.375 in. x .375 in.)
76.2 (3.0 in.)	12.7 x 12.7 (.500 in. x .500 in.)
114 (4.5 in.)	12.7 x 12.7 (.500 in. x .500 in.)
165 (6.5 in.)	12.7 x 15.9 (.500 in. x .625 in.)
254 (10.0 in.)	19 x 19 (.750 in. x .750 in.)

(2) Circumferential seal envelopes are similar in construction and size to radial lip seals.

(3) Radial face seals normally require a greater axial length than lip or circumferential seals. Table 2.4 represents pertinent envelope dimensions from the ASLE standard for face seals (Reference 5). These should be regarded only as suggested sizes for standardization purposes.

TABLE 2.4

Face Seal Envelopes (mating ring not included)

Shaft Diameter		Housing Bore		Width	
(mm)	(in.)	(mm)	(in.)	(mm)	(in.)
25.4	(1.000)	41.2	(1.625)	16.6	(.656)
31.8	(1.250)	47.6	(1.875)	16.6	(.656)
38.1	(1.500)	55.5	(2.1875)	17.5	(.688)
44.5	(1.750)	63.5	(2.500)	19.1	(.750)
50.8	(2.000)	70.	(2.750)	19.1	(.750)
57.2	(2.250)	77.6	(3.0625)	23.8	(.938)
63.5	(2.500)	85.7	(3.375)	24.6	(.968)
70.	(2.750)	92.	(3.625)	24.6	(.968)
76.3	(3.000)	100.	(3.9375)	27.	(1.062)
82.6	(3.250)	108.	(4.250)	27.	(1.062)
89.	(3.500)	114.	(4.500)	27.	(1.062)
95.3	(3.750)	122.	(4.8125)	27.	(1.062)
101.6	(4.000)	130.	(5.125)	30.2	(1.88)

2.1.9 Leakage Limitations

A successful seal design is not necessarily one that does not leak, but one whose leakage does not effect component operation or cause inconvenience to operators and maintenance personnel. Current leakage limitations are as high as 5 cc/hr. The acceptable limit depends on the seal location, and a current design goal is to keep leakage rates less than 1 cc/hour.

Face seals with lubricant at the outside diameter of the primary seal (Figure 2.7) are used more often than face seals with lubricant at the inside diameter of the primary seal, (Figure 2.8). The reason for this is that centrifugal forces on the lubricant tend to retard leakage if the lubricant is at the primary seal outside diameter but will increase primary seal leakage if the lubricant is at the inside diameter.

2.1.10 Life Considerations

Currently, helicopter transmission seal life requirements are from a few hundred house to 2000 hours. Future transmission overhaul interval goals will be 3000 hours or more. Life requirements for seals are based on the scheduled overhaul interval of each transmission component. Transmission seals are seldom reused, since seal removal generally produces damage; therefore, design lives higher than maximum transmission overhaul life would not be beneficial. The selection of materials for the primary seal components is of extreme

PRIMARY SEAL

ROTATING SHAFT

CARBON (NON-ROTATING)

LUBRICANT SIDE

SEAT (ROTATING)
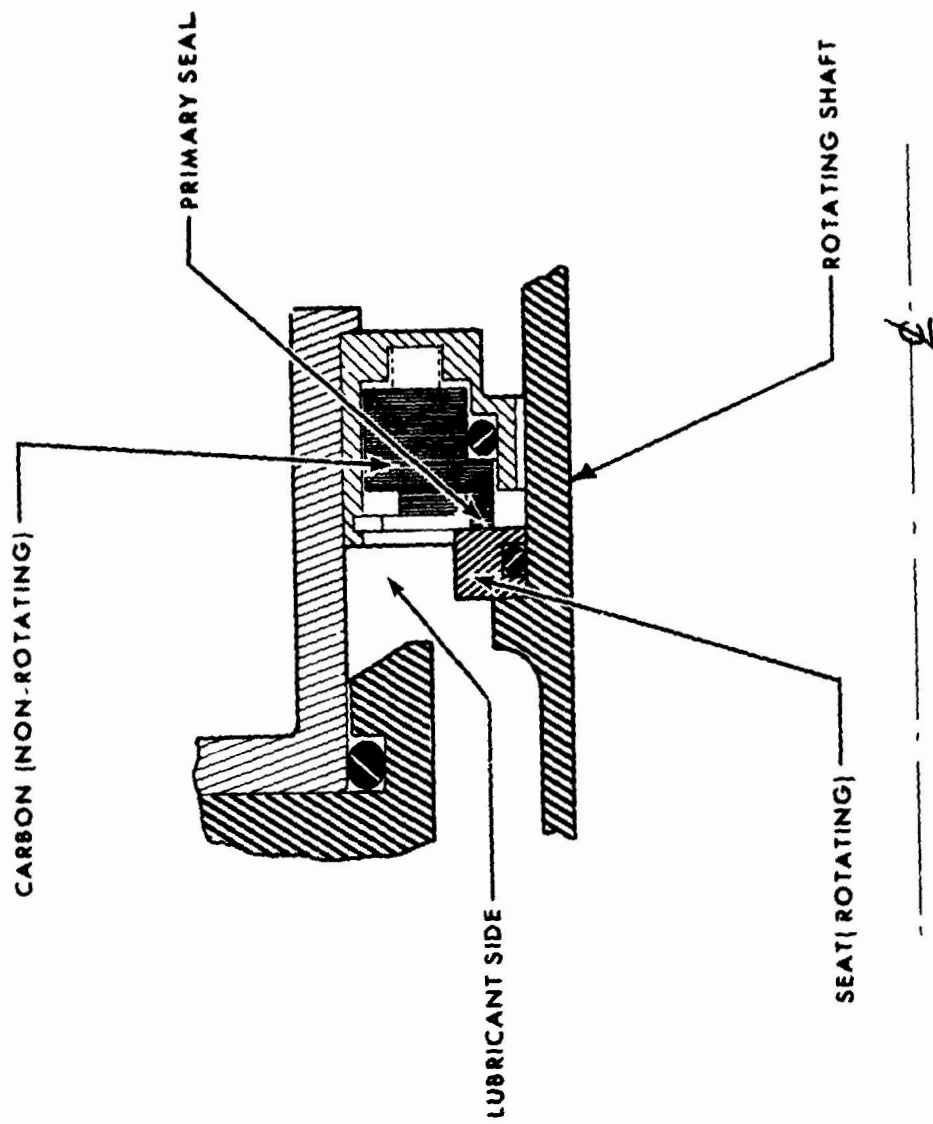
FIGURE 2.1 FACE SEAL WITH LUBRICANT AT O.D. OF PRIMARY SEAL (*1)

33

CARBON (ROTATING)

ROTATING SHAFT

S

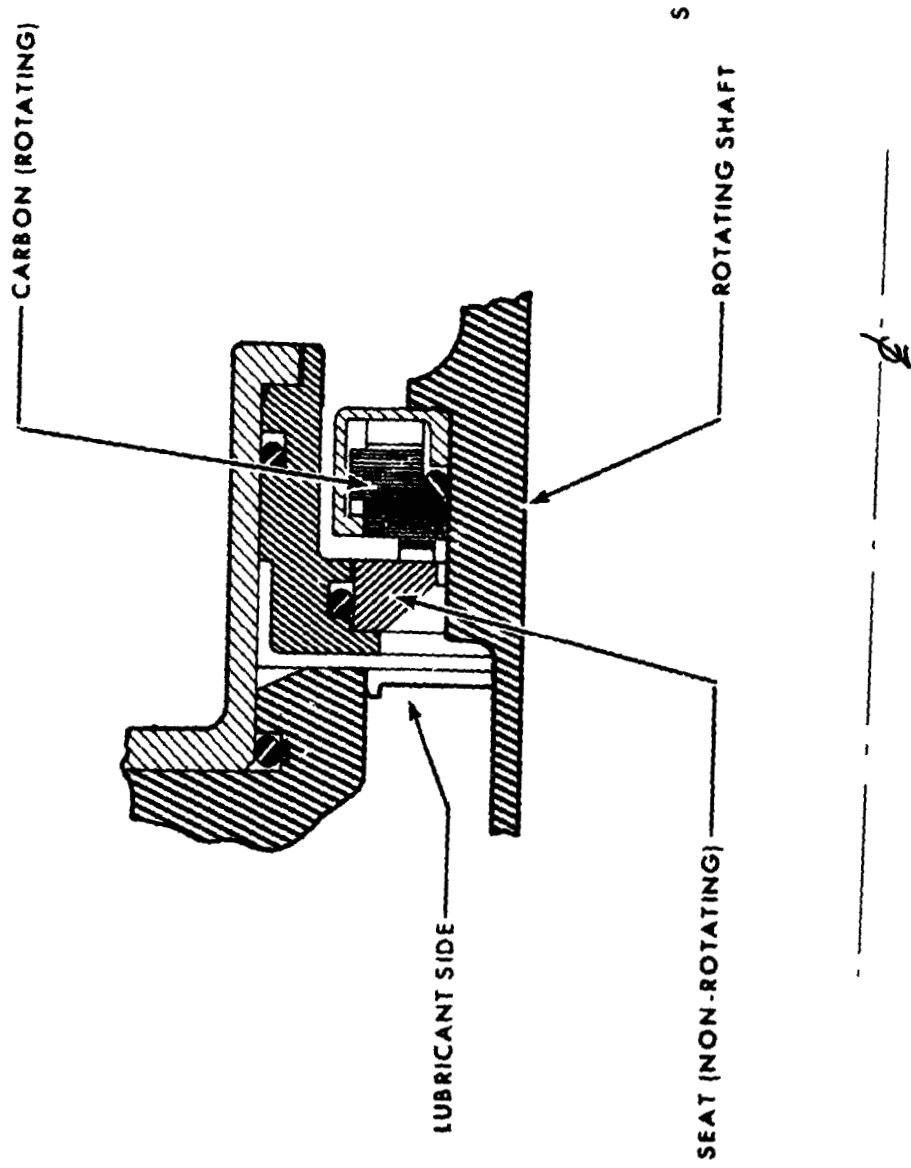LUBRICANT SIDE

SEAT (NON-ROTATING)

FIGURE 2.8 FACE SEAL WITH LUBRICANT AT I.D. OF PRIMARY SEAL (*1)

importance. In helicopter transmissions, the two most often used
material couples are carbon-graphite/hardened stainless steel and
fluoroelastomer/hardened steel. In each case, the materials in each
couple have different hardnesses.

Reference 6 presents a mathematical theory of rubber lip seal life
and correlation with controlled laboratory tests. The theory states
that the life of a lip seal is directly proportional to its capacity
to absorb frictional power dissipation and inversely proportional to
the rate of energy generation. The life equation developed is

$$L_s = \frac{BD}{M\omega^2} \qquad\qquad (2.3)$$

where L_s = life, hours
B = life constant
D = shaft diameter, mm
M = friction torque, newtons xmm
ω = angular velocity, radians/hour

The seal life constant, B, which depends on seal material, shaft
material, shaft finish, eccentricity, runout, temperature and type
of lubricant, must be determined from test. Then equation (2.3)
can be used to compare the seal life of a population of seals under
similar operating conditions once the life of one lip seal of that
population is determined from test.

The problem of seal wear is mitigated if the sliding surfaces are
not in rubbing contact. Sealing in this manner is done by either
establishing a full fluid film or a close clearance in the primary
seal. Since no rubbing occurs, seals of this type have wear only at
start and stop and from abrasion of hard particles and debris in the
lubricant. Most successful lip circumferential and face seals operate
as a full fluid film lubrication with sliding contact only at start
and stop. (This is discussed in paragraph 2.1.11.)

2.1.11 Thermal and Lubrication Considerations

Seal performance in regard to leakage and wear is affected by lubrica-
tion and by related heat generation. Methods of predicting the heat
balance are currently dependent on prior experience at similar operat-
ing conditions. The "measuring stick" for performance limits is
usually expressed as a PV factor, in which P is pressure and V is
rubbing speed. The PV factor should be considered only as a guide
in regard to heat generation. In the literature, there is general
disagreement on PV factor limiting values, because PV factor is
dependent on material, seal design, application, fluid properties
and other parameters.

Pressure is sometimes defined as the pressure of the sealed fluid
and sometimes as the contact pressure. Contact pressure seems the

logical choice for helicopter applications since the sealed pressure is very low.

There are basically three different models of lubrication (energy dissipation) in the primary seal. These are as illustrated in Figure 2.9: (1) full film lubrication, which is characteristic of slow-speed containment of viscous fluids; (2) dry sliding contact, which is found when sealing nonviscous fluids, such as in gas turbine shaft seals; and (3) mixed or boundary lubrication, which is usually exhibited in high-speed applications. Boundary lubrication represents a combination of the first two and is not easily amenable to analysis or usable mathematical models. The general case of full film lubrication can be divided into two special cases: fill film lubrication, in which the boundary surfaces are rigid and lubricant viscosity is not a function of film pressure, and the second is elastohydrodynamic lubrication, where high contact pressure causes increased viscosity and deformation between contact surfaces. Both of these types of lubrication propably occur in lip. circumferential, and face seals as applied to helicopter transmissions.

Seal designed for full film lubrication generally must operate for long periods. Seals of this type may leak since no direct contact of seal surfaces takes place; however, the leakage may only be in vapor form (due to boiling) and therefore not noticeable.

In typical geared transmissions for helicopters, the shaft seals are exposed to spray or splash of a lubricant. This lubricant comes into contact with the edge of the primary seal. For circumferential and face seals, surface energy forces then cause the lubricant to migrate (infiltrate) between the surfaces of the primary seal. It is important to visualize to what extent the lubricant film has migrated at static conditions, that is, before start-up. For example, the primary sealing faces of a face seal, even though lapped flat, do not form a solid barrier to migration of the lubricant. Actually, at static conditions, the primary faces will be touching only on a few local areas, while the majority of the primary seal surfaces will not be in solid contact. Surface energy causes the lubricant to migrate into the gap between the surfaces of the primary seal. Thus, a continuous lubricant film is present at the start of rotation. (The extent to which this migration exists for a lip seal has not been determined.)

There is considerable evidence that the lubricant film between the primary surfaces of a properly lubricated seal (lip, hydrodynamic lip, circumferential and face types) is very thin, in the range of 1.25 to 2.5 microns (50 to 100 microinches). Thus, any shaft motion or surface variation on this same order can affect the lubrication of the primary face. For example, a wavy surface could induce hydrodynamic lubrication forces that act to maintain separation of the sealing surfaces; or small seal mutating motion could exert a pumping effect that may either increase leakage or work against leakage (Reference 4).

FILMTHICKNESSS
SEVERAL TIMES
GREATER THAN
SURFACE
ROUGHNESS

OIL FLOW
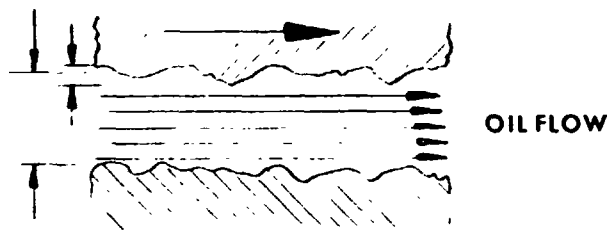
FULL FILM LUBRICATION

FILM THICKNESS
SAME ORDER
AS SURFACE
ROUGHNESS
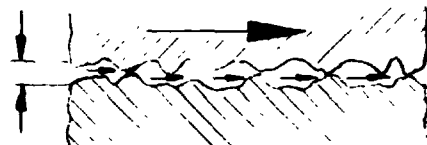
BOUNDARY LUBRICATION

BOUNDARY
FILMS

VISCOUS PROPERTIES
OF OIL
!NEFFECTIVE

DRY SLIDING LUBRICATION

FIGURE 2.9   SEAL LUBRICATION MODELS

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Although there is a vast amount of data suggesting that a fluid film does exist, the mechanisms by which the fluid film maintains surface separation have not been definitely identified.

The energy dissipation or heat generation for shear of a fluid film between two surfaces under full film lubrication is

$$P_e = \mu \left(\frac{D}{2}\mu\right)^2 \frac{A_e}{H} \qquad (2.4)$$

where P_e = energy dissipation
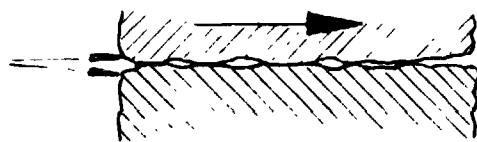 μ = fluid viscosity
        A_e = wetted area of the sealing face
        H = film thickness
        D = primary seal diameter
        ω = angular velocity

There are some restrictions that must be placed on this equation, since the formula states that as the film thickness approaches zero, the power approaches infinity. Actually, as the film thickness becomes very small and the speed remains high, the lubricant will start to vaporize and the formula is no longer valid.

Some of the parameters governing equation 2.7 which are not known exactly are:

1. Fluid viscosity
2. Film thickness
3. Wetted area

The fluid viscosity in the primary seal probably varies considerably throughout the film. The film temperature probably varies with film thickness, as well as across film width, which makes the determination of the viscosity difficult. The fluid film thickness is usually unknown. However, it probably ranges between 1.25 and 2.5 microns (50 and 100 microinches). Also, the wetted area of the sealing surfaces can be smaller than the apparent contact area. Thus, computations on energy dissipation will contain uncertainties and therefore should be correlated with experimental data.

Dry sliding energy dissipation is

$$P_e = \pi \zeta FDN \qquad (2.5)$$

where P_e = energy dissipation
        π = coefficient of friction in dry sliding
        F = normal force on seal surfaces
        D = mean diameter of seal surfaces
        N = revolutions per minute

The normal force on the mating surfaces can be fairly well predicted for a low-pressure environment such as a helicopter transmission. The force is basically made up of the spring load (radial lip loads with lip seals), inertial forces, and secondary seal friction, if any. (In high-pressure applications, the geometry of the leakage path markedly affects the normal load.)

The parameter that is most difficult to predict is coefficient of friction. Its variation with material couple, operating temperature, load and speed is not well known in most cases. A good example of this variability is the carbon-graphite coefficient of friction. It is dependent upon the type of graphite (natural or artificial), particle size, graphitization temperature, impurities, binder type and impregnants. Quality control of carbon-graphite is difficult to maintain, and variations within grades could vary with time of manufacture. In general, carbon-graphite coefficient of friction ranges between .1 and .25, although figures as high as 1.90 at 1110 K (650°C) ambient temperature have been recorded. It can be seen that since coefficient of friction varies over a wide range, so does energy dissipation.

Various models of boundary lubrication are described in a Department of the Army publication (Reference 7) entitled "Boundary Lubrication". Because of the extensive treatment of boundary lubrication in Reference 8, no attempt is made herein to summarize the various models.

The heat transfer mechanism must be capable of dissipating heat fast enough to maintain a low stabilized temperature. This could be accomplished by directly lubricating the seal interface with a jet of oil.

Other methods of heat removal, especially in high-speed applications, are:

1. Radial holes or face slots, Figure 2.10, in rotating mating rings. Cooled lubricating oil is centrifugally fed through the shaft into and out of the radial slots in the mating ring. The cooling effect on the mating ring and interface is substantial, lowering the temperature in the seal area to workable limits.

2. Under-runner cooling of circumferential seals and lip seals, Figure 2.11. This method is commonly used for shaft seals in gas turbine engines.

If adequate cooling is not supplied, thermal distortion of the primary seal occurs. Since heat is first conducted away by the primary seal elements and then convected away by lubricant, a temperature profile exists across the seal. The temperature profile causes coning in face seals, and causes taper in shaft-riding seals. (Clamping can also cause these effects.)
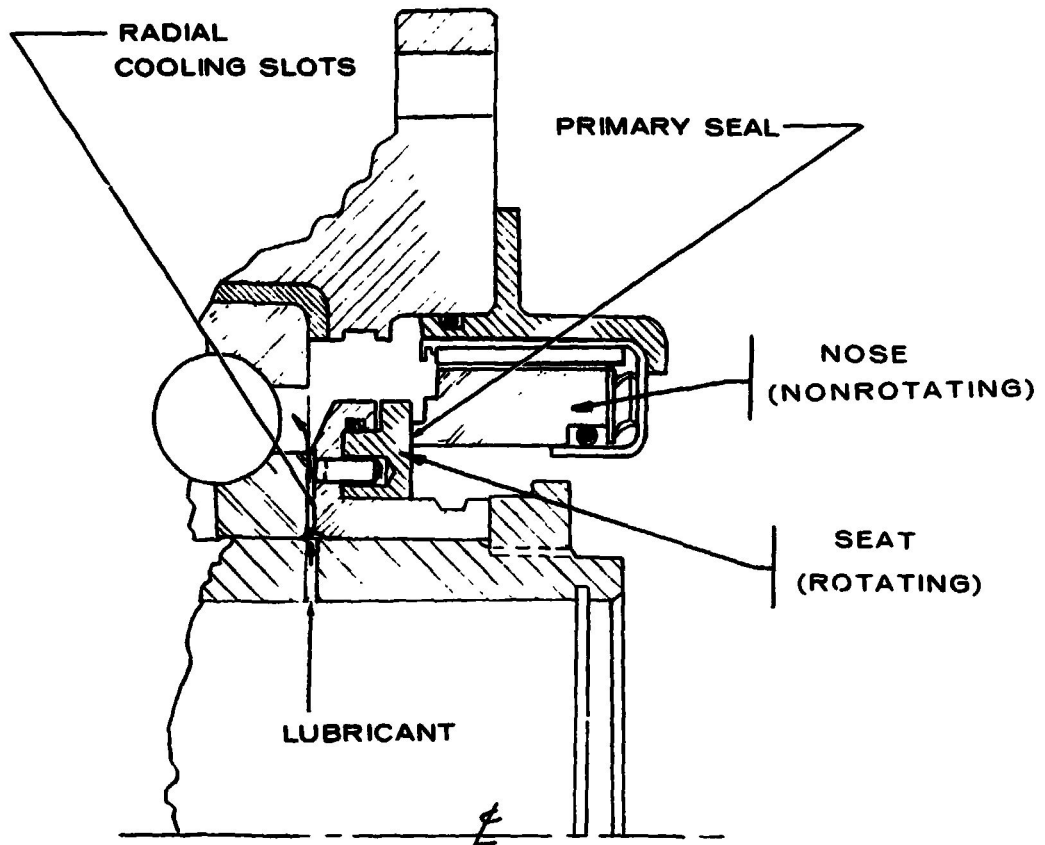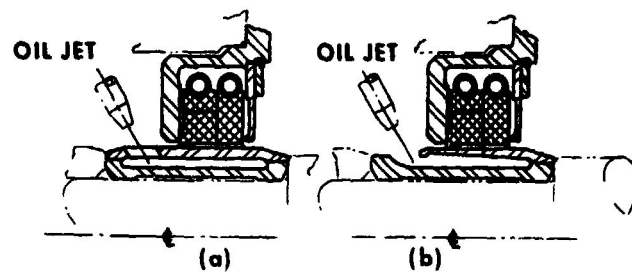
FIGURE 2.10 MATING RING RADIAL
COOLING SLOTS.

FIGURE 2.11 UNDER-RUNNER COOLING (*5)

Coning and taper can cause excessive wear due to high load on a re-
duced contact area. The problem would be solved by wearing in, but
coning and taper change rapidly with temperature, causing different
coning angles.

Coning of face seals in helicopter transmissions has been a problem
in past applications. Excessive rubbing contact is usually found
at the nose inside diameter. The current approach to the problem is
to taper lap the seal face to fit the expected coning, or to cool
the mating ring and minimize coning.

2.1.12 Drainage Requirements

Although gearboxes are usually vented, inadequate or improper drain-
age in the seal area will create a flooding condition and a localized
pressure buildup. In a flooded condition, oil trapped in the seal
area is quickly heated up due to the churning effect of the rotating
seal component and adjacent hardware. Lip seals will have a limited
wear life in a high-pressure environment caused by flooding, and
circumferential seals will leak excessively. A properly designed
face seal is not affected by flooding.

How much drainage is adequate? Gravity-feed drainage is the usual
means of oil removal. Therefore, a head of lubricant must be built
up to a height that is sufficient for drainage.

Using Bernoulli's equation, the theoretical flow through an orifice
is equal to

$$Q = C_v \; C_c A \sqrt{2gh_s} \qquad (2.6)$$

where Q = lubricant flow, m^3/sec
C_v = velocity coefficient
C_c = contraction coefficient
A = drainage area, m^2
g = constant of acceleration, $9.8 m/sec^2$
h_s = head of lubricant, m

For drainage flow, C_v and C_c are assumed to be 1.0 and 0.65
respectively.

Substituting in equation (2.4), the equation for drainage area be-
comes

$$A = \frac{.35 \; Q}{\sqrt{h_s}} \qquad (2.7)$$

From experimental data obtained in both full-scale transmission tests
and bench tests, where values of h_s range from 1.0 to 5.C cm, equa-
tion 2.7 has been shown to produce adequate drainage requirements,
although the effects of turbulence and windage are neglected. If
excessive turbulence does exist, tangential drain holes have been

shown to be effective in the evacuation of lubrication.

2.1.13 Quality Control

Although a seal may be properly selected and designed, the final
product may not function satisfactorily due to poor quality control.
Recent studies of quality control procedures for lip type seals,
especially by automobile manufacturers, has shown a marked improve-
ment of seal performance when extensive controls are in effect.

The Lip Seal Subcommittee of the ASME was formed to establish lip
seal specifications, which include the following:

1. Compound specifications
2. Physical properties
3. Dimensional properties
4. Acceptance testing and quality control
5. Quality control equipment

The compound specification describes the elastomer material and its
mechanical properties. An example of the components of the specifi-
cation is shown below:

The compound is specified as xxxx(N6518C5DX203EX107)

–	Seal Material	xxxx
–	Compound	N (Nitrile)
–	Hardness	65 (Durometer)
–	Tensile Strength	18 (1800 psi)
–	Compression Set	C5 (50% max.)
–	Hardness Change	DX203 (ASTM Oil 1; –10% to+15% change in hardness)
–	Volume Change	EX 107 (ASTM Oil 1; –5% to +35% change in volume)

A more detailed description of this compound specification can be
found in Reference 8.

The physical properties specification provides standard tests for
corrosion and material hardness.

The dimensional specifications have standards for:

1. Checking lip diameters
2. Checking lip opening pressures
3. Specifying radial lip-type oil seals
4. Checking lip seal garter springs

The acceptance testing and quality control procedures will be dis-
cussed in Section 2.1.16. The quality control equipment standard
specifies the necessary equipment to perform tests of various seal

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parameters. This equipment includes:

1. Lip size and lip opening pressure rig
2. Life testers
3. Friction characteristics
4. Eccentricity gage

Not only must the seal be inspected thoroughly, but the runner surface and case outside diameter should also be checked. Component inspections should include:

1. Runner material and hardness
2. Runner out-of-roundness and lobing
3. Runner surface finish
4. Runner surface imperfections (nicks, pits, scratches)
5. Runner dimensions
6. Case outside diameter variance
7. Case out-of-roundness
8. Case imperfections (nicks, pits, scratches)

Quality control procedures for circumferential and face seals are mainly concerned with a visual inspection of carbon-graphite elements for nicks, pits and scratches. The following are some general guidelines:

(a) Nicks (small imperfections caused by mechanical means) are acceptable if they do not extend more than 10% across the sealing face and are limited to 20 in number.

(b) Pits (small, irregular cavities in the surface) have the same restrictions as nicks.

(c) Scratches (a linear depression caused by drawing a sharp object across the surface) are not acceptable if they are of discernible depth.

The flatness of the primary seal surfaces of a face seal should be checked with optical flats. Generally the seal specifications call for flatness within three helium lightbands.

Mating rings for face seals and runners for circumferentials seals should have the same limits on surface imperfections as the carbon-graphite element. It is also important to inspect mating rings for material hardness and surface finish, squareness, flatness and runout. Runners for circumferential seals should be checked for material hardness and surface finish, taper and out-of-roundness.

Dimensional inspection should be made of the seal case, carbon-graphite elements and mating ring or runner. For face seals, a sample inspection of spring load versus axial operating length should be made.

2.1.14 Packaging and Shelf Life

Seals should be packaged to provide maximum protection at a reasonable cost. Damage from handling and from contamination can result in unnecessary scrappage before installation in the assembly. Seal manufacturers can provide many different types of packaging, ranging from individually sealed cartons to bulk wrap.

Lip seals are usually packaged as follows:

1. Each seal is placed on a corrugated pad.
2. A cardboard disc is placed on top of the seal.
3. The seal is packaged in a polyethylene film .3mm (.012 in.) thick, which combined with the corrugated board, provides for a well cushioned and contamination proof packaging. Lip type seals are sometimes packaged in a sealed plastic bag. This method, although corrosion proof, does not provide protection from handling damage.

Face seals are usually packaged as follows:

1. With two-piece designs (seal and mating ring), the carbon-graphite seal nose is cushioned by a cardboard ring that protrudes above the seal nose surface and circumscribes the seal nose outside diameter.
2. Each piece is wrapped in VPI-MP (Volatile Phase Inhibitor-Multi-Purpose) to protect against corrosion.
3. Each piece is individually cushioned and then placed side-by-side and cushioned as a unit.
4. The unit is then securely placed in a carton. Face seals and circumferential seals are sometimes packaged in a sealed plastic bag.

Handling of seals should be kept to minimum. After inspection, a seal should be removed from its package only for installation. Seals should be stored in a temperature-and humidity-controlled area. Seals should be permanently marked and cartons identified with:

1. Part number
2. Manufacturer's identification
3. Cure date of any elastomeric components

The shelf life of an elastomeric element (primary element of a lip seal, secondary seal element of a face seal) depends on the storage conditions and the compound use. Although there is much disagreement on shelf life of various elastomers, Table 2.5 lists shelf lives of the common elastomers under good storage conditions.

TABLE 2.5

Shelf Lives of Common Elastomers

| Compound	Shelf Life
Nitrile	5-10 years
Fluoroelastomers	>10 years
Silicones	>10 years

2.1.15 Installation and Disassembly

Proper installation techniques will aid in the successful performance of a transmission seal. Damage caused by installation can cause premature seal failure. The most critical assembly operation is the installation of a lip or circumferential seal on the runner. The rubber lip and the carbon rings are very susceptible to damage caused by improper runner design or faulty assembly techniques.

The normal installation method is to press the lip or circumferential seal into a housing against a locating shoulder with an installation tool as shown in Figure 2.12. If there is no locating shoulder, a method as shown in Figure 2.13 can be used. In some cases, heating the housing to produce thermal expansion will reduce the interference fit and the axial force required to press the seal into the housing. For assembly of the seal and shaft, the shaft should have a large enough chamfer to prevent seal damage and provide for easy assembly. The chamfer or radius recommended is shown in Figure 2.14. A common problem is damage of the seal by a sharp corner or burr at the junction between the chamfer and the shaft outside diameter. Therefore, it is good practice to radius blend this corner.

Unitized (nonseparable) carbon-graphite elements and mating rings require different installation techniques. In a unitized design, the seal must be installed in the housing and on the shaft simultaneously using a tool comprised of two concentric cylinders of different length. The smaller cylinder contacts the mating ring face; the larger contacts the seal case, with the difference in length equal to the axial operating length between the two faces as shown in Figure 2.15. With a separable seal, the mating ring is first installed on the shaft and then the seal case is installed into the housing, with the case being seated against a shoulder or located by a predetermined reference surface.

Seal removal usually produces damage which eliminates reuse of the seal. Lip seals, with draw holes drilled in the outside face of the case, are removed as shown in Figure 2.16. This method can also be used for circumferential and radial face seals. The incorporation of puller groves in circumferential seals, (Figure 2.17) will
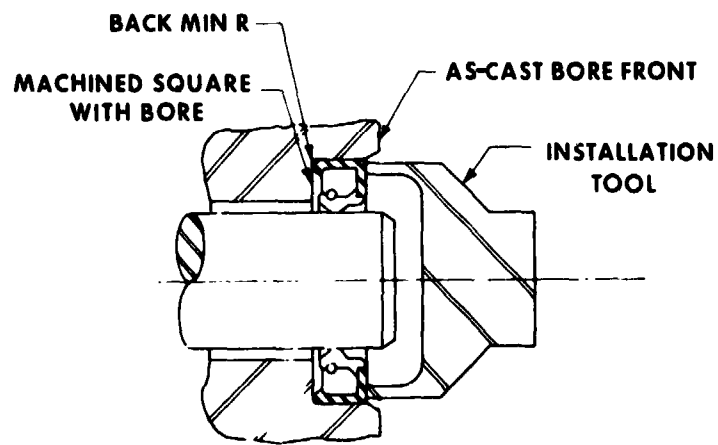
FIGURc 2.12 INSTALLATION OF LIP SEAL
AGAINST REFERENCE SHOULDER (*4)
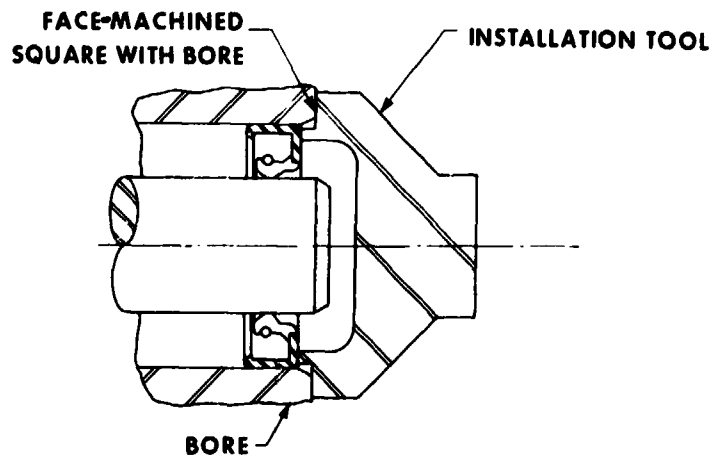
FIGURE 2.13 INSTALLATION OF LIP SEAL
BY A REFERENCE HOUSING FACE (*4)

THESE CORNERS MUST BE BURR FREE

1.75/3.05 MM
(070/.120 inch) .317 mm(⅛ in) R.

15/30 DEG
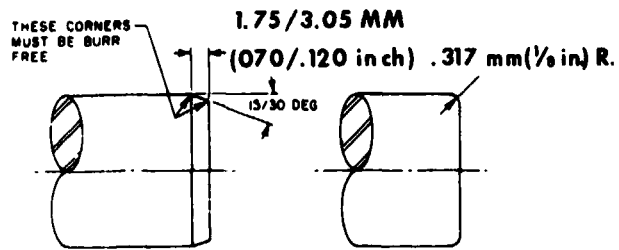
FIGURE 2.14 RECOMMENDED RUNNER
(OR SHAFT) CHAMFER OR RADIUS (*4)

OUTER CYLINDER

SEAL CASE, PRESS
FIT TO HOUSING

NOSEPIECE

INNER CYLINDER

ROTATING SEAL,
PRESS FIT TO SHAFT
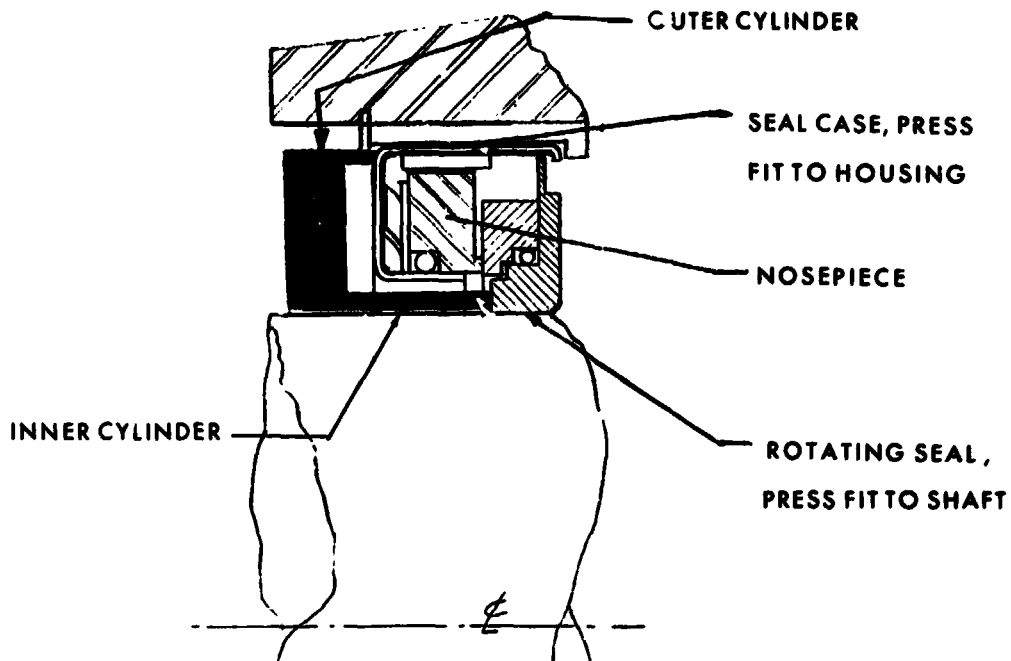
FIGURE 2.15 INSTALLATION OF UNITIZED FACE SEAL

TRANSMISSION HOUSING

SEAL

PULLER HOLES IN SEAL HOUSING

FIGURE 2.16   REMOVAL OF AN INTERFERENCE FIT SEAL

PULLER GROOVE

FIGURE 2.17 CIRCUMFERENTIAL SEAL
WITH PULLER GROOVE (*6)

eliminate damage caused by the above method. But the puller groove
method can distort the housing flange if the force to remove the seal
is excessive. Face seals are often removed with a puller tool con-
tracting the inside face of the seal case bore (Figure 2.18). Puller
grooves are often added to mating rings in order to remove them
from the shaft.

2.1.16 Test Criteria

All sea s used in helicopter transmission are subjected to qualifica-
tion tests prior to being accepted as a production item. This test-
ing can be accomplished either using a ????? bench rig or using a
full-scale transmission. Currently, most ?????-????d and accessory
seals are qualified during initial transm?????? ??velopment tests.
Seals used in high-speed applications are usually developed prior
to full-scale transmission testing. The high-speed seal development
tests can be conducted on a seal test machine ????? ically built with
actual transmission components or a universa ??. ine that can be
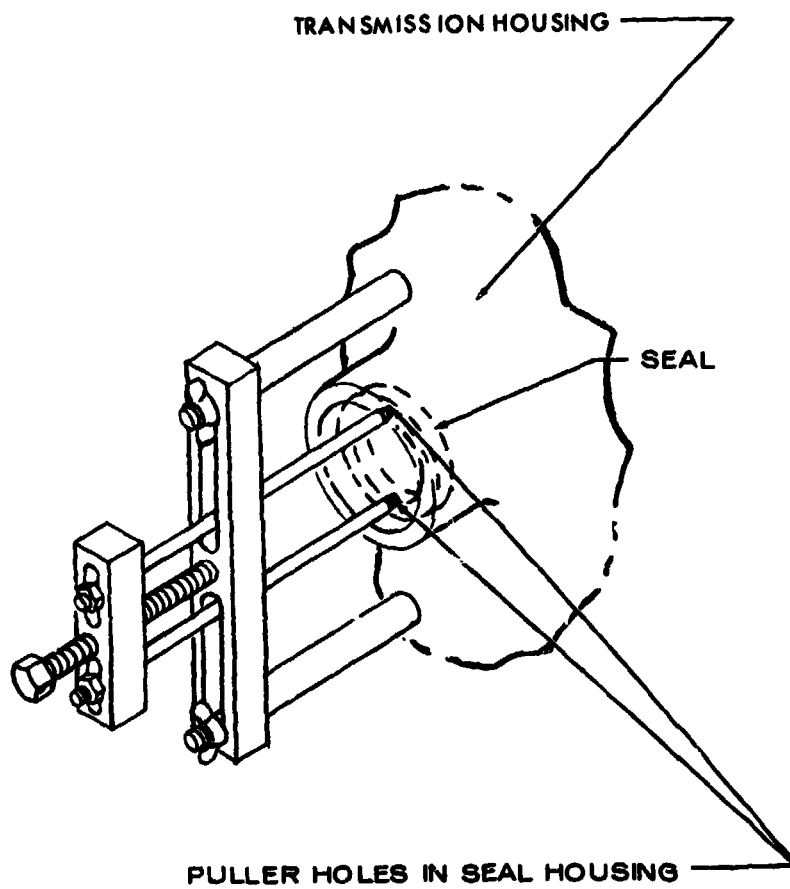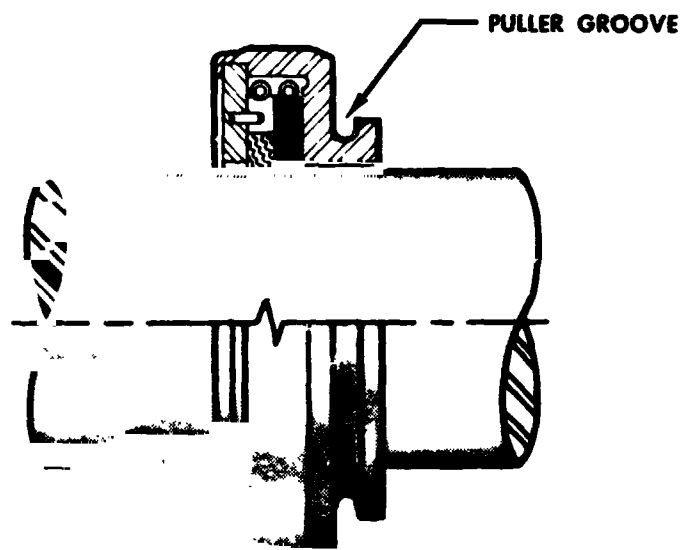adapted to various seal configurations and ????es. These machines
are marketed by numerous companies. To obtain significant results
from high-speed seal bench tests, all parameters affecting seal
performance must be included in the test. Simulation of the param-
eters is more easily accomplished using actual transmission hardware,
although proper use of a commercial test machine can produce the
needed conditions. The goal of these tests is to develop a seal that
will successfully perform in service. Development might include
changes to the seal (e.g., spring loads, material changes) and to
surrounding hardware (e.g., added drainage, added cooling).

Environmental testing should be incorporated into qualification test-
ing. The effect of environmental conditions must be evaluated before
a seal can be properly qualified. Low-temperature testing has not,
until recently, been included in the qualification test procedure.
Low-temperature effects can be obtained in bench testing by using a
low-temperature environmental chamber or by cold soaking the seal
area.

What criteria are used to determine acceptable seal leakage and life?
Since seal life requirements in helicopter transmissions are high
(Section 2.1.), leakage is the primary concern in development and
acceptance testing. Acceptable leakage should be determined by the
effect of leakage on the component. Some components may require
near zero leakage, in which case, any seal leakage will have to be
trapped by a secondary drain or an absorbent packing; other appli-
cations allow for some leakage. An acceptable limit on leakage
should be based on system requirements of the helicopter. In general,
current leakage limitations in helicopter transmission applications
range from 5 cc/hr to leakage that is not visually detectable.

Figure 2.19 is a layout of a typical development seal test rig for
a high-speed input application. Since this rig uses actual transmis-
sion hardware, the test will duplicate most parameters experienced
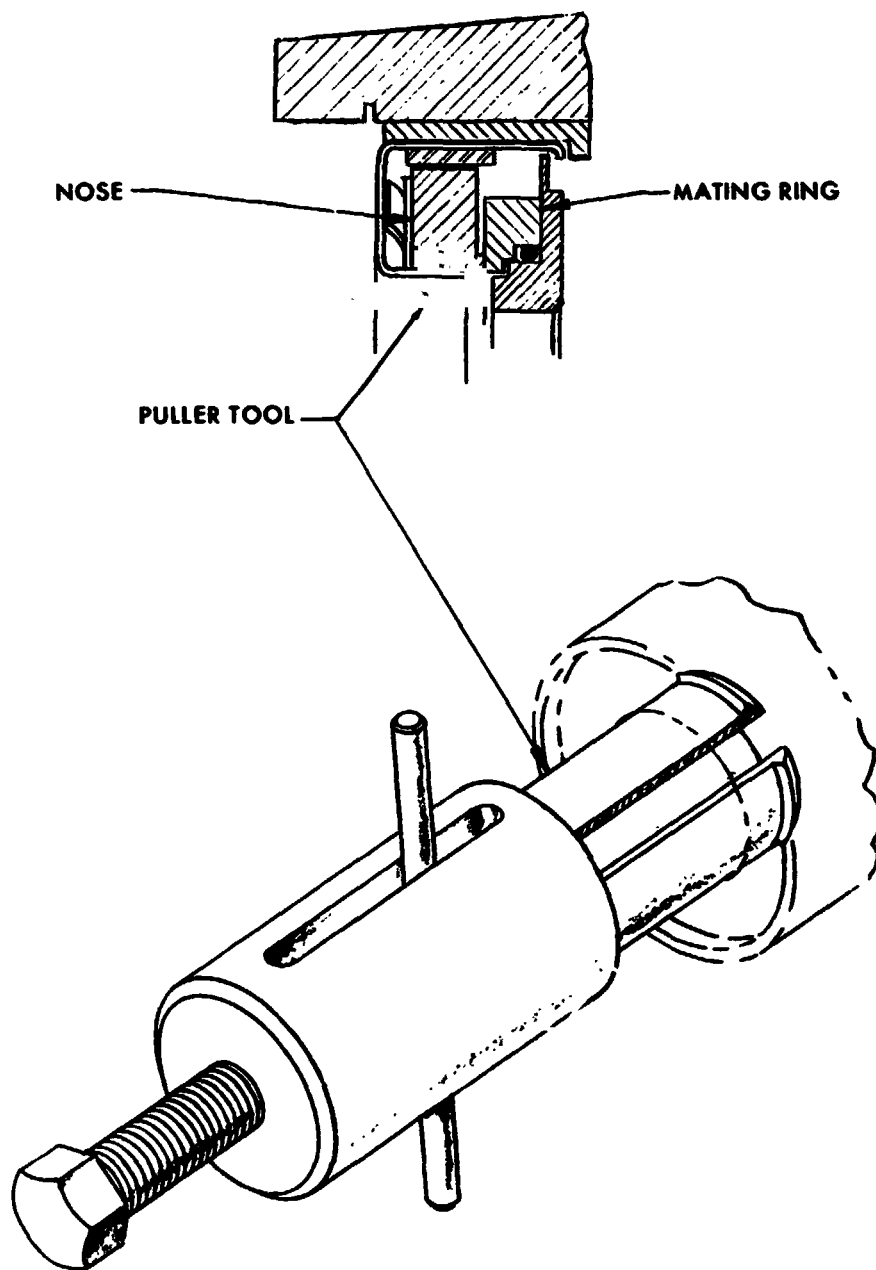
50

NOSE

MATING RING

PULLER TOOL

FIGURE 2.18 REMOVAL OF FACE SEAL

DRIVE MOTOR

DRIVE SHAFT
(TRANSMISSION COMPONENT)

TEST SEAL

GEAR AND BEARING ASSEMBLY
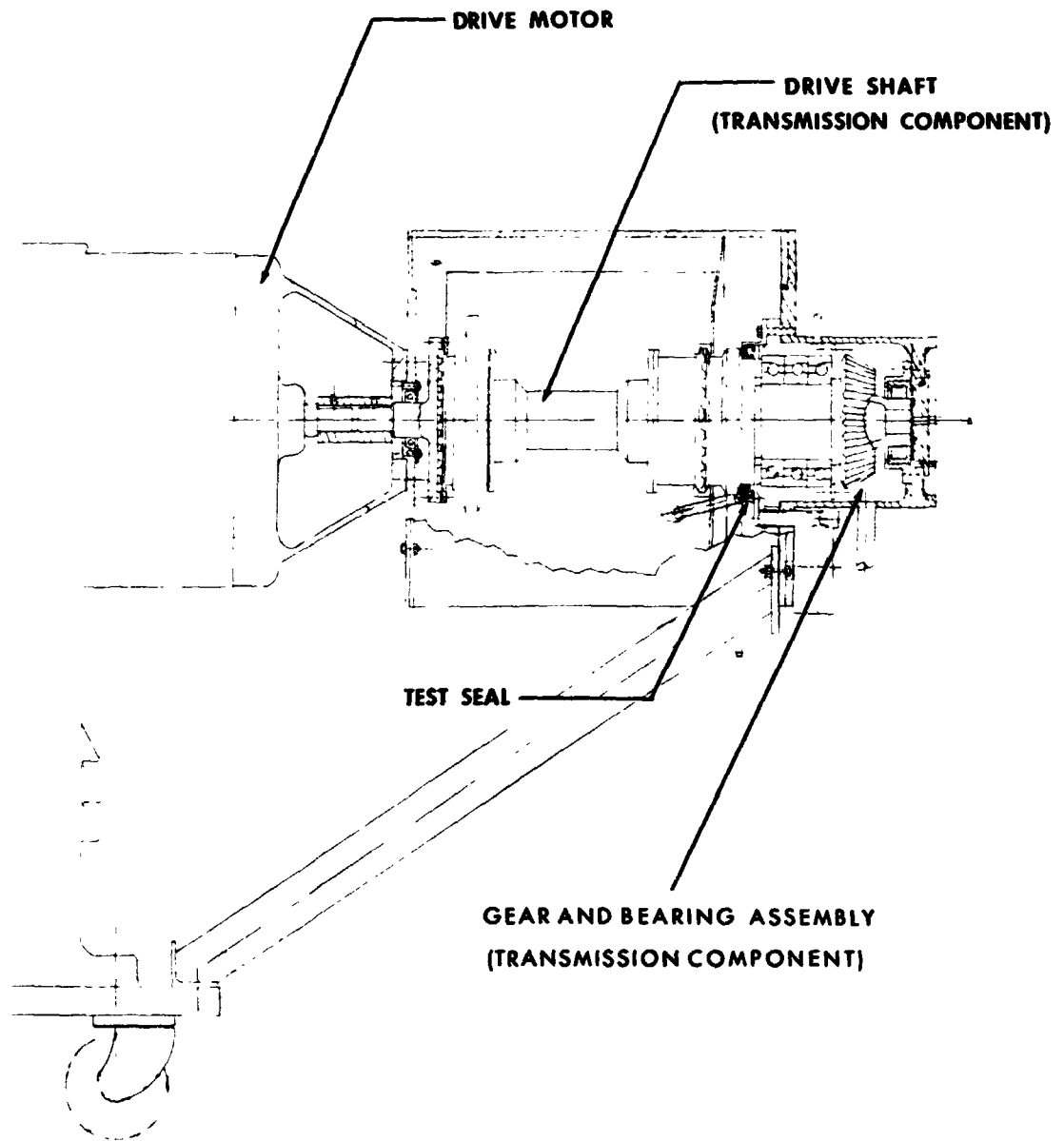(TRANSMISSION COMPONENT)

FIGURE 2.19 HIGH-SPEED SEAL TEST RIG

in operation. The effect of aircraft vibration, and deflection or misalignment caused by torque and gear loads can not be evaluated by this rig. Final qualification of the seal is accomplished with full-scale qualification testing of the transmission system.

2.2   Seal Component Design

The operating conditions and performance requirements determine the type of seal and its detail component design (lubrication, drainage, envelope, environment exclusion device, etc.). The components of the seal include a primary element, mating element, case, loading device, secondary seal element and, in some cases, an external exclusion device.

2.2.1 Primary Seal Element Design

The most critical component of a seal design is the primary sealing element. In lip seals, the primary element is the flexible elastomer lip. Face and circumferential seals contain carbon-graphite primary elements. Each element is discussed below.

2.2.1.1   Conventional Lip Seals

Successful lip seal performance requires the proper selection of material and control of geometry. There are numerous materials available for lip seal use, although only three-fluorelastomer, silicone and nitrile-are currently being used in helicopter applications. The choice of lip materials is limited when operation with the ester lubricants (MIL-L-7808 and MIL-L-23699) is required. For these lubricants, the seal manufacturer usually recommends a nitrile lip material for service below 380 K (225°F) and a fluoroelastomer for service above 380 K (225°F).

Fluoroelastomer is a compound containing fluorine, hydrogen and carbon and is currently used in most helicopter lip seal applications. Its advantage over other elastomers is its ability to operate at higher temperatures. Its resistance to most oils, fuels and chemical solvents is good. These elastomers also have excellent resistance to ozone, oxygen and weathering. Mechanical properties include low compression set, high modulus and good tensile strength. Two important disadvantages are its brittleness at low temperature and its high cost.

Silicone rubber is a semiorganic synthetic compound comprised of a flexible, but weak, chain of silicone and oxygen. Greater tensile strength can be achieved with reinforcing materials. Silicone compounds can successfully perform over a temperature range of 213 K to 450 K (-75 to +350°F). The flexibility and relatively low damping capacity of silicone make it extremely effective when shaft runout is excessive. Silicone will swell in some synthetic oils used in helicopter transmissions.

Nitrile compounds were used extensively in older transmissions

where rotational speeds and temperatures were lower. This
material can be used at temperatures up to 380 K (225°F).
However, exposure for extended periods to a combination of air
and oil at temperatures above 356 K (180°F) will cause harden-
ing.

Table 2.6 is a selection chart for the three elastomers discussed
above and for acrylics, TFE fluorocarbons, and leather.

Some caution is needed in regard to various formulations that
meet specifications for MIL-L-7808 and MIl-L-23699 lubricants.
The ester base stock is usually compatible with nitrile, silicone
and fluoroelastomers. However, various oil companies use differ-
ent additive packages with the ester base stock (these additives
are for antioxidation, extreme pressure lubrication, anticorrosion
and defoaming), and these additives can cause compatibility
problems. Thus, all formulations meeting the MIL-L-7808 and
MIL-L-23699 specifications do not have the same compatibility
with polymers: one can be suitable; the other not. Also, not
all elastomers with the same name have similar compatibility.
For instance, it is possible to have one nitrile sample shrink
and the other expand when immersed in the same fluid (Reference
9), because different manufacturers use different copolymer ratios,
polymerization techniques, plasticizers and fillers. Therefore,
in determining compatibility, the investigation should include
a detailed look into the lubricant additive and the polymer con-
stituents.

Reference 9 gives a method of determining the compatibility of
elastomers in fluids. Briefly, the procedure involves determin-
ing the solubility parameter (the ratio of molecular energy of
vaporization to molar volume) and correcting for hydrogen bond-
ing and dipole moment effect. In addition to checking for
possible catasthropic effects due to compatibil'ty, the magnitude
of polymer swell should be considered in a seal application
(Reference 10). For example, a nitrile rubber had a 27 percent
change in volume after 70 hours at 422 K (300°F) in MIL-L-7808
(a diabasic ester). The hardness of the polymer can also change.
The effect of these volume and hardness variations is to change
the lip loading and thus the lubricant film. When selecting a
lip seal design, probable changes in hardness and volume should
be checked; the change should be less than 25 percent.

Figure 2.20 is a descriptive representation, with nomenclature
for a lip type seal (hydrodynamic ribs are shown but will be
discussed in Section 2.2.1.2), published with permission of ASLE
from Reference 11. Extensive research has been undertaken to
determine optimum lip cross section; from this research, it has
been determined that, although all lip geometry parameters are
important, the following four parameters are the most significant:

    1.   Interference (either molded or trimmed lip) of an

TABLE 2.6 - COMMON LIP SEAL MATERIALS

Property	Fluoroelastomer	Silicone	Nitrile	Acrylate	Fluorocarbon (TFE)	Leather
Durometer Range	65-90	25-80	20-90	40-90	----	----
Temperature Range	233 K to 477 K (-40°F to 400°F)	213 K to 45° K (-75°F to 350°F)	219 K to 380 K (-65°F to 225°F)	244 K to 422 K (-20°F to 300°F)	188 K to 477 K (120°F to 400°F)	200 K to 366 K (-100°F to 200°F)
Compression Set	Fair	Good	Good	Fair	Good	----
Dry Running	Fair	Poor	Good	Fair	Good	Good
Swell	Low	Medium	Low	Low	None	None
Abrasion Resistance	Fair	Poor	Good	Fair	Fair	Good
Tear Resistance	Good	Fair	Good	Good	Good	Good
Fluid Compatibility						
MIL-L-7808	Good	Fair	Fair	Poor	Good	Fair
MIL-L-21260	Good	Good	Good	Good	Good	Good
MIL-L-23699	Good	Fair	Fair	Poor	Good	Fair
SAE 90	Good	Good	Good	Good	Good	Good
MIL-L-6082A	Good	Good	Fair	Poor	Good	Good
MIL-L-5606	Good	Fair	Good	Good	Good	'ood
MIL-L-21058	Good	Poor	Fair	Good	Good	lr
MIL-G-10924	Good	Poor	Good	Good	Good	ᴜod
Water (Salt or Fresh)	Good	Good	Good	Poor	Good	Poor

LEGEND

1 – CASE WIDTH
2 – CLINCHED CASE
3 – CONTACT POINT
4 – FLEX SECTION
5 – FREE–LIP INNER DIAMETER
6 – GARTER SPRING
7 – HEAD SECTION
8 – HEAL SECTION
9 – HOUSING
10 – HOUSING BORE
11 – INNER CASE
12 – INSIDE FACE
13 – INSIDE–FACE INNER DIAMETER
14 – INSIDE LIP ANGLE
15 – INSIDE LIP SURFACE

16 – LIP INNER DIAMETER
17 – LIP LENGTH
18 – OUTER CASE
19 – OUTER–CASE INNER DIAMETER
20 – OUTSIDE FACE
21 – OUTSIDE LIP ANGLE
22 – OUTSIDE LIP SURFACE
23 – RADIAL HEIGHT DIMENSION
24 – RIB
25 – SECONDARY LIP
26 – SPRING GROOVE
27 – SPRING POSITION
28 – SPRING–RETAINING LIP
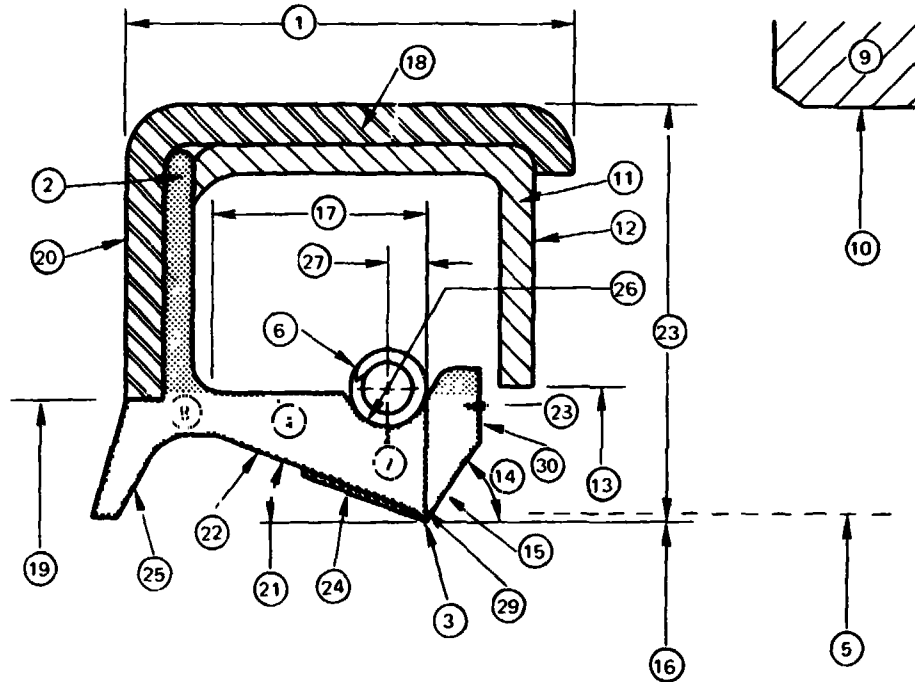29 – STATIC LIP
30 – TOE FACE

FIGURE 2.20 NOMENCLATURE FOR DETAILS OF RADIAL LIP SEAL (HELIXSEAL TYPE) (*2)

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unsprung seal should be between .508 and 1.27 mm (.020
to .050 in.) diametral. The interference controls the
lip contact width, which for an optimum seal should be
less than .254 mm (.010 in.).

2. The R-value (spring position; see item 27 in Figure
   2.20) of a lip is the distance between the spring center-
   line and the lip contact point. The centerline of the
   spring should always be on the air side of the seal.
   Placing it on the oil side will create an instability
   of the sealing element. An R-value of .508 to 1.0 mm
   (.020 to .040 in.) has been shown to be a good working
   tolerance.

3. The seal flex section is bounded by the seal head and
   heel sections (see items 7 and 8 in Figure 2.20). Its
   primary function is to permit radial runout between the
   seal lip and the case. Flex section thickness should
   be approximately .89 mm (.035 in.).

4. The lip length or beam length is the axial distance
   between the thinnest part of the flex section and the
   lip contact point. A short lip will respond more
   quickly to radial runout than a longer lip, but it will
   allow for less runout.

Corrosion of the rubber lip and shaft, a factor until recently
attributed to the material, is electrochemical in nature. The
action occurs in a high-humidity environment and is accelerated
by heat. The corrosion process is slightly retarded by lubrica-
tion in the seal area. It has been established that almost all
elastomers corrode steel shaft surfaces although some noncorroding
compound studies are currently underway to reduce elastomer
corrosion and provide standard noncorroding compounds.

The corrosion process that occurs under lip seals is of the
electrochemical type and acts in the same way as a concentration
cell. It is actually a small battery with electrical potentials
of up to 0.3 volts between the seal case and shaft. Although
the base material is often a cause of the corrosion properties,
additives such as carbon blocks, fillers and curing agents have
been found to add substantially to the corrosiveness of the
elastomer.

Most seals, until recently, were trimmed seals. With a trimmed
seal, excess material is mechanically removed by a knife trimming
operation (Figure 2.21). This procedure introduces machining
tolerance variations in trim diameter, eccentricity, R-values
and other lip variables. In a molded seal, the lip contact point
is formed in the mold, with the excess rubber being trimmed away
from the toe of the lip (Figure 2.22). This process has led to
good reproducibility of lip variables, which was absent in the

AIR SIDE · u.L SIDE
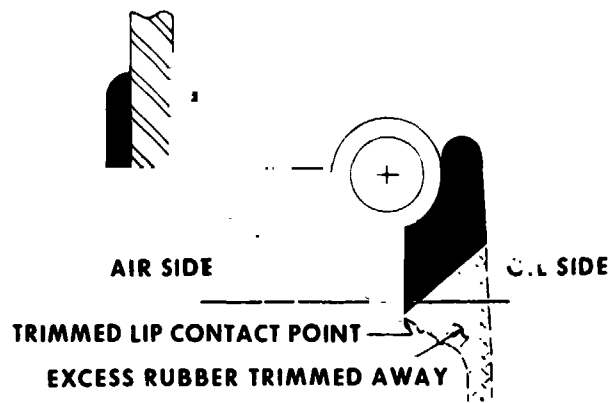
TRIMMED LIP CONTACT POINT

EXCESS RUBBER TRIMMED AWAY

FIGURE 2.2 1 TRIMMED LIP SEAL (*4)

MOLDED LIP CONTACT POINT

EXCESS RUBBER TRIMMED AWAY
(THE FUNCTIONAL AREA OF SEAL
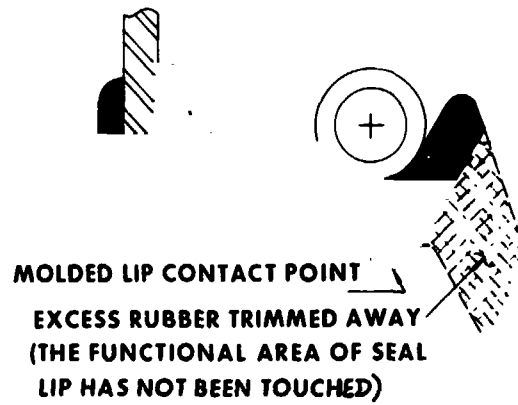LIP HAS NOT BEEN TOUCHED)

FIGURE 2.2 2 MOLDED LIP SEAL (*4)

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trimmed design. The only disadvantage to the molding technique
is that a radius is molded at the contact point (a radial sharp
lip is produced in the trimming process).

2.2.1.2   Hydrodynamic Lip Seals

A recent development in seal technology is the hydrodynamic lip
seal (Figure 2.23). The seal combines a conventional lip seal
with the pumping action of a viscoseal. The hydrodynamic lip
seal currently used in helicopter transmissions have a circum-
ferential sharp lip to form the primary seal, with the runner
and a helical groove pattern extending from the primary seal into
the air side of the seal to pump back any leakage from the primary
seal. The optimum lip seal parameters established in References
12, 13 and 14 are applicable for the hydrodynamic lip seal. The
development of this seal is described in Reference 13.

In the present hydrodynamic lip seal, the rib heights are low,
generally between .0508 and .1016 mm (.002 to .004 in.). The
helix angle ranges between 20 and 30 degrees. Figure 2.24 is a
chart of fluid flow vs. helix angle for a typical hydrodynamic
lip seal. A 20-degree helix angle is shown to produce the most
pumping action, but it is sometimes necessary to compromise pump-
ing action by increasing the helix angle to decrease the circum-
ferential contact that the ribs make with the runner. Nomen-
clature for a hydrodynamic lip seal is similar to that for a
standard lip seal except for the addition of the pumping ribs
(Figure 2.20). Nomenclature for the outer-lip surface and con-
tact surface is shown in Figure 2.25, published with permission
of ASLE from Reference 11. There are various rib configurations
currently available from seal manufacturers. These include a
bead shape, V shape, and ratchet shape (Figure 2.26). There
appears to be little difference in pumping action of the various
rib configurations. Hydrodynamic seals used in helicopter trans-
missions are designed to prevent leakage for a specific direction
of shaft rotation, although bi-directional seal designs are
available. Current emphasis in hydrodynamic seal development is
to produce a bi-directional seal as efficient as the undirection-
al design. Current bi-directional seals include sinusoidal ribs,
diamond-shaped recessed pattern, and a cross helix pattern.

2.2.1.3   Circumferential Seal

Circumferential seals have primary elements that are split or
segmented, single or multiple rings.

The ring's inside diameter mates with a runner, or shaft. The
rings are restrained from rotating by pins in the housing. These
seals are considered to be better for sealing gas than liquid
since viscous fluids may cause hydrodynamic lift of the primary
elements, causing each segment to "water ski" over an oil film.
Experience has shown that by increasing the radial spring load
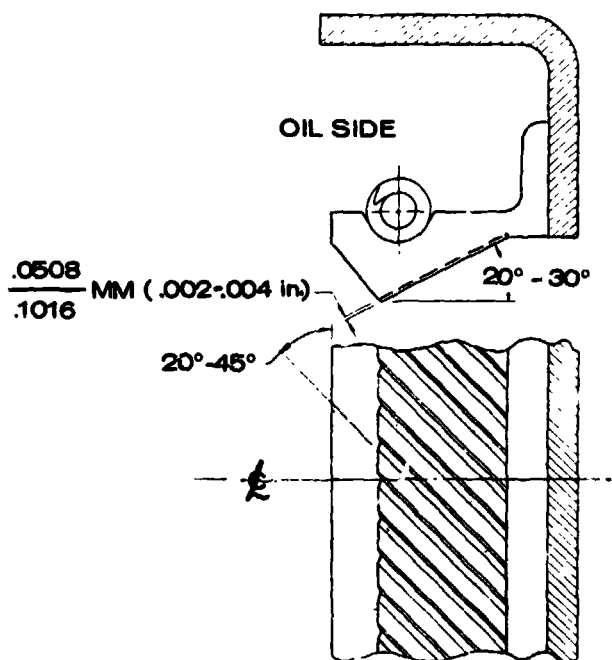
FIGURE 2.23 HYDRODYNAMIC LIP SEAL (*3)
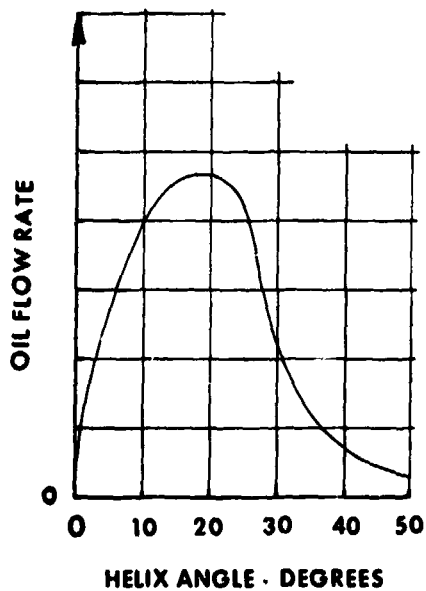
FIGURE 2.24 FLUID FLOW VS HELIX
ANGLE FOR A TYPICAL
HYDRODYNAMIC LIP SEAL (*4)
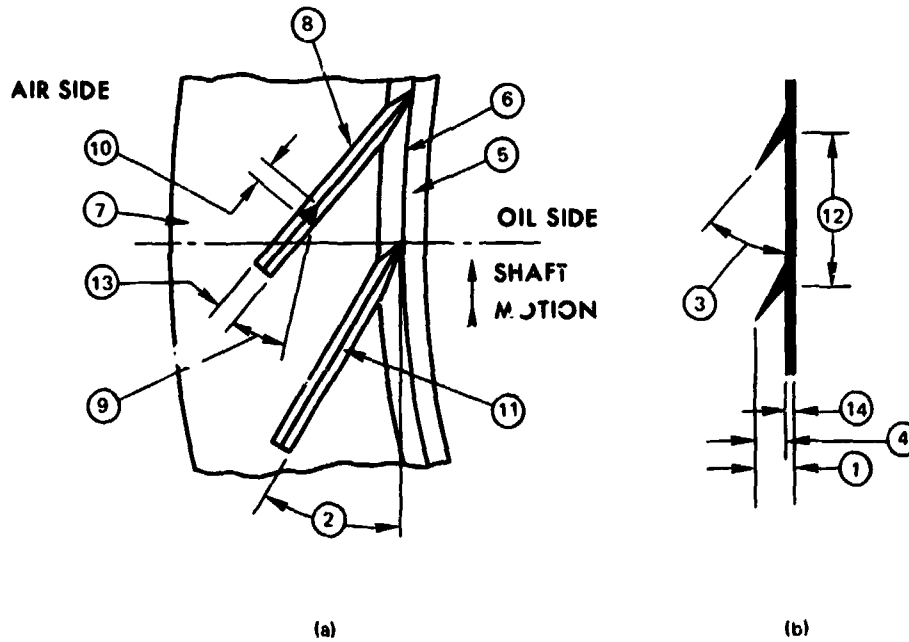
FIGURE 2.25 NOMENCLATURE FOR OUTER-LIP-SURFACE AND
CONTACT-SURFACE DETAILS OF RADIAL LIP SEAL (HELIXSEAL TYPE)
(a) VIEW IN PLANE OF OUTSIDE LIP SURFACE
(b) VIEW OF CONTACT SURFACE. (*2)

(a) BEAD SHAPE
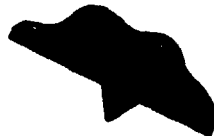
(b) V SHAPE

(c) RACHET SHAPE

FIGURE 2.26 RIB CONFIGURATIONS (*4)

or by reducing the bearing area, circumferential seals can be effective oil seals.

It is necessary in single-ring designs to provide step cut gaps (ring joint) in the axial and radial directions to prevent leakage through the gap. In the step seal design, the machine work performed on the ring joint is extremely critical, with lapped surfaces necessary for good sealing.

The two-ring design requires a step seal joint on the inner ring. The joint on the outer ring can be straight cut. With this type of design, the machine work on the step is still critical but is easier to perform than the single-ring design.

Figure 2.27 shows a three-ring circumferential or segmented ring shaft riding seal. This type of seal is radially loaded to a shaft runner for dynamic sealing and axially loaded to the seal housing flange for static sealing. Carbon-graphite is usually used for a ring material (see Section 2.2.1.4) with the case fabricated from steel, stainless steel or a lightweight metal.

The three-ring design is the most common type of circumferential seal used in helicopter transmissions. All joints are straight cut with static sealing accomplished by overlapping of rings. The three-ring design consists of a primary ring, cover ring and back ring. Each ring is cut into a number of segments, the number depending upon the size of the seal. The back and cover rings have the same circumferential orientation of gaps, while the primary ring gaps are oriented at a different angle. Thus all static leak paths are eliminated. A primary seal is formed by loading the rings against the runner with garter springs, and a secondary seal is formed by loading the rings against the housing flange with a wave or with coil springs.

The primary ring and cover ring must be of the same width and are therefore, usually lapped together. Lapping of ring sides is necessary to establish a static seal. Conformity of the primary ring outside diameter and of the cover ring inside diameter is also necessary for good static sealing.

The ring assembly is seated against the housing flange by an axial spring. Since no rotation occurs at this surface, only fretting wear has been observed. Hang-up of the seal rings does occur if the friction force caused by the axial force is excessive. This will not allow the primary elements to respond to runout and compensate for wear of the cylindrical sealing surfaces (primary seal). The amount of wear allowed at the primary surface depends on the clearance between antirotation pins and the ends of the ring segments. Wear beyond this clearance cannot be compensated for by ring inward movement.

Taper and out-of-roundness of the primary seal surface can cause

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LEGEND:

1-BACK RING

2-AXIAL SPRING

3-COVER RING

4-RADIAL GARTER SPRINGS

5-PRIMARY-SEAL RING

6-RUNNER

7-SEAL CASE

8-CASE FLANGE

9-SPRING ADAPTER

10-ANTIROTATION PINS

11-PRIMARY SEAL

12-SECONDARY SEAL
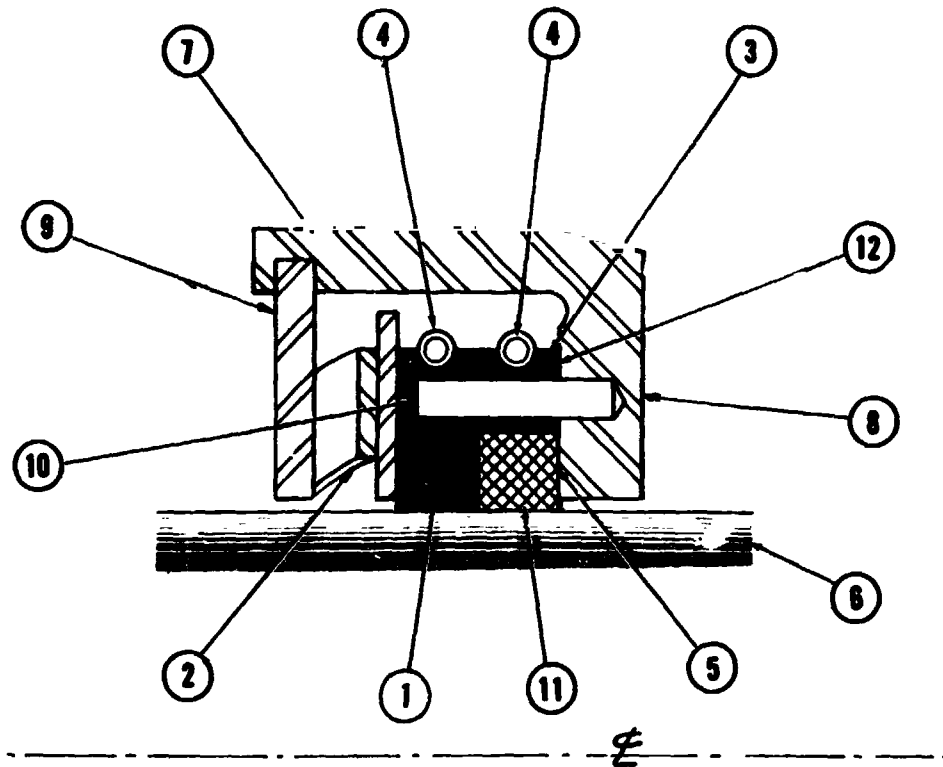
FIGURE 2.27 NOMENCLATURE FOR THREE RING
CIRCUMFERENTIAL SEAL (*6)

excessive unit pressure and high wear. These conditions can be caused by axial temperature gradients as well as by poor-quality manufacturing.

Ring axial widths of less than 2.54 mm (0.1 in.) are currently being used in helicopter applications.

2.2.1.4   Face Seals

Figure 2.28 shows a standard face seal reproduced from Reference 11 with permission from ASLE. This type of seal, as usually applied, has the lubricant on the outside diameter of the primary seal. The seal consists of a rotating mating ring and a non-rotating carbon-graphite seal ring flexibility attached to the seal case. A secondary static seal between the carbon-graphite seal ring and case prevents leakage and permits axial movement of the carbon-graphite ring (due to thermal expansions, mating ring face runout and wear). In Figure 2.28, the secondary seal is a standard type O-ring. This design can be used in many low-temperature, low-pressure applications typical of helicopter transmissions. Other types of rubber molded secondary seals which can be considered for normal helicopter applications are V-rings, U-cups, wedges, quad-rings and elastomeric bellows. For higher temperatures, secondary seals such as bellows, piston rings and teflon seals should be considered. A metal bellows seal, Figure 2.29, reproduced from Reference 11 with permission from ASLE is the newer development in secondary seals, that performs the function of a secondary seal, a spring, an antirotation device and a pilot. However, the configuration of the O-ring type seal ring, Figure 2.28, is common to most face seals used in helicopter transmissions.

The primary ring can also be composite design, where a carbon-graphite nosepiece is retained by a metal carrier with an interference fit and/or an adhesive band.

The face width of the primary seal ring is dependent on the seal size and Table 2.7 presents standard face widths for different seal sizes. The height of the seal nose usually varies from 1mm to 1.75 mm (.040 to .070 in.), which is more than that needed for wear allowance.

TABLE 2.7 - Seal Face Width

Shaft Diameter —mm	Face Width —mm
25.4 (1 inch)	1.58 (.062 inch)
50.8 (2 inches)	1.58 (.062 inch)
76.2 (3 inches)	1.98 (.078 inch)
101.6 (4 inches)	2.39 (.094 inch)

LEGEND

1 – ANTIROTATION DEVICE
2 – CUP
3 – EXTENDED LENGTH
4 – PRIMARY–SEAL CARRIER
5 – PRIMARY–SEAL RING
6 – SEAL BORE
7 – SEALING FACES
8 – SEALING–FACE INNER DIAMETER
9 – SEALING–FACE OUTER DIAMETER

10 – SEALING–FACE WIDTH
11 – SEAL NOSE
12 – SEAL OUTER DIAMETER
13 – SECONDARY SEAL
14 – SECONDARY–SEAL LAND
15 – SECONDARY–SEAL–LAND DIAMETER
16 – WASHER
17 – WAVE SPRING

SHEAR-NITCH TYPE

MATING RING
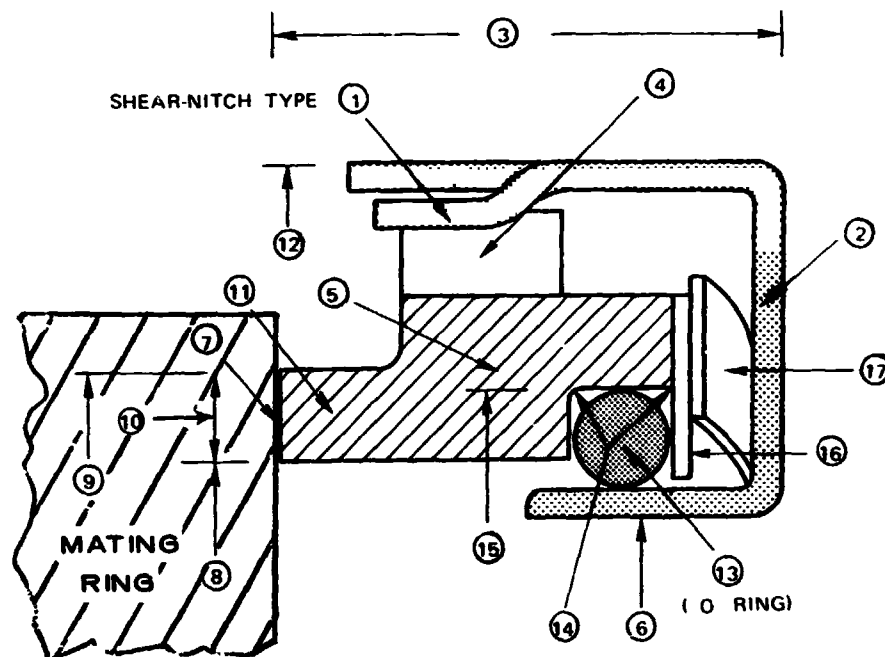
(O RING)

FIGURE 2.28 NOMENCLATURE FOR TYPICAL FACE SEAL
WITH O -RING SECONDARY SEAL. (*2)

2 – CUP
3 – EXTENDED LENGTH
4 – PRIMARY–SEAL CARRIER
5 – PRIMARY–SEAL RING
6 – SEAL BORE
7 – SEALING FACE
8 – SEALING–FACE INNER DIAMETER
9 – SEALING–FACE OUTER DIAMETER

10 – SEALING–FACE WIDTH
11 – SEAL NOSE
12 – SEAL OUTER DIAMETER
13 – SECONDARY SEAL
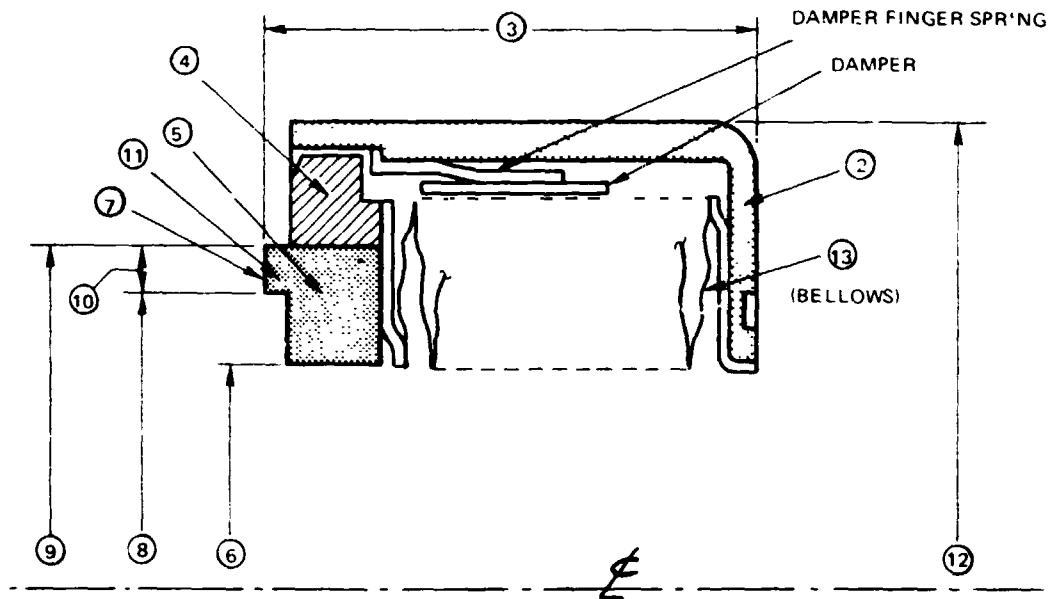
FIGURE 2.29 NOMENCLATURE FOR TYPICAL
METAL BELLOWS FACE SEAL (*2)

To prevent rotation between the primary seal ring and the seal case, antirotation locks (usually two shear nitches or welded blocks) on the inside diameter of the case, mate with slots on the outside diameter of the primary seal ring. The length of the slots must be sufficient to allow for the axial travel of the seal.

The outside diameter of the seal bore acts to pilot the primary seal ring through the secondary seal. This pilot must be long enough to maintain concentricity of the primary seal ring and the shaft, and insure O-ring contact over the full operating length, but its length must be limited to allow for some misalignment capability of the primary seal ring.

The material usually used for the primary seal ring is carbon-graphite. Its almost universal use as a face seal material is because of its self-lubrication, chemical inertness, compatibility with other metals, thermal characteristics and temperature resistance. Carbon-graphite is a brittle material and is easily chipped. It has a low tensile strength and a low modulus of elasticity which can sometimes lead to deflection problems.

Carbon-graphite is chemically inert with all lubricating oils used on helicopter transmissions. It has an inherent quality of self-lubrication and non-galling, although long term operation of carbon-graphite at high speed without any external lubrication presents special design problems. Carbon-graphite compositions exhibit low coefficient of friction. Coefficient of friction is highly dependent on the material mated with carbon-graphite. In general, carbon-graphite, mated with a good bearing surface, exhibits a coefficient of friction of .1 to .25. With the presence of a hydrodynamic fluid film, friction coefficients of less than .05 can be realized.

Carbon-graphite exhibits good thermal characteristics. Common carbon-graphite compositions have thermal conductivities varying from 12.4×10^{-6} to 74.3×10^{-6} joule/hr/m^2/K/m (10 to 60 btu/hr/ ft^2/$^\circ$F/in.). When impregnated with various metals, the conductivity increases. They have the ability to radiate heat rapidly for dissipation to the surrounding environment.

As compared to elastomers, carbon-graphites physical properties change slowly with temperature. Oxidation of carbon-graphite will occur at temperatures over 533 K (500°F) if special treatments such as impregnations with resins, inorganic salts, polymers or metals, are not applied. (Some grades are still functional at temperaturss up to 755K (900°F).) Dimensionally, carbon-graphite rings are less affected by temperature than are ferrous metals. Carbon-graphites coefficient of thermal expansion is 1.22×10^{-6} m/m/ K (2.2 x 10-6 in./in./$^\circ$F) as compared to 3.9×10^{-6} m/m/ K (7.0 x 10^{-6} in./in./$^\circ$F) for ferrous metals.

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Carbon-graphites are porous materials because during sintering, volatiles escape from the binder, creating voids. The amount of permeability is expressed in darcys, which are defined as

$$D_a = \frac{Q \, L\mu}{A_d \, P_d} \qquad (2.8)$$

where D_a = permeability, darcys
Q = quantity of fluid passed, cm^3/sec
μ = viscosity, centipoises
L = flow path length, cm
A_d = area normal to flow direction, cm^2
P_d = pressure drop, atm

It is desirable for carbon-graphite seal rings to have low permeability. This is accomplished by impregnating the matrix with various materials, which can produce a permeability from 15,000 x 10^{-6} to .001 x 10^{-6} darcys.

Carbon-graphites are molded to the general size and shape of the final product. Final machining is performed with regular machine tools for soft carbon-graphites and with carbide tools and grinding equipment for harder materials.

The seal face has special requirements for flatness and surface finish. Flatness of the seal surface is obtained by lapping. Helicopter transmission seals are usually lapped to within three helium light bands. The flatness, in terms of helium light bands, is measured with monochromatic light against an optical flat of transparent quartz or Pyrex glass.

Surface finish cannot be specified as a root-mean-square (rms) or arithmetic average (AA) as with most metals. The porosity of the material makes this measurement meaningless.

Table 2.8 lists some common carbon-graphite grades used in seal rings and their properties.

2.2.2 Mating surface design

Mating surface design is as critical as the primary element design. Cylindrical runners and face seal mating rings will be discussed separately.

2.2.2.1   Cylindrical Runners

This type of mating surface is utilized for shaft riding seals such as lip and circumferential seals. For either type of seal, the material selection and design limitations are similar. Material selection should consider wear, corrosion and thermal properties.

TABLE 2.8 - COMMON CARBON-GRAPHITE GRADES

Grade	Scleroscope Hardness	Compressive Strength, N/cm² (PSI)	Transverse Breaking Strength N/cm² (PSI)	Tensile Strength N/cm² (PSI)	Modulus of Elasticity X 10⁶ N/cm² (PSI)	Coefficient of Thermal Expansion X 10⁶ m/m/ K (in/in/ F)	Maximum Recommended Operating Temperature K (°F) in Air	Apparent Density Gm/cc	Porority (Vol. %)
1	105	26,200 (38,000)	7,900 (11,500)	6,200 (9,000)	1.72 (2.5)	1.89 (3.4)	533 (500)	1.82	1.00
2	100	24,800 (36,000)	9,300 (13,500)	6,540 (9,500)	2.06 (3.0)	1.28 (2.3)	533 (500)	1.90	.25
3	100	24,100 (35,000)	7,230 (10,500)	5,850 (8,500)	1.72 (2.5)	2.0 (3.6)	533 (500)	1.80	1.70
4	85	19,300 (28,000)	5,500 (8,000)	4,480 (6,400)	1.51 (2.2)	1.78 (3.2)	533 (500)	1.80	2.60
5	90	25,800 (37,500)	7,570 (11,000)	5,500 (8,000)	2.13 (3.1)	1.22 (2.2)	533 (500)	1.80	2.00
6	75	20,600 (30,000)	7,570 (11,000)	5,500 (8,000)	2.27 (3.3)	1.89 (3.4)	505 (450)	1.85	1.00
7	75	20,600 (30,000)	7,570 (11,000)	4,820 (7,000)	2.34 (3.4)	1.39 (2.5)	477 (400)	1.85	1.00
8	85	22,400 (32,500)	6,890 (10,000)	5,170 (7,500)	2.13 (3.1)	1.67 (3.0)	477 (400)	1.80	2.50

Currently most runners are made from a case carburized material (usually AISI 9310 steel), a hard stainless steel (usually 440 C), or a standard low alloy steel (typically AISI 4130 or 4340) with a hard chrome plate. The corrosion-resistance properties of the latter two (stainless steel or chrome plate steel), make them preferred. Stainless steel is seldom used as a runner because runners in most cases are structural members and stainless has less strength than standard steels. Also, the cost of high-strength stainless steels is a factor that works against its use.

Most seal manufacturers recommend a shaft or runner hardness of Rockwell C30 minimum (for lip type seals), but lip seal runners currently used in helicopter transmissions are usually designed to have a surface hardness of Rockwell C55 to C58 minimum. This is done to prevent abrasive scoring of the runner during operation in a gritty environment. Hardness of circumferential seal runners is similar to that of lip seal runners.

The thermal conductivity of the runner is important, since it serves as a heat sink. Heat generated at the seal interface is conducted away through the runner and the lubricant. It is an advantage, even in relatively low temperature transmission applications, to have a runner material with a high thermal conductivity.

Runner geometry has been a major concern in recent years. Studies by lip seal manufacturers and users have resulted in more stringent controls on runner surface finish and out-of-roundness. Tests reported in Reference 14 demonstrate that with a mirror-smooth finish, hydrodynamic effects were decreased since the lubricant could not be trapped in pockets formed by the surface roughness. This produced increased lip torque. With very rough surfaces, since the fluid film is thin, shaft irregularities penetrate the film, causing rubber-metal contact. The optimum finish is about .38 micron (15 microinches); in Table 2.9 the seal torque for this optimum finish is compared to that of .125 and .89 micron (5 and 35 microinches) on a 100 mm (4 in.) shaft operating at 1000 rpm. The Rubber Manufacturers Association (RMA) and the Society of Automotive Engineers (SAE) recommend a surface finish of .254 to .508 micron (10 to 20 microinches). The technique used to produce and measure the finish is also important. Reference 15 presents a summary of the various techniques used to produce a lip seal runner surface finish. The experimental results of Reference 16 shows that plunge grinding and glass-bead-blasting can be successfully used to produce the required surface asperities needed for lip seal applications. However, glass-bead-blasting procedures are not sufficiently developed to warrant current use; therefore, plunge grinding should be specified. It is important that no machine lead be introduced. A common belief among seal users is that plunge grinding does not produce machine lead. This is true only if:

1. The wheel is dressed slowly with a shallow cut.

2. A cluster-head diamond dresser is used.

3. The ratio of grind wheel/workpiece rotational speed
   is not a whole number.

4. The grinding wheel is not withdrawn until spark-out
   has occurred.

TABLE 2.9

SEAL TORQUE VERSUS SURFACE FINISH

Surface Finish, microns	Torque, cm-N	Temperature, K
.125 (5 microinches)	29. (41.2 in.-oz)	338 (148°F)
.38 (15 microinches)	13.2 (18.7 in.-oz)	336 (143°F)
.89 (35 microinches)	21.1 (30.0 in.-oz)	338 (148°F)

The measurement of surface finish is also subject to question.
Surface roughness can be specified as root-mean-square (rms) or
arithmetic average (AA). RMS values are approximately 11 percent
higher than AA values. Surface finish is usually measured by a
profilometer in the axial direction. With lip seals, the differ-
ence in values between circumferential readings and axial read-
ings is dependent on machining method and material. With plunge
grinding in the .254-to .508-micron (10-to 20-microinch) range,
the axial readings will be approximately .051 micron (2 micro-
inches) higher than the circumferential reading.

A maximum roughness of .2 micron (8 microinches) is used for
circumferential seal runners. Rubbing contact with this type of
seal can be tolerated, whereas it is detrimental to lip seals.

Out-of-roundness and lobing are discussed in Section 2.1. Exces-
sive out-of-roundness of runners causes a high-frequency runout,
hydrodynamic lift, and leakage.

Taper of the runner surface for circumferential seals has the
same effect as taper of carbon-graphite elements. It can cause
high contact pressure and excessive wear.

2.2.2.2   Face Seal - Mating Rings

The mating ring requirements are similar to those of a circuferential seal runner. The sealing surface is on a radial face compared to a circumferential seal's cylindrical diameter. In regard to material properties, a hard, wear-resistant and corrosion-resistant material with a high thermal conductivity is desirable.

Both thermal and mechani al distortion can disrupt the sealing mechanism. Thermal distortion will always be present in face seals, but it will be more prevalent in high-speed, high-temperature seals. Proper selection of high-conductivity materials and, in extreme cases, additional lubrication to the mating ring, Figure 2.10, can keep thermal distortion within acceptable limits.

Mechanical distortion is more common in transmission seals. In most cases, the mating ring is a rotating member and is also used to clamp the support bearings. This clamping force can be substantial. Presently, success in controlling distortion caused by clamping has been achieved by using a two-piece mating ring, (Figure 2.30). With the inner ring solidly fixed to the shaft, clamping forces are transmitted through the inner ring to the support bearing. The seal ring is loosely keyed or pinned to the clamping ring and internally sealed. Since the seal ring is flexibily mounted, distortion from the clamping ring does not affect the seal ring. This design also permits small amounts of shaft to housing misalignment without affecting parallalism of the sealing surfaces. The material of the inner ring should be similar to that of the seal ring, but it is not necessary that they be the same. It is imperative that the inner ring material:

    1.   Have nearly the same thermal coefficient of expansion
         as the seal ring (outer ring).

    2.   Be corrosion resistant.

    3.   Be strong enough to carry the clamping load.

Usual seal ring materials are hardened 440C stainless steel, or a standard steel with tungsten-carbide, or chromium plating. A 300 series stainless steel is frequently used as the clamping ring material.

The flatness requirement for the mating ring is similar to the primary carbon-graphite element and is usually held to three helium light bands. Surface roughness is usually specified as .1 micron (4 microinches) AA maximum.

2.2.3 Case Design

The seal case is fixed to the housing and usually contains the

O-RING

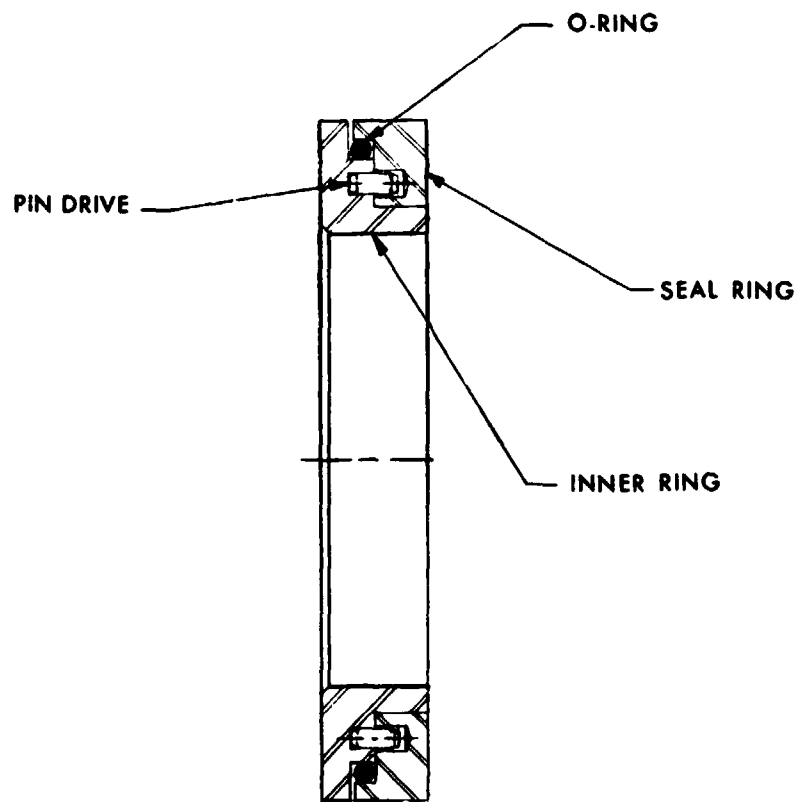PIN DRIVE

SEAL RING

INNER RING

FIGURE 2.30

TWO PIECE MATING RING

primary seal element. Its functions are:

1. Static sealing between the transmission housing and seal case.

2. Providing a mating surface for secondary sealing.

Most seal cases used in helicopter transmissions are manufactured from a 300 series stainless steel. Other materials are standard carbon steel (usually cadmium plated) and aluminum. Aluminum is used because its coefficient of thermal expansion (7.0×10^{-6} m/m/K (12.6×10^{-6} in./in./$^{\circ}$F)) is similar to magnesium (7.8×10^{-6} m/m/K (14.0×10^{-6} in./in./$^{\circ}$F)), a common transmission housing material. Using materials with similar thermal coefficients will eliminate some of the distortion caused by the high press fits needed when the seal case material has a coefficient of thermal expansion significantly different than that of the transmission housing.

Case thickness and outside diameter out-of-roundness can have a substantial effect on case distortion and possibly secondary seal leakage. An application in an auxiliary power plant (APP) clutch illustrates this point. Two face seals made by different manufacturers were tested. One seal had a case thickness of 1.65 mm (.065 in.) and was .0125 mm (.0005 in.) out-of-round, while the other was 1.00 mm (.040 in.) thick and .05 mm (.002 in.) out-of-round. All other design parameters were approximately equal. Six seals of each type with varying degrees of interference fit were tested. All six seals with the thicker case and reduced out-of-roundness exceeded the performance criteria. Five of the thinner case seals leaked excessively. Case distortion can cause the secondary seal land surface to become out-of-parallel with the case outside diameter. With face seals, the seal ring pilot or the case flange on circumferential seals is usually manufactured parallel or square to the case outside diameter to within .0125 mm (.0005 in.). Case distortion disturbs this accuracy, causing secondary seal leakage, excessive frictional drag, or misalignment of the primary seal element.

The case outside diameter is usually a ground cylindrical surface with a chamfer or radius on the corners. An O-ring groove is sometimes machined on the outside diameter to form the static seal. Bonded lip seals can be made with a rubber outside diameter or with a rubber nose gasket.

2.2.4 Loading Devices

All seals discussed in the design guide are positive contact seals since one element is energized against another to form the lubricant barrier. The mode of energizing is usually by spring loading the stationary flexible primary element against the mating surface. Commonly used springs are finger, garter, coil and wave springs.

2.2.4.1   Lip Seal Loadings

Both garter and finger springs are used to load the lip against
the runner. Garter springs are usually used, with the total
radial lip load equal to the load caused by circumferential ten-
sion of the spring and by the lip interference. High spring
loads will cause excessive wear and heat, and low spring loads
will result in excessive film thickness and leakage.

With hydrodynamic lip seals, the spring load required is less
than with standard lip seals, since the spring is needed only to
insure a static seal. Dynamic sealing is accomplished by the
pumping action of the ribs. The position of the spring in
relation to the lip contact point is called the R-value and is
discussed in Section 2.2.1.

The radial lip load can be determined by a variety of methods.
In Reference 17 the radial lip load is calculated as a function
of lip opening pressure. The radial load is

$$P_r = \frac{\pi p D h}{2 \cos^2 \alpha} = \frac{P_p}{? \cdot s \alpha} \tag{2.9}$$

where P_r = radial lip load, newtons

P_p = beam load, Figure 2.31, newtons

p = lip opening pressure, newtons/cm^2

D = shaft diameter, cm

h = beam length, Figure 2.31, cm

α = lip angle, Figure 2.31, radians

This method is used mainly for quality control after production.
The method is quickly performed, although a precise measurement
of the compressive force of the seal cannot be obtained since
lip opening pressure is dependent on many material and geometric
variables.

A device described in Reference 17 measures radial lip load with
an error of less than ±5 percent. The lip seal is placed on a
split mandrel of the normal shaft diameter. One part of the
mandrel is held stationary, while the other part is fixed to a
leaf spring. The radial lip load deflects the leaf spring and
causes a displacement of one-half of the mandrel to the stationary
part. The measurement of this deflection by a dial indicator or
an electrical length dial gage is linearly related to the radial
lip load.

In Reference 18, the radial lip load is described as having two
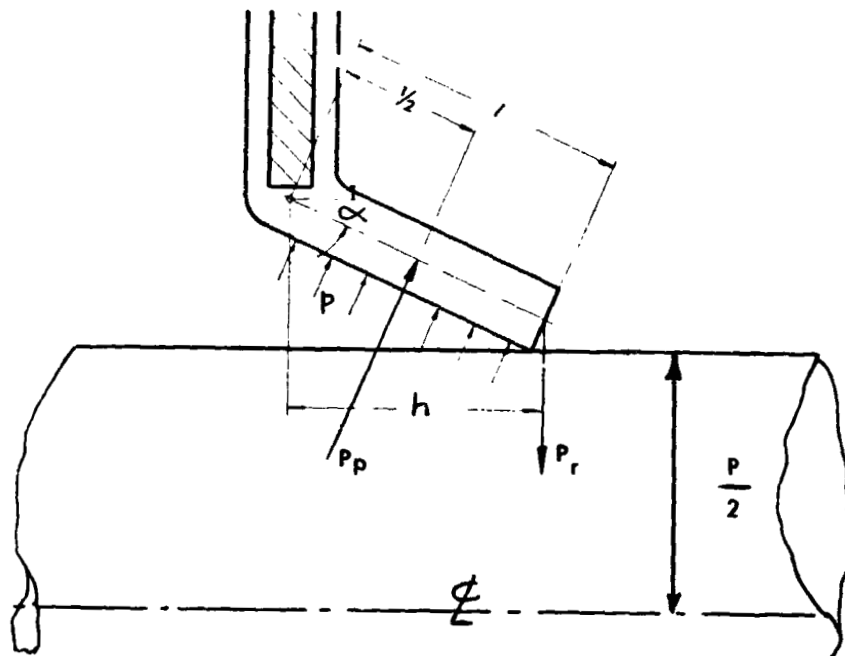components, as shown in the following equation:

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FIGURE 2.31   SCHEMATIC OF LIP
CROSS SECTION FOR LIP LOAD
CALCULATION (*3)

$$L_R = L_H + L_B \qquad\qquad (2.10)$$

where L_R = Total radial load, newtons

   L_H = Load due to lip interference and spring tension, newtons

   L_B = Beam load, newtons

Each is further described by:

$$L_R = 2\pi\,(sA_h + T) + \frac{3\,E\,I}{h^3}\,\delta \qquad\qquad (2.11)$$

where s = hoop stress in the lip element, newtons/cm^2

   A_h = effective hoop area of lip element, cm^2

   T = spring tension, newtons

   E = Young's modulus or apparent modulus, newtons/cm^2

   I = Moment of inertia of beam portion of the seal element, cm^4

   δ = Deflection of the lip element, cm

   h = Effective lip length, cm

These above methods will determine the initial lip load, which is usually more than the running load. The lip load is diminished due to stress relaxation of the rubber lip and the spring, and lubricant swell of the rubber lip over an extended period of operation. Also, the lip will wear during operation, causing a decrease in lip load. In addition, a change in Young's modulus of the rubber may occur with time, and this causes additional changes in the seal lip load.

Although the speed, runout, lip material and other factors will determine the optimum lip load for an application, it is generally recommended to have a total load of 87.5 to 123 N/meter (.5 to .7 lb/in.) of circumference for a conventional lip seal and 52.5 to 70 N/meter (.3 to .4 lb/in.) of circumference for a hydrodynamic lip seal.

2.2.4.2   Circumferential Seal Loading

Primary seal carbon-graphite elements are loaded axially against the housing flange to form the static seal and radially against the runner to form the dynamic seal. The axial spring is usually a wave washer, finger spring or several helical springs in parallel. Loading in both directions is also necessary to prevent internal leakage between seal rings and through segment gaps. Garter springs are used to impose a radial load. The axial and radial spring loads should be selected to prevent frictional hang-up on

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the flange face. Typical values of axial and radial load for a successful transmission circumferential seal are shown in Table 1.2.

The radial load in a low-pressure transmission application is due almost entirely to the spring load. (In high-pressure applications, the spring load is negligible compared to the hydraulic load.). The radial spring load is usually specified in pounds per inch of circumference in tension. For low-pressure seals, a load of 17.5 N/meter (.1 lb/in.) of circumference has proven to be adequate in successful applications.

2.2.4.3   Face Seal Loads

The axial loading of the primary element against the mating ring is usually done by a wave washer, coil springs or a metal bellows (see Section 2.2.1.4). In the low-pressure transmission applications, hydraulic loading is neglected and the spring load alone forces the seal faces together. The load must be sufficient to make the seal ring follow the runout motion of the mating ring face. The spring load is opposed by frictional restraints of the secondary seal and antirotation lock. It is usually necessary to have spring loads which will provide a primary element nose pressure of 6.89 to 26.5 N/cm^2 (10 to 30 psi). The type of spring is determined by the axial operating range of the seal. The operating range of wave washers is limited since small changes in height will cause large changes in load. If a small variation in load is required over a large operating range, either coil springs, multiple wave springs or a metal bellows should be used. Spring materials vary, but usually a 300 series stainless steel is used in helicopter transmission seals.

2.2.5   Secondary Sealing Elements

Of the seals commonly used in helicopter transmissions, only face seals require separate secondary sealing devices. The secondary seal provides a seal between the seal ring and the stationary case and allows the seal ring to move in the axial direction, compensating for shaft movement, wear or runout (see Figure 2.28). The secondary seal should exhibit low friction or drag to minimize the seal spring load. Most commonly used is an O-ring, although the following other devices can be used:

Quad - ring	Teflon lip
V-ring	Metal bellows
U-cup	Piston rings
Wedges	

Most of the above operate in the same manner as an elastomeric O-ring. The flexibility of the material allows sufficient radial squeeze to seal, yet prevents hang-up from dynamic response. The gland design usually follows standard procedures for military or industrial

reciprocating O-rings and can be found in Reference 19.

Elastomeric secondary seals offer simplicity and economy. They can be molded in a variety of materials having different mechanical and thermal properties (see Section 2.2.1.1). Generally, a nitrile or fluoroelastomer is used. The main disadvantage of elastomers is variation of coefficient of friction, elastic modulus, and mating surface diameter creating uncertain conditions with regard to frictional drag and necessary seating force. These problems can be lessened by using a spring-loaded teflon ring where the frictional drag is considerab less than elastomers and offers a higher temperature limit (477 K (500°F) maximum).

An elastomeric bellows, commonly used as a secondary seal in commercial face seals (water pumps), has been considered for transmission use. The axial movement of the seal ring is taken by the bellows flexure. Because no rubbing contact is necessary for it to perform, its frictional drag is considerably less than with packing type seals.

Piston ring secondary seals are used where extreme temperatures and pressure of the sealed medium eliminates elastomeric seals from consideration. They are not used in helicopter face seal applications since extreme conditions do not exist and leakage through the gap cannot be tolerated. These seals have been replaced by a metal bellows element, (Figure 2.29) in extreme applications.

Although not currently used in helicopter transmissions, the metal bellows will replace elastomeric secondary seals in future designs which require higher operating temperatures (above 477 K (400°F)).

Due to all-metal construction, temperature limits are comparable to those of other metal components of the seal. The basic types of bellows used are fabricated by hydraulic or rolled forming and welding preformed plates or washers.

A welded bellows requires less space and has lower spring rates than the formed type, but prediction of fatigue life of the welded joint is difficult.

2.2.6 Environmental Exclusion Devices

Currently, most helicopter transmission seals have no element designed specifically for excluding environmental contaminants. As discussed in Section 2.1.7, a seal as a complete unit for a dusty location should include:

1. A primary seal which is the initial lubricant leakage barrier.

2. An environmental exclusion device which will block atmospneric contaminants from entering the primary seal area.

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2.2.6.1   Lip Seals

Environmental exclusion is accomplished by a wiper seal, which
is molded integral to the seal (Figure 2.5) or mechanically
assembled in the seal case as shown in Figure 2.32 which is re-
produced with permission of ASLE from Reference 11.

The wiper seal must perform satisfactorily with little lubrica-
tion and various amounts of abrasion. Special materials are
usually needed for this type of service.

Wiper materials include leather, rubber (polyurethane, carboxy-
lated nitrile) and Teflon. Leather is an excellent wiper
material with good abrasion resistance, low friction and ability
to absorb or feed out lubricant when needed. Porosity can be
adjusted with impregnants to control its self-lubricating pro-
perty. Leather will polish the rubbing surface better than
elastomers, thus reducing the possibility of establishing a
lubricating film. Leather is resistant to all lubricating oils
used in helicopter transmissions. A major disadvantage of
Leather as a wiper material is a maximum temperature limit of
366 K (200°F), making it unsuitable for high-speed, high-temper-
ature applications. Elastomers, basically polyurethane and
carboxylated nitrile, exhibit a greater high-temperature limit,
up to 383 K (230°F), but have no self-lubricating features.
These elastomers are usually manufactured with a Shore hardness
of 75 to 90 and are resistant to most transmission oils.

The exclusion lip is usually short and sturdy with sufficient
flexibility to follow shaft radial runout. Since springs are
usually not used, the radial load results from lip interference
only. The amount of interference will depend on the radial run-
out and the type of service. Excessive interference may cause
the seal edge to fold over, allowing foreign material to penetrate
the contact surface. This will score the runner, abrade the lip,
and ultimately may cause complete seal failure.

2.2.6.2   Circumferential and Face Seals

Most applications with face or circumferential seals do not
provide elements designed specifically for environmental exclusion.
In some applications external drains and viscoseals are used for
leakage entrapment and external shrouds for foreign material
exclusion. In most cases, this has proven to be adequate in
meeting performance requirements.

Other industries (machine tool) have successfully used absorbent
packings, and their use in helicopter applications should be
considered. These packings can be used for both leakage entrap-
ment and exclusion. The packing must be capable of absorbing
leakage of the primary seal and must remain in intimate contact
with the runner to prevent contaminants from entering the seal

LEGEND

1 BONDED WASHER

2 GARTER SPRING

3 INNER CASE

4 OUTER CASE

5 PRIMARY SEALING ELEMENT

6 ENVIRONMENT EXCLUSION LIP
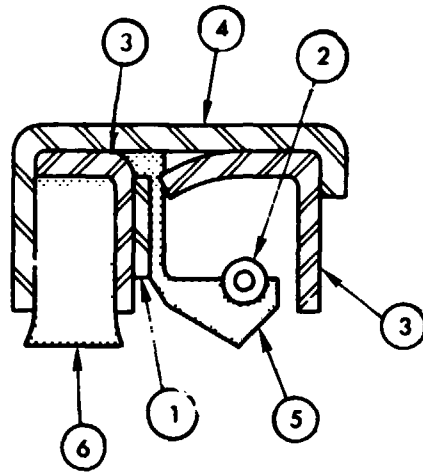
FIGURE 2.32 NOMENCLATURE
FOR MULTIPLE -LIP RADIAL LIP
SEAL (*2)

cavity. Another requirement is easy field replacement between gearbox overhauls.

Felt, a homogeneous buildup of interlocking fibers, is the most common form of absorbent packing. It is formed by mechanical work, chemical action, and heat, without spinning, weaving, knitting, thermal bonding or adhesives. There re two basic classes of felt: wool or part wool felts and synthetic fiber felts. Felt is often impregnated to improve its properties.

The major advantages of felt for entrapment and exclusion are:

1. Oil Absorption: Although its storage capacity is dependent on density, approximately 78% (varies with felt type and impregnation) of its volume can be utilized for oil absorption.

2. Chemical Resistance: Its resistance to lubricating oil is excellent.

3. Thermal Stability: Wool felt can be used at temperatures up to 356 K (180°F), while synthetic fibers can tolerate temperatures up to 422 K (300°F).

4. Resiliency: Felts are natural springs. They can maintain intimate contact in spite of wear, runout or minor misalignment.

5. Low Friction: Felt against a smooth steel surface has a dry coefficient of friction at approximately .22. Saturated with oil the friction coefficient is reduced to .015.

6. Filtration: Dry felts will provide nearly 100% blockage of foreign material particles of .7 micron or greater. Wet filtration will trap and retain smaller particles.

7. Dirt Absorption: Because of the porous, fibrous structure of felt, abrasive particles are absorbed into the voids of the felt matrix. This will prevent runner scoring.

8. Oil Wicking: Leakage past the primary seal will be absorbed by the felt. Due to the capillary properties of felt, oil feed will lubricate the felt/metal interface.

Felts can be accurately cut and fitted into any envelope. They can be retained in the transmission housing with a steel case for easy removal or mechanically fastened to the primary seal case. Although successful operation of felt at 1220 meters per minute (4000 ft/min) has been recorded, it is usually limited to a surface speed of 610 meters per minute (2000 ft/min). A problem

of shaft corrosion due to the hydroscopic nature of felt has been found in some applications.

Further research will be required to develop high-speed absorbent packings. Many types of compression packings exist(Reference 20), but testing of these packings with regard to speed capability, oil absorption, filtration, resilency, etc., is required to determine their suitability to the entrapment and exclusion principle. Present efforts are concerned with various types of asbestos packings.

2.2.7 Ring Type Seals

The need for low heat generation in high-speed transmission seals has led to a recent investigation of multiple-series split-ring seals. The ring type seal operates as a labyrinth seal in the low-pressure environment of a helicopter transmission. Its success depends on good drainage of the compartment since ring seals will leak if flooded. Ring seals usually require additional devices such as slingers or viscoseals to prevent splash oil from contacting the ring seal. Figure 2.33 shows the seal with a stationary sleeve fixed to the housing, a rotating carrier attached to the shaft, and a series of expanding split rings. The rings are self-energized radially outward, forming a static seal at the sleeve bore. The carrier side grooves and the ring sides form a dynamic labyrinth, in addition to ring gap clearance. Similar to a circumferential seal, the split rings can be used in a single or multiring design. Usually a one-piece or two-piece ring is used, since space requirements for additional rings are excessive.

Multiple-ring type seals are good dirt excluders and are easily adaptable to the secondary leakage control principle. The external ring serves to block the entry of dirt, while the inner rings retain lubricant. Lubricant leakage passing the first and second rings is drained back into the gearbox. For a more detailed discussion of ring seals, the reader is referred to Reference 21.

SLEEVE

INTERNAL DRAIN

INTERNAL SLINGER
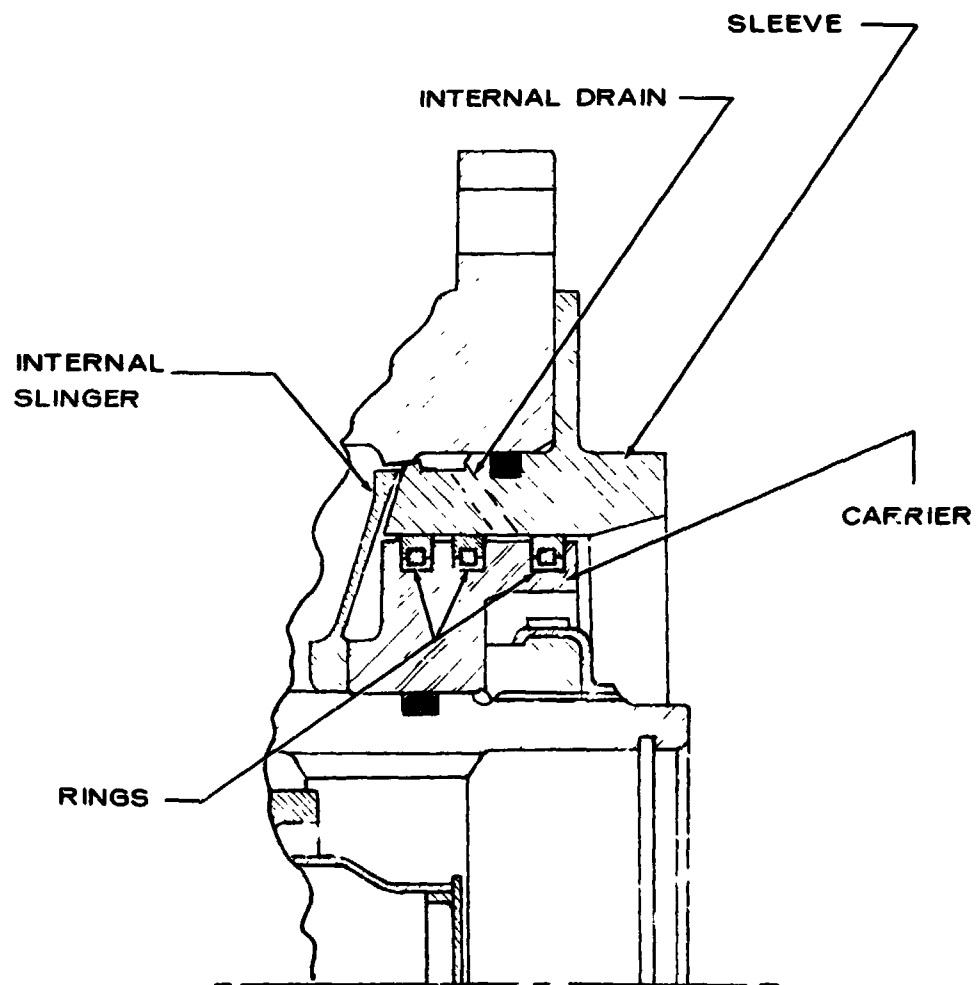
CARRIER

RINGS

FIGURE 2.33 TRIPLE-ELEMENT SPLIT RING SEAL

3.0 DESIGN CRITERIA AND RECOMMENDED PRACTICES

3.1 Seal Assembly Design

3.1.1 Speed Capability

THE SEAL SHALL HAVE A SLIDING SPEED CAPABILITY, AT THE MAXIMUM PRES-
SURE AND ECCENTRICITY EXPECTED, THAT IS EQUAL TO OR GREATER THAN THE
MAXIMUM OPERATING SPEED ANTICIPATED.

In seal selection, a primary consideration is the seal speed cap-
ability at the maximum transmission cavity pressure and runout that
will be encountered in the application. The seal should be capable
of operating satisfactorily for the full transmission life between
overhauls. The recommended practice based on experience to be follow-
ed is:

1. Conventional lip seals should be limited to a maximum speed
   of 1220 meters per minute (4000 ft/min) for a zero pressure
   differential across the seal. For seal pressures greater
   than zero, the recommendations of Table 3.1 should be
   followed.

TABLE 3.1

SEAL PRESSURE VERSUS LIP SEAL SPEED

Seal Pressure, N/cm^2 (psi)	Maximum Recommended Speed, meters per minute (ft/min)
3.44 (5) 2.06 (3) .689 (1)	458 (1500) 610 (2000) 915 (3000)

The recommended seal limiting speeds for various shaft
eccentricities at zero pressure differential are shown in
Table 3.2.

TABLE 3.2

SHAFT ECCENTRICITY VERSUS MAXIMUM LIP SEAL SPEED

Total Eccentricity - Offset & Runout, mm (in.)	Shaft Speed, rpm
0.381 (.015) 0.254 (.010) 0.203 (.008) 0.178 (.007)	1000 2000 3000 4000

2. The hydrodynamic lip seal with no special means of under lip
   cooling should be limited to a maximum speed of 2135 meters
   per minute (7,000 ft/min) for zero pressure differential
   across the seal. For seal pressures greater than zero the re-
   commended speeds are less than 2135 meters per minute (7,000
   ft/min); and the following approximate guide is recommended:

TABLE 3.3

SEAL PRESSURE VERSUS MAXIMUM HYDRODYNAMIC LIP SEAL SPEED

Seal pressure, N/cm^2	Maximum Recommended Speed, meters per minute (ft/min)
3.44 (5 psi) 2.06 (3 psi) 0.689 (1 psi)	610 (2000) 1220 (4000) 1830 (6000)

The allowable shaft-to-housing eccentricity of a nydrodynamic
lip seal is less than that of a conventional lip seal since
hydrodynamic lip seals are usually designed with less inter-
ference than a conventional lip seal. In high-speed applica-
tions of a hydrodynamic lip seal (between 1220 to 2135 meters
per minute (4000 ft/min to 7000 ft/min.)), a low life is
anticipated if the eccentricity is greater than .127 mm
(.005 in.).

3. The circumferential seal has a maximum recommended speed
   that is dependent on shaft runout because leakage tends to
   increase as shaft runout increases.

TABLE 3.4

SHAFT RUNOUT VERSUS LIMITING   CIRCUMFERENTIAL SEAL
SPEED(FOR ZERO DIFFERENTIAL PRESSURE)

Shaft runout (TIR), mm(in.)	Maximum recommended speed, meters per minute (ft/min)
.0254 (.001) .127 (.005) .254 (.010) .381 (.015) .508 (.020)	4560 (15,000) 3660 (12,000) 2440 (8,000) 1830 (6,000) 1220 (4,000)

For pressures above zero the above allowable speeds should
be reduced; not enough data is available to provide a guide.

Note that the circumferential seal is used only for very low
pressure differential (0.689 N/cm^2 (1 psi)); therefore,

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speed dependence on pressure is not an important considera-
tion for these applications.

4. The <u>face seal</u> is relatively insensitive to shaft runout and
pressure condition in a helicopter seal application. There-
fore, the maximum recommended speed is imposed by structural
or seal heat generation considerations. Operation to 6096
meters per minute (20,000 ft/min) and greater is possible
with proper cooling.

3.1.2 Pressure Capabilities

THE SEAL SHALL HAVE A PRESSURE CAPABILITY THAT MEETS OR EXCEEDS THE
MAXIMUM OPERATING PRESSURE.

In helicopter transmissions, the pressure differentials are usually
less than 3.44 (N/cm^2 (5 psi) across the seal. The seals should be
capable of sustaining the maximum anticipated pressure for the full
transmission life during overhauls. Paragraph 3.1.1 contains the
recommended pressure-speed relations for <u>lip type seals</u>.

Because of leakage, <u>circumferential seals</u> should not be used at
pressures over 0.689 N/cm^2 (1 psi) and should not be used where a
pressure head builds up due to lubricant flooding of the seal. Use
of a windback on the oil side of the seal mitigates the seal leakage
problem due to flooding and pressure.

<u>Face seals</u> can readily operate at the maximum pressure expected in the
helicopter transmission application.

3.1.3 Temperature Capability

ALL SEAL COMPONENTS SHALL BE CAPABLE OF FUNCTIONING (FOR THE FULL
TRANSMISSION LIFE) OVER THE FULL TEMPERATURE RANGE EXPECTED IN OPERA-
TION.

<u>Conventional lip seal</u> temperature limits are primarily determined by
the type of elastomer used for the lip. The heat generated in the
thin lubricating film under the lip must be considered in addition to
the ambient temperature. The recommended temperature limits are
shown in Table 3.5.

TABLE 3.5

RECOMMENDED AMBIENT TEMPERATURES FOR LIP SEALS

Material Type	Material Temperature Limit	Underlip Temperature Rise (approx.)		Recommended Ambient Temperature Limit	
		305 meters per minute (1000 ft/min)	610 meters per minute (2000 ft/min)	305 meters per minute (1000 ft/min)	610 meters per minute (2000 ft/min)
1. Nitrile	380 K (225°F)	28 K (50°F)	39 K (70°F)	352 K (175°F)	341 K (155°F)
2. Silicone	450 K (350°F)	28 K (50°F)	39 K (70°F)	422 K (300°F)	411 K (280°F)
3. Fluoro-elastomer	477 K (400°F)	28 K (50°F)	39 K (70°F)	449 K (350°F)	438 K (330°F)

At speeds above 610 meters per minute (3000 ft/min) the allowable ambient temperatures are lower, but not enough underlip temperature rise data is available to set limits.

Hydrodynamic lip seals operate with lower underlip temperature rises than lip seals because of the better lubrication and lower lip loads. Therefore. the recommended ambient temperature limits are higher than those of conventional lip seals. At high speeds, the underlip temperature probably reaches the boiling points of the lubricant (approximately 505 K (450°F)). The recommended temperature limits are shown in Table 3.6.

TABLE 3.6

RECOMMENDED AMBIENT TEMPERATURES
FOR HYDRODYNAMIC LIP SEALS

Material Type	Material Temperature Limit	Underlip Temperature Rise (approx.)		Recommended Ambient Temperature Limit	
		305 meters per minute (1000 ft/min)	610 meters per minute (2000 ft/min)	305 meters per minute (1000 ft/min)	610 meter per minute (2000 ft/min)
1.Nitrile	380 K (225°F)	17 K (30°F)	28 K (50°F)	363 K (195°F)	352 K (175°F)
2.Silicone	450 K (350°F)	17 K (30°F)	28 K (50°F)	433 K (320°F)	422 K (300°F)
3.Fluror-elastomer	477 K (400°F)	17 K (30°F)	28 K (50°F)	450 K (370°F)	449 K (350°F)

All speeds above 610 meters per minute (2000 ft/min), the allowable ambient temperatures are lower, but not enough data is available to provide a guide.

Circumferential seals are composed of carbon-graphite and metal components and contain no elastomers; therefore, temperature capability is higher than that of lip type seals. Coking of the lubricant, rather than seal material thermal degradation, will be the limiting temperature consideration, and this is near 505 K (450°F) for conventional lubricants. It is recommended that the circumferential seals be limited to ambient temperatures of 477 K (400°F) when used with currently available lubricants at speeds below 610 meters per minute (2000 ft/min). At higher speeds coking will be a problem (due to heat generated in the lubricating film) if the ambient temperature is not lower than 477 K (400°F)

Those face seals that contain elastomeric secondary eals are limited by thermal degradation of the elastomer. In face seals, no heat is generated at the secondary seal; therefore, the ambient temperature establishes the temperature limits. It is recommended that face seals with elastomeric secondary seals be limited as shown in Table 3.7.

TABLE 3.7

RECOMMENDED AMBIENT TEMPERATURES FOR FACE SEALS
WITH ELASTOMERIC SECONDARY SEALS

Secondary Seal Material Type	Recommended Ambient Temperature
1. Nitrile	380 K (225°F)
2. Silicone	450 K (350°F)
3. Fluoroelastomer	477 K (400°F)

Those face seals that contain carbon or bellows secondary seals are
limited by the coking of the lubricant. Therefore, the recommended
temperature limit is 505 K (450°F) for currently used lubricant at
speeds below 610 meters per minute (2000 ft/min). At higher speeds
the allowable ambient temperature will be reduced due to heat
generated at the primary seal.

3.1.4 Misalignment, Shaft Runout and Shaft Roundness

THE SEAL SHALL BE CAPABLE OF SUCCESSFUL OPERATION AT THE MAXIMUM
SHAFT-TO-HOUSING MISALIGNMENT, SHAFT RUNOUT AND SHAFT ROUNDNESS

Excessive misalignment has detrimental effects on performance for all
types of seals. Misalignment tends to disturb the lubricating fluid
film, and this causes increased leakage and wear rates. It results
from dimensional tolerances of the support components and support
bearing internal clearance. By tightly controlling tolerances and by
rigidly supporting the shaft, misalignment can be reduced.

Data in the literature, Reference 22, indicates that conventional lip
seals will operate successfully if the speed-eccentricity relation-
ship of Table 3.2 is satisfied.

In addition to eccentricity, the shaft roundness is a major considera-
tion. It is recommended (based on work of Reference 13) that the
shaft be round within 5.08 microns (200 microinches).

Hydrodynamic Lip Seals have been operated successfully at about the
same eccentricity of a conventional lip seal, although hydrodynamic
lip seals are designed with less interference. Not enough data is
available at the higher speed limit of hydrodynamic lip seals to
state eccentricity limits, but from some limited tests it has been
determined that it should be less than .127 mm (.005 inch). Shaft
geometry considerations are usually the same as with conventional lip
seals.

Circumferential seals are sensitive to runout in that as runout increases, leakage increases. A rule of thumb has been to limit shaft runout to .0005 m/m (.0005 in./in.) of diameter. It is recommended that Table 3.4 be followed with regard to limiting runout. Offset (static eccentricity) is not a limiting factor with circumferential seals since the primary carbon-graphite elements float radially in the seal case.

Face Seals are less sensitive to shaft runout than circumferential seals. But as speed increases, the allowable runout decreases. It is recommended that the runouts be as follows:

TABLE 3.8

SPEED VERSUS SHAFT RUNOUT FOR FACE SEALS

Speed, meters per minute (ft/min)	Allowable Runout of Shaft, mm(TIR)
0 to 1522 (0 to 5000)	.508 (.020 in.)
1522 to 3044 (5000 to 10,000)	.38 (.015 in.)
3044 to 6090 (10,000 to 20,000)	.254 (.010 in.)

As with circumferential seals, face seals are not limited by static offset of housing and shaft centerlines.

3.1.5 Axial Location and Axial Runout

THE SEAL SHALL BE CAPABLE OF SUCCESSFUL OPERATION THROUGHOUT THE FULL AXIAL DISPLACEMENT OF THE SHAFT DUE TO TOLERANCE BUILDUP AND THERMAL GROWTH.

Shaft-riding seals (lip and circumferential seals) are not affected by axial displacement of the shaft, as long as sufficient length of shaft surface has been prepared to mate with the lip or carbon element.

Radial face seals are manufactured to tolerate a certain amount of axial displacement, dependent on the type of spring and the spring rate. For a small operating range (approximately ± .76 mm (±.030 in.)), wave washers can be used. For larger displacement (up to ±1.27 mm (±.050 in.)), coil springs, multiple wave springs, and metal bellows are suitable. Above ±1.27 mm (±.050 in.)), shaft-riding seals should be considered.

3.1.6 Vibration Considerations

FLEXIBILITY MOUNTED SEAL COMPONENT SHALL BE DAMPED TO PREVENT
RESONANT VIBRATION.

Lip seals and circumferential seals are usually insensitive to vibra-
tion. Also, face seals with elastomer secondary seals are usually
insensitive, but the design of bellows face seals should consider
natural frequency of the bellows and damping properties. An analysis
of the natural frequency of the primary element with its bellows
attached to the stationary housing serves as a simple model of the
vibration properties. Damping can be achieved by dampers that operate
against the bellows or primary ring; also, oil collecting (by surface
tension) between the bellows plates serves to dampen the seal.

3.1.7 Environment Limitations

THE SEALS SHALL BE PROTECTED FROM FOREIGN CONTAMINATION, AND
THE EXCLUSION DEVICE SHOULD BE CAPABLE OF OPERATION AT BOTH HIGH
AND LOW AMBIENT TEMPERATURES.

Seals exposed to the environment must be protected from contamination
and abrasion of the seal primary surfaces. Exclusion devices
include:

  1. Sheet-metal shrouds
  2. Wiper seals
  3. Slingers
  4. Labyrinths
  5. Absorbent packings

The selection of seal primary and secondary materials should include
consideration of the temperature extremes that the seal might exper-
ience. Often overlooked are the effects on elastomeric elements with
low-temperature environments. Silicone rubber has been shown to be a
superior material for cold-weather operation. (See Table 2.6.)

3.1.8 Seal Envelope Requirements

Shaft Requirements

THE SHAFT SHALL LOCATE AND HOLD THE ROTATING SEAL COMPONENT
WITH A MINIMUM OF DISTORTION.

Lip type seals often run on a thin shaft spacer that is pressed onto
the shaft. Any shaft out-of-roundness or notch in the shaft is re-
flected into the thin spacer. Therefore, the shaft under the spacer
should be round within 5.08 microns (200 microinches). It is recom-
mended that if a runner is used with circumferential seals, the runner
O.D. be concentric with the shaft centerline within .005 m/m (.0005 in./
in.) of diameter and round within 12.7 microns (500 microinches).

Mating rings for <u>face seals</u> may be a loose fit with an O-ring installed to prevent leakage along the shaft and to allow for minimum distortion. If a one-piece mating ring is used, the clamping surfaces (bearing race, spacer, shaft shoulder) should be lapped to mitigate clamping distortions.

Housing Requirements

THE HOUSING SHALL LOCATE, HOLD AND SEAL THE STATIONARY
SEAL COMPONENT.

The stationary seal cases are usually press fitted into the transmission housing. However, in some applications the housing is held in place and sealed by an O-ring. The amount of interference fit needed is determined by the differential expansion rates of the materials of the seal case and the transmission housing and the operating temperature.

To obtain squareness of the seal case with respect to the bore centerline, the seal case is located by a housing shoulder or in an axial location by dimensional control from a reference face on the transmission housing. As a precautionary measure, retaining rings should be used to prevent the seal from moving axially.

Radial and Axial Space Requirements

THE SEAL ENVELOPE SHALL BE LARGE ENOUGH TO PERMIT THE
USE OF OPTIMUM SEAL COMPONENTS.

As shown in Tables 2.3 and 2.4, each seal type requires a different space envelope. The use of oil windbacks and exclusion devices will increase the space required. By not providing sufficient space, the seal design will be compromised, while providing too much will add weight and cost to the gearbox. The envelopes illustrated in the aforementioned tables provide data for gearbox preliminary design. Detail design of the space envelope will depend on other special requirements for the seal; e.g., increased operating range, and puller groove for seal removal.

3.1.9 Leakage Limitation and Control

THE LEAKAGE FROM THE SEAL SYSTEM SHOULD BE LESS THAN
5 CC/HOUR.

Leakage of 1 cc/hour is a goal that is acceptable for most applications; higher leakage is tolerated in some cases, but it should be less than 5 cc/hour.

Shaft out-of-roundness and runout promote leakage in conventional lip

seals. However, hydrodynamic lip seals, because of helix grooves which pump the leakage flow back as fast as it leaks out, have a greater tolerance to imperfect shaft geometry and motion.

Circumferential seals will leak when operating under a flooded condition if the carbon segments "hydroplane" on the lubricating fluid. Shart runout increases this "hydroplaning" tendency. Thus, it is recommended that circumferential seals be operated without flooding the seal. This can be accomplished with generous drainage capacity, slinger and helical groove windbacks.

Face seals will leak if the lubricant is on the inside diameter of the primary seal (dam); centrifugal force tends to drive the lubricant through the primary seal. Therefore, it is recommended that the lubricant be on the outside diameter of the primary seal. In this case centrifugal force will tend to inhibit leakage.

3.1.10 Life Considerations

THE LIFE OF THE SEAL SHALL BE CONSISTENT WITH THE SCHEDULED GEARBOX OVERHAUL INTERVALS.

There is no known method to determine seal life from mathematical models. The only reliable approach in determining life is by extensive testing.

A method for determining the wear life of conventional lip seals has been published (Reference 6). It is based on the seal's capacity to absorb work. This is termed the "life factor" and must be determined experimentally. Once determined for specific fluid, fluid temperature, eccentricity and shaft finish, the life factor can be used to predict seal life for various shaft sizes and speeds. A similar approach probably can be used with hydrodynamic seals.

The preceding method for predicting life does not include the effects of contaminants such as air-entrained dust. Unfortunately, hydrodynamic lip seals tend to "pump" the air-entrained dust into the primary seal, and this can be very detrimental to seal life.

Some evidence suggests that lip seal life is also a function of the change in elastomer properties which, in turn, are a function of the fluid film temperature in the primary seal. Thus, a further check on the feasibility of lip seal life is provided if there exists some knowledge of the change in elastomeric properties with temperatures and with a specific fluid.

The life of circumferential seals and face seals is not amenable to calculation by mathematical models. However, because of the excellent wear characteristics of carbon seal materials, the life of these seals should be more than suitable. Air-entrained dirt will tend to reduce life. But data on dirt effects are not available.

3.1.11 Thermal and Lubrication Considerations

THE SEAL SHALL BE SUPPLIED WITH AN ADEQUATE AMOUNT OF
LUBRICANT FOR COOLING AND LUBRICATION.

An adequate oil supply should be provided for lubrication and cooling
to keep the seal temperature within the limits specified in section
3.1.3.

Although the amount of heat generated at the seal interface is not
readily calculated, an approximation can be made with full film models
by assuming a viscosity and film thickness. This will provide an
approximate value of heat generation. From this approximation the
cooling flow necessary can be estimated. The heat transfer paths of
conduction and convection should be evaluated to determine if special
oil heat transfer features (e.g., mating ring radial slots, under-
runner cooling) are required.

Conventional lip seals and hydrodynamic lip seals operate with fluid
film lubrication. Therefore, these seals must be supplied with
adequate oil for lubrication and cooling. Oil spray, splash or partial
flooding has been proved to be suitable. Under-runner cooling would
be particularly effective in controlling oil film temperate in high
speed applications.

Circumferential seals can run without lubrication. For high speeds,
under-runner cooling is a particularly effective means of seal cool-
ing. For high-speed operation without lubrication the radial spring
loading must be less than that of lubricated operation.

Face seals can run without lubrication, but in the usual transmission
application they are lubricated. For high speeds, outward radial
oil flow through holes drilled in the rotating mating ring is an
effective means of seal cooling. For high speed operation without
lubrication the axial spring loading must be less than that of lubri-
cated operation.

3.1.12 Drainage Requirements

THE SYSTEM SHALL HAVE ADEQUATE DRAINAGE TO AVOID PRESSURE
BUILDUP ON THE SEAL.

Pressure buildup due to improper drainage shall be avoided. However
proper cooling and lubrication of seal should be provided. The
drainage area should allow for the maximum amount of oil that can
enter the seal area.

Although adequate drainage is needed to prevent flooding, lip seals
must be lubricated, since they will not run "dry". Both circumferen-
tial seals and face seals are capable of operating without any lubrica-
tion.

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3.1.13 Quality Control

THE SEAL SHALL BE INSPECTED THOROUGHLY BEFORE BEING
INSTALLED.

Seal surfaces should be inspected for damage, variation of geometry,
dimensions and finishes on a 100% basis. Of particular importance
are the following:

. Lip Seals

    1. Nicks and scratches on the primary seal
    2. Runner out-of-roundness
    3. Radial lip load (or lip opening pressure)

  Circumferential Seals

    1. Nicks and scratches on the primary seal
    2. Runner out-of-roundness and taper
    3. Axial and radial spring loads

  Face Seals

    1. Nicks and scratches on the primary seal
    2. Primary seal flatness
    3. Axial spring load

Inspection of material properties and static testing can be done on
a sample basis from each production lot. Many of these inspection
requirements are performed by the seal manufacturer since special
equipment not usually available to the seal user is required.

In some applications it may be necessary to perform special material
tests of elastomers. These tests are specified as ASTM test standards
and shown in Table 3.9.

TABLE 3.9

ASTM STANDARD TESTS FOR ELASTOMERS

Requirement	Test Method
Heat aging	D-573, D-865
Compression set	D-395(B)
Weather resistance	D-1171
Load deflection	D-575
Tear resistance	D-624
Flex resistance	D-430
Abrasion resistance	D-394
Adhesion by curing	D-429
Water resistance	D-471
Resilience	D-945

3.1.14 Packaging and Shelf Life

THE SEAL SHALL BE PACKAGED AND STORED TO MINIMIZE DAMAGE
AND DETERIORATION.

It is recommended that all seal types be polyethylene skin packaged
for minimum contamination, cushioned, and placed in a folding carton.
The seals shall be stored in a temperature/humidity-controlled area.
Shelf life will depend on elastomers used and storage conditions.
Approximate shelf lives are given in Table 2.5. Additional data con-
cerning shelf lives can be obtained from Reference 23.

3.1.15 Installation and Disassembly

THE SEAL SHALL BE INSTALLED AND REMOVED WITH SPECIALLY
DESIGNED TOOLS TO PREVENT DAMAGE.

The housing must be manufactured with a burr-free lead-in chamfer to
prevent axial scoring of the case. Bore sealers can be used as a
precautionary measure to fill any voids. The runner must also have
a burr-free lead-in chamfer or radius to prevent tearing or nicking
of the primary element.

Tools designed specifically for seal housing installation should be
provided. These tools shall be designed to produce the least amount
of press-fit distortion and accurately locate the primary seal surface.

A universal lip seal removal tool, (Figure 2.16) is recommended for
convenience, although after removal the seal is not reusable. Special
removal tools, (Figure 2.18) for face and circumferential seals will
permit the reuse or rework of some.

All O-rings and rubber lips should be lubricated with a grease type
lubricant before assembly to prevent tear damage. The assembly pro-
cedure should not require that O-rings pass over drilled holes or
local cored areas.

3.1.16 Test Criteria

SEALS SHALL BE SUBJECTED TO A DYNAMIC TEST BEFORE ACCEPTANCE
AS A QUALIFIED COMPONENT.

In general, most slow-speed and moderate-speed seals are qualified in
initial full-scale transmission tests. Some high-speed seals require
bench development testing prior to testing in the gearbox. It is
recommended that test rigs be comprised of actual transmission com-
ponents in lieu of using a commercial seal test machine. To obtain
meaningful data, the maximum expected runout and misalignment shall
be built into the test rig. Also, the maximum expected cavity
pressure should be used in the test.

The criteria for acceptable seal leakage depends on the application, but shall be limited to less than 5 cc/hr leakage for most high-speed applications.

3.2   Seal Component Design

3.2.1 Primary Seal Element Design

3.2.1.1   Lip Seals

THE LIP MATERIAL AND DESIGN SHALL PROVIDE FOR ACCURATELY CONTROLLED RADIAL LIP LOADING FORCE THROUGHOUT THE SEAL DESIGNED LIFE.

It is recommended that the lip material be compatible with the lubricant and temperature variations of the application. Fluoroelastomers and silicone should be considered for high-speed, high-temperature applications, while nitriles and acrylics should be used only in lower temperature accessory applications. Fluoroelastomers are able to withstand high temperatures and are generally resistant to lubricating oils. Silicone is less expensive and has excellent high and low thermal properties.

The lip design should conform to optimum parameters set forth in Section 2.2.1.1 and in References 12, 13 and 14. Completely molded seals are recommended for new applications in lieu of trimmed seals.

3.2.1.2   Hydrodynamic Lip Seals

THE HYDRODYNAMIC LIP SEAL SHALL BE USED FOR INCREASED RELIABILITY AND/OR HIGH-SPEED OPERATION.

Hydrodynamic seals provide improved leakage control and longer life (in the absence of air-entrained dirt) than conventional lip seals. This is particularly true under the effects of shaft imperfections. Therefore, hydrodynamic seals are a recommended choice over conventional seals. In general, the lip loading is less than that of conventional lip seals (this lower loading can be used because of the beneficial effect of the helical groove pumping). In general, this lip design is similar to that of the conventional seal; however, published design data is scarce. Therefore, seal manufacturers should be consulted with questions relating to lip design.

A new lip design currently being investigated features a sinusoidal lip instead of a straight circumferential lip. Initial test results have shown that the lip design offers low heat generation at the shaft/seal interface, pumping efficiency less than a hydrodynamic rib design, but no reduction in pumping action

after normal wear of the seal has occurred. This design has potential for future helicopter transmission seal applications.

3.2.1.3   Circumferential Seals

THE PRIMARY ELEMENT SHALL HAVE WEAR LIFE AND THERMAL PROPERTIES ADEQUATE FOR HIGH-SPEED OPERATION OVER THE FULL OVERHAUL LIFE OF THE TRANSMISSION.

Circumferential seals are recommended in lieu of face seals for seal pressures less than .689 N/cm^2 (1 psi) and when axial motion is excessive or space envelope is limited. The three-ring segmented design is preferred to the one-ring or two-ring design, because of cost.

The primary ring and cover ring are lapped flat together and then mated to a backer ring to prevent leakage. Also, the outside diameter of the primary ring and inside diameter of the cover ring must be in intimate contact.

The material of the primary rings should be carbon-graphite. The bore of the carbon-graphite rings, which forms the primary seal with the runner outside diameter, should be made round within   25.4 microns (1000 microinches). The bore finish should be ground to .254 to .508 micron (10 to 20 microinches). Finer finishes will improve initial wear-in performance and can be obtained by polishing.

3.2.1.4   Face Seals

THE PRIMARY ELEMENT SHALL HAVE WEAR LIFE AND THERMAL PROPERTIES ADEQUATE FOR HIGH-SPEED OPERATION OVER THE FULL TRANSMISSION OVERHAUL LIFE.

Carbon-graphite, having exceptionally good wear and thermal properties, is the recommended material for the primary element of face seals. It is readily available and easily machined, and it has a low coefficient of friction, high thermal conductivity, and good stability at high temperatures.

The grades of carbon-graphite as shown in Table 2.8 are recommended for the temperature conditions currently experienced in helicopter transmissions. Higher temperature carbon-graphites are available, but these grades are more expensive and in some cases exhibit other deficiencies such as exuding or absorbing water.

The functional dynamic surface, the seal nose, must be lapped flat to at least three helium light bands and have a good surface finish (.254 to .508 micron (10 to 20 microinches) AA). Waviness

deviations from flatness will result in excessive film thickness and leakage.

The primary element must be capable of moving axially and provide for some angular misalignment. The element is stopped from circumferential movement by antirotational devices, such as keys, pins, tangs or a metal bellows. An internal O-ring or metal bellows allows axial movement and provides a seal between the primary element and seal housing.

3.2.2 Mating Surface Design

3.2.2.1   Cylindrical Runners

THE RUNNER OUTSIDE DIAMETER SHALL MEET RECOMMENDED ROUNDNESS AND SURFACE FINISH SPECIFICATION.

The cylindrical runners used for lip and circumferential seals should be made from a hardened, corrosion-resistant or plated steel with high thermal conductivity. Generally, <u>lip seal</u> runners should have:

1.   Hardness of Rockwell C45 minimum, or greater.

2.   Surface roughness of .38 micron (15 microinches) AA (.254 to .508 micron (10-20 microinches) AA is a normal tolerance).

3.   Surface to be plunge ground with no lead.

4.   Out-of-roundness less than 5.08 microns (200 microinches) with number of lobes to be minimum practical.

<u>Circumferential seal</u> runners should have:

1.   Minimum hardness of Rockwell C55, a hard chromium-plated surface recommended per AMS 2406D.

2.   Maximum surface roughness of .203 micron (8 microinches) AA.

3.   Surface free of taper.

4.   Out-of-roundness less than 12.07 microns (500 microinches) with number of lobes to be minimum practical.

All runners should be designed for minimum distortion at installation and shall be free of surface defects.

3.2.2.2   Face Seals - Mating Rings

THE MATING RING SHALL BE CONSTRUCTED TO PROVIDE MINIMUM
CLAMPING DISTORTION OF THE SURFACE THAT FORMS THE PRIMARY
SEAL.

If clamping through the mating ring is desired, then it is
recommended that the mating ring be of the two-piece type con-
sisting of a clamping ring that is fixed to the shaft and a
seal ring that is loosely pinned or keyed to the clamping ring
so that no clamping distortion will be transferred between rings.
An O-ring radially locates the seal ring to the clamping ring
and provides an internal seal. Both rings should be corrosion-
resistant, have high thermal conductivity, and have approximately
the same coefficient of thermal expansion. The seal ring mating
surface should be lapped flat to a minimum of three helium light
bands with a surface roughness of .1016 micron (4 microinches)
AA maximum. Waviness of the surface can cause excessive film
thickness and a pumping action if runout is present. A chronium-
plated (AMS 2406D) seal surface is recommended if hardenable
stainless steel is not used.

3.2.3 Case Design

THE CASE SHALL SUPPORT THE PRIMARY SEAL ELEMENT AND EFFECTIVELY
SEAL TO THE HOUSING BORE.

The case shall be fabricated from stainless steel or a lightweight
alloy. The case should have sufficient thickness to prevent gross
distortion, especially with circumferential seals where the case
flange serves as a secondary sealing surface. Distortion due to
press fits can be minimized by matching housing and case coefficient
of expansions or by using a loose fit with O-rings to seal at the
case outside diameter.

3.2.4 Loading Devices

3.2.4.1   Lip Seal Loading

THE SEAL SHALL BE LOADED TO PROVIDE INTIMATE CONTACT WITH
THE RUNNER (OR SHAFT).

The garter spring is required to maintain adequate lip pressure
on the runner. To prevent excessive leakage during dynamic
operation, the spring must maintain a lip load which will provide
a thin film thickness between the lip and runner.

With hydrodynamic seals, the lip load can be reduced since
dynamic sealing is accomplished by visco pumping action of the
helical grooves. The load required is determined by the force

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necessary to form a static seal. In general, conventional lip loads should be 87.5 to 123 N/meter (.5 to .7 lb/in) of circumference, which usually produces a contact width of .254 to .762 mm (.010 to .030 in.). Hydrodynamic lip loads should be 52.5 to 70 N/meter (.3 to .4 lb/in.) of circumference.

3.2.4.2   Circumferential Seal Loading

THE SPRING LOADS SHALL PERMIT THE PRIMARY ELEMENTS TO OVER-COME RADIAL FRICTIONAL HANG-UP ON THE CASE FLANGE.

The spring loads, both axial and radial, should be of sufficient magnitude to allow the primary element bore to follow the runner eccentricity without any frictional hang-up caused by the axial load. The axial load is required to seal the primary element against the case web to form an internal seal and is in the range of 70 N/meter (.4 lb/in.) of circumference.

The radial garter spring tension load for transmission seals is usually two times greater than loads used for gas film seals because the higher viscosity oil increases hydrodynamic lift and is in the range of 17.5 to 35.0 N/meter (.1 to .2 lb/in.) of circumference.

3.2.4.3   Face Seal Loads

THE LOAD SHALL KEEP THE SEAL SURFACES IN INTIMATE CONTACT THROUGHOUT THE OPERATING RANGE

A spring load producing a face pressure of 6.89 to 20.7 N/cm^2 (10 to 30 psi) should be used. The spring stiffness should be low enough to permit a deviation of spring height due to axial tolerance of the assembly and wear. In general, wave washers can be used for most applications; however, coil springs provide lower spring rates. A welded metal bellows, which also provides the secondary seal function, can be used in place of coil springs.

The spring load must be of sufficient magnitude to prevent face separation due to secondary seal friction coupled with axial runout.

3.2.5 Secondary Sealing Elements (Face Seals)

THE SECONDARY SEAL ELEMENT SHALL PROVIDE ZERO INTERNAL LEAKAGE WHILE ALLOWING THE PRIMARY ELEMENT AXIAL AND ANGULAR FREEDOM.

Face seals can use elastomeric elements in many applications since fluoroelastomers maintain their physical properties up to 477 K (400°F). The more expensive metal bellows type is used where elastomer

degradation is expected to be excessive.

With elastomeric secondary seals, particular attention must be paid
to the radial squeeze between the primary carbon-graphite element
and the case. Excessive squeeze will cause frictional hang-up of the
primary element and possible separation of the primary seal. With
inadequate squeeze, internal leakage past the secondary seal will
occur. It is also recommended that attention be given to possible
changes in radial squeeze due to elastomer swell.

3.2.6 Environmental Exclusion Devices

FOR OPERATION IN EXTERNAL ENVIRONMENTS HAVING A HIGH CON-
CENTRATION OF AIRBORNE DEBRIS, THE SEAL ASSEMBLY SHALL HAVE
DIRT EXCLUSION DEVICES TO PROTECT THE PRIMARY SEAL.

The following is the recommended practice for each seal type:

1. Lip seals - Environmental exclusion is accomplished by a
   wiper seal or a dust lip molded integral with the primary
   lip.

2. Face seals and circumferential seals - Environmental
   exclusion is provided by an absorbent packing. The packing
   should be attached to the seal case and easily removable.

3.2.7 Ring Type Seals

MULTIPLE-ELEMENT RING SEALS SHALL BE USED IN HIGH-SPEED APPLICATIONS
TO PROVIDE MINIMUM HEAT GENERATION.

A minimum of three rings should be used to control leakage. The one-
piece straight gap ductile iron ring offers the lowest cost. The
two-piece Teflon-shrouded step-gap design should be considered if
excessive leakage occurs with the straight gap design.

Both sleeve and carrier material should have similar coefficients of
thermal expansion and be preferably a hardened 400 series stainless
steel or a plated standard steel.

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GLOSSARY

L_H	load due to lip interference and spring tension, newtons
L_R	total radial load, newtons
L_s	seal life, hr
M	friction torque, newtons x mm
N	speed, rpm
P_d	pressure drop, atm
P_e	energy dissipation
P_p	beam load, newtons
P_r	radial lip load, newtons
Q	lubricant flow, m^3/sec
S	sliding speed, meters per minute
T	spring tension, newtons
α	lip angle, rad
δ	deflection of the lip element, cm
ζ	coefficient of friction in dry sliding
μ	fluid viscosity, centipoises
ω	angular velocity, rad/hr

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