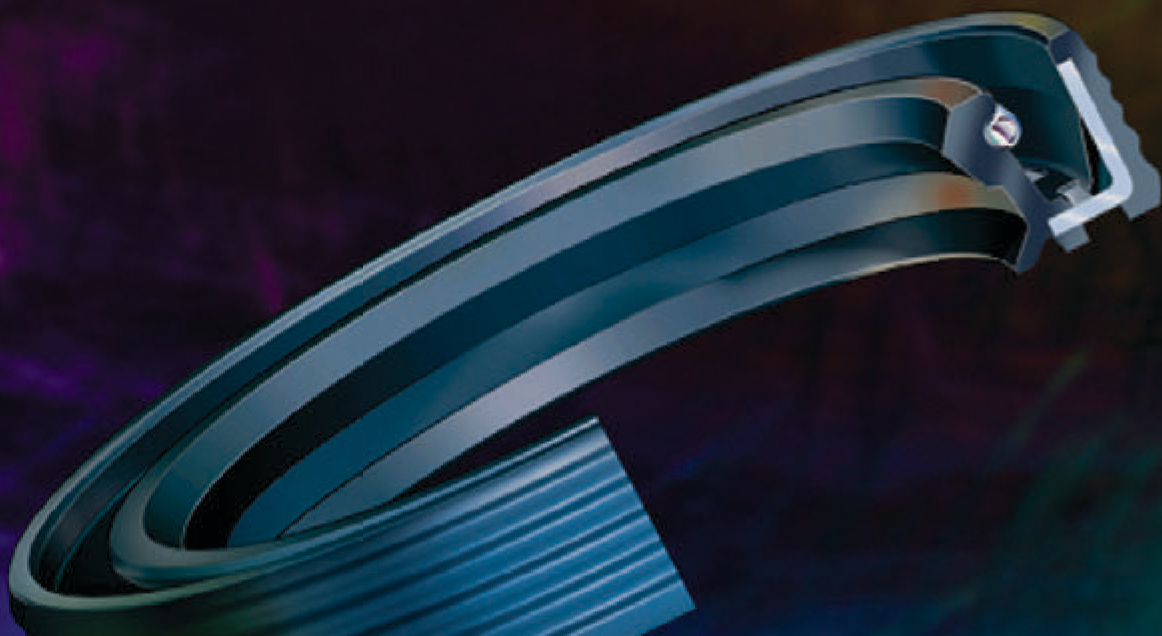
R.L. HUDSON & COMPANY

SHAFT SEAL DESIGN & MATERIALS GUIDE



We Supply Solutions.

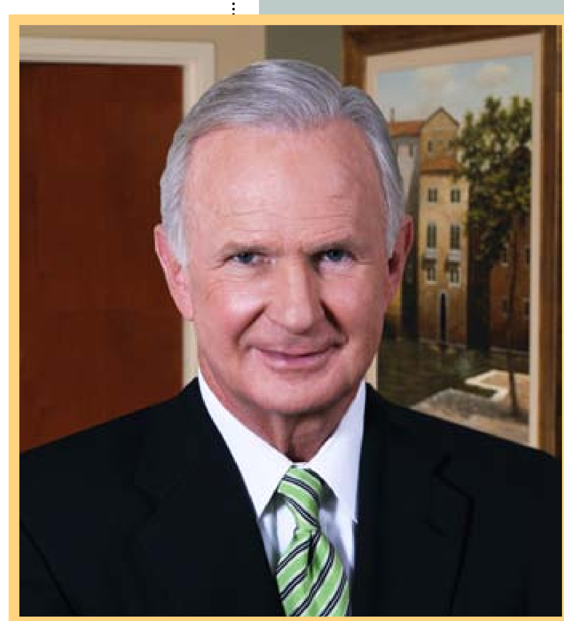
Over two decades ago, RL Hudson began with a commitment to bring our customers a unique combination of quality products, innovative design solutions, and excellent customer service. We started small and built the business one satisfied customer at a time. And although we now supply O-rings, shaft seals, rubber hoses, custom-molded products, metal components, and assemblies to leading manufacturers all across America, our commitment to quality, design assistance, and customer service is as strong as ever.

We work in partnership with our customers, always looking for the solutions that best fit their needs. Our territory managers, account managers, and in-house engineering and quality staff work together to provide the highest level of customer service to accounts of all sizes.

What is it that really makes RL Hudson unique? It is our fierce dedication to anticipating and meeting the needs of our customers. This ongoing commitment to customer satisfaction includes a desire to share our technical knowledge. As we did with our *O-Ring Design & Materials Guide*, we've prepared this *Shaft Seal Design & Materials Guide* to help you select the right seals for your projects. If you have questions, please call us. We'll do whatever it takes to make sure you get the sealing solutions you need.

Rick Hudson
CEO

"What is it that really makes R.L. Hudson & Company unique? It is our fierce dedication to anticipating and meeting the needs of our customers."



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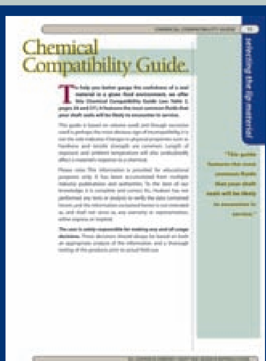
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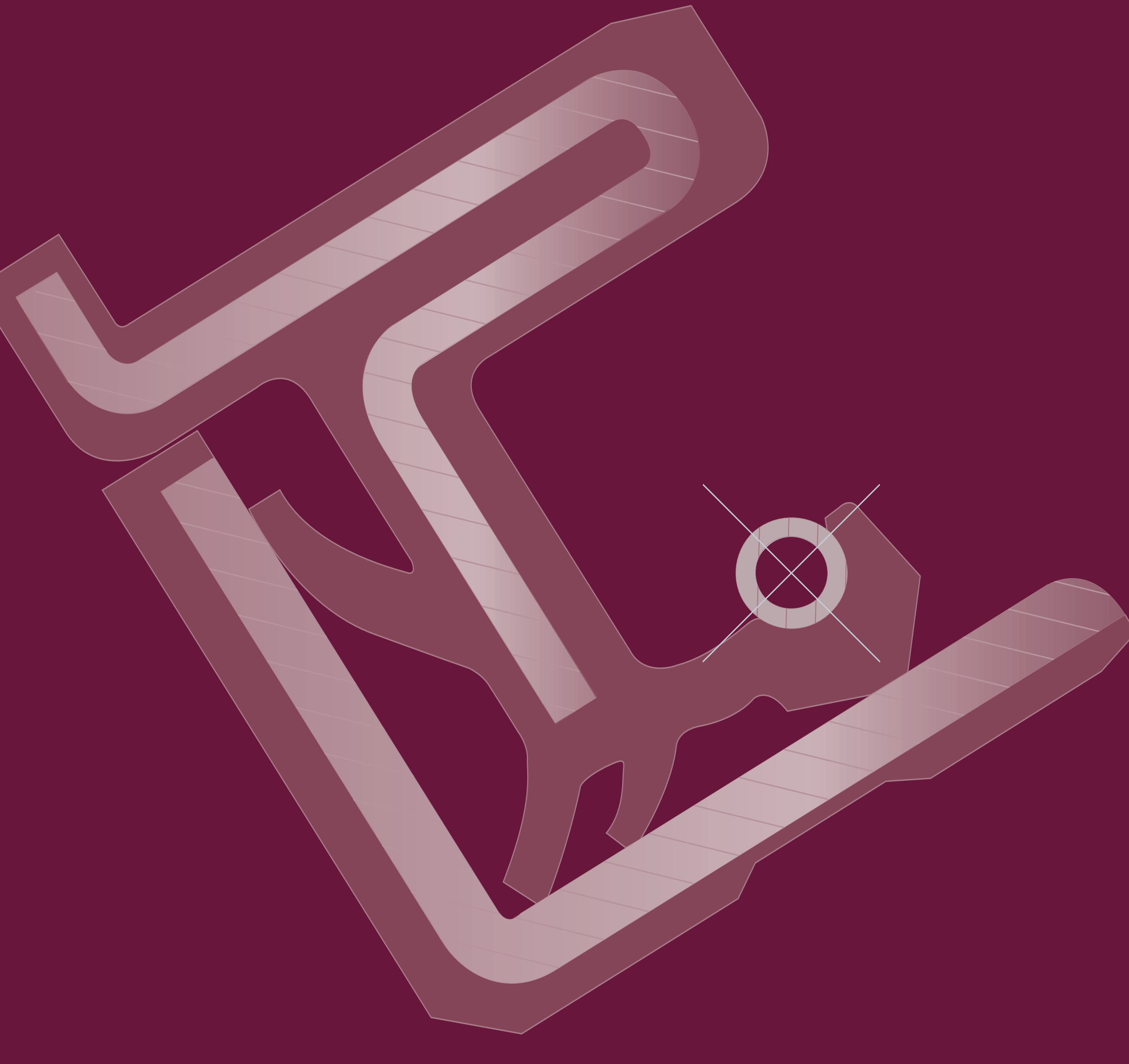


introduction

introduction

Using this Design Guide.

This design guide provides data on a widely used type of seal: the shaft seal. As you review the contents, please keep in mind that the many materials and designs featured in this guide are just part of the wide variety of sealing solutions offered by RL Hudson.

Though we have tooling for over 20,000 standard shaft seal sizes, we recommend that each seal be designed for its specific application. For example, a gearbox seal designed for low speed and low temperature use will not necessarily work in a gasoline engine application. To determine the best seal for your project, consider the following factors: the materials (both elastomeric and metallic) to be used for the seal; the type of seal application; the operating conditions; the seal dimensions; and the design of the shaft and housing bore. Follow the steps below to design your shaft seal.

DESIGNING THE BEST SHAFT SEAL FOR YOUR APPLICATION

1. Review the discussion of material properties beginning on ***page 19*** to determine which of these factors are most important to your project.
2. For chemical compatibility questions, see the guide starting on ***page 35***.
3. Review the material profiles beginning on ***page 41***.
4. For information on standard seal designs, see ***page 94***. Non-standard designs are discussed starting on ***page 99***.
5. For recommendations regarding shaft and bore finishes, see the discussion starting on ***page 116***.
6. Review the discussion of seal environment beginning on ***page 186***.

Before you begin, however, you may find it helpful to quickly review a few of the basics. "A Shaft Seal Primer" (starting on ***page 8***) gives a brief overview of sealing concepts. "Back to Basics" (beginning on ***page 13***) takes a look at why elastomeric materials make good seals.

"The many materials and designs featured in this guide are just part of the wide variety of sealing solutions offered by R.L. Hudson & Company."

introduction

A Shaft Seal Primer.

Also known as oil seals and radial lip seals, shaft seals are widely used in conjunction with rotating, reciprocating, and oscillating shafts to contain fluids and to exclude contaminants. In some applications, shaft seals are designed to contain pressure or to separate fluids. Shaft seals have several key strengths: They are economical, easy to install, and effective in a wide range of environments.

There are many factors to carefully consider as you select or design a shaft seal for a specific application. This *Shaft Seal Design & Materials Guide* provides detailed information on the many factors that influence the design of an effective shaft seal.

"The purpose of a seal is to block the clearance gap so that nothing passes through it."

PURPOSE OF ANY SEAL

Any mechanical assembly containing fluids must be designed so that these substances flow only where intended and do not leak out of the assembly. Seals are incorporated into mechanical designs to prevent such leakage at the points where different parts of an assembly meet. These meeting points are known as *mating surfaces*, and the space between them is called a *clearance gap*. The purpose of a seal is to block the clearance gap so that nothing passes through it.

EVOLUTION OF SHAFT SEALS

Consumer needs drive the development of most products, and shaft seals are no exception. Technological advances and the ever-more-demanding needs of end-users have spurred the development of increasingly sophisticated radial lip seals over the past century. In actuality, the first "shaft seals" (such as those found on the axles of low-speed frontier wagons) were nothing more than leather strips attempting (typically with very limited success) to contain the animal fat that served as lubrication.

As time wore on and industry revolutionized society, motorized vehicles replaced wagons, and leather strips were replaced by rope packings made of flax, cotton, and hemp (see ***Figure 1***). Though still relatively crude, such packings worked because lubricants tended to be very viscous (thick), operating speeds were still low, and temperatures never got high enough to degrade the lubricants or seal materials.

As the 1920s arrived, application speeds and temperatures further increased. Thinner, more environmentally unfriendly lubricants became common, and sealing them adequately became more difficult. Rope packings were superseded by *assembled leather seals* (see ***Figure 2***). A leather lip was chemically-treated to improve oil resistance, then clamped into a metallic case to facilitate installation and removal. The metal case allowed for a pressfit seal to prevent bore leakage, and the leather lip rode a region of the shaft that had been ground to a prescribed roughness.

Technical improvements to machinery, vehicles, and road surfaces caused shaft speeds and application temperatures to increase. New oils were developed to withstand these higher temperatures. Unfortunately, these higher temperatures and the new lubricants caused swelling and degradation of leather sealing lips. These difficulties were overcome in the 1940s with the development of oil-resistant polymers. *Assembled synthetic rubber seals* featuring lips made of nitrile (NBR) rather than leather became the norm (see ***Figure 3***).

By the 1950s, technology allowed for the chemical bonding of rubber to metals. This made possible a seal in which the rubber lip was chemically bonded to the case (rather than clamped in place). Seals of the 1960s began to feature lips made of materials other than nitrile. Silicone and polyacrylate materials were developed and used for *bonded seals* (see ***Figure 4***). Polytetrafluoroethylene (PTFE) has great chemical and temperature resistance in combination with good low frictional properties. As a result, PTFE was used to replace leather and NBR materials in assembled lip seals. Methods of bonding PTFE

EVOLUTION

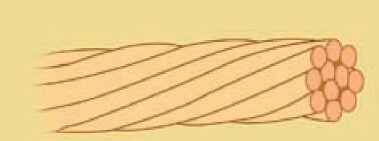


Figure 1: Rope Packing

Figure 2: Assembled Leather Seal

Figure 3: Assembled Rubber Seal

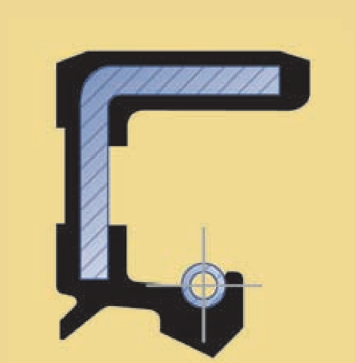


Figure 4: Bonded Seal

EVOLUTION

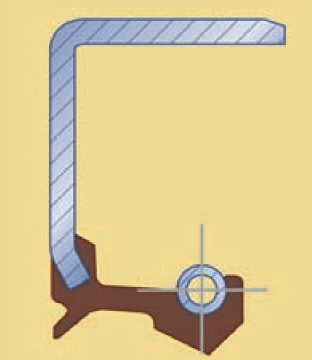


Figure 5: Reduced Bonding Area Seal

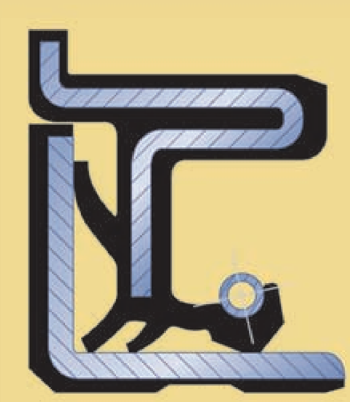


Figure 6: Unitized Seal

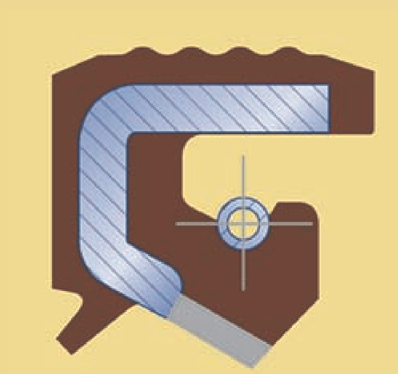


Figure 7: Composite Seal

to rubber or metal did not exist at this time. Fluoroelastomer also started being used in the 1970s. Though all of these "alternative" materials could be useful, they were also more expensive than nitrile, so seal designers sought ways to minimize material usage and reduce costs. This resulted in the production of seals with *reduced bonding areas.* An example of this is shown in ***Figure 5***.

Seal designers also began to look beyond the seal for ways to further improve performance and to extend reliability. They turned their attention to the sealing surface itself, and by the 1980s, seals that incorporated running surfaces into their designs became common. These *unitized* (or *cassette*) *seals* (see ***Figure 6*** for one example) took some of the worry away from the end user, who no longer had to be concerned about preparation of a proper running surface on the shaft. Should a shaft surface become damaged (for example, severely scratched) during service, replacement of a standard shaft seal with a unitized seal can often prevent (or postpone) the costly alternative of shutting down the application for either remachining or replacement of the shaft.

Thanks to the development of improved cements, *composite seals* featuring PTFE bonded to rubber also became possible. An example of a composite seal is shown in ***Figure 7***.

The most recent evolution of seal design has come about in the last decade. Seal designers are now combining the seal with other components from the sealing area (such as filters, reinforcing inserts, and excluders). The resulting *value-added seals* (such as the one shown in ***Figure 8***) make life easier for the user by reducing the number of components and thus simplifying both purchasing and assembly.

EVOLUTION

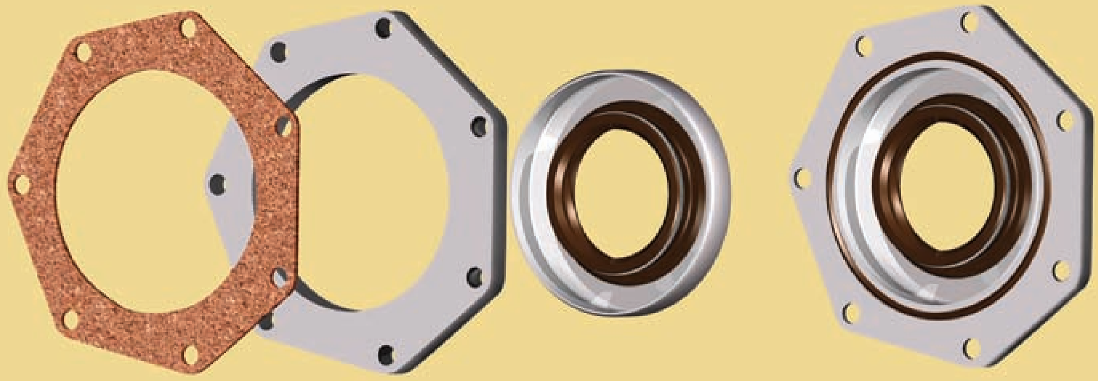


Figure 8: Value-Added Seal

USE OF A SHAFT SEAL

A shaft seal is but one part of a three-part system. Part two is the shaft itself, which is in motion. This motion may be rotary (round and round), reciprocating (in and out), or oscillating (rotating back and forth). Part three is the housing into which the seal is installed. ***Figure 9*** shows a shaft seal installed into a housing bore and onto a shaft.

SHAFT SEAL INSTALLED

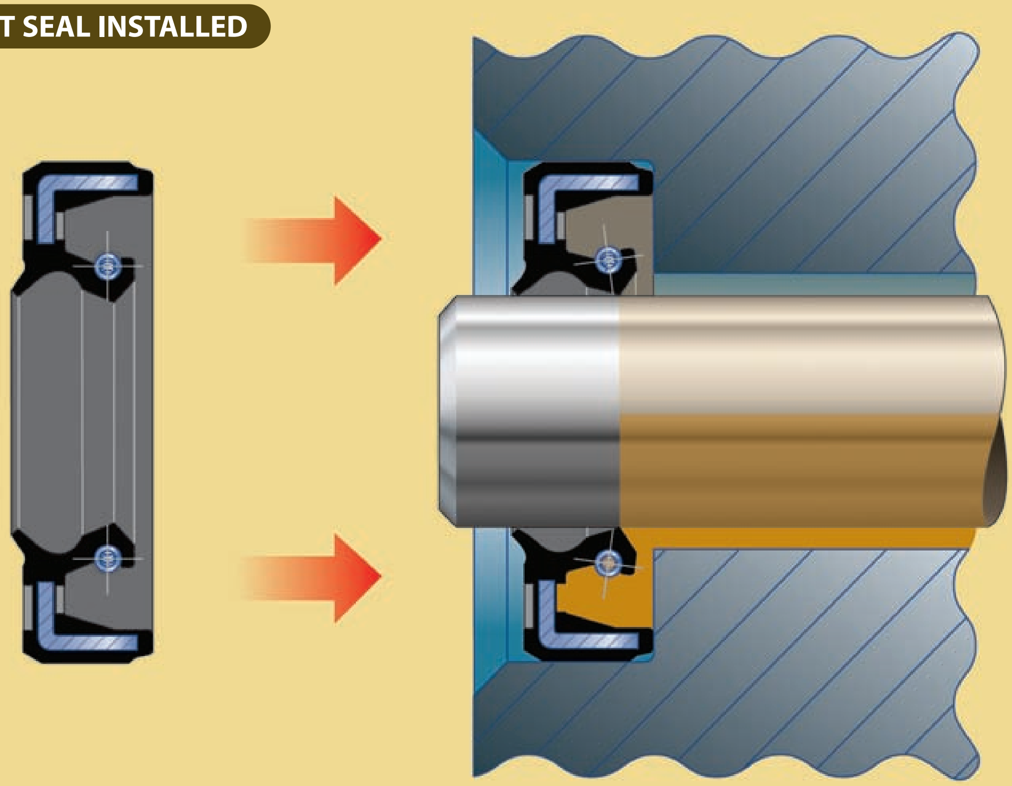
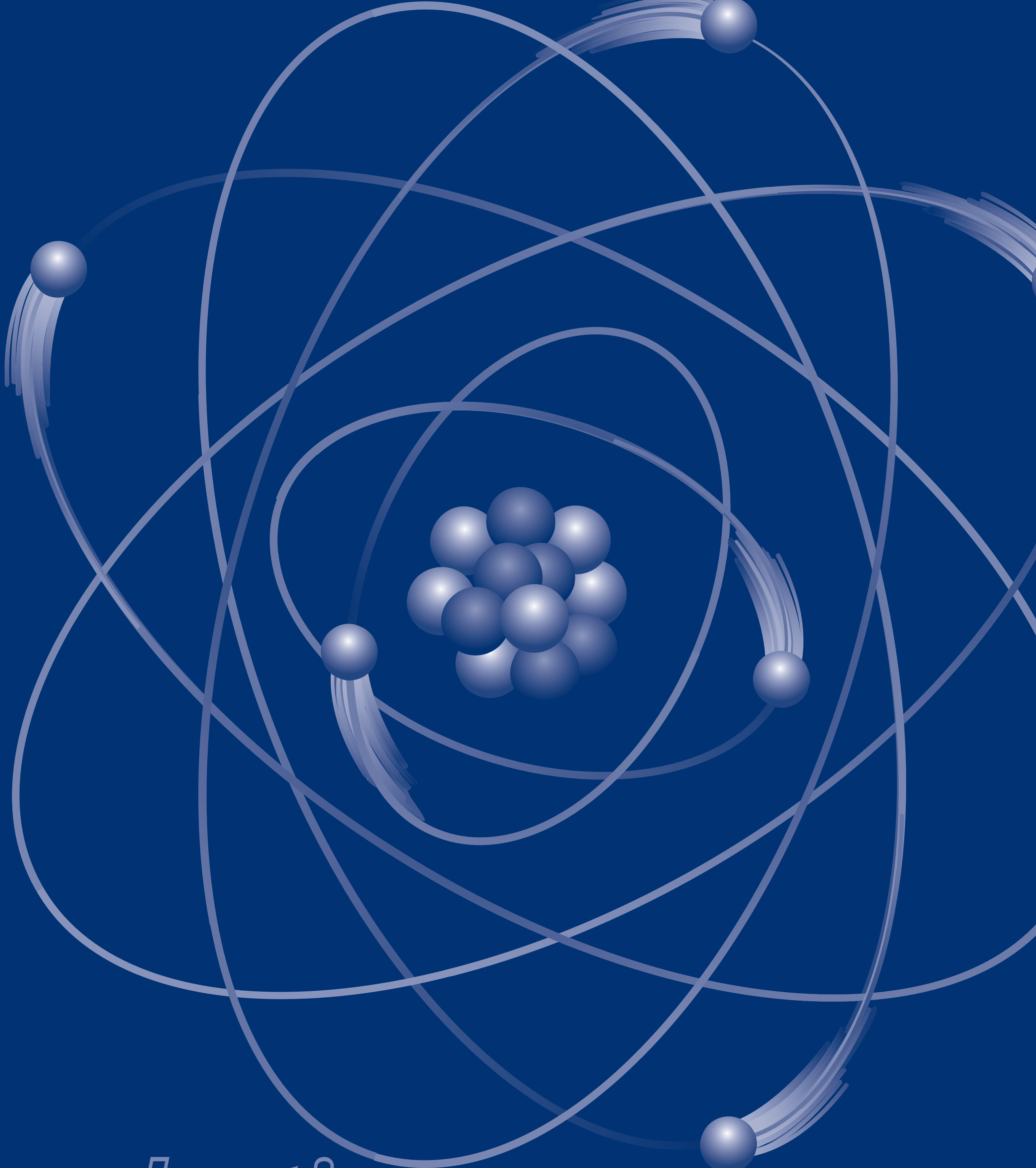


Figure 9: Rubber O.D. Shaft Seal In An Assembly



selecting the material

Back to Basics.

Anyone designing a shaft seal must answer a multitude of questions. In a sense, that's really what good design work is all about: determining which questions to ask and where to find the answers. Though the sealing industry offers its practitioners a multitude of exotic terms and explanations with which to grapple, a clear understanding of a few simple concepts will help you ask the most pertinent questions and find the most productive answers. Since the essence of a successful shaft seal is the lip material, here's a quick review, beginning with the basic building block of all materials, the atom.

"A clear understanding of a few basic concepts will help you ask the most pertinent questions and find the most productive answers."

ATOMS

Strictly defined, an atom is the smallest unit of an element that 1) retains all the element's distinctive properties and 2) can enter into a chemical reaction. In other words, anything less than an atom of carbon (C) is no longer carbon. We could split a carbon atom into its component parts (see ***Figure 10***), but the resulting subatomic particles (positively-charged *protons*, non-charged *neutrons*, and negatively-charged *electrons*) would not reflect the properties of carbon. Though their number and arrangement vary from element to element, subatomic particles alone tell you nothing about the atoms from which they came. A proton from a carbon atom is identical to an oxygen (O) proton.

Subatomic particles are important, however, in that they determine one of the defining characteristics of any given atom: its *atomic weight*, or the total mass of the protons and neutrons within its nucleus (orbiting electrons are of negligible weight and don't figure into this total). For example, the

CARBON ATOM

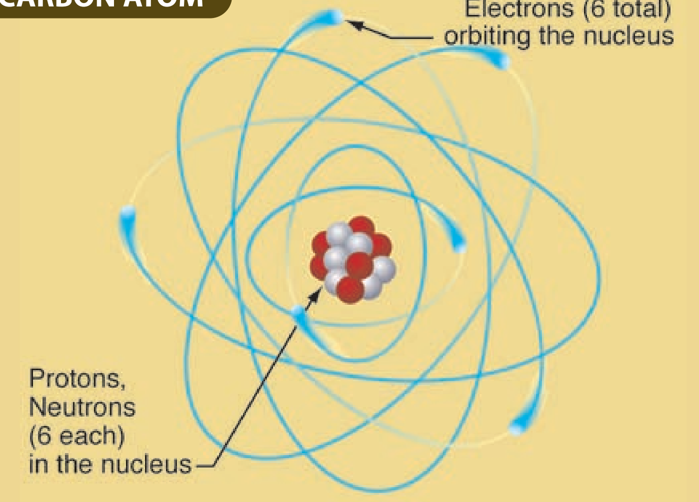


Figure 10: Subatomic Particles

nucleus of a hydrogen (H) atom contains one proton and no neutrons, so its atomic weight is 1. Carbon is composed of six protons and six neutrons, for an atomic weight of 12.

Each individual atom is also distinguishable by the one or more energy bonds it can form with neighboring atoms. This ability to combine is known as *valence*, and the amount of valence varies with each element. For example, an atom of hydrogen has a valence of 1, meaning it can form only one such energy bond. Oxygen has a valence of 2 and carbon has a valence of 4, meaning they can form two and four bonds, respectively. To be more precise, atoms need to form these energy bonds to be "satisfied" or "stable." The interaction of differing valences is what allows a group of atoms to join together into a *molecule*.

MOLECULES

The kind of molecule is determined by the exact type and number of atoms. For example, a water molecule is composed of three atoms: two of hydrogen and one of oxygen. The components of a water molecule are most simply expressed by the well-known chemical formula "H_20" or by the structural diagram: H-O-H (see ***Figure 11***). A water molecule is considered stable because the valences of each of its constituent atoms are satisfied: the two hydrogen atoms have formed one bond each, and the single oxygen atom has formed the two bonds it needs.

WATER MOLECULE

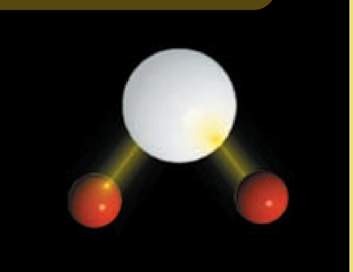


Figure 11: A Simple Inorganic Compound

When dissimilar atoms join together (as with water), the resulting molecule is called a *compound*. There are two major types of compounds: *organic* and *inorganic*. Generally speaking, organic compounds contain carbon and inorganic compounds do not, though a few carbon-containing compounds (such as metallic cyanides, carbon dioxide, carbides, and carbonates) are studied as part of inorganic chemistry.

The specific way in which a molecule is formed depends on which type of compound it is. Inorganic compounds are formed when an atom gives up, or transfers, one of its orbiting electrons to a nearby atom. Because of the rules of valence, this electron transfer can help both the donor and the recipient attain greater stability. Because of the compact structure of the carbon atom, it is much less inclined to give up an electron. It will, however, share an electron with a nearby atom (such as hydrogen) to attain a more stable compound.

As previously stated, each atom has its own atomic weight. When atoms unite to form a molecule, the sum of these atomic weights is then known as the *molecular weight*. For example, a methane molecule (CH_4) combines the atomic weight of one carbon atom (12) with the atomic weight of four hydrogen atoms (1 x 4), for a total molecular weight of 16. In addition to hydrogen, oxygen, and carbon, there are a handful of other atomic elements that form the basis for the majority of chemical raw materials used in the sealing industry. These include nitrogen (N), fluorine (F), silicon (Si), sulfur (S), and chlorine (Cl). The atomic weights and valences of each of these elements are listed in ***Table 1***.

ELEMENT	ATOMIC WT	VALENCE (ENERGY BONDS)
Hydrogen	1	1
Carbon	12	4
Nitrogen	14	3
Oxygen	16	2
Fluorine	19	1
Silicon	28	4
Sulfur	32	2
Chlorine	35	1

Table 1: Atomic Elements Most Commonly Used as Building Blocks in the Sealing Industry

POLYMERS

Small, individual molecules are known as mers, or *monomers* (literally, “single mers”). When conditions are right, these small molecules can chemically “link” together to form long, chainlike structures. The *macromolecules* (giant molecules) that result may incorporate thousands of the original monomers. These long chain macromolecules are therefore known as *polymers* (“many mers”). The linking process itself is called *polymerization*. An example of this process is shown in ***Figure 12***. Methane monomers can combine to form ethane, and eventually, polyethylene. Rubber and plastics are polymer-based materials.

MONOMER TO POLYMER

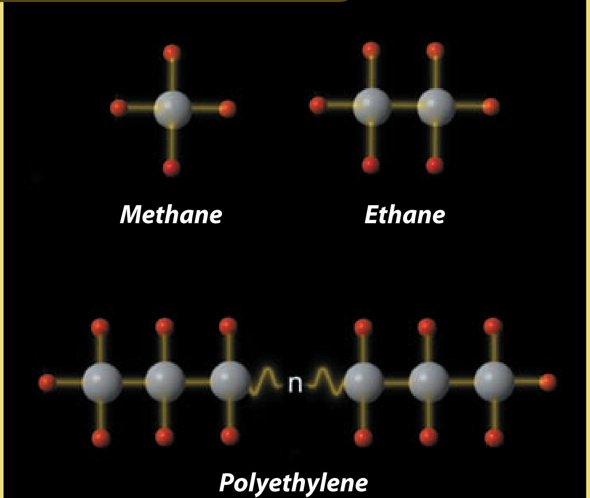


Figure 12: Effect of Polymerization

Changes in physical properties as a result of polymerization are largely a factor of molecular weight. When molecules (each with their own total weight) join together to form a polymer, the sum of the molecular weights has a huge impact on the polymer's physical properties. As a general rule, an increase in chain length (and thus molecular weight) also means an increase in strength and viscosity (resistance to flow).

Long polymeric chains are held in place by intermolecular forces (known as *van der Waals forces*) and by chain entanglement (as in a bowl of spaghetti). The intermolecular forces are heat-sensitive, so that as a polymer is heated, the molecular motion increases and the attractive forces between the molecules decrease. The polymer chains can then slide past one another.

Oil seal polymers are composed of branched, non-symmetrical molecules that cannot fit closely to one another. Because of this increased distance between the molecules, the van der Waals forces will be at their weakest, resulting in a random mass of twisted and entwined polymer chains. These polymers are said to be *amorphous* (see ***Figure 13***).

AMORPHOUS

Figure 13: Random, Non-Symmetrical Chains

Because their intermolecular forces are not very strong, amorphous polymers can be thought of as very viscous (thick) liquids that appear to be solids. All rubbers or elastomers are amorphous at room temperature.

ADDITIVES

Because many seals will face potentially detrimental service conditions (such as extreme cold or heat), a base polymer alone is seldom an effective seal material. Polymers are chosen for their heat and oil resistance, then other ingredients are added to make the material easier to process and to augment its physical and/or chemical traits. These other ingredients may include:

- **Fillers** (to reinforce or extend the material)
- **Plasticizers** (to aid flexibility and processibility)
- **Cure activators and accelerators** (to initiate and speed processing)

- **Inhibitors** (to ensure the reaction does not proceed too quickly)
- **Anti-Degradants** (to help resist environmental elements like oxygen or ozone)
- **Pigments** (to color the material)

The combination of a base polymer and additives is called an *elastomeric compound*.

VULCANIZATION

After a compound has been formulated, it must still be processed into a useful form (such as the lip of a shaft seal). Under normal conditions, an elastomer's amorphous chain segments are free to move relative to one another. This is not true only when the chains meet mechanical entanglement (as with the spaghetti effect), or when the separate chains are chemically connected. *Vulcanization* (also known as *cure*) is a heat-induced process whereby the long chains of the rubber polymers are permanently cross-linked to one another, thus forming three-dimensional elastic structures (see ***Figure 14***). Aided by curing agents in the original compound, vulcanization transforms soft, weak, non-cross-linked materials into strong elastic products. In addition to making the compound stronger, the vulcanization process is also generally the point at which the material is molded into a useful shape that it can retain thanks to its memory.

EFFECT OF VULCANIZATION

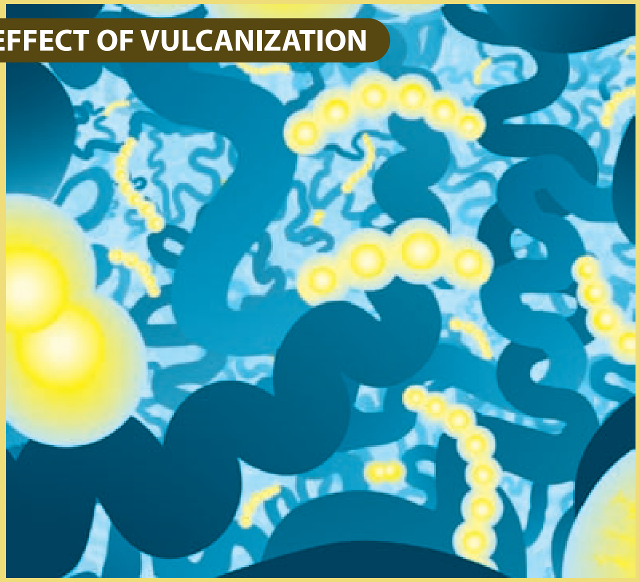


Figure 14: A 3-D Elastic Structure

Though every effort has been made to simplify the preceding discussion, it's important to realize that putting together an elastomeric compound can get quite complex. Decisions made in compounding will ultimately impact the processing and performance of any seals produced from the compound. Depending on the type and degree of additives in use, a single base polymer can generate hundreds of different compounds, each with unique characteristics.

VISCO-ELASTICITY

Though the term elastomer was initially used to denote a synthetic form of Natural Rubber, "elastomer" and "rubber" are now more or less synonymous. To be officially considered an elastomer by the American Society for Testing and Materials (ASTM), a polymer must not break when stretched 100%, and it must return to within 10% of its original length within five minutes after being held for five minutes at 100% stretch.

An elastomer is perhaps best described as a *visco-elastic* material, in that it goes through both a viscous phase and an elastic phase. The visco-elastic behavior of elastomers can be simulated using a spring coupled with a dashpot (damper). The spring illustrates the elastic phase; the dashpot exemplifies the viscous phase (see ***Figure 15***).

But why is an elastomer elastic and resilient, able to undergo high strain and yet recover its original shape? Put simply, it's the tangled nature of its long molecular chains. When pressure (in the form of either a compressive load or a stretching force) is applied to the elastomer, the chains rotate around their chemical bonds. This rotation tends to uncoil the entangled mass and straighten the chains. When the pressure is removed, the chains coil up again, reverting to their normal state of entanglement. This tendency to return to its original configuration helps explain an elastomer's rubbery, resilient nature.

VISCO-ELASTICITY

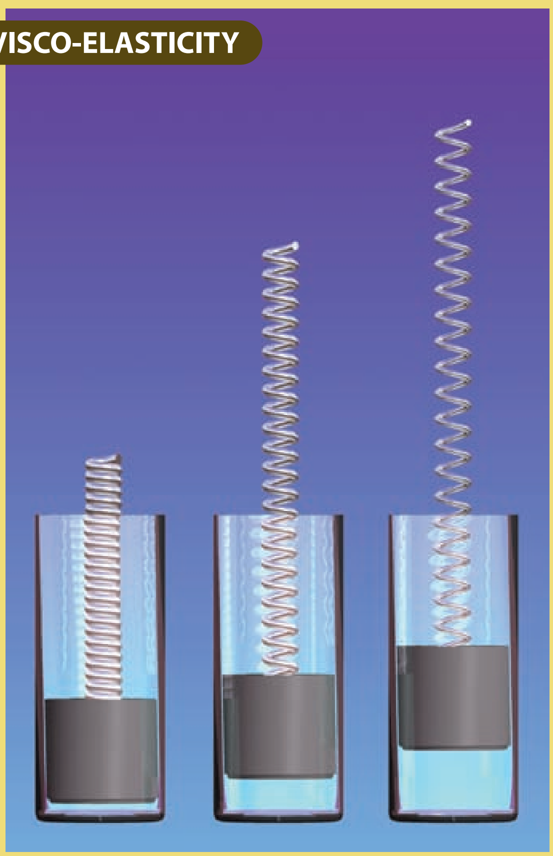


Figure 15: A spring-dashpot combination can illustrate the visco-elastic nature of elastomers.

Under certain conditions, a few elastomers will have their molecules align and form crystalline regions. An elastomer that crystallizes due to cold temperatures becomes harder and less able to stretch. This can be detrimental to shaft seal performance.

Since choices made during compounding directly determine the properties of an elastomeric seal, let's look at these physical, thermal, and chemical properties next.

Physical Properties.

here are a number of physical properties you should consider when choosing an elastomeric compound for your shaft seal application. These include hardness, tensile strength, modulus, elongation, tear resistance, abrasion resistance, compression set resistance, and resilience. The extent to which these properties are present in a material has a huge impact on the material's ability to function effectively as part of a shaft seal.

HARDNESS

Typically defined as resistance to indentation under specific conditions, the hardness of an elastomer is more accurately thought of as two related properties: inherent hardness and processed hardness. As a result of chemical structure, each elastomer has its own inherent hardness. This inherent hardness can be modified (and is typically supplemented) via compounding and vulcanization. Hardness in molded rubber articles (processed hardness) is a factor of cross-link density (and the amount of fillers). The more cross-linking a given material undergoes during vulcanization, the harder the final molded part will be. When judging the potential effectiveness of a molded seal, (processed) hardness is one of the most common criteria in the rubber industry.

Unfortunately, hardness is also one of the least consistent concepts in that the most-used measurement scales have only limited comparability. There is no single "universal hardness" unit, so it is often impossible to draw a clear and easy correlation between readings on two different scales, even when the samples being measured are absolutely identical. There are currently two hardness tests that predominate in the rubber industry: Shore durometer and International Rubber Hardness Degrees (IRHD) (see ***Figure 16***).

"The extent to which these properties are present in a material has a huge impact on the material's ability to function effectively as part of a shaft seal."

HARDNESS TESTERS

Figure 16: Some instruments generate both durometer and IRHD readings.

Because Shore Instruments led the way in the marketing of durometer gauges, the words "Shore" and "durometer" have become virtually synonymous within the rubber industry. Now a division of Instron Corporation, Shore Instruments offers a wide range of durometer scales conforming to the ASTM D 2240 standard.

The Shore A durometer is a portable and adaptable device that uses a frustum (truncated) cone indentor point and a calibrated steel spring to gauge the resistance of rubber to indentation. When the durometer is pressed against a flat rubber sample, the indentor point is forced back toward the durometer body. This force is resisted by the spring. Once firm contact between the durometer and the sample has been made, a reading is taken within one second unless a longer time interval is desired. Five readings are typically taken, then an average value calculated. The amount of force the rubber exerts on the indentor point is reflected on a gauge with an arbitrary scale of 0 to 100. Harder substances generate higher durometer numbers. A reading of 0 would be indicative of a liquid, whereas 100 would indicate a hard plane surface (e.g. steel or glass).

DUROMETER USE

Figure 17: An average of several readings is preferable.

Though the elastomeric lips of most standard shaft seals fall in the 70 to 90 Shore A range, the application in question will always govern the necessary hardness. Softer compounds offering less resistance may be perfectly fine for low-pressure seals, but high-pressure seals will likely require a harder, tougher lip material. Making decisions about a property such as hardness often demands compromise, however, in order to ensure the long-term usefulness of the seal. For example, a relatively hard compound may resist high pressure, but its use can also lead to increased frictional buildup. Increased friction leads to increased heat, which can, in turn, degrade the sealing lip and decrease the seal's life span.

It is also important to realize that measuring the hardness of a rubber sample is an imprecise art (see ***Figure 17***). Depending on both the specific gauge in use and the expertise of its operator, it is possible (even probable) that the same sample will yield two or more different readings. The rate at which the durometer is applied to the sample, the force used, the amount of time that elapses before taking the reading, and the temperature of the specimen at the time of testing can all impact a test result. For this reason, all durometer readings normally include a tolerance of ± 5 points, but sometimes even this may not be enough to fully anticipate all of the variances to be seen in testing. Technological advances have reduced many of the discrepancies, but sometimes at the expense of the simplicity and portability that initially made durometers popular. It is generally a good idea to test a given specimen several times and average the results to ensure accuracy.

Despite the long-standing close association between "Shore" and "durometer," there are a number of other companies that manufacture high-quality durometers, including PTC Instruments and Rex Gauge Company. And while durometers are fine for measuring the hardness of material samples, measuring the hardness of actual shaft seal lips is a different matter. The cross-sections of sealing lips tend to be too small, and the lips themselves too thin, for hardness to be accurately measured using a Shore device. Gauging the rubber hardness of a sample taken from an actual sealing lip generally requires use of a microhardness tester such as those manufactured by Wallace (see ***Figure 18***).

The Wallace tester is similar in design and function to a Shore durometer, with the main difference being that the indentor point on the microhardness tester is smaller. This smaller point can more accurately reach into small places (such as the spring groove of the lip) to take measurements.

Microhardness readings are generally expressed in terms of International Rubber Hardness Degrees (IRHD) and come with the same tolerance (± 5 points) as Shore readings. Though not as common in the United States as abroad, all IRHD testers are designed to conform to the ASTM D 1415 standard.

MICROHARDNESS TESTER

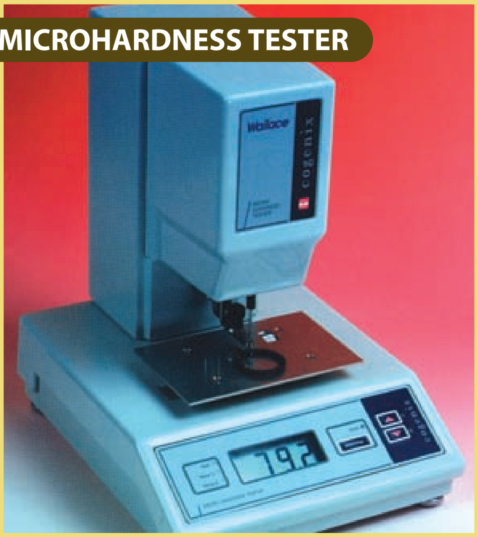


Figure 18: Can be used with samples from actual sealing lips.

TENSILE STRENGTH

Typically noted in either pounds per square inch (psi) or megapascals (MPa), tensile strength is the amount of force required to break a rubber specimen. To convert from MPa to psi, simply multiply the MPa figure by 145. For example, 14 MPa converts to 2,030 psi. Converting from psi to MPa is just a matter of dividing the psi number by 145.

TENSILE SPECIMEN

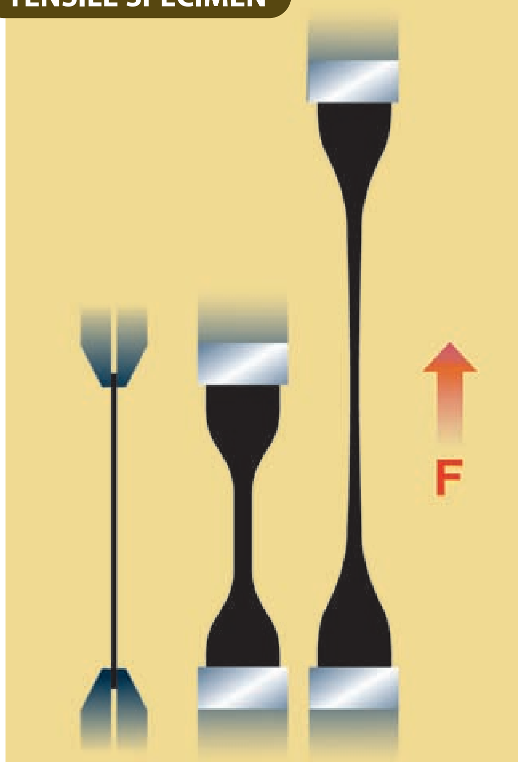


Figure 19: Molded Dumbbell

Per ASTM D 412, a compound's tensile strength is generally tested using a molded dumbbell (see ***Figure 19***). The dumbbell is placed in the grips (jaws) of a tensile tester (see ***Figure 20***). The best-known tester is Instron, but tensiTECH and others are also used. When the tester is activated, the dumbbell is pulled steadily at a rate of 20 inches per minute until it breaks. The force being exerted on the sample at the time of rupture is said to be the sample's tensile strength.

Minimum tensile strength is typically used as both a qualification criterion when specifying a new material and as a control criterion (with a ± 15% production tolerance) when testing batches of mixed material. A shaft seal's elastomeric lip will undergo a small amount of stretch as the seal is placed on the shaft, but a shaft seal will not be exposed to high degrees of stretch in normal operation. Unfortunately, people often specify tensile strength that is larger than required for a given application.

MODULUS

Perhaps the best single gauge of a compound's overall toughness and extrusion resistance, modulus is the force (stress) in pounds per square inch (psi) required to produce a certain elongation (strain). This elongation might be 50%, 100%, or even 300%, though 100% is the most widely used figure for testing and comparison purposes. Industry literature typically refers to 100% elongation as "M100" (or modulus 100). Compounds with a higher modulus are more resilient and more resistant to extrusion. Generally speaking, the harder a compound, the higher its modulus. Because it is basically a measure of tensile strength at a particular elongation (rather than at rupture), modulus is also known as tensile modulus or tensile stress.

As described in ASTM D 412, modulus is typically gauged simultaneously with tensile strength on the same dumbbell specimen shown in ***Figure 19***. As the specimen is being stretched, the tester records the psi (for example, 836.7) needed to achieve a given elongation (for example, 100%). This figure in psi is considered to be the sample's modulus at that elongation. Minimum modulus is typically used as both a qualification criterion when specifying a new material. Keep in mind that the elastomeric lip of a shaft seal will never undergo 100%, or even 50%, elongation. Rather, elongation (actually, expansion) of the sealing lip as it is placed onto the shaft is typically no more than 5%. For this reason, it's not generally possible to make a direct correlation between modulus data for a given material and actual performance of a sealing lip molded from that material.

TENSILE TESTER

Figure 20: Stretching a Sample

ELONGATION

Elongation is the percentage increase in original length (strain) of a rubber specimen as a result of tensile force (stress) being applied to the specimen. Elongation is inversely proportional to hardness, tensile strength, and modulus. That is, the greater a material's hardness, tensile strength, and modulus, the less it will elongate under stress. It takes more force to stretch a hard material with high tensile strength and high modulus than to stretch a soft material with low tensile strength and low modulus.

Ultimate elongation is the elongation at the moment the specimen breaks. Per ASTM D 412, ultimate elongation is generally noted along with tensile strength and modulus during tensile testing. Some elastomeric materials are much more forgiving in this area than others. Natural rubber can often stretch up to 700% before breaking. Fluorocarbons typically rupture at about 300%. Keep in mind that these figures highlight relative failure modes only and do not reflect shaft seal installation values. As noted in the discussion of modulus, the elastomeric lip of a shaft seal is typically stretched no more than 5 % during installation. Greater stretch can be problematic, not because it pushes the bounds of the material's properties, but because greater lip stretch upon installation means the lip will exert increased radial force on the shaft. This increased force will generate greater friction and heat, both of which can decrease the useful life span of the seal.

TEAR RESISTANCE

Noted in kilonewtons per meter (kN/m) or pound force per inch (lbf/in.), tear resistance (or tear strength) is resistance to the growth of a cut or nick in a vulcanized (cured) rubber specimen when tension is applied. Tear resistance is an important consideration, both as the finished article is being removed from the mold *and* as it performs in actual service.

Tear resistance can be gauged via the same ASTM D 412 apparatus used in the testing of tensile strength, modulus, and elongation. As described in ASTM D 624, different specimen types can be used to measure both *tear initiation* (resistance to the start of a tear, see ***Figure 21***) and *tear propagation* (resistance to the spread of a tear, see ***Figure 22***). Either way, the sample is placed in the tester's grips, which then exert a uniform pulling force until the point of rupture. This force may then be divided by the specimen's thickness to arrive at the tear resistance for that particular sample. Three separate samples are typically tested and an average calculated.

TEAR INITIATION

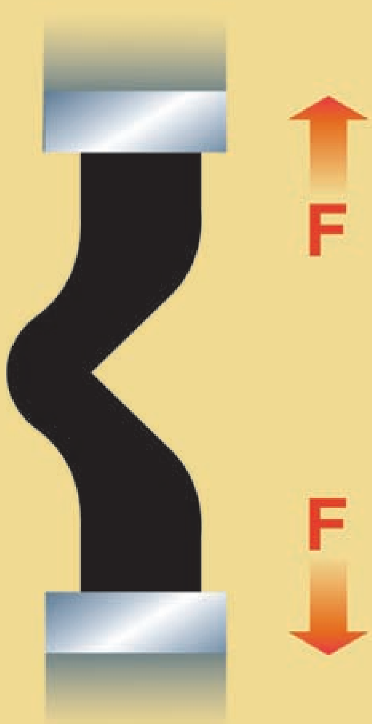


Figure 21: Unnicked 90° Angle

Though some materials have excellent tear resistance, many materials are not very strong in this area. Silicone has notably poor tear resistance. Though it might seem logical, it is in fact a misconception that hardness automatically equals good tear resistance. Compounds whose tear resistance is less than 100 lbf/in. are most at risk for installation damage, especially in designs featuring non-smooth areas (as with burrs, slots, threads, etc.) and/or sharp, non-radiused (non-rounded) corners. Once damaged, materials with poor tear resistance will quickly fail in service. This is especially true for dynamic seals. Poor tear resistance is linked to poor abrasion resistance.

TEAR PROPAGATION

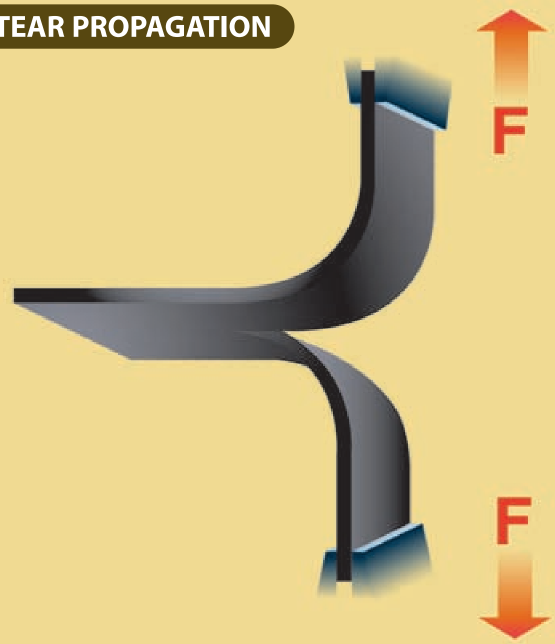


Figure 22: Slit ("Trouser") Specimen

ABRASION RESISTANCE

Measured as a loss percentage based on original weight, abrasion resistance is the resistance of a rubber compound to wearing away by contact with a moving abrasive surface. Whereas the cutting or nicking of a sealing lip is an instantaneous event, abrasive rubbing or scraping is much more of a progressive phenomenon that develops over time. Seals in motion are most susceptible to abrasion. Hard compounds tend to exhibit less abrasive wear than soft compounds, but use of a harder compound can also increase friction in dynamic seals, and increased friction generates seal-degrading heat.

Because of the many potential variables (including heat fluctuation and surface contamination), abrasion resistance is hard to accurately measure. Testing typically involves the uniform application of an abrasive material (such as sandpaper) to the surface of a sample. ASTM standards describe three different abraders: D 1630 relies on a National Bureau of Standards (NBS) abrader (see ***Figure 23***); D 2228 uses a Pico abrader (see ***Figure 24***, next page); and D 3389 (also known as Taber Abrasion) employs a double-head abrader and a rotary platform (see ***Figure 25***, next page). Regardless of the specific test method, the relative amount of sample material that is lost due to abrasion is a good indication of abrasion resistance.

NBS ABRADER

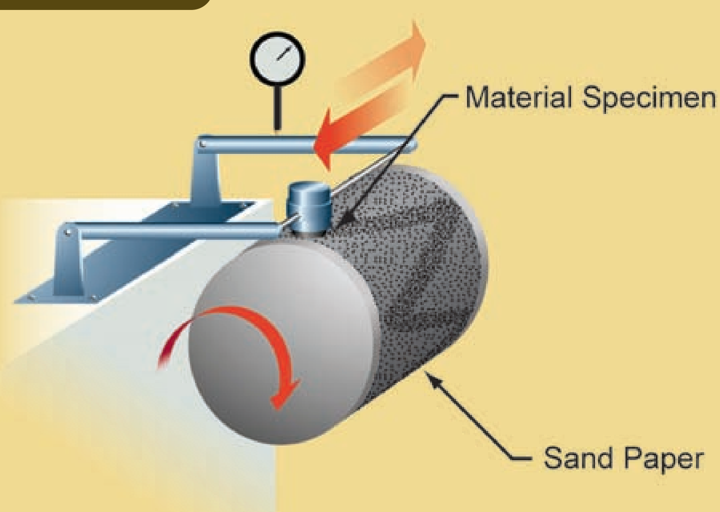


Figure 23: Rotating Drum

PICO ABRADER

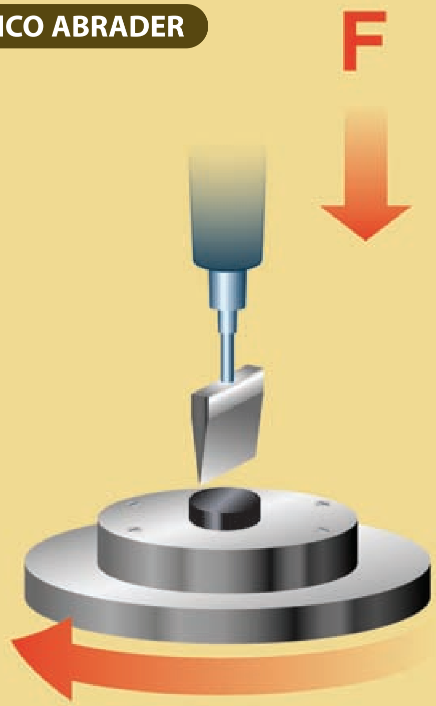


Figure 24: Rotating Platform

Generally speaking, hydrocarbon-based elastomers tend to offer better abrasion resistance than fluoroelastomers. Carboxylated nitrile and hydrogenated nitrile offer abrasion resistance that is superior to other hydrocarbon-based elastomers. Keep in mind that the tests described here are conducted under dry conditions; material performance will vary in lubricated applications.

COMPRESSION SET

Compression set is the end result of progressive stress relaxation, which is the steady decline in sealing force that results when an elastomer is compressed over a period of time. In terms of the life of a seal, stress relaxation is like dying, whereas compression set is like death.

Though it is very difficult to accurately quantify stress relaxation, compression set is easy to measure. ASTM D 395 details compression set testing for rubber that will be compressed in air or liquid media. Two methods are described ("A" for constant force; "B" for constant deflection), but the basic methodology is substantially the same. Testing generally involves use of cylindrical disk compression set test buttons (0.49" thick by 1.14" diameter) taken from molded slabs. In lieu of buttons, die-cut plied (stacked) samples (0.070" thick by 1.14" diameter) may be substituted. The buttons or plied samples are placed between steel plates. In method A (see ***Figure 26***), the plates are then forced together using either a calibrated spring or a pre-defined external force. In method B (see ***Figure 27***), a bolt-tightened device and steel spacers are used. Either way, compression (normally 25% of original thickness) is held for a given time (e.g. 22 hours) at a specific temperature (e.g. 100° C), these last two variables based on anticipated service conditions.

TABER ABRADER

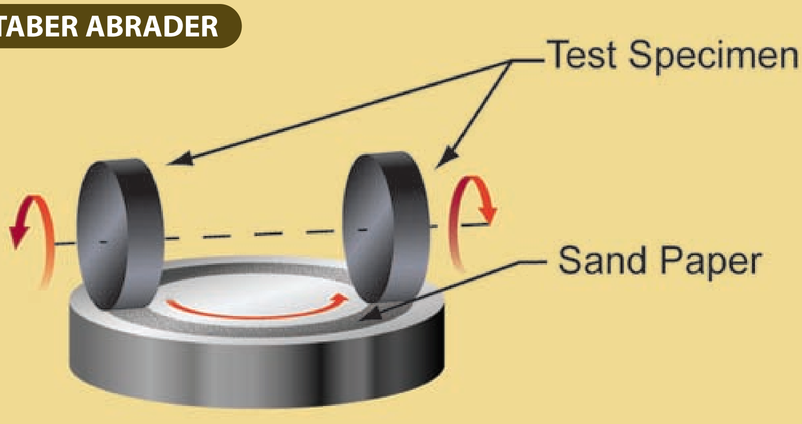


Figure 25: Rotating Samples

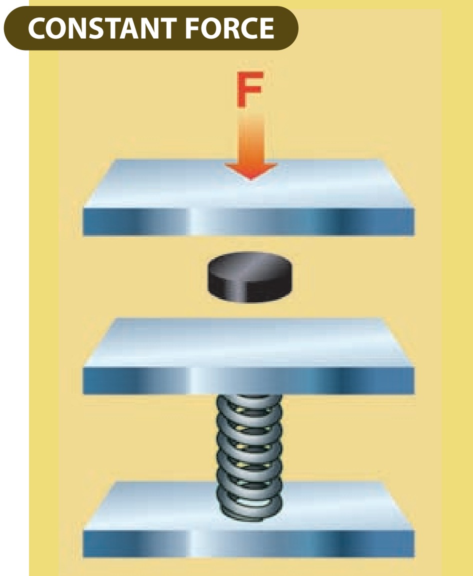


Figure 26: Compression Set Test Method A

Figure 27: Compression Set Test Method B

After removal from the compression device and a 30-minute cooling period, the specimens are measured using a dial micrometer. Compression set can then be calculated as either a percentage of original specimen thickness or as a percentage of original deflection. Because ASTM D 395 primarily describes the testing of materials to be used in high temperatures, a similar test procedure for materials to be used in low temperatures is outlined in ASTM D 1229.

Though a high degree of compression set is to be avoided, other service variables (such as inadvertent fluid swell or the intentional application of greater squeeze) may compensate. Seals are most likely to fail when there is both high compression set and shrinkage. ***Table 2*** (next page) shows how several of the most commonly used sealing materials respond to increasing temperatures.

Because the elastomeric lip of a shaft seal is not normally compressed, compression set tests have limited applicability when it comes to choosing a lip material. In relation to shaft seals, compression set tests are most helpful when a seal design calls for a rubber-covered outside diameter. This O.D. will need to maintain tight contact with the housing bore after installation, so being able to anticipate the degree to which the rubber may set is important. Of course, compression set tests find their widest use in selecting materials for use as O-rings and other static seals that rely on an optimal degree of compression.

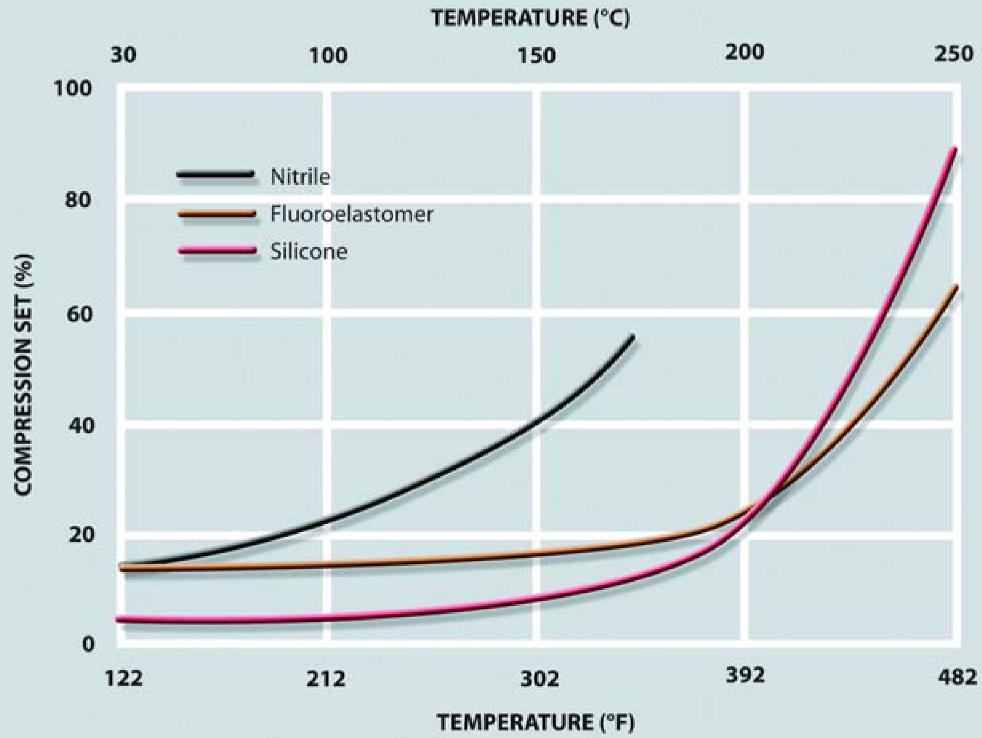


Table 2: Compression Set in Increasing Temperatures

RESILIENCE

As detailed in ASTM D 2632, resilience (also known as rebound) refers to a compound's ability to regain its original size and shape following temporary deformation. Resilience testing typically involves the dropping of a small weight onto a test specimen (such as a compression set button, see ***Figure 28***). The extent to which the weight bounces back is then noted as a percentage of the initial drop height. A highly resilient material (one that can rapidly regain its dimensions) might engender a 70% rebound value, but values in the range of 40 to 50% are more typical for the majority of elastomers tested. Though compounding may improve an elastomer in this area, it can also detract from good resilience, which is largely an inherent property. As a general rule, resilience is most critical in dynamic seals (such as shaft seals) because it allows them to regain their original shape following deformation. This

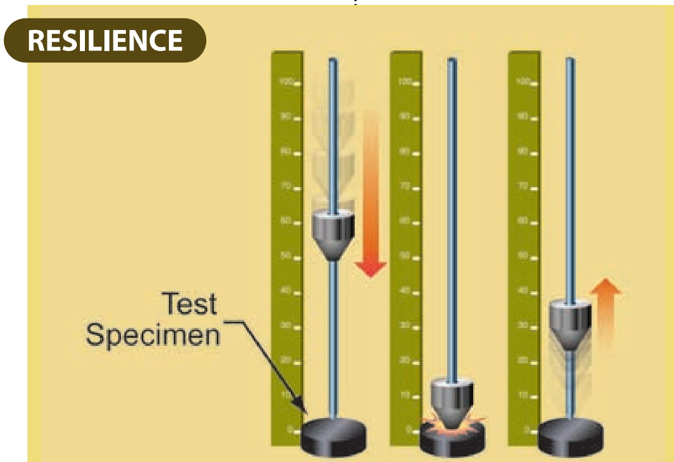


Figure 28: Bashore Resilience Testing

can be seen, for example, when a shaft seal's elastomeric lip flexes (is distorted) to follow a shaft imperfection, then returns to its original ("resting") position. Resilience in a shaft seal lip is closely linked to flexibility.

FLEXIBILITY

Defined as the ability to flex or bend when necessary without being damaged, the flexibility of a given rubber material is particularly important in dynamic seals. In combination with resilience, flexibility is what allows a shaft seal's elastomeric lip to adjust for shaft imperfections. Flexibility determines *followability*, or the ability of the sealing lip to maintain contact with the shaft despite vibrations or dynamic runout. In other words, the more flexible the material, the better the lip can adjust to conditions without losing the ability to seal.

Testing a material's ability to maintain flexibility over time is known as flex fatigue testing, and there are two main tests related to flex fatigue. As described in ASTM D 430, material samples can be subjected to repeated motion (such as bending, see ***Figure 29***) to see how long it takes for surface cracks to appear. Because such cracks would eventually lead to lip rupture in a shaft seal, the length of time it takes for them to appear can be a good indicator of the material's overall suitability for lip usage. In some instances, it may also help to know how resistant a material is to the growth of a pre-existing crack. ASTM D 813 describes flex fatigue testing for a material sample with a crack already in it.

FLEXIBILITY

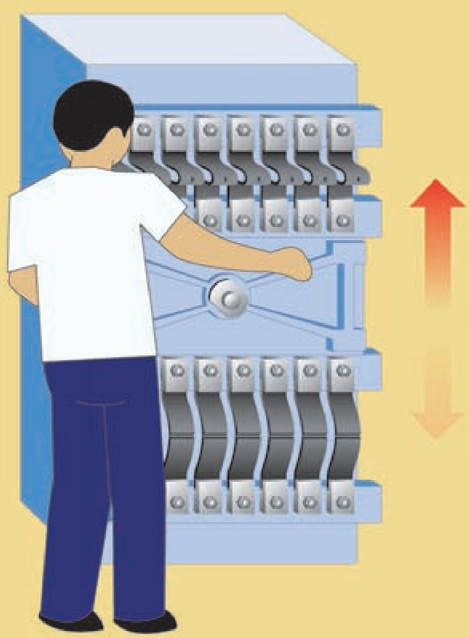


Figure 29: Bending Tests

ADHESION

Because most shaft seals feature an elastomeric lip bonded (during molding) to a metallic (or, in some cases, plastic) case, the ability of a rubber material to bond fully (and permanently) is very important to the life span of the seal. As described in ASTM D 429, adhesion tests for rubber bonded to a rigid substrate (such as metal) are typically performed using the same type of tester used to stretch dumbbells and gauge tensile strength (see ***Figure 20***). In an adhesion test, the rubber portion of a rubber-bonded-to-metal sample is pulled at a constant rate until either the rubber peels away

ADHESION

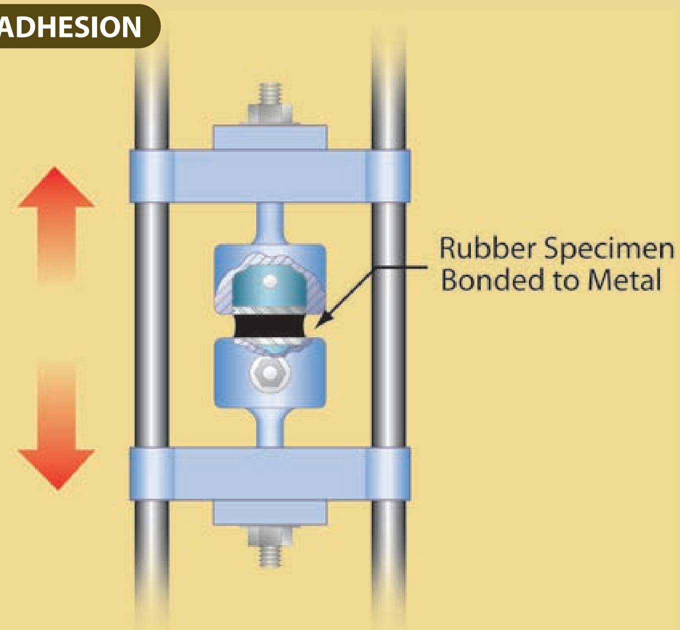


Figure 30: Testing of a Rubber-To-Metal Bond

from the metal (the bond fails) or the rubber itself ruptures (though the bond stays intact). An example of adhesion testing is shown in ***Figure 30***.

Because adhesion testing cannot be performed on completed seals, some manufacturers have devised ad-hoc methods for testing the adhesion properties of finished parts. One such method uses pliers to grip the sealing lip and pull it away from the case. If the lip separates cleanly, the bond is not strong enough. If rubber residue is left behind on the case, the bond was satisfactory. Under normal circumstances, a sealing lip in service will never be subjected to stresses so great, but adhesion tests can be good indicators of bond strength.

As important as they are, the physical properties of a given material are not the end of the story. Chemical properties are also critical, so let's take a closer look at them next.

Chemical Properties.

In addition to the physical properties discussed in the preceding section, there are also a couple of very important chemical properties you should consider when choosing an elastomeric compound. Primarily, the compound must be chemically compatible with the substance(s) to be sealed. You must also anticipate any volume changes that the compound may undergo as a result of contact with system fluids.

COMPATIBILITY

As used by the sealing industry, the term *compatibility* refers to a seal material's resistance to having its chemical (and, by extension, its physical) properties degraded (either temporarily or permanently) as a result of contact with a liquid or gas. Because "likes dissolve likes," the true key to compatibility between the seal and the fluid(s) being sealed is dissimilar chemical structure. For example, a shaft seal lip made from an all-hydrocarbon rubber (such as natural rubber) will be severely compromised when put in contact with petroleum-based oils or fuels.

In addition to being resistant to the primary system fluid, the seal must also be resistant to any and all additives that may be encountered during the course of operation. For example, oil-field applications often utilize film-forming amine inhibitors to coat tubular goods and prevent metal corrosion. Unfortunately, amine inhibitors act as curing agents for some fluoroelastomers, causing seal hardening and failure. In such an application, a shaft seal would need to be resistant to the fluid(s) being sealed and to the added amine inhibitors in order to provide an effective and long-lasting seal.

Even if they do not degrade the elastomeric compound directly, some fluids degrade surfaces adjacent to the seal (as with metal corrosion), thus reducing the effectiveness of the seal itself. You should also keep in mind that while some compounds formulated from a particular polymer may be okay for use in a given fluid, not *all* compounds of that polymer will be appropriate for use in that fluid. Since a compound's properties are a direct result of its interactive

"Because 'likes dissolve likes,' the true key to compatibility between the seal and the fluid(s) being sealed is dissimilar chemical structure."

constituents (e.g. reinforcing agents, plasticizers, etc.), each unique formulation should be tested in service conditions to determine its appropriateness for an application.

There is no single ASTM test method for "chemical compatibility." Rather, compatibility is understood to be a wider concept incorporating changes (or the lack thereof) in a number of material properties, each of which have its own test method. Hardness, tensile strength, modulus, and elongation can all be compromised if a compound is not compatible with (resistant to) a given fluid. Perhaps the most visible evidence of chemical incompatibility is a change in the material's volume.

VOLUME CHANGE

Volume change is the increase (swell) or decrease (shrinkage) in the volume of a specimen that has been in contact with a fluid. This contact may range from occasional "splashing" to constant immersion. Any resulting volume change can range from minor (indicating there is a relative compatibility between the fluid and the specimen) to major (indicative of incompatibility). Volume change is typically noted as a percentage of the original volume. For example, a specimen that swells to twice its original volume is said to have undergone a 100% increase. ***Figure 31*** shows the swelling of a shaft seal's elastomeric lip.

An elastomeric material typically becomes softer as a result of swell, whereas shrinkage generally hardens the material. A slightly swollen seal is, in most cases, still functional. A limited amount of swell may compensate for other variables, such as compression set. Shrinkage, on the other hand, can

LIP SWELL

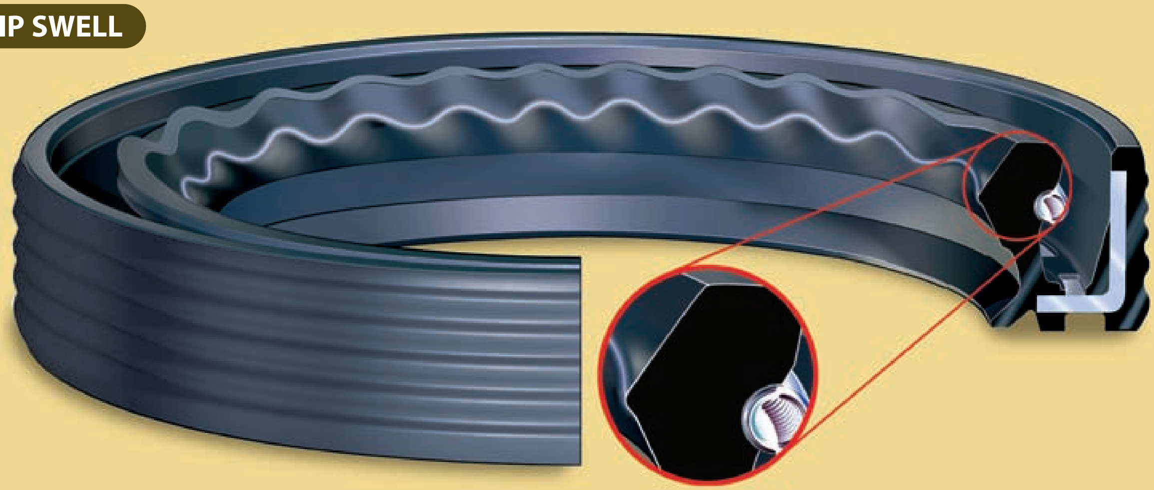


Figure 31: Impaired Sealing Due To Swell

exacerbate an already-existing compression set problem. With some of its soluble components (such as plasticizer) having been extracted by system fluid, a rubber O.D. seal that has undergone shrinkage is more prone to leaks between the seal O.D. and the housing I.D.

As described in ASTM test method D 471, volume change testing typically employs ASTM and Industry Reference Material (IRM) oils, as well as ASTM Reference Fuels, Service Liquids, and Type IV Reagent Water. Regardless of the liquid in use, testing involves immersing a material sample (of known properties) in the liquid for a specific period of time (e.g. 70 hours) at a specific temperature (e.g. 100° C ± 2°), both variables based on the conditions expected in service. Material deterioration (if any) is then determined based on changes in physical properties, including volume.

ENVIRONMENTAL FACTORS

Seals with an elastomeric element may also be susceptible to degradation by environmental factors such as ozone and oxygen. The combination of material stress and ambient ozone can cause cracks to develop in some materials. ***Figure 32*** shows cracking of a shaft seal's elastomeric lip.

Nitriles, for example, are at high risk for this type of degradation. This is because the chemical backbone of nitrile contains a double bond, and double bonds are the primary attack sites for ozone. In actuality, environmental cracking is most often seen in nitrile seals prior to installation. To help avoid this, seals should be stored away from sunlight, UV light, and ozone-generating devices such as welders and electric motors. In addition to compromising a sealing lip's

LIP CRACKING

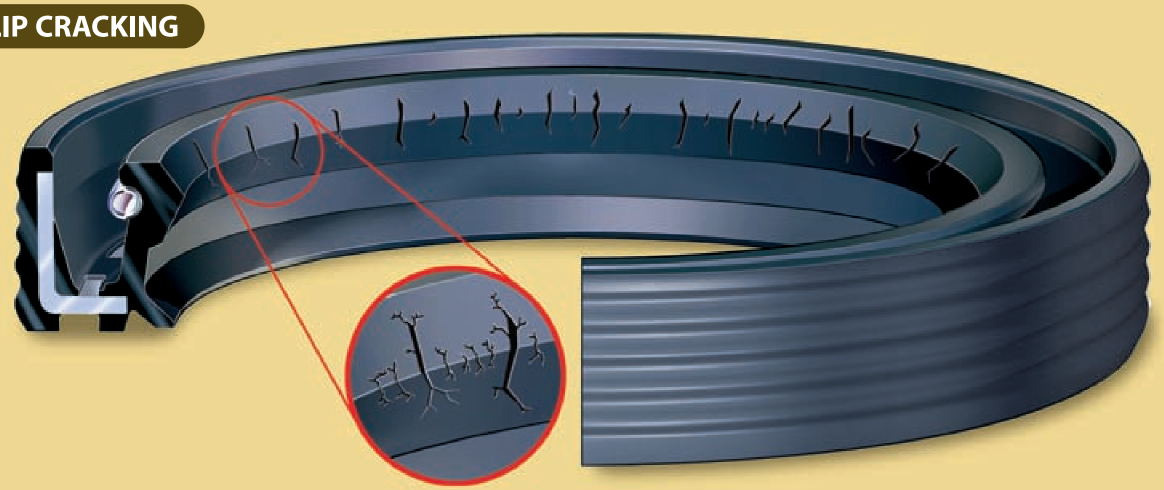


Figure 32: Impaired Sealing Due To Cracking

ability to maintain proper contact with the shaft, cracks also act as minute leak paths through which fluid can escape.

There are several different ASTM tests designed to gauge a material's resistance to ozone degradation. ASTM D 1149 calls for 20% elongation of a material sample that is then exposed to a prescribed ozone/air mixture. This exposure takes place within an enclosed ozone chamber heated to 104° F. Following a specified length of exposure, the sample is examined for the presence of surface cracks. ASTM D 1171 utilizes largely the same procedure, with the difference that the material specimens are triangular and are wrapped around a mandrel (rather than elongated).

selecting the lip material

Chemical Compatibility Guide.

To help you better gauge the usefulness of a seal material in a given fluid environment, we offer this Chemical Compatibility Guide (see *Table 3*, pages 36 and 37). It features the most common fluids that your shaft seals will be likely to encounter in service.

This guide is based on volume swell, and though excessive swell is perhaps the most obvious sign of incompatibility, it is not the sole indicator. Changes in physical properties such as hardness and tensile strength are common. Length of exposure and ambient temperature will also undoubtedly affect a material's response to a chemical.

Please note: This information is provided for educational purposes only. It has been accumulated from multiple industry publications and authorities. To the best of our knowledge, it is correct for shaft seal applications. R.L. Hudson has not performed any tests or analysis to verify the data contained herein, and the information contained herein is not intended as, and shall not serve as, any warranty or representation, either express or implied.

The user is solely responsible for making any and all usage decisions. These decisions should always be based on both an appropriate analysis of the information and a thorough testing of the products *prior* to actual field use.

"This guide features the most common fluids that your shaft seals will be likely to encounter in service."

	NBR	HNBR	FKM	ACM	VMQ	PTFE
Automatic Transmission Fluid	2	1	1	2	4	1
Engine Oil (Petroleum-Based)	2	1	1	2	4	1
Engine Oil (Synthetic-Based)	3	2	2	4	4	1
Fuel Oil	2	1	1	2	4	1
Gasoline	2	1	1	4	4	1
Gear Lubricant (EP)	4	1	1	4	4	1
Gear Lubricant (Non-EP)	2	1	1	4	4	1
Grease (Petroleum-Based)	1	1	1	3	4	1
Grease (Synthetic-Based)	3	2	2	4	4	1
Hydraulic Oil (Petroleum-Based)	1	1	1	1	3	1
Hydraulic Oil (Synthetic-Based)	3	2	2	4	4	1
Kerosene	2	1	1	2	4	1
Skydrol 500	4	4	4	4	4	1
Water	1	1	3	4	4	1

Table 3: Chemical Compatibility Guide

CHEMICAL COMPATIBILITY GUIDE RECOMMENDATION CODES

1 = Excellent: *No more than 15% swell. Any softening or surface degradation is minimal. Material should perform well in all but the most extreme conditions.*

2 = Good: *Up to 30% swell. Minor softening or surface degradation is likely, though static applications may not be jeopardized.*

3 = Poor: *Up to 50% swell. Chemical resistance is definitely compromised, making the material unsuitable for some static and many dynamic applications.*

4 = Not Recommended: *More than 50% swell. Major softening and material degradation are very likely. Usage should be avoided and an alternative material sought.*

Be aware that a material's response to a fluid is largely dependent on the application in question. A material may respond to a fluid differently when molded as an O-ring than when used for a shaft seal.

Our online Chemical Compatibility Guide features over 2,000 different fluids seen in applications of all types. It is available at www.rlhudson.com.

selecting the lip material

Thermal Properties.

"Because shaft seals may be asked to perform in extreme heat or extreme cold, there are several important thermal properties to consider."

Because shaft seals may be asked to perform in extreme heat or extreme cold, there are several important thermal properties to consider when choosing an elastomeric compound for the sealing lip. These include both high and low temperature effects. The coefficient of thermal expansion must also be considered when selecting a case material.

HIGH TEMPERATURE EFFECTS

Most of the physical and chemical properties discussed thus far are impacted when an elastomeric compound is exposed to high temperatures. Whether these temperatures are inherent to the application or the result of frictional buildup, they can be dangerous, especially if the elastomeric lip is exposed for a prolonged period of time. Affected properties can include hardness, tensile strength, modulus, elongation, compression set, and lip volume.

Unless specially formulated, elastomers will typically soften when first exposed to high temperatures. Extended heat exposure can cause irreversible changes in tensile strength and elongation, as well as alterations in the chemical makeup of the compound such that it progressively and permanently hardens. This hardening is the result of additional cross-linking, plasticizer evaporation, and / or oxidation. A hardened seal is less flexible and therefore less able to follow any eccentricities in the shaft. This progressive inability to maintain proper contact with the shaft can result in leakage past the lip.

Two ASTM test methods for gauging high temperature effects are most used in relation to shaft seals. Both are designed to gauge the amount of material degradation that results from exposure to a heated environment. The difference between these two tests is mainly the device used to maintain pressure and heat on the specimen. ASTM D 573 details testing in an air oven, and ASTM D 865 describes heat and air testing within a test tube enclosure.

LOW TEMPERATURE EFFECTS

Unlike the changes that result from exposure to high temperatures, changes brought about by low temperature exposure are generally not permanent and can often be reversed if heat returns. For example, extended exposure to low temperatures will increase an elastomer's hardness, but the material can soften again if the temperature rises (due, for example, to ambient changes or frictional build-up). Of course, the ambient temperature may not increase, and frictional heat may not warm the lip sufficiently to offset the low temperature effects. In such instances, the lip will become progressively stiff and brittle. The brittleness will make it more susceptible to fracturing, which will result in immediate seal failure. Even if the lip doesn't fracture, its modulus will continue to increase. As modulus increases, followability decreases. A stiff lip is less able to flex and follow shaft eccentricities. A gap will develop between the lip and the shaft, and leakage may result.

There are two main tests related to low temperature effects. The first is detailed in ASTM D 2137 (Method A) as a way to determine a sample's "brittleness point," or the lowest temperature at which the sample will not fracture or crack when struck once. The second test is described in ASTM D 1329. Better known as a "TR-10," this temperature retraction test (see ***Figure 33***) is considered by many within the rubber industry to be the most useful indicator of a material's low temperature performance.

In a nutshell, the TR-10 measures material resilience. Samples are frozen in a stretched state, then gradually warmed until they lose 10% of this stretch (i.e. retract by 10%). The results of such tests are believed to provide a good basis for evaluating the effects of crystallization and the impact of low temperatures on visco-elastic properties. TR-10 results are generally thought to be consistent with the capabilities of most dynamic seals, including shaft seals.

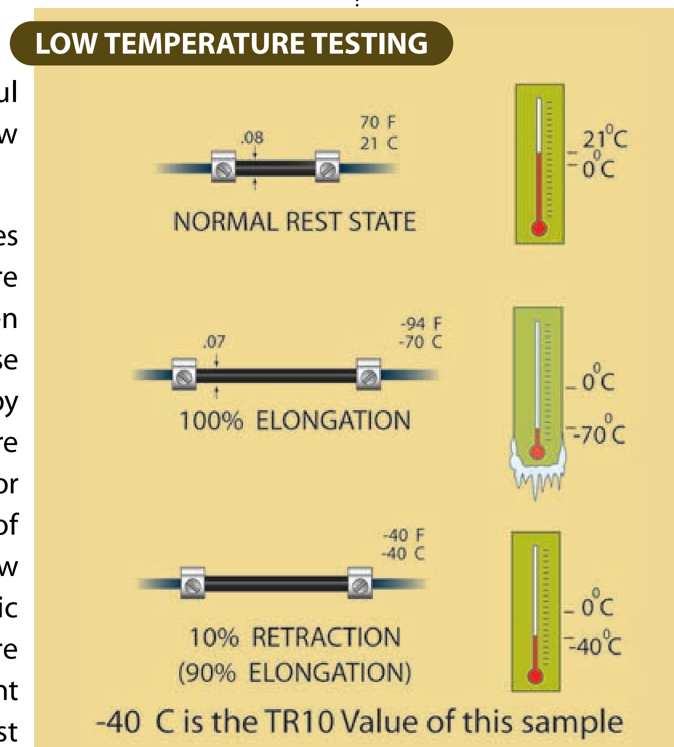


Figure 33: Temperature Retraction, or "TR-10"

DIFFERENTIAL THERMAL EXPANSION

The elastomeric lip isn't the only part of a shaft seal subject to temperature-induced changes. The metal case can also be affected, and this can be a serious issue, particularly if the case and the housing bore into which it fits are made of different materials. For example, when subjected to high temperatures, an aluminum bore will expand more quickly than a carbon steel case. Known as *differential thermal expansion*, this disparity can turn what was initially a tight fit between the case and the bore into a loose fit. A loose fit (like that shown in ***Figure 34***) can allow fluid to escape between the case and the bore (around the seal's O.D.). In extreme cases, the seal can start rotating in the bore and/or pop out.

DIFFERENTIAL THERMAL EXPANSION

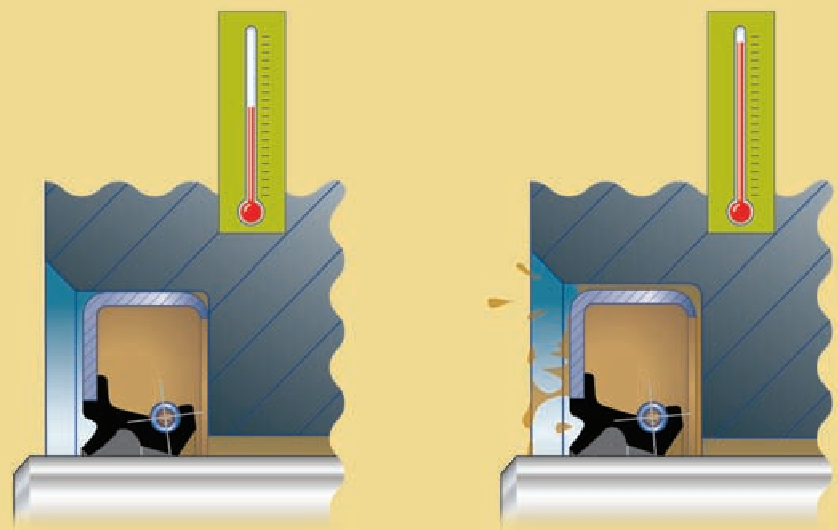


Figure 34: Development of a Leak Path

The opposite effect is also possible. Low temperatures cause different metals to contract at different rates. If the housing contracts at a faster rate than the seal's metal case, deformation of the housing bore or seal case may occur. Designs in which both the bore and the case are made of the same metal can help eliminate concerns about either expansion or contraction. When using the same metal isn't possible, a seal with a rubber covered O.D. may be required to maintain a proper seal and fit. See ***page 87*** for more information on how a shaft seal O.D. may be treated to combat differential thermal expansion.

With these physical, chemical, and thermal properties in mind, let's now look more specifically at the relative strengths and weaknesses of the most commonly used shaft seal lip materials.

selecting the lip material

Material Profiles.

The following profiles of the most commonly used shaft seal polymers are intended to serve as general guidelines only. Inherent strengths and weaknesses are noted, but keep in mind that these properties may be enhanced or diminished through compounding.

For this reason, selecting the best material for a given application will inevitably require both comparison and compromise. When possible, you should always test the final formulation (the base polymer and all additional modifying agents) in actual service conditions prior to field use. Table 4 shows the color coding for the individual atoms depicted in the molecular illustrations of each polymer.

"Selecting the best material for a given application will inevitably require both comparison *and* compromise."

Hydrogen
Carbon
Nitrogen
Oxygen
Fluorine
Silicon
Sulfur
Chlorine

Table 4: Atomic Color Coding

Nitrile (Buna N).

ASTM D 1418 Designation: NBR, XNBR

ASTM D 2000, SAE J200 Type / Class: BF, BG, BK, CH

RELATIVE COST: Low

GENERAL TEMPERATURE RANGE: -25° to +225° F

Nitrile rubber is the most commonly used elastomer in the manufacture of shaft seals and other sealing devices. Also known as Buna N, nitrile (see ***Figure 35***) is a copolymer of butadiene and acrylonitrile (ACN). The name Buna N is derived from butadiene and natrium (the Latin name for sodium, the catalyst used in the polymerization of butadiene). The letter "N" stands for acrylonitrile.

"Nitrile rubber is the most commonly used elastomer in the manufacture of shaft seals and other sealing devices."

The butadiene segment imparts elasticity and low temperature flexibility. It also contains the "unsaturated" double bond that is the site for crosslinking, or vulcanization. This unsaturated double bond is also the main attack site for heat, chemicals, and oxidation.

The acrylonitrile segment imparts hardness, tensile strength, and abrasion resistance, as well as fuel and oil resistance. Heat resistance is also improved through increased ACN content, which typically ranges from 18% to 45%. A standard, general-purpose nitrile compound usually contains 34% ACN. Low temperature performance is reduced as ACN level increases.

NBR

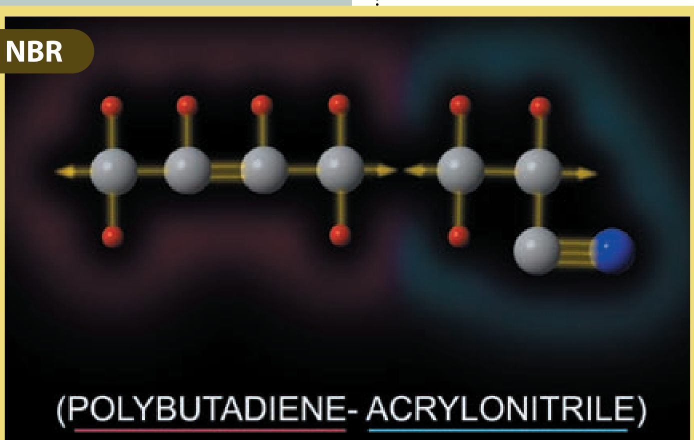


Figure 35: Molecular Structure of Nitrile

The relationship between the ACN content, volume swell in IRM 903 oil and the brittle point of the elastomer is illustrated in ***Table 5***.

General-purpose nitrile compounds with a 34% ACN content have a recommended temperature range of -25° F to +225° F (-32° C to +107° C). The low temperature capability can be extended to -40° F (-40° C) by reducing the ACN content to about 30%. Nitrile compounds with an even lower ACN content of 18% to 20%

remain flexible at temperatures down to -65° F (-54° C). Be aware that reducing ACN content improves low temperature properties but reduces high temperature properties, increases material swell, and reduces fluid resistance.

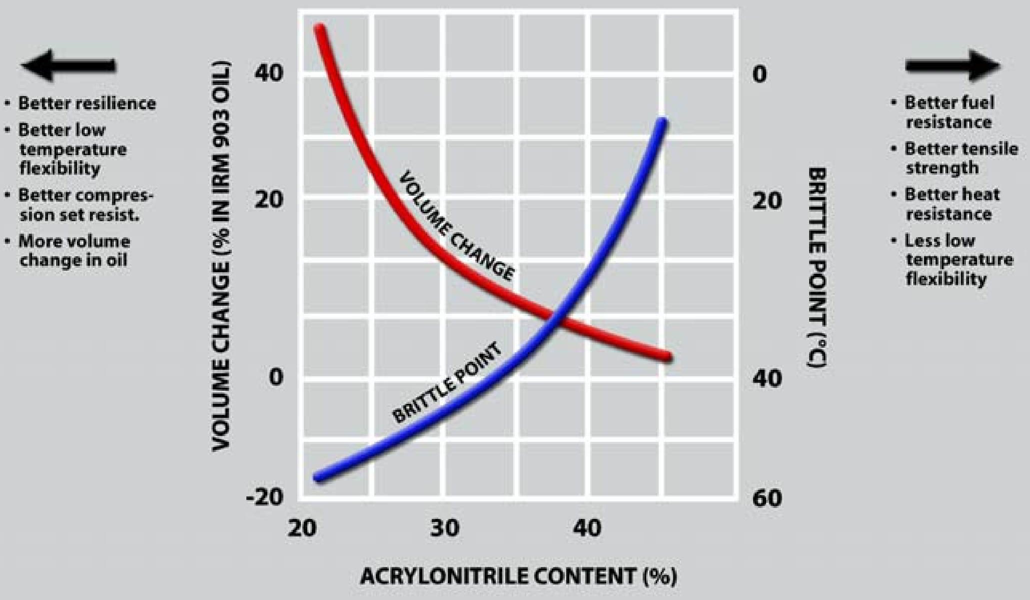


Table 5: Relationship Between ACN content, Volume Swell in IRM 903 Oil, and Brittle Point

Unfortunately, compounding ingredients and polymers that offer the best low temperature properties are usually adversely affected by high temperatures. A general-purpose compound is cured with sulfur, but as the ambient temperature in an application exceeds +225° F, free sulfur in the compound finds other unsaturated double bonds and forms additional crosslinks. This results in compression set and hardening of the compound. To improve high temperature properties, a peroxide cure system and/or mineral fillers must be used. Peroxide-cured compounds have both better high temperature properties (up to +250° F, +121° C) and improved compression set characteristics, but they are also more difficult to process and more expensive than sulfur-cured compounds.

Nitrile compounds have excellent tensile strength, as well as excellent abrasion, tear, and compression set resistance. Because of the double bonds present in the polybutadiene segments of the chemical backbone, nitrile compounds have poor resistance to ozone, sunlight, and weathering. They should not be stored near ozone-generating electric motors, welding equipment, or in UV light.

NBR PERFORMS WELL IN:

- Petroleum oils & fuels
- Silicone oils & greases
- Ethylene glycol
- Dilute acids
- Water (below 212° F)

NBR DOES NOT PERFORM WELL IN:

- Aromatic hydrocarbons (benzene, toluene, xylene)
- Halogen derivatives (carbon tetrachloride, trichloroethylene)
- Ketones (MEK, acetone)
- Phosphate ester hydraulic fluids (Skydrol®, Pydraul®)
- Strong acids
- Extreme pressure (EP) lubes at elevated temperatures

Carboxylated nitrile rubber compounds (XNBR) provide even better strength properties, especially abrasion resistance. Carboxylated nitriles are produced by the inclusion of carboxylic acid groups on the polymer during polymerization. These carboxylic acid groups provide extra “pseudo” crosslinks, producing harder, tougher compounds with higher abrasion resistance, modulus, and tensile strength than standard nitriles. Carboxylated nitriles are, however, less flexible at low temperatures and less resilient than non-carboxylated compounds. Also, the “pseudo” crosslinks (being ionic in nature) are thermally sensitive. As temperatures increase, the ionic bonds lose strength.

Other nitrile variations are possible, including internally lubricated compounds with improved friction and wear properties, as well as Food and Drug Administration (FDA) and National Sanitation Foundation (NSF) formulations for food and potable water applications.

Hydrogenated Nitrile.

ASTM D1418 Designation: HNBR

ASTM D 2000, SAE J200 Type / Class: DH

RELATIVE COST: High

GENERAL TEMPERATURE RANGE: -25° to +300° F

Though the double bonds within nitrile's butadiene segments are needed for cross-linking, they are also the predominant attack sites for heat, chemicals, and oxidation. As part of an ongoing effort to engineer more resistant compounds, a different class of nitrile was developed in the 1980s. Initially known as highly saturated nitrile (HSN), this class is now more commonly called hydrogenated nitrile butadiene rubber (HNBR), or just hydrogenated nitrile (see ***Figure 36***).

Hydrogenated nitrile results from the *hydrogenation* of standard nitrile. Hydrogenation is the process of adding hydrogen atoms to the butadiene segments. Adding hydrogen greatly reduces the number of carbon-to-carbon double bonds that would otherwise be weak links in the polymer chain. Why are double bonds weak? It stems from *valence*, or the ability of an atom to form one or more energy bonds with neighboring atoms. A carbon atom can form four distinct covalent bonds. Because carbon has this valence of four, it is most "satisfied" when it has actually formed four single bonds (a state known as *saturation*) rather than two single bonds and a double bond. A satisfied, saturated atom is more stable, so a compound composed largely of saturated carbons is less reactive and more resistant to chemical attack.

"As part of an ongoing effort to engineer more resistant compounds, a different class of nitrile was developed in the 1980s."

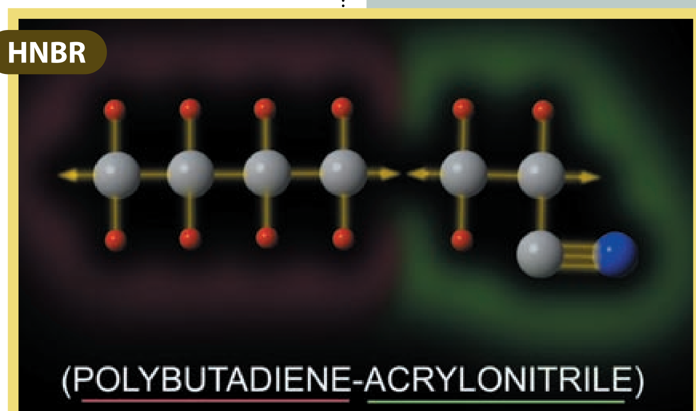


Figure 36: Molecular Structure of Hydrogenated Nitrile

As shown in ***Figure 36***, HNBR's main chain is primarily composed of highly saturated hydrocarbons and acrylonitrile (ACN). Thanks to their saturation, the hydrocarbon segments impart heat, chemical, and ozone resistance. On the down side, increased hydrogenation also leads to decreased low temperature elasticity. As with standard nitrile, the ACN content of HNBR imparts toughness, as well as fuel and oil resistance. This ACN content can be modified for specific uses. There are also a few remaining unsaturated butadiene segments (typically well under 10%) to facilitate peroxide curing or, in some instances, sulfur vulcanization. Peroxide-cured HNBR has improved thermal properties and will not continue to vulcanize like sulfur-cured nitriles.

HNBR is used in automotive air conditioning systems where R134a refrigerant gas has replaced the chlorofluorocarbon (CFC)-containing R12 refrigerant. HNBR is used in fuel parts due to its increased resistance to sour gasoline and ozone. It is used in engine applications because of its resistance to elevated temperatures and oil additives.

HNBR has excellent abrasion resistance, making it a viable alternative to FKM. HNBR also has better low temperature properties and tear resistance than FKM.

HNBR PERFORMS WELL IN:

- Petroleum oils & fuels
- Silicone oils & greases
- Water & steam (up to 300° F / 149° C)
- Ozone

HNBR DOES NOT PERFORM WELL IN:

- Chlorinated hydrocarbons
- Polar solvents (esters & ketones)
- Strong acids

Fluoroelastomer.

ASTM D1418 Designation: FKM

ASTM D 2000, SAE J200 Type / Class: HK

RELATIVE COST: High

GENERAL TEMPERATURE RANGE: -15° to +300° F

Fluoroelastomers are thermoset elastomers containing fluorine (see ***Figure 37***). Fluoroelastomers make excellent general-purpose seals due to their exceptional resistance to chemicals, oil, and temperature extremes (-15° to +300° F). Specialty compounds can further extend the low temperature limit down to about -25° F for dynamic seals.

Fluoroelastomers usually have good compression set resistance, good oil resistance, and resistance to ozone and sunlight. FKM compounds are widely used in the automotive, appliance, fluid power, and chemical processing industries.

Three main factors contribute to the remarkable heat (see ***Table 6***, next page) and fluid resistance of fluoroelastomers. First, there are extremely strong bonds between the carbon atoms comprising the polymer backbone and the attached (pendant) fluorine atoms. Under most circumstances, these bonds cannot be broken, and thus the polymer is not prone to undergo chain scission (division of the macromolecular chains into smaller, weaker, more susceptible segments).

Second, fluoroelastomers feature a high fluorine-to-hydrogen ratio. In other words, fluorine (rather than hydrogen) atoms fulfill the majority of the available bonds along the material's carbon backbone. Polymers with a high level of fluorination have proven to be extremely stable. A stable compound is less inclined to either react to, or be broken down by, its environment.

"Fluoroelastomers make excellent general purpose seals due to their exceptional resistance to chemicals, oil, and temperature extremes."

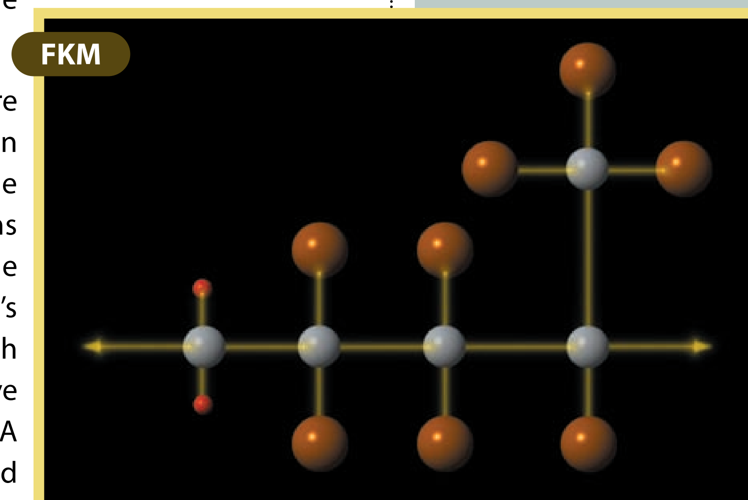


Figure 37: Molecular Structure of Fluorocarbon (Type A)

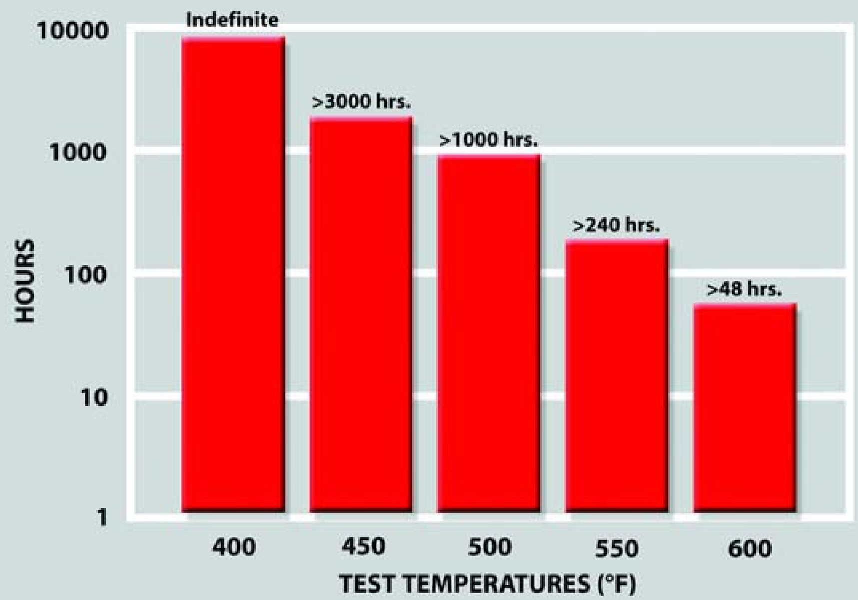


Table 6: Heat Resistance of FKM (As Conducted on Test Slabs)

Third, the carbon backbone is fully saturated. That is, it contains only single bonds between the carbon atoms. It does not contain any of the covalent double bonds present in unsaturated compounds. Since double bonds are the focus for chemical attack, the saturated structure of fluoroelastomers renders them impervious to harmful agents (such as oxygen, ozone, and UV light) that typically degrade unsaturated materials.

FKM PERFORMS WELL IN:

- Petroleum oils & fuels
- Acids
- Aircraft engine applications
- Hard vacuum applications
- Silicone oils & greases
- Solvents

FKM DOES NOT PERFORM WELL IN:

- Amines
- Methanol
- Hydrocarbons (nitro)
- Ketones
- Low molecular weight esters & ethers
- Fireproof hydraulic fluids (e.g. Skydrol®)

Depending on the needs of your application, there are a number of different fluoroelastomer formulations available. Though they may share some common characteristics, these different types are distinguished by their processing and end-use properties. The best-known fluoroelastomer manufacturer is DuPont Dow Elastomers; the trade name for their compound, Viton®, is often used as if it were a generic term for FKM. In the interests of simplicity, the following descriptions of some of the most common FKM formulations will make use of the DuPont "type" names.

The original commercial fluoroelastomer, **Viton A**, is the general-purpose type and is still the most widely used. It is a copolymer of vinylidene fluoride (VF_2) and hexafluoropropylene (HFP). Generally composed of 66% fluorine, Viton A compounds offer excellent resistance against many automotive and aviation fuels, as well as both aliphatic and aromatic hydrocarbon process fluids and chemicals. Viton A compounds are also resistant to engine lubricating oils, aqueous fluids, steam, and mineral acids.

Viton B fluoroelastomers are terpolymers combining tetrafluoroethylene (TFE) with VF_2 and HFP. Depending on the formulation, the TFE partially replaces either the VF_2 (raising the fluorine level to about 68%) or the HFP (keeping the fluorine level steady at 66%). Viton B compounds offer better fluids resistance than Viton A copolymers.

Viton F fluoroelastomers are terpolymers combining TFE, VF_2, and HFP (as in Viton B), except the fluorine level is approximately 69%. Viton F compounds have the best fluid resistance of the various Viton types, but Viton F compounds (and Viton GF compounds, see below) also have the poorest low temperature properties of the various Viton types.

Viton GF fluoroelastomers are tetrapolymers composed of TFE, VF_2, HFP, and small amounts of a cure site monomer. Presence of the cure site monomer allows peroxide curing of the compound, which is normally 70% fluorine. As the most fluid resistant of the various FKM types, Viton GF compounds offer improved resistance to water, steam, and acids. As mentioned above, Viton F and GF compounds have the poorest low temperature properties of the Viton types.

Viton GLT fluoroelastomers are engineered to combine the heat and chemical resistance of general Viton types (like A,B, and F) with improved low temperature properties. The "LT" in

PERCENT FLUORINE	VITON A 66	VITON B 68	VITON GF 70	VITON GFLT 67
FUEL SWELL (70 hrs. at 23° C):				
Fuel C, % volume	+5	+4	+3	+3
Methanol, % volume	+70	+22	+3	+5
LOW TEMPERATURE FLEXIBILITY:				
TR-10, °C/°F	-17/1	-14/7	-6/21	-24/-11
COST (Relative to Viton A)	1	1	1.25	2.75

Table 7: Comparison of Standard Fluorocarbons to Specialty Type GFLT

Viton GLT stands for "low temperature." The *glass transition temperature* (Tg) is generally accepted as an indicator of low temperature capabilities; the Tg of Viton GLT is typically 8° to 12° C lower than for the general-use Viton types.

Viton GFLT fluoroelastomers combine VF_2, perfluoromethylvinyl ether (PMVE), TFE, and a cure site monomer. The result retains both the superior chemical resistance and high heat resistance of the GF-series fluoroelastomers. Viton GFLT compounds (typically 67% fluorine) also offer the lowest swell and the best low temperature properties of the types discussed here (see ***Table 7***). As with Viton GLT, the "LT" in Viton GFLT stands for "low temperature." ***Table 8*** compares both the low temperature flexibility and fuel swell for several Viton formulations.

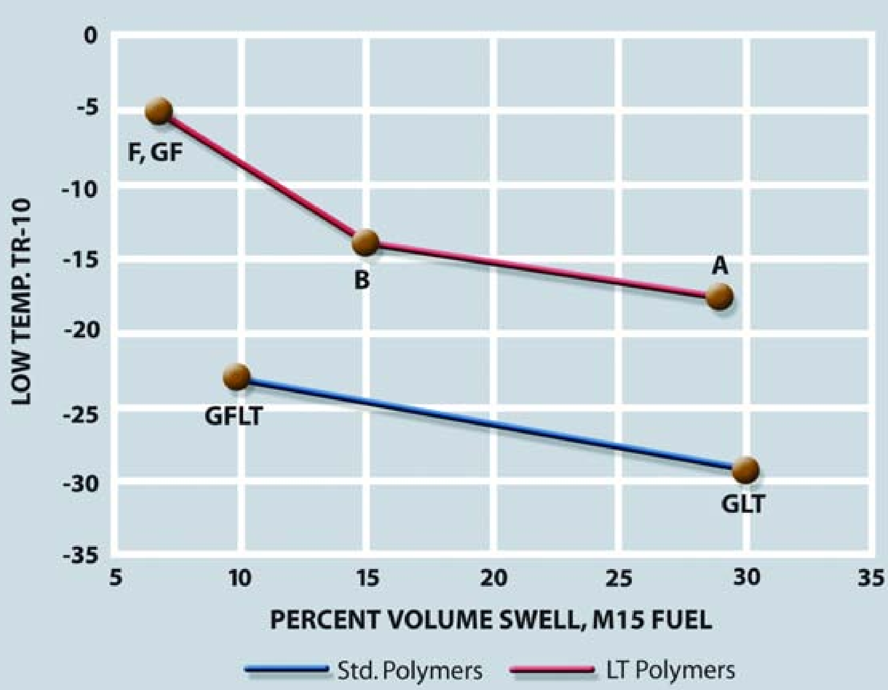


Table 8: Comparison of Low-Temp Flexibility & Fuel Swell For Viton Types

Polyacrylate.

ASTM D 1418 Designation: ACM

ASTM D 2000, SAE J200 Type / Class: DF, DH

RELATIVE COST: Medium

GENERAL TEMPERATURE RANGE: -25° to +275° F

Polyacrylate is a copolymer (ethyl acrylate, see ***Figure 38)*** which offers good resistance to petroleum fuels and oils. Resistant to flex cracking, polyacrylate also resists damage from oxygen, sunlight, and ozone (due to main chain saturation).

For low temperature applications, particular care must be taken to ensure that the polymer used is cold resistant; some polyacrylate polymers will crack when flexed at temperatures of 0° F or lower. Though it is marginally more resistant to hot air than nitrile, polyacrylate falls short in strength and compression set resistance, as well as in resistance to water and low temperatures. For these reasons, polyacrylate is slowly being replaced by fluoroelastomers and hydrogenated nitriles (though both are more expensive).

"Polyacrylate offers good resistance to petroleum fuels and oils."

ACM PERFORMS WELL IN:

- Petroleum oils & fuels
- Automatic transmission fluid
- Type A power steering fluid

ACM DOES NOT PERFORM WELL IN:

- Alcohol
- Alkalies
- EP fluids
- Gear oils
- Glycols
- Hydrocarbons (aromatic, chlorinated)
- Water

ACM

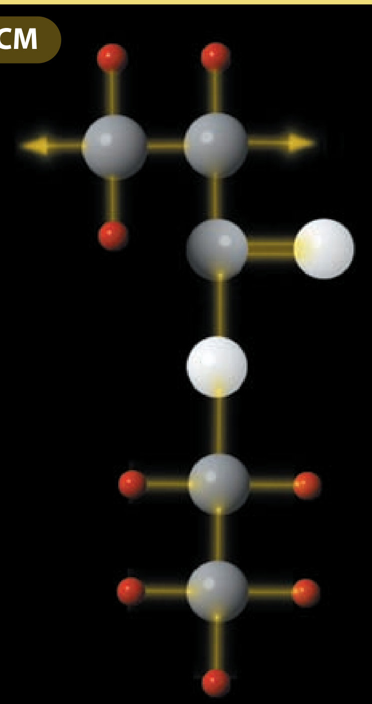


Figure 38: Molecular Structure of Polyacrylate

selecting the lip material

Silicone.

ASTM D 1418 Designation: VMQ

ASTM D 2000, SAE J200 Type / Class: FC, FE, GE

RELATIVE COST: Medium

GENERAL TEMPERATURE RANGE: -65° to +300° F

Though carbon and hydrogen are part of their chemistry, silicones are primarily based on a strong sequence of silicon and oxygen atoms (see ***Figure 39***) rather than a long chain of carbon atoms (as with many hydrocarbons). This silicon-oxygen backbone is much stronger than a carbon-based backbone, making silicones more resistant to extreme temperatures (-65° F to +300° F, -54° C to 149° C), chemicals, and shearing stresses.

"Silicones are primarily based on a strong sequence of silicon and oxygen atoms rather than a long chain of carbon atoms."

Due to saturation in the polymer's main chain, silicones are very resistant to oxygen, ozone, and UV light. Of course, this same saturation also demands that the material be peroxide cured since it is not possible to sulfur cure a saturated polymer. In addition to being generally inert (non-reactive), silicones are odorless, tasteless, non-toxic, and fungus resistant. They also have great flexibility retention and low compression set.

VMQ

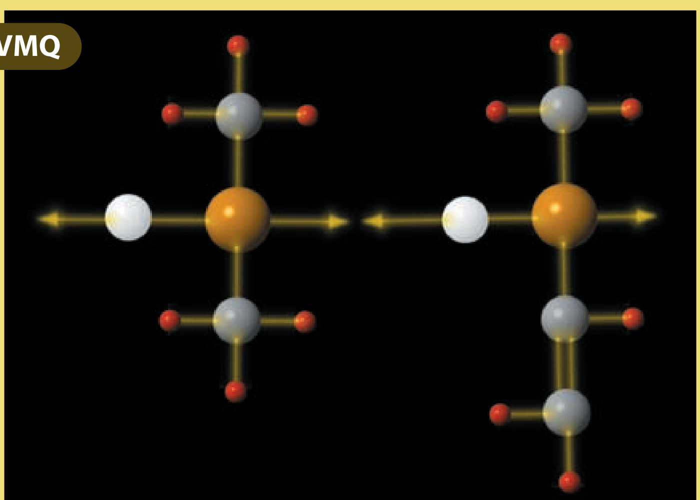


Figure 39: Molecular Structure of Silicone

As the compositions of modern lubricants have become more complex (i.e. incorporating more additives), use of silicone as a material in radial lip seals has declined. In actuality, silicones are not well suited for dynamic use due to their high friction characteristics, low abrasion resistance, and poor tensile and tear strength; silicones see only limited use as shaft seals for these reasons. (Fluoroelastomers and hydrogenated nitriles are often used instead.) Silicones typically undergo a large amount of shrinkage after molding, so special mold designs are required to compensate for this. Silicones swell considerably in both aliphatic and aromatic hydrocarbon fuels unless a special compound is formulated. Silicones are also very gas permeable.

SILICONE PERFORMS WELL IN:

- Mineral oils with low additive contents
- Ozone
- Dry heat

SILICONE DOES NOT PERFORM WELL IN:

- Concentrated acids
- EP fluids
- Fuels
- Gear oils
- Ketones (MEK, acetone)
- Steam

Polytetrafluoroethylene.

ASTM D 1418 Designation: FEP

ASTM D 2000, SAE J200 Type / Class: None

RELATIVE COST: High

GENERAL TEMPERATURE RANGE: -65° to +325° F

Polytetrafluoroethylene (PTFE) is a completely fluorinated polymer produced when the monomer tetrafluoroethylene (TFE) undergoes free radical vinyl polymerization. Because of PTFE's unique properties, it is used for high pressure, high speed, and non-lubricated applications, as well as in applications involving aggressive chemicals.

"Because of PTFE's unique properties, it is used for high pressure, high speed, and non-lubricated applications, as well as in applications involving aggressive chemicals."

As a monomer, TFE is made up of a pair of double-bonded carbon atoms, both of which have two fluorine atoms covalently bonded to them. Thus the name: "tetra" means there are four atoms bonded to the carbons, "fluoro" means those bonded atoms are fluorine, and "ethylene" means the carbons are joined by a double bond as in the classic ethylene structure. (Ethylene has hydrogen atoms attached to the carbons, as in ***Figure 40***, but TFE has fluorine in place of the hydrogen, as in ***Figure 41***.) When TFE polymerizes into PTFE, the carbon-to-carbon double bond becomes a single bond and a long chain of carbon atoms is formed, as in ***Figure 42***. This chain is the polymer's backbone. PTFE powders are combined with a variety of fillers to improve strength, wear resistance, and lubricity. PTFE is often used in composite high-pressure seals as a backup / support ring.

With a ratio of four fluorine atoms to every two carbon

FOUR FLUORINES

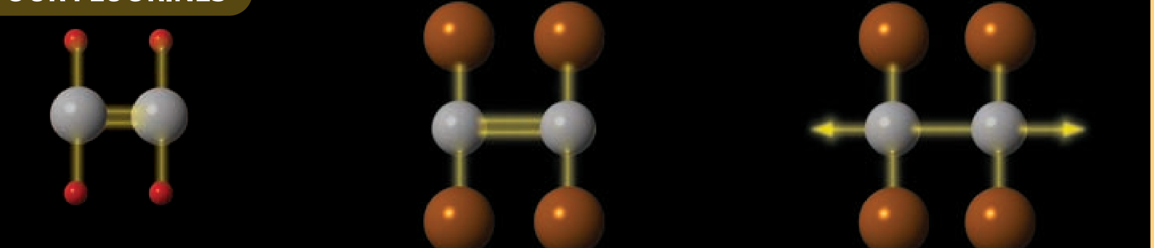


Figure 40: Ethylene *Figure 41: Tetrafluoroethylene* *Figure 42: Polytetrafluoroethylene*

atoms, the backbone is essentially shielded from contact. It's almost impossible for any other chemical structure to gain access to the carbon atoms. Even if an agent could gain access, the carbon-to-fluorine bonds have high bond disassociation energy, so they're almost unbreakable. This makes PTFE the most chemically resistant thermoplastic polymer available. PTFE is inert to almost all chemicals and solvents, so PTFE parts function well in acids, alcohols, alkalies, esters, ketones, and hydrocarbons. Only a few substances harm PTFE, notably fluorine, chlorine trifluoride, and molten alkali metal solutions at high pressures.

PTFE is also very slippery. By its very nature, the fluorine in PTFE repels everything. As part of a molecule, fluorine is decidedly "anti-social." Anything getting close is repelled, and repelled molecules can't stick to the PTFE surface. This makes PTFE perfect for applications requiring a low coefficient of friction. The only thing slicker than PTFE is ice! Because they are essentially self-lubricating, PTFE parts are ideal for applications in which external lubricants (such as oils and greases) can't be used. PTFE's inherent anti-stick characteristics do, however, make it difficult to bond to other materials (such as the metal of a shaft seal case). For this reason, the PTFE lips in PTFE shaft seals may sometimes need to be clamped in place. Some seals bond the PTFE to rubber, which is then bonded to the case.

PTFE can withstand a wide range of temperatures (-65° to 325° F, -54° to 163° C). On the down side, PTFE has poor cut resistance, so extra care must be taken during installation so as not to damage shaft seal lips made of PTFE. PTFE also has poor flexibility, and special designs are required when it is used as a shaft seal lip material. Sealability decreases noticeably in the presence of shaft vibrations or eccentricities.

PTFE PERFORMS WELL IN:

- Petroleum oils & fuels
- Ozone
- Chemical applications
- Solvents (MEK, acetone, xylene)
- Weather

PTFE DOES NOT PERFORM WELL IN:

- Highly fluorinated greases & oils

E = Excellent
G = Good
F = Fair
P = Poor

MATERIAL NAME / ASTM D1418 DES.	ASTM D 2000, SAE J200 Type / Class	Relative Cost	General Temperature Range (°F) *	Abrasion Resistance	Acid Resistance	Base Resistance	Chemical Resistance
NBR NBR, XNBR	BF, BG, BK, CH	Low	-25 to 225	G	F	G	F-G
Hydrog. Nitrile HNBR	DH	High	-25 to 300	G	E	E	F-G
Fluoroelastomer FKM	HK	High	-15 to 300	G	E	P-F	E
Polyacrylate ACM	F, DH	Med	-25 to 275	G	P	P	P
Silicone VMQ, PVMQ	FC, FE, GE	Med	-65 to 300	P	F-G	E	G-E
Teflon® FEP	None	High	-65 to 325	P-G	E	F-E	E

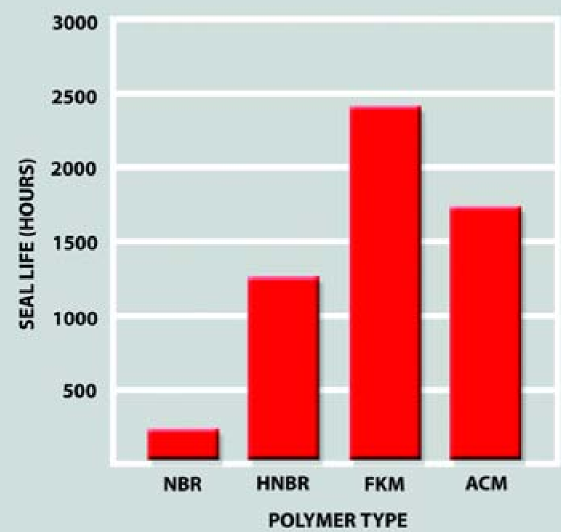
* temperature limits specific to shaft seal applications; upper and lower temperature limits may be extended for other applications

Table 9: Base Polymer Properties Summary (Specific properties may be enhanced through compounding.)

E = Excellent
G = Good
F = Fair
P = Poor

MATERIAL NAME ASTM D1418 DES.	Cold Resistance	Dynamic Properties	Electrical Properties	Flame Resistance	Gas Impermeability	Heat Resistance	Oil Resistance	Ozone Resistance	Set Resistance	Steam Resistance	Tear Resistance	Tensile Strength	Water Resistance	Weather Resistance
Nitrile NBR, XNBR	G	G-E	F	P	G	G	E	P	G-E	P	F-G	G-E	G	F
Hydrog. Nitrile HNBR	G	G-E	F	P	G	E	E	G	G-E	G	F-G	E	G	G
Fluoroelastomer FKM	P-F	G-E	F	E	G	E	E	E	G-E	P	F	G-E	G	E
Polyacrylate ACM	P	F	F	P	E	E	E	E	F	P	F-G	F	P	E
Silicone VMQ, PVMQ	E	P	E	F	P	E	F-G	E	G-E	F-P	P	P	E	E
Teflon® FEP	E	P	E	E	G	E	E	E	G	E	P	F	E	E

Table 10 *shows the results of life tests on shaft seals with the same design and size but with lips molded from different polymer types.*



Shaft Size: 76.2 mm (3.000 inch); Shaft Speed: 2165 RPM; Lubricant: SAE 30 DRO: 0.13 mm (0.005 inch); STBM: 0.13 mm (0.005 inch); Seal Cock: Zero Pressure: Zero; Sump Level: Centerline; Cycle: 20 hours on, 4 off; Sump Temperature: 135° C (275° F)

Table 10: Effect of Polymer Type Upon Seal Life

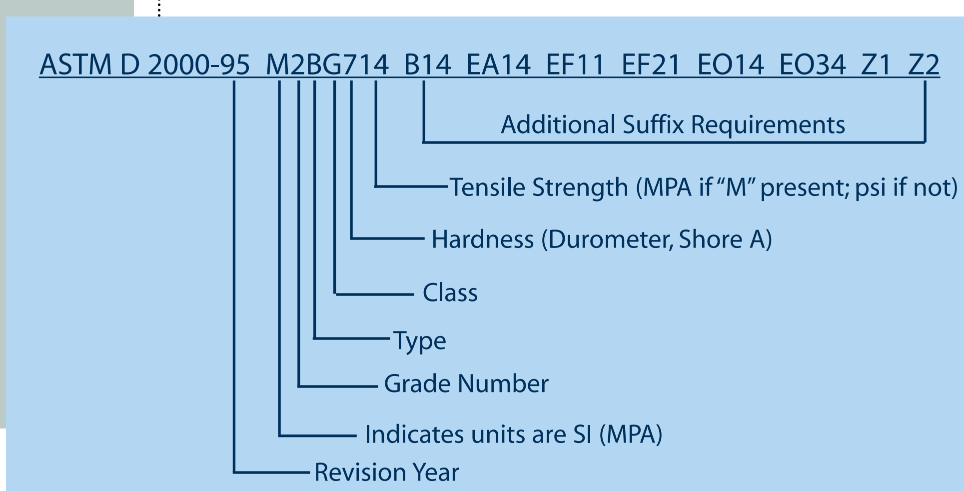
Understanding ASTM D 2000 / SAE J200.

Having discussed the properties and uses of the elastomers most commonly used in shaft seal manufacturing, the question then becomes: How can these properties be succinctly specified when an existing compound is being selected or when a new compound must be formulated?

In order to provide guidance in the selection of vulcanized rubber materials, and to provide a method for specifying these materials by the use of a simple line call-out specification, the American Society for Testing and Materials (ASTM) and the Society of Automotive Engineers (SAE) established ASTM D 2000 / SAE J200. Though these standards are virtually identical, J200 finds its widest use within the automotive industry. D 2000 is the more common tool among rubber manufacturers. Specifying your elastomer choice via a standardized line call-out is a good idea because it allows the flexibility of using different manufacturers' compounds while ensuring that the material quality and performance stay consistent.

"Specifying your elastomer choice via a standardized line call-out is a good idea because it allows the flexibility of using different manufacturers' compounds while ensuring that the material quality and performance stay consistent."

D 2000 is based on the premise that the properties of all rubber products can be arranged into characteristic material designations. These designations are determined by types, based on resistance to heat aging, and classes, based on resistance to swelling. Here is the line call-out, or specification, for "N470," a 70 (Shore A) durometer nitrile:



ASTM D 2000-95 M2BG714 B14 EA14 EF11 EF21 EO14 EO34 Z1 Z2

This line call-out contains the following:

A. The document name (ASTM D 2000-95). The two-digit number following the hyphen indicates the revision year (in this case, 1995).

B. The letter "M" may or may not be present. Since it is present in our example, the units of measure in the line call-out (and in any other documentation, such as a test report) are understood to be stated in SI (metric) units. For example, tensile strength is in megapascals (MPa). If the "M" were not present, English units would be in use. For example, tensile strength would be in pounds per square inch (psi).

C. The Grade Number defines specific added test requirements that are desirable in cases where the basic requirements do not always sufficiently ensure an acceptable material. Grade 1 indicates that only the basic requirements are compulsory; no suffix requirements are permitted. All other grades and test requirements are listed in Table 6 of the D 2000 document. In our example, the material is Grade 2.

D. The Type is based on changes in tensile strength of not more than ±30%, elongation of not more than –50%, and hardness of not more than ±15 points after heat aging for 70 hours at a given temperature. The temperatures at which these materials shall be tested for determining type are listed in ***Table 11***. In our example, the material is Type B, which corresponds to a 100° C test temperature.

Type	Test Temperature (° C)
A	70
B	100
C	125
D	150
E	175
F	200
G	225
H	250
J	275
K	300

Table 11: Basic Requirements for Establishing Type by Temperature

ASTM D 2000-95 M2BG714 B14 EA14 EF11 EF21 EO14 EO34 Z1 Z2

E. The Class is based on the material's resistance to swelling in Industry Reference Material (IRM) 903 Oil (now used in lieu of ASTM Oil Number 3, which was discontinued due to requirements by the Occupational Safety and Health Administration, or OSHA). Testing involves immersion for 70 hours at the temperature previously determined from ***Table 11*** (100° C), after which swell is calculated. Limits of swelling for each class are shown in ***Table 12***. In our example, the material is Class G, indicating a maximum swell of 40%. Be aware that ASTM Oil Number 3 and IRM 903 Oil are similar but not identical, so complete equivalency among results is not possible. For information on converting ASTM oil swell values to IRM values, refer to ASTM Emergency Standard ES 27-94. ***Table 13*** lists the D 2000 material designations (type and class) and the elastomers that are most often used for each.

Class	Volume Swell (Maximum %)
A	No requirement
B	140
C	125
D	100
E	80
F	60
G	40
H	30
J	20
K	10

Table 12: Basic Requirements for Establishing Class by Volume Swell

Material Designation	Most-Used Elastomer(s)
AA	Natural Rubber
BA	Ethylene Propylene
BC	Neoprene®
BE	Neoprene®
BF	Nitrile
BG	Nitrile, Polyurethane
BK	Nitrile
CA	Ethylene Propylene
CH	Nitrile
DA	Ethylene Propylene
DF	Polyacrylate
DH	Polyacrylate
EF	Vamac®
FC	Silicone
FE	Silicone
FK	Fluorosilicone
GE	Silicone
HK	Viton®

Table 13: Material Designations & Most-Used Elastomers

ASTM D 2000-95 M2BG714 B14 EA14 EF11 EF21 EO14 EO34 Z1 Z2

F. The next three digits (in this case, "714") specify the hardness and tensile strength. The first digit indicates Shore A durometer. For example, 7 for 70 ±5. The next two numbers indicate the minimum tensile strength. For example, 14 for 14 MPa. Remember, this will be in SI units if the letter "M" is in the call-out, and English units if not. To convert to psi, simply multiply the MPa number by 145. In this case, 14 MPa would convert to 2,030 psi.

MPa X 145 = psi

psi / 145 = MPa

G. Suffix letters and suffix numbers follow the hardness and tensile strength specifications to provide for additional testing requirements. The meaning of each suffix letter is shown in ***Table 14***. For example, the "B" of "B14" specifies a compression set test. Suffix letters are typically followed by two suffix numbers. The first number always indicates the test method, and the second indicates the test temperature. The suffix numbers are covered by Tables 4 and 5 of the D 2000 document. For example, the "1" specifies a 22-hour compression set test as detailed in D 395 (Method B) for solid test specimens, and the "4" specifies testing at 100° C. Keep in mind that in some cases, the second suffix number may be two digits, which means you might see something like "F110." F110 would indicate a 3-minute low temperature resistance test as detailed in ASTM D 2137 (Method A) and conducted at a temperature of -65° C.

A	Heat Resistance
B	Compression Set
C	Ozone or Weather Resistance
D	Compression Deflection Resistance
EA	Fluid Resistance (Aqueous)
EF	Fluid Resistance (Fuels)
EO	Fluid Resistance (Oils & Lubricants)
F	Low Temperature Resistance
G	Tear Resistance
H	Flex Resistance
J	Abrasion Resistance
K	Adhesion
M	Flammability Resistance
N	Impact Resistance
P	Staining Resistance
R	Resilience
Z	Any Special Requirement (e.g. "Resistance to Marking")

Table 14: The Meaning of Suffix Letters

That's all there is to understanding the D 2000 / J200 call-out system. It is one of the most versatile specifications in the rubber industry. In addition to helping you specify compounds, familiarity with the system will also help you make sense of material test reports. Let's take a closer look at a sample report next.

Anatomy of a Test Report

“Provided these tests mirror the anticipated service conditions, you can use them to make an informed decision regarding the compound’s suitability for your application.”

Many manufacturers provide material test reports (also known as technical reports or specification sheets) as a service to their customers. These reports typically show the performance of a given vulcanizate (cured rubber compound) when subjected to a variety of standardized ASTM tests. Provided these tests mirror the anticipated service conditions, judgments can be made as to the compound’s suitability for use in a particular application.

To help you better understand what test reports can tell you, let’s take a closer look at a sample report (see ***pages 68*** and ***69***) whose subject is the same 70 (Shore A) durometer nitrile compound we dealt with in “Understanding ASTM D 2000 / SAE J200.” As we go through the report line by line, you’ll find references to many of the most commonly used ASTM tests. Keep in mind, however, that not every report you see will (or should) cover all of these tests. We’re including them here simply to help you get better acquainted with as many tests as possible.

A. This line tells you the absolute basics: you’re looking at a report on “N470,” a nitrile compound with a durometer of 70 (Shore A).

B. The next item lists all of the ASTM specifications to which the N470 material conforms. Each of these are defined individually during the course of the report, but for now, just recall from “Understanding ASTM D 2000 / SAE J200” that each line call-out entry corresponds to a particular test. For example, “EA14” is an ASTM D 471 70-hour water resistance test conducted at 100° C.

C. “Original properties” are just that: the initial attributes of the material prior to testing. Information in this and all subsequent entries is broken into two columns: the “specification” (what is required to be acceptable) and the properties (or response) of the “N470” nitrile. There are six different original properties on this report: 1) Hardness, 2) Tensile Strength, 3) Elongation, 4) Modulus at 100%, 5) Tear

Resistance, and 6) Specific Gravity. Note that specific gravity (S.G.) is not specified on the report; rather, the S.G. of N470 (1.25) is understood relative to water's S.G. of 1.00. N470 is thus 25% heavier than water.

D. The first test on this report is "heat resistance" (also known as heat aging or air aging). Per the line call-out, our nitrile is a Grade 2 "BG" compound. This would normally send you to the D 2000 or J200 documents, where you'd turn to the "BG Materials" section of Table 6 and see data similar to that shown here in ***Table 15***. You'll see there that "A14" is the

	Suffix Requirements	Grade 1	Grade 2	Grade 3	Grade 4	Grade 5	Grade 6	Grade 7	Grade 8
A14	Heat resistance, Test Method D 573, 70 h at 100° C: • Change in hardness, max, points • Change in tensile strength, max, % • Change in ultimate elongation, max, %				 ± 5 ±15 -15	 +15 -20 -40	 +15 -20 -40		
B14	Compression set, Test Methods D 395, Method B, max, %, 22 h at 100° C		25	50	50	25	25	25	
B34	Compression set, Test Methods D 395, Method B, max, %, 22 h at 100° C		25			25	25		
C12	Resistance to ozone, Test Method D1171, quality retention rating, min, %			*	*				
EA14	Water resistance, Test Method D 471, 70 h at 100° C: • Change in hardness, points • Change in volume, %		 ±10 ±15					 ±10 ±15	
EF11	Fluid resistance, Test Method D 471, Reference Fuel A, 70 h at 23° C: • Change in hardness, points • Change in tensile strength, max, % • Change in ultimate elongation, max, % • Change in volume, %		 ±10 -25 -25 -5 to +10					 ±10 -25 -25 -5 to +10	
EF21	Fluid resistance, Test Method D 471, Reference Fuel B, 70 h at 23° C: • Change in hardness, points • Change in tensile strength, max, % • Change in ultimate elongation, max, % • Change in volume, %		 0 to -30 -60 -60 0 to +40					 0 to -30 -60 -60 0 to +40	
EO14	Fluid resistance, Test Method D 471, No. 1 Oil, 70 h at 100° C: • Change in hardness, max, points • Change in tensile strength, max, % • Change in ultimate elongation, max, % • Change in volume, %		 -5 to +10 -25 -45 -10 to +5	 -7 to +5 -20 -40 -5 to +10	 -7 to +5 -20 -40 -5 to +5	 -5 to +15 -25 -45 -10 to +5	 -5 to +15 -25 -45 -10 to +5	 -5 to +15 -25 -45 -10 to +5	
EO34	Fluid resistance, Test Method D 471, IRM 903 Oil, 70 h at 100° C: • Change in hardness, points • Change in tensile strength, max, % • Change in ultimate elongation, max, % • Change in volume, %		 -10 to +5 -45 -45 0 to +25	 -10 to +5 -35 -40 +16 to +35	 -10 to +5 -35 -40 0 to +6	 0 to -15 -45 -45 0 to +35	 0 to -20 -45 -45 0 to +35	 -10 to +5 -45 -45 0 to +25	
F16	Low-temperature brittleness, Test Methods D 2137, Method A, 9.3.2, nonbrittle after 3 min at -35° C							pass	
F17	Low-temperature brittleness, Test Methods D 2137, Method A, 9.3.2, nonbrittle after 3 min at -40° C		pass				pass		
F19	Low-temperature brittleness, Test Methods D 2137, Method A, 9.3.2, nonbrittle after 3 min at -55° C			pass	pass	pass			

* Values not yet established.

Table 15: Sample suffix and grade requirements

suffix designation for "heat resistance" as determined by ASTM D 573, a 70-hour test conducted at 100° C.

Why, then, is A14 not listed among the additional suffix requirements in this material's line call-out? It is omitted from the call-out because there are no A14 specifications for Grade 2 BG compounds. In Table 15, the Grade 2 column across from row A14 is empty, so the air aging specifications column in our sample report is blank. When there are no specifications, a material cannot be said to "conform" to a given test, and the corresponding suffix designation is not listed in the call-out. We've chosen to include "air aging" on this report because it is a common test used to gauge resistance to oxidation and thermal attack over time. You'll no doubt see it regularly on test reports, and it will likely be specified in three properties: 1) Hardness Change, 2) Tensile Change, and 3) Elongation Change.

IN THE LAB

Figure 43: Our Blue M oven and specialized test fixtures allow for compression set testing.

E. The second test is "compression set" (B14 in the line call-out) as determined by ASTM D 395, a 22-hour test conducted at 100° C. This report lists one property specification related to compression set: Percent of original deflection, which is specified at a 25% maximum. In this instance, the N470 test specimen takes a 14% set. A number of factors other than the compound itself can greatly affect compression set results, including test temperature and sample thickness.

F. The third test is "water immersion" (EA14 in the line call-out) as determined by ASTM D 471, a 70-hour test conducted at 100° C. This report lists two property specifications related to water immersion: 1) Hardness Change and 2) Volume Change.

G. The next four tests gauge fuel and oil resistance (EF11, EF21, EO14, and EO34 in the line call-out). In each case, there are four property specifications: 1) Hardness Change, 2) Tensile Change, 3) Elongation Change, and 4) Volume Change. Per J200 / D 2000, EF 11 is the suffix designation for ASTM D 471, a 70-hour test conducted at 23° C using Reference Fuel A. That's good to know, but you're probably wondering what EF11 and the other fluid resistance tests can really tell you about a compound.

Put simply, fluid resistance tests give you an indication of how the compound will react when brought in contact with fuels and oils. In most cases, the primary concern is swelling, though compound degradation is also common. Recall that volume changes (either swell or shrinkage) are typically accompanied by changes in physical properties, including hardness, tensile strength, modulus, elongation, tear resistance, and compression set.

Reference Fuel Type	Composition (Volume %)
A	Isooctane (100)
B	Isooctane (70), Toluene (30)
C	Isooctane (50), Toluene (50)
D	Isooctane (60), Toluene (40)
E	Toluene (100)
F	Diesel Fuel, Grade 2 (100)
G	Fuel D (85), anhydrous denatured ethanol (15)
H	Fuel C (85), anhydrous denatured ethanol (15)
I	Fuel C (85), anhydrous methanol (15)
K	Fuel C (15), anhydrous methanol (85)

Table 16: ASTM Reference Fuels

ASTM Reference Fuels A through K (see ***Table 16***) have been specifically selected to test compounds in contact with gasolines or diesel fuels. Which tests are called for depends on which fluid(s) the seal will encounter. For example, Reference Fuel A (used in the EF11 test) is a 100% isooctane fluid which mirrors the shrinking or low-swell effects of gasolines composed primarily of straight-chain aliphatic (rather than ringed aromatic) hydrocarbons. If the compound in question will be used around gasolines with a very high aliphatic content, then an EF11 test is a good idea. Reference Fuel B (used in the EF21 test) is a 70% isooctane-30% toluene mixture. The toluene content lends the mixture a level of aromaticity, enabling Reference Fuel B to more closely approximate the swelling effects of commercial gasolines.

The other two fluid resistance tests on this report are based on shrinking or swelling in lubricating oils rather than fuels. EO14 is the suffix designation for another ASTM D 471 test, this one lasting 70 hours and conducted at 100° C using

Number 1 Oil. EO14 is commonly used to gauge elastomer shrinkage. The time and temperature requirements for EO34 are identical to EO14, with the exception that Industry Reference Material (IRM) 903 is used rather than Number 1 Oil. EO34 is a common tool for gauging elastomer swell. As with the Reference Fuels, the choice of oils in testing is not arbitrary. Rather, Number 1 Oil and IRM 903 are used because they have an aniline point similar to the aniline point of a fluid to be found in service.

The *aniline point* is the lowest temperature at which equal volumes of aniline (an oily, colorless, and poisonous organic liquid derived from benzene) and the oil will completely dissolve in one another. The aniline point is actually a good measure of the aromatic content, or the amount of unsaturated hydrocarbons present in the oil. The higher the level of unsaturants, the more easily the organic aniline can "step in" to combine with the oil, and thus the aniline point will be low. A low aniline point translates to a higher potential for swelling certain rubber compounds.

IN THE LAB

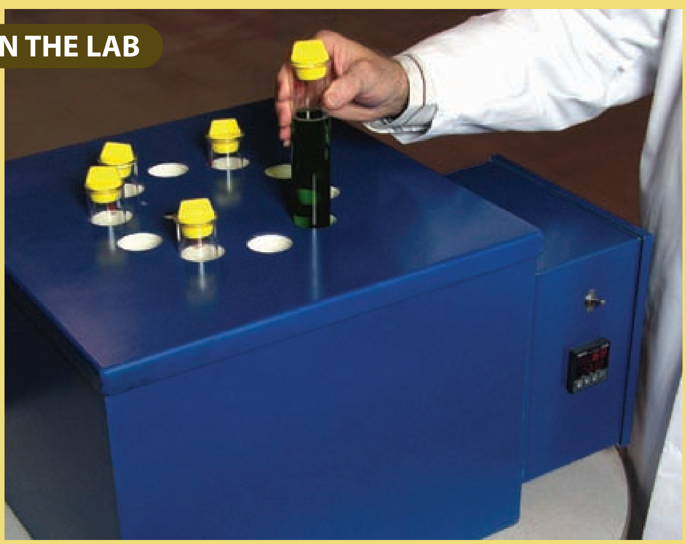


Figure 44: Our heating block provides us with precise temperature control during fluid resistance tests.

Number 1 Oil has the highest aniline point (124° C ± 1°) of the ASTM test oils, meaning it typically causes the least amount of rubber swell. As is clear by looking at the EO14 volume change specification (-10% to +5%), Number 1 Oil actually has the potential to cause more shrinkage than swell. Testing with Number 1 Oil is thus a common tool for gauging oil-induced shrinkage due to plasticizer extraction. IRM 903, on the other hand, has the lowest aniline point (70° C ± 1°) among the test oils and typically causes the greatest swell. Be aware that IRM 903 is used in lieu of the now-obsolete Number 3 Oil for EO34 testing.

H. The eighth test is "impact brittleness" (also known as low-temperature brittleness; Z1 in the line call-out). Note that this is a three-minute test conducted at -25° C. Per ASTM D 2137 (Method A), low temperature tests are normally conducted at -35° C, -40° C, or -55° C. For example, if this test had been conducted at -40° C, F17 would have been noted in the line call-out. Because this test was conducted at a non-standard temperature (-25° C), it is noted in the line call-out using a special "Z" suffix. (Per D 2000 / J200, special suffix requirements begin with a "Z" and must be specified in detail, including test methods.) Our report has one specification related to Z1, which is conducted on a pass-fail basis only: No cracks in the material after it is struck once. N470 passes this test.

On some reports, you may also see a "temperature retraction" TR-10 listing. Though TR-10 is not covered by a D 2000 suffix, ASTM D 1329 does detail TR-10 as a way to gauge a compound's crystallization and visco-elastic properties at low temperatures. In this case, specification is for the material to remain viable at -25° C. N470 passes this test. For more on TR-10 testing, see "Low Temperature Effects" on page 37.

I. The ninth test is another special stipulation required by the user of the material (Z2 in the line call-out). In our example, "Z2" is "resistance to marking." There is one specification related to this test, which is conducted on a pass-fail basis only: Non-marking by the material. That is, the compound should not leave any mark when wiped on white paper with a 0.03 MPa contact pressure. N470 passes this test.

In some instances, a Z suffix may be used for something as basic as a hardness reading, as with the specification for a 75 (Shore A) durometer fluorocarbon (Viton®). Because the line call-out system only allows three digits for both durometer and tensile strength (as with "714" indicating a 70 durometer material with a tensile strength of 14 MPa), it is not possible to specify a 75 durometer material in this way. Thus, a special Z suffix would be needed.

A.

B.

TEST REPORT

Compound: N470 Nitrile 70 Durometer

C. Conformance to: ASTM D 2000-95, M2BG714, B14, EA14, EF11, EF21, EO14, EO34, Z1, Z2

	ORIGINAL PROPERTIES	SPECIFICATION	N470
	Hardness, Durometer A	70 ± 5	70
	Tensile Strength, MPa	14 min	15.9
	Elongation, %	250 min	370
	Modulus @ 100%, MPa	11 min	12.1
D.	Tear Resistance, kN/m	20 min	35
	Specific Gravity	-	1.25
	A14 - HEAT RESISTANCE – 70 hrs @ 100° C		
	Hardness Change, Points	-	+2
E.	Tensile Change, %	-	+14
	Elongation Change, %	-	-11
F.	B14 - COMPRESSION SET – 22 hrs @ 100° C		
	% of Original Deflection	25 max	14
	EA14 - WATER IMMERSION – 70 hrs @ 100° C		
G.	Hardness Change, Points	± 10	-2
	Volume Change, %	± 15	+3
	EF11 - ASTM FUEL A – 70 hrs @ 23° C		
	Hardness Change, Points	± 10	-2

Tensile Change, %	-25 max	-10	
Elongation Change, %	-25 max	-10	
Volume Change, %	-5 to +10	+1	G.

	SPECIFICATION	N470	
EF21 - ASTM FUEL B – 70 hrs @ 23° C			
Hardness Change, Points	0 to -30	-10	
Tensile Change, %	-60 max	-56	
Elongation Change, %	-60 max	-45	
Volume Change, %	0 to +40	+26	
EO14 - ASTM OIL # 1 – 70 hrs @ 100° C			
Hardness Change, Points	-5 to +10	+2	
Tensile Change, %	-25 max	+21	
Elongation Change, %	-45 max	-2	
Volume Change, %	-10 to +5	-5	
EO34 - IRM903 OIL – 70 hrs @ 100° C			
Hardness Change, Points	-10 to +5	-4	
Tensile Change, %	-45 max	+7	
Elongation Change, %	-45 max	-3	
Volume Change, %	0 to +25	+7	H.
Z1 - LOW TEMPERATURE – 3 mins @ -25° C			
Impact Brittleness	No cracks	Pass	I.
TR-10	-25° C	-25° C	
Z2 - RESISTANCE TO MARKING			
Marking by the material on white paper when wiped at 0.03 MPa contact pressure	Non-marking	Pass	

Batch Testing.

"Batch testing is vital in ensuring consistency among finished parts."

When one of our suppliers mixes or buys a batch of rubber, a batch number is automatically assigned. But before it can be molded into usable parts (such as shaft seals), the batch must be tested to ensure that it is a "good batch," i.e. its physical properties meet specifications. Batch testing is vital in ensuring consistency among finished parts.

To test a batch of rubber's physical properties, a sample of the material is molded into 6" x 6" x .070" slabs. These slabs are then cut into the various shapes needed to test for hardness, tensile strength, modulus, elongation, and compression set. All of these tests are described in the Physical Properties section of "Selecting the Lip Material."

Specific gravity is also often measured, though more as a check on compounding consistency than as a physical test. Per ASTM D 792, measurement of specific gravity (or relative density) compares the weight of a molded sample to the weight of an equal volume of water. Specific gravity (SG) is noted without units. If a material is twice as heavy as water, its specific gravity is 2. Using the specific gravities of previously-molded compounds for comparison (e.g. ethylene propylene might have a specific gravity of 0.86, or less than that of water), a manufacturer can see if a sample is consistent with prior batches.

If the tested physical properties of a batch of rubber meet all specifications, the batch is approved for production of shaft seals or other articles. If the properties are not satisfactory, the batch must either be reworked (broken down and reformulated) or scrapped. Scrapping an entire batch of rubber and starting over can be very costly and is thus a last resort. But even if the compound's physical properties are acceptable, it must still meet processing requirements in order to be ready for use in a specific molding facility.

VISCOMETERS

At one time, the Mooney Viscometer was the most common tool used to determine the processing characteristics for a given batch of rubber. Many compounders still use the

Mooney to verify viscosity (which is indicative of molecular weight) when obtaining raw polymer stock. This works because a sample's resistance to being moved by the Mooney's internal rotor is directly linked to its viscosity. The viscometer is also helpful in determining scorch time, or how long a material can be exposed to heat before it starts to cure. The viscometer's role as the chief indicator of processing traits, however, was usurped by the rheometer.

RHEOMETERS

There are two main types of rheometers currently in use: the ODR and the MDR. The older of these, the Oscillating Disk Rheometer (ODR), builds on the Mooney Viscometer's rotor-based design. An ODR gauges the amount of torque (twisting force in pounds per inch, lb./in, or decinewtons per meter, dN/m) needed to oscillate a rotor within the rubber sample. Whereas a viscometer rotor relies on full rotation, the ODR rotor only moves back and forth across a small arc.
This oscillation is less degrading to the material than in the viscometer, where destruction of the sample is typical.

ODR test results are also more reflective of actual cure conditions because constant high pressure and the desired vulcanization temperature are maintained on the sample. As testing progresses, the sample begins to behave in predictable ways. Viscosity briefly drops as the sample first heats up, but the chemical reaction soon starts. The rubber becomes more viscous due to crosslinking of the macromolecular chains. As a result, the amount of torque that is required to internally shear (deform) the sample increases. Using this increasing torque as a gauge, the ODR plots a *cure curve* (see ***Table 17**, next page*) illustrating the state of cure for a given time and temperature.

Though the Monsanto ODR was for many years the most-used rheometer, a more recent development is the Moving Die Rheometer (MDR, see ***Figure 45***). Whereas the ODR uses an embedded rotor to torque the rubber sample, an MDR holds the sample between a pair of heated dies (metal plates forming a cavity). As one of the dies moves across a small arc, the other die gauges the reaction torque generated in the sample. This again results in a cure curve that can show the optimum cure

MDR

Figure 45: Moving Die Rheometer

time for the desired blend of properties. Since the MDR does not insert a rotor into the sample, many molders feel the MDR is less intrusive to the curing process and thus more objective and accurate than the ODR.

RUBBER PROCESS ANALYZERS

More recently, rheometers have begun to be supplanted by the rubber process analyzer (RPA). Whereas viscometers measure properties before cure, and rheometers gauge properties both before and during cure, the RPA is designed to measure properties of polymers and rubber compounds before, during, *and* after curing. Processability is a key parameter gauged by the RPA, along with cure characteristics and final cured properties.

DETERMINING CURE TIMES

Cure characteristics are typically reflected using what is known as a "cure curve." As shown in Table 17, a cure curve is essentially "torque versus time (at a given temperature)." The torque value is a direct indication of the sample's shear modulus (resistance to shearing deformation). A number of processing characteristics can also be read, including the minimum pressure needed to make the material flow properly into the mold cavity, *scorch time* (prior to vulcanization), *optimum cure time* (typically 85 to 95% of maximum cure), and *maximum cure* (prior to over cure). Keeping the initial cure slightly below the maximum helps

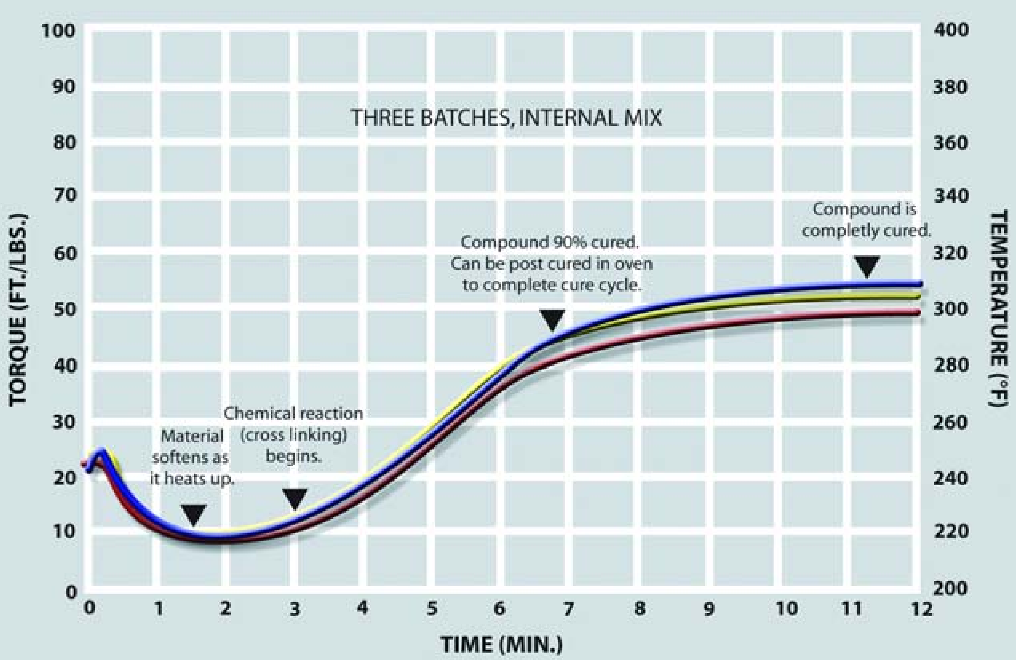


Table 17: ODR-Plotted Cure Curve

avoid over cure by allowing leeway for any necessary post cure (controlled continuation of vulcanization to finish cure, drive off byproducts, and stabilize) or inadvertent *after cure* (uncontrolled continuation of vulcanization after heat is removed).

Though specific vulcanization questions can be answered via a cure curve, rheometers also help molders address more general concerns about processibility and consistency. No matter what the cure curve says, "optimum" cure time is a matter of economics and logistics. There is no "universal" cure time for a given compound. A batch of rubber may have different cure times if given to different manufacturers, depending on their capabilities. Ultimately, the cure time for a specific molded part is based upon the design of that part.

The old adage about time being money is especially true when it comes to *cycle time* (the time between a given point in one molding cycle and the same point in the next cycle; e.g. loading of raw stock, through molding and unloading of finished parts, then to reloading; see ***Figure 46***). Generally speaking, the longer the cycle time, the more expensive the process and the more costly the part. As a cost-cutting measure, manufacturers may increase mold temperature to decrease cure time. A 20° F boost can cut cure time in half, but this is not always advantageous. Sometimes the ratio of the time the mold is open (for unloading and reloading) to the time the mold is closed and in the press allows the mold temperature to dip below what is needed for full vulcanization. Partially-vulcanized, unusable parts can result.

GOING TO PRESS

Figure 46: Multiple cavity mold in use.

Again, consistency among different batches of the same material is always a concern. The cure curve can serve as a "fingerprint" for a given batch of rubber. By comparing cure curves, it is possible to see if the properties present in one batch are present in another. Because wasted processing can be costly in terms of both time and money, compounding errors are much more economically spotted in batch testing than in subsequent stages of quality control, such as vulcanizate (cured rubber compound) testing.

selecting the lip material

Aging & Shelf Life.

As they age, shaft seals and other rubber products can undergo changes in physical properties. They may even become unusable due to excessive hardening, softening, cracking, crazing, or other surface degradations. These changes may be the result of a single factor or a combination of factors, such as the action of oxygen, ozone, light, heat, humidity, oils, water, or other solvents. Detrimental effects can be minimized, however, by proper storage conditions (such as those outlined below). As shown in Figure 47, we make it a priority here at R.L. Hudson & Company to maintain these proper conditions in all of our warehouse facilities.

"Detrimental effects can be minimized by proper storage conditions."

TEMPERATURE

The optimum storage temperature is between 40° F and 80° F. High temperatures accelerate the deterioration of rubber products. Heat sources in storage rooms should be arranged so that the temperature of stored items never exceeds 120° F. Low temperature effects are neither as damaging nor as permanent, but rubber articles will stiffen. Care should be taken to avoid distorting them at temperatures below 30° F.

HUMIDITY

Expressed as a percentage, *relative humidity* is the ratio of the amount of water vapor present in the air to the greatest amount that could be present at a given temperature. Ideally, the relative humidity in the storage area should be below 75%. Very moist or very dry conditions must be avoided. Where ventilation is necessary, it should be kept to a minimum. Condensation cannot be allowed to occur. Some materials, such as polyester-based polyurethanes, are hygroscopic; they absorb moisture from the air. This moisture attacks the polymer's chemical backbone, resulting in *chain scission* (division of the polymer chain into smaller, weaker segments). Over time, the material becomes soft and cheesy. In humid environments, this can happen in a matter of mere weeks unless precautions are taken.

STORAGE

Figure 47: Our warehouse is designed to provide proper storage conditions.

LIGHT

Shaft seals and other rubber products should always be protected from light, especially natural sunlight. Strong artificial light with a high ultraviolet (UV) content is also dangerous. Regardless of the source, UV rays can cause chain scission. Use of polyethylene bags stored inside large cardboard containers is recommended. Polyethylene-lined craft bags can also offer good protection.

OXYGEN & OZONE

Both oxygen (O_2) and ozone (O_3) are very damaging to rubber products. Whenever possible, shaft seals and other molded articles should be stored in *hermetic* (airtight) containers to protect them from circulating air. Oxygen (especially in combination with heat) causes rubber articles to form additional cross-links, leading to unwanted hardening of the seal. As with water and UV light, ozone is capable of causing chain scission. Rubber products should be kept away from ozone generators such as electric motors, mercury vapor lamps, welding equipment, and high voltage electrical equipment.

DEFORMATION

Rubber articles should be stored in a relaxed condition, free from tension, compression, or other deformation, all of which can cause cracking or permanent shape change. Shaft seals should not be stored on pegs or wires.

SHELF LIFE

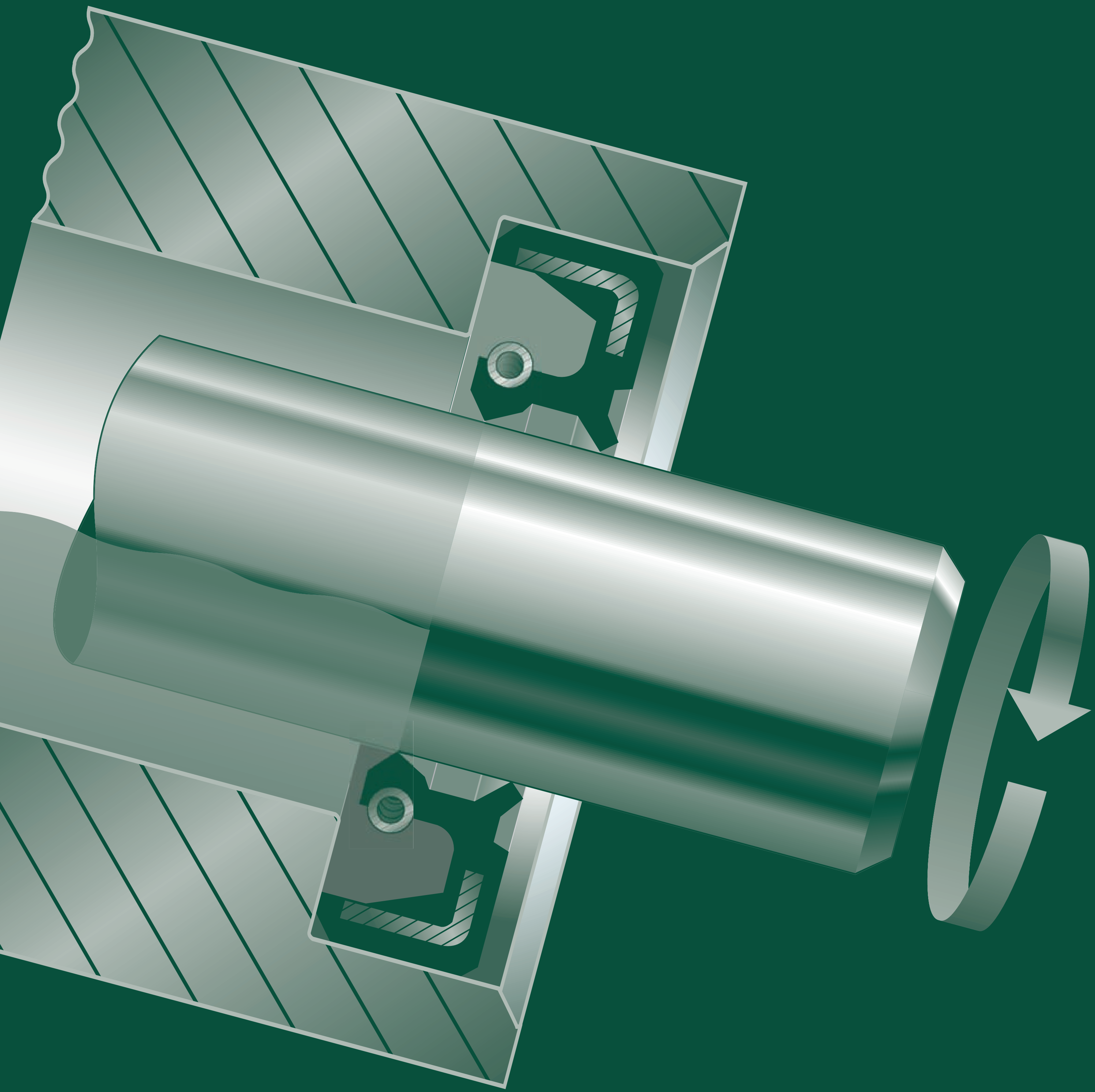
In normal warehouse conditions, the shelf life of even relatively age-sensitive elastomers is considerable. This is largely due to advances in compounding. ***Table 18*** lists the generally recommended limitations of many materials. Taken from Military Handbook 695, this table is quite conservative.

Type	ASTM Desig.	Shelf Life
Nitrile (Buna N)	NBR	3 to 5 years
Styrene Butadiene (Buna S)	SBR	3 to 5 years
Polybutadiene	BR	3 to 5 years
Polyisoprene	NR, IR	3 to 5 years
Hypalon®	CSM	5 to 10 years
Ethylene Propylene	EPDM, EPM	5 to 10 years
Neoprene®	CR	5 to 10 years
Polyurethane (polyether)	EU	5 to 10 years
Epichlorohydrin	CO, ECO	5 to 10 years
Fluoroelastomer (Viton®)	FKM	up to 20 years
Perfluoroelastomer	FFKM	up to 20 years
Silicone	Q	up to 20 years
Fluorosilicone	FVMQ	up to 20 years
Polyacrylate	ACM, ANM	up to 20 years
Polysulfide	T	up to 20 years

Table 18: Shelf Life of Common Elastomers

STORAGE

All shaft seals inventoried and shipped by R.L. Hudson & Company are stored in either zipper-locking or heat-sealed plastic bags.



designing the seal

Anatomy of a Shaft Seal.

The cross-sections of typical shaft seals are made up of many variable features. Being familiar with each of these features and understanding the roles they play in effecting a successful seal should be of paramount importance to anyone designing a shaft seal. Figures 48 and 49 illustrate many of these variable features.

THE SEALING LIP

The most important design feature of the seal is the elastomeric sealing lip. The *beam length* is the axial distance from the thinnest portion of the lip (the *flex thickness*) to the point at which the lip contacts the shaft. For a given flex thickness, a short lip exerts more force on the shaft (with a corresponding increase in friction and wear) than a long lip. A short lip also has better resistance to deformation caused by high pressure than a long lip. A longer lip (with the same flex thickness) exerts less force on the shaft, thus reducing friction and wear. A longer lip is also more flexible and can thus more easily follow any shaft eccentricities, such as shaft-to-bore misalignment (STBM) or dynamic runout (DRO). Increasing the flex thickness while maintaining the same lip length will increase force on the shaft; decreasing the flex thickness decreases force on the shaft.

"The cross-sections of typical shaft seals are made up of many variable features."

RUBBER O.D. SEAL

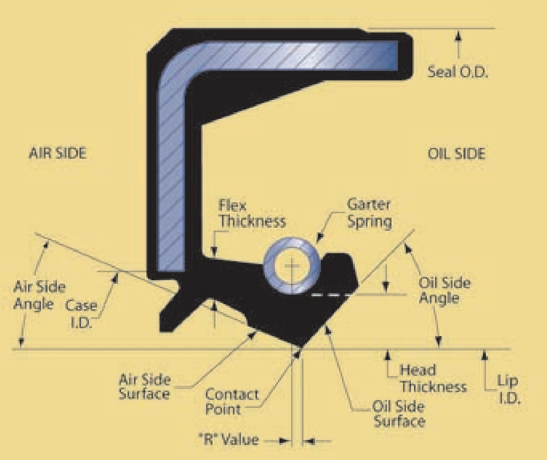


Figure 48: Shaft Seal Features

METAL O.D. SEAL

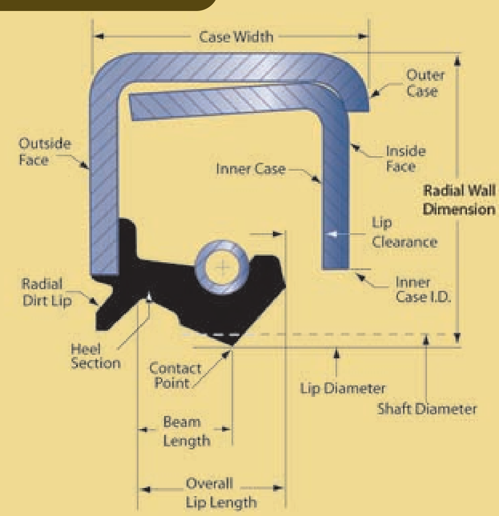


Figure 49: Shaft Seal Features

Two other important lip-related variables are the angles that meet at the head of the lip (portion nearest the shaft) to form the contact point. The angle facing the fluid being sealed is known as the *oil side* (or *scraper*) *angle*. The angle facing away from the fluid being sealed is known as the *air side* (or *barrel*) *angle*. To prevent leakage, the oil side angle must always be greater (steeper) than the air side angle.

In order to ensure contact between the lip and the shaft, the lip must always be designed to have a smaller inside diameter (I.D.) than the diameter of the shaft. The difference between the shaft diameter and the seal lip I.D. is known as *interference*. Increasing the interference (e.g. making the lip I.D. even smaller relative to a given shaft diameter) increases the amount of force on the shaft, thus also increasing friction and wear. Decreasing the interference (e.g. enlarging the lip I.D. such that it is closer to the diameter of the shaft) reduces the force on the shaft but also reduces the lip's ability to follow shaft dynamics.

THE SPRING

Lip interference is often augmented through use of a *garter spring*. A garter spring is a helically coiled spring formed into a ring (see ***Figure 50***). If present, the garter spring rests in a radiused groove molded into the head section of the lip. Seals without springs are common in applications in which the fluid being sealed has relatively high viscosity (such as grease). Because thick fluids don't flow very readily (and thus require a fairly large leak path to be problematic), a sealing lip without a spring will generally suffice. If the fluid is thin, however, it can flow more quickly though a much tinier space. A spring-loaded lip may be needed to make sure less viscous fluids such as water and oil don't escape.

SPRING DIMENSIONS

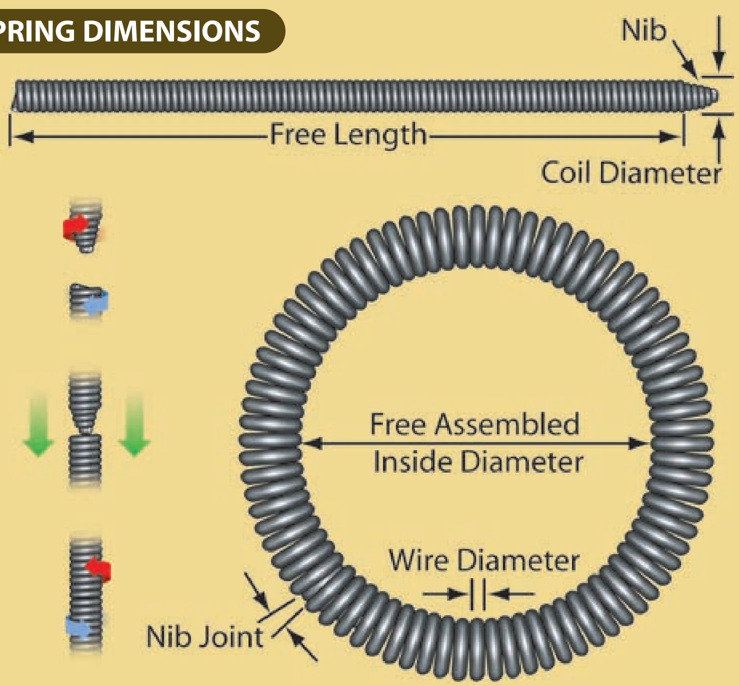


Figure 50: Garter springs are formed into a ring for insertion into the lip of a shaft seal.

The axial distance between the centerline of the garter spring and the contact point is known as the *R value*. A positive R value means the spring centerline is located toward the air side of the seal relative to the contact point, and this is desirable. A negative R value means the spring centerline is located toward the fluid side, which will result in immediate leakage if the R value is still negative after the seal has been installed on the shaft.

A spring fulfills two main functions. First, it contributes to the total radial sealing force, or *load*, between the lip and the shaft. (Load is also based on the sealing lip's inherent "beam force," as well as the "hoop force" generated when the lip is stretched outward slightly during installation onto the shaft.) Second, the spring also helps make sure the desired amount of load is maintained even when the lip material itself might swell and soften due to, for example, chemical exposure at high temperatures. An example of radial load loss is shown in ***Table 19***. (Note: 1 N/mm = 5.71 lb/in)

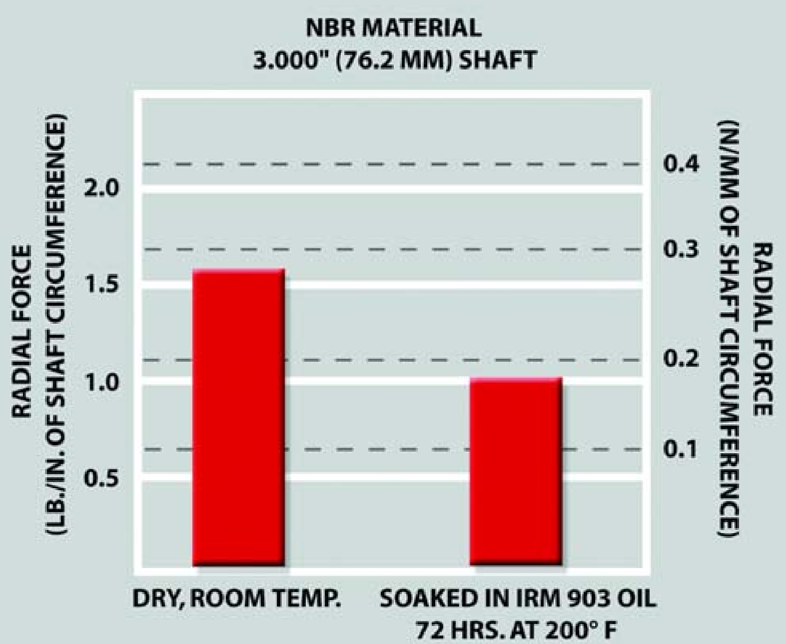


Table 19: Loss of Radial Load Due To Chemical Exposure

A lip that has swollen away from the shaft is less able to maintain consistent contact with the shaft without the aid of a spring. Inconsistent contact makes the development of a leak path likely. The spring artificially stiffens the lip, and this helps hold the lip in place. Seals without springs will leak sooner than seals with springs. For example, consider an NBR seal for a 76.2 mm (3.000 inch) shaft operating at 2165 RPM in SAE 30 engine oil at a sump temperature of 107° C (225° F). Without a spring the average life of the seal is 480 hours; the life more than doubles to almost 1000 hours when a spring with an eight-ounce tension (2.224 N) is added to the seal.

The standard spring material is hard-drawn carbon steel wire, an economically priced, general-purpose material. This material is typically designated using a four-digit number, such as 1070, that corresponds to a system developed by the Society of Automotive Engineers (SAE). This system, which mirrors the system of the American Iron and Steel Institute (AISI), assigns a four- or five-digit number to each type of steel. This number is based on the differing levels of carbon and other elements present in the steel. The first digit denotes the primary alloying element (such as a "1" for plain carbon). The second digit indicates the presence of other elements. The last two digits specify the amount of carbon in the steel (in hundredths of a percent).

For example, a designation of 1070 indicates plain carbon steel (1) with no alloying elements (0) and a 0.70 percent carbon content (70). This may not seem like much carbon, but a little bit goes a long way in adding toughness to a material. High carbon steels (such as are used for hammers and chisels) might contain up to 0.95 percent carbon, but that's still less than 1 percent! The standards for wire used in the production of shaft seal springs call for a carbon content in the 0.50 percent to 0.95 percent range (SAE 1050 to 1095, AISI C1050 to C1095).

It is recommended that the garter springs be heat treated, particularly if the spring will be exposed to temperatures of 100° C (212° F) or higher in service. Wire for use in seals that will face exposure to the elements are typically treated with a rust preventative. In highly corrosive environments, stainless steel wire (SAE 30302 to 30304, AISI 302 to 304) may be needed. In some cases, this stainless steel may be treated using a nitric acid solution (a process known as *passivation*) to further reduce the chemical reactivity of the metal. Springs incorporated into shaft seals that will be used around food or water are required by the Food and Drug Administration (FDA) to be made of stainless steel to prevent development of–and contamination by–rust. Stainless steel wire is more expensive than carbon steel wire, thus adding to the overall cost of the shaft seal.

Wire Diameter (W_d) (mm)	Tolerances (mm)	Wire Diameter (W_d) (inch)	Tolerances (inch)
0.15 - 0.25	± 0.008	0.006 - 0.010	± 0.0003
0.28 - 0.38	± 0.010	0.011 - 0.015	± 0.0004
0.41 - 0.48	± 0.013	0.016 - 0.019	± 0.0005
0.50 - 0.69	± 0.015	0.020 - 0.027	± 0.0006
0.71 - 0.86	± 0.018	0.028 - 0.034	± 0.0007

Table 20: Wire Diameter Tolerances

Whether made of carbon steel or stainless steel, garter springs are produced in three stages. First, reels of metal wire are coiled and cut to length to produce straight springs. The tolerances for the wire diameter are shown in ***Table 20***.

One end of each spring is tapered during the coiling operation to form a nib. The ends of each straight spring are joined together by back-winding each end of the spring, then inserting the nib end into the open end and screwing them together, creating what is called a *nib joint*. (Back-winding is required to eliminate a twisting tension generated when the ends are joined together (see ***Figure 50, page 78***). If the ends have not been back-wound properly, the spring may twist into a figure eight shape after assembly.) Circular springs with a specific assembled inner diameter (AID) result. The tolerances for the AID are shown in ***Table 21***.

The manufacturing process creates an *initial tension* (N) in the spring by backwinding the coils so that force is required to pull them apart. *Spring rate* (N/m) is defined as the force required to elongate (deflect) a spring a given amount. Initial tension and spring rate combine to determine the total

Wire Diameter (W_d) (mm)	AID (mm)	Wire Diameter (W_d) (inch)	AID (inch)
0.15 - 0.28	± 0.20	0.006 - 0.011	± 0.008
0.30 - 0.48	± 0.30	0.012 - 0.019	± 0.012
0.50 - 0.76	± 0.40	0.020 - 0.030	± 0.015
0.80 - 1.40	± 0.50	0.031 - 0.055	± 0.020

Table 21: Assembled Inner Diameter (AID) Tolerances

tension (or *spring load*) generated by a given spring. The result can be shown as a spring load versus deflection chart (see ***Table 22***). It is recommended that the initial tension be 50 to 80% of the total spring load at the working spring deflection. The tolerance for total spring load is the greater of ± 0.14 N (± 0.5 oz) or ± 20% of the nominal load at the desired working deflection.

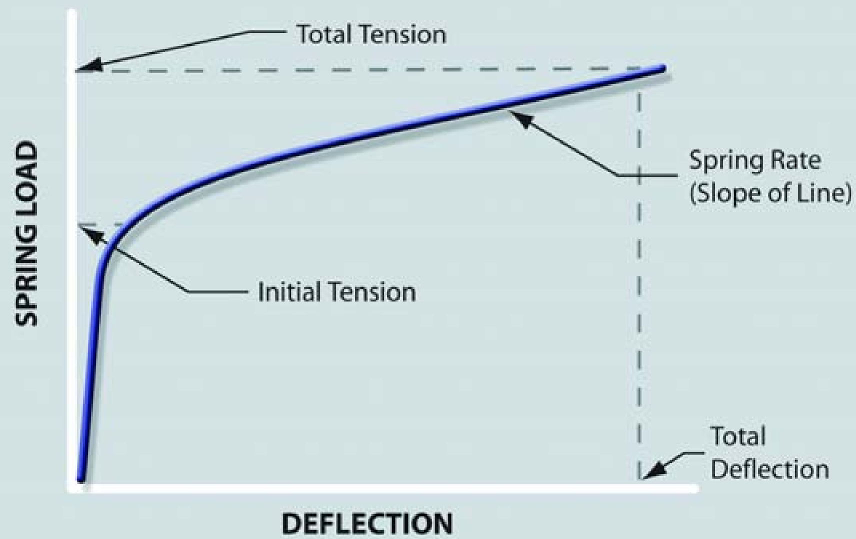


Table 22: Spring Load Versus Deflection

A shaft seal lip without a spring is said to be unsprung; thus, the amount of interference between an unsprung lip and a shaft is known as *unsprung interference*. With a spring in place and the seal installed, the interference between the lip and shaft is known as *sprung interference*.

Because the springs exert inward tension on the lips, the inner diameter of the springs determines the final diameter of the sealing lip. When the seal is installed, the spring stretches and contributes to the amount of load applied to the shaft. As shown in ***Table 23***, the tolerances for lip I.D. are dependent on shaft diameter.

A SECONDARY LIP

In addition to the primary sealing lip, many designs also incorporate a smaller, *secondary lip* to exclude dust, dirt, and other contaminants. Unlike the primary lip, this secondary lip typically faces the application's air side (since dirt and other unwanted matter may try to migrate in from outside the assembly). If present, a secondary lip generally originates away from the primary lip, at the opposite end of the

Shaft Diameter (D_S) (mm)	Primary Lip I.D. (mm)	Shaft Diameter (D_S) (inch)	Primary Lip I.D. (inch)
$D_S \leq 75$	± 0.50	$D_S \leq 3.000$	± 0.020
$75 < D_S \leq 150$	± 0.65	$3.000 < D_S \leq 6.000$	± 0.025
$150 < D_S \leq 250$	± 0.75	$6.000 < D_S \leq 10.000$	± 0.030

Table 23: Primary Lip I.D. Tolerances

elastomeric beam (in what is known as the heel, rather than the head). Depending on the needs of the application, a secondary lip can be oriented either radially (facing the shaft; known as a *radial dirt lip*) or axially (facing away from the shaft; an *axial dirt lip*). An axial dirt lip will require a vertical component against which to seal.

THE CASE

In most shaft seals, the elastomeric portion is chemically bonded to a stamped metal case (also known as a shell). Non-elastomeric members (made of materials such as PTFE, which is more difficult to bond) may be mechanically clamped in place inside the case (such as through the use of metal spacers).

Either way, the case does two things for the seal. First, it provides stability, allowing the outside diameter (seal O.D.) to pressfit snugly into a housing bore. Second, the case also provides protection, preventing damage to the lip during installation. The total axial width of the case is the case width (also known as the seal width). Tolerances for seal width are shown in ***Table 24.***

Depending on the needs of the application, a variety of different case configurations are possible. The most common

Seal Width (W_S) (mm)	Tolerance (mm)	Seal Width (W_S) (inch)	Tolerance (inch)
$W_S < 10$	± 0.3	$W_S < 0.400$	± 0.015
$W_S > 10$	± 0.4	$W_S > 0.400$	± 0.020

Table 24: Seal Width Tolerances

and least expensive design is the *L-cup case* (see ***Figure 51***). Applications requiring added seal strength may utilize an *inner cup* (see ***Figure 52***) inserted into the outer case. This inner cup protects both the lip and the spring during handling and installation. Applications in which there is large clearance between the shaft and the bore often utilize a *stepped case* (see ***Figure 53***). This type of design also offsets the lip if there are parts of the assembly that might otherwise interfere. A *reverse channel case* (see ***Figure 54***) also helps bridge large shaft-to-bore gaps. Case strength is increased, and a pry-out flange can help with removal. Strength and ease of removal are also helped by a *shotgun case* (see ***Figure 55***), though this design is costly to make and may trap drawing oils during the metal forming operation. These oils may bleed out and cause bond problems during molding.

This case is typically stamped from strip steel stock using a stamping press. The standard case material is mild carbon steel, an economically priced, general-purpose material. This material is typically designated using a four-digit number, such as 1005, that corresponds to a system developed by the Society of Automotive Engineers (SAE). This system, which mirrors the system of the American Iron and Steel Institute (AISI), assigns a four- or five-digit number to each type of steel. This number is based on the differing levels of carbon and other elements present in the steel. The first digit denotes the primary alloying element (such as a "1" for plain carbon). The second digit indicates the presence of other elements. The last two digits specify the amount of carbon in the steel (in hundredths of a percent). For example, a designation of 1005 indicates plain carbon steel (1) with no alloying elements (0) and a 0.05 percent carbon content (05). Most shaft seal cases are formed from steel with carbon

CASE CONFIGURATIONS

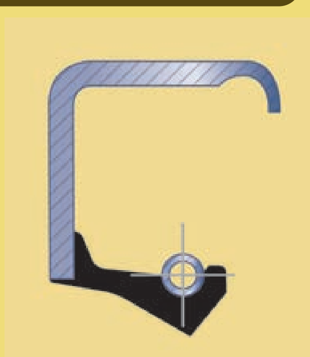


Figure 51: L-Cup Case

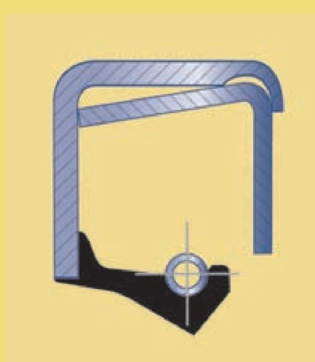


Figure 52: Inner Cup

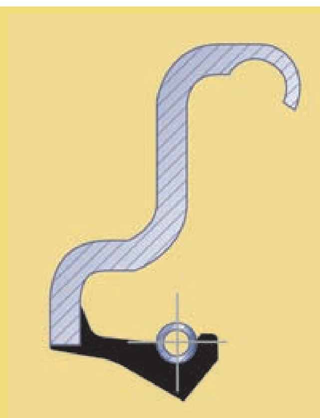


Figure 53: Stepped Case

CASE CONFIGURATIONS

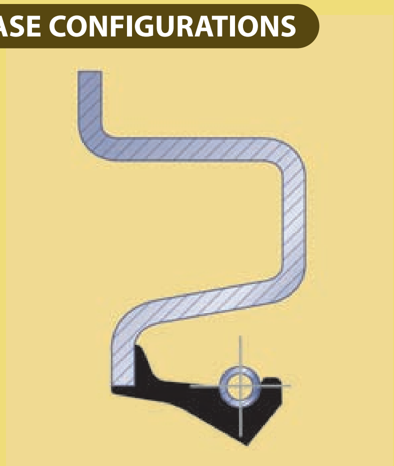


Figure 54: Reverse Channel Case

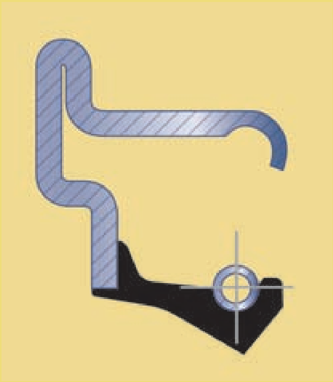


Figure 55: Shotgun Case

content in the 0.05 percent to 0.20 percent range (SAE 1005 to 1020, AISI C1005 to C1020).

Once the case has been stamped, it is often coated with zinc phosphate to protect against corrosion and to provide an uneven surface to which an adhesive will adhere. This adhesive will facilitate bonding between the metal case and the elastomeric lip during the subsequent molding operation. Grit blasting is also used to prepare the surface for bonding; however, care must be taken because grit blasting can affect the dimensions of the metal case. Once applied, the adhesive is "set" by running the cases through an oven prior to sending them to the production line for molding.

In many applications, the outside of the case is exposed to the elements. Steel cases that have been zinc phosphated and coated with cement will resist these elements in most applications. In highly corrosive environments, a stainless steel case (SAE 30302 to 30304, AISI 302 to 304) may be needed. Shaft seals for use around food or water are also required by the Food and Drug Administration (FDA) to have stainless steel cases so as to prevent development of—and contamination by—rust. Stainless steel cases are more expensive than carbon steel cases. Sometimes the entire metal component is completely encased in rubber to prevent corrosion.

A properly designed seal featuring an undamaged metal case will not leak if installed into a steel housing bore meeting material and finish specifications. However, O.D. leakage can result from use of a seal with a damaged case or from installation into a bore with surfaces that don't meet specifications. Axial scratches or corrosion on the bore can be particularly troublesome. Differential thermal expansion (due to unlike bore and case materials) can also cause a leak path to develop around the seal O.D. Depending on the specific application, it may be necessary to alter the seal O.D. in order to prevent leakage. For shaft seal O.D.s, there are three basic categories: metal O.D. seals, rubber O.D. seals, and seals in which the O.D. is a combination of metal and rubber.

Metal O.D. seals are economical and well suited for a variety of standard uses, including non-pressure fluid sealing and grease sealing. Metal O.D. seals have proven very effective when placed in steel and cast iron housings. Metal O.D. seals may be treated in various ways to further improve their performance. The entire metal case is usually coated with an adhesive used to bond the seal's elastomeric member to the case. This coating makes the seal O.D. resistant to corrosion and also assists retention of the seal in the housing bore. **Figure 56** shows an example of a metal O.D. seal with a coating of adhesive.

Other possibilities include spraying the O.D. with a polyurethane-based bore sealant to a thickness of .001" to .003". Bore sealants applied to metal O.D. seals are useful when there are (at most) minute scratches or marks on the bore surface. Deep scratches will necessitate use of a secondary adhesive such as Permatex®. The seal shown in ***Figure 57*** is a good example of a metal O.D. seal with a coating of bore sealant.

A third option is to grind the metal case O.D. This process

ADHESIVE COATING

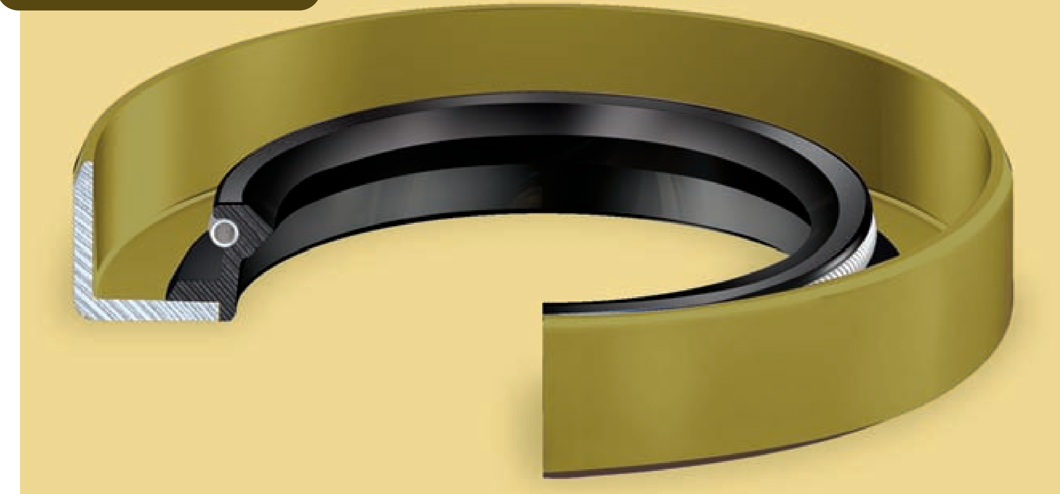


Figure 56: Metal O.D. Seal Coated With Adhesive

Figure 57: Metal O.D. Seal Coated With Bore Sealant

provides a straight wall with a very accurate outside dimension. The seal will have uniform retention strength after installation. If pressed into a bore with a good surface finish (80 to 100 Ra), a precision ground O.D. can be very effective. If the bore surface is rough, however, a secondary adhesive / sealant will be needed. ***Figure 58*** (next page) shows a seal with a precision ground O.D.

The O.D. specifications for metal O.D. seals are shown in ***Table 25*** (next page). The nominal pressfit is the difference between the seal O.D. nominal dimension and the bore I.D. nominal dimension. Maximum out-of-round (O.O.R.) for the seal O.D. is the maximum deviation from a perfect circle.

Rubber O.D. seals are often used in applications where metal O.D.s will not work. For example, what if the housing in your application is aluminum rather than steel? A metal O.D. seal won't be your best bet. The reason: differential thermal expansion of the metals in use. When heated, aluminum expands at roughly twice the rate of steel. Progressive expansion as a result of thermal cycling will decrease the *interference* (retention force) between a steel O.D. and an aluminum bore. Less retention force means the seal will be allowed to "walk" (move) within the housing. Leakage becomes a possibility. In such an instance, you may benefit from a rubberized coating on the seal O.D. The seal shown in ***Figure 59*** (page 89) is a good example of a rubber coated O.D. seal.

At .010" to .050" thick, this rubber coating encapsulates the seal's metal case and ensures good contact between the O.D.

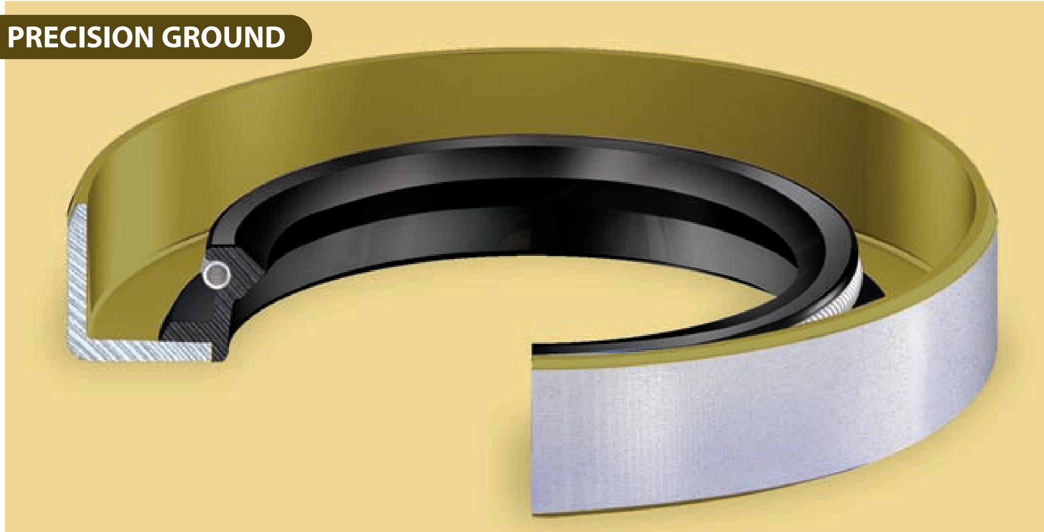


Figure 58: Metal O.D. Seal With Precision Ground Finish

and the bore. In actuality, a rubber O.D. allows for a higher pressfit than a metal O.D. does, and less force is exerted on the housing. In addition, the rubber coating is capable of maintaining a tighter, more "reactive" fit during thermal expansion (and later contraction) of the aluminum housing. Rubber O.D. seals are also good in corrosive environments; the rubber coating shields the metal case. ***Figure 60*** shows the distribution pattern of the contact pressure between a rubber O.D. and a housing bore.

Though the rubber O.D.s shown in ***Figures 59*** and ***60*** are straight, it is also possible to mold in small, round ribs along

Seal O.D. (D_O) (mm)	Seal O.D. Tolerance (mm)	Nominal Pressfit (mm)	Diametrical Max. Out of Round (O.O.R.) (mm)	Seal O.D. (D_O) (inch)	Seal O.D. Tolerance (inch)	Nominal Pressfit (inch)	Diametrical Max. Out of Round (O.O.R.) (inch)
$D_O \leq 50$	+ 0.20 + 0.08	0.12	0.18	$D_O \leq 2$	± 0.002	0.005	0.007
$50 < D_O \leq 80$	+ 0.23 + 0.09	0.14	0.25	$2 < D_O \leq 3$	± 0.0025	0.0055	0.010
$80 < D_O \leq 120$	+ 0.25 + 0.10	0.15	0.30	$3 < D_O \leq 5$	± 0.003	0.0065	0.012
$120 < D_O \leq 180$	+ 0.28 + 0.12	0.17	0.40	$5 < D_O \leq 7$	± 0.003	0.007	0.016
$180 < D_O \leq 300$	+ 0.35 + 0.15	0.21	0.25% of D_O	$7 < D_O \leq 12$	± 0.0035	0.0085	*
$300 < D_O \leq 440$	+ 0.45 + 0.20	0.28	0.25% of D_O	$12 < D_O \leq 20$	± 0.005	0.012	*
				$20 < D_O \leq 40$	± 0.005	0.013	*
				$40 < D_O \leq 60$	± 0.006	0.016	*

* 0.0025 inch per inch of seal O.D. (D_O)

Table 25: O.D. Specifications For Metal O.D. Seals

Figure 59: Flat Rubber Coating on the Seal O.D.

the seal O.D. These ribs can be advantageous because they provide high point-of-contact unit loading to increase sealability and retention. An example of a seal with a ribbed O.D. is shown in ***Figure 61*** (next page). The distribution pattern of the contact pressure between a ribbed rubber O.D. and a housing bore is shown in ***Figure 62*** (next page).

Though it offers many advantages, a rubber coated O.D. seal does have drawbacks. The rubber portion can be damaged during installation if proper lead-in chamfers are not built into the design. Care must also be taken due to a phenomenon known as *springback*. Springback is the tendency of a shaft seal with a rubber O.D. to unseat itself

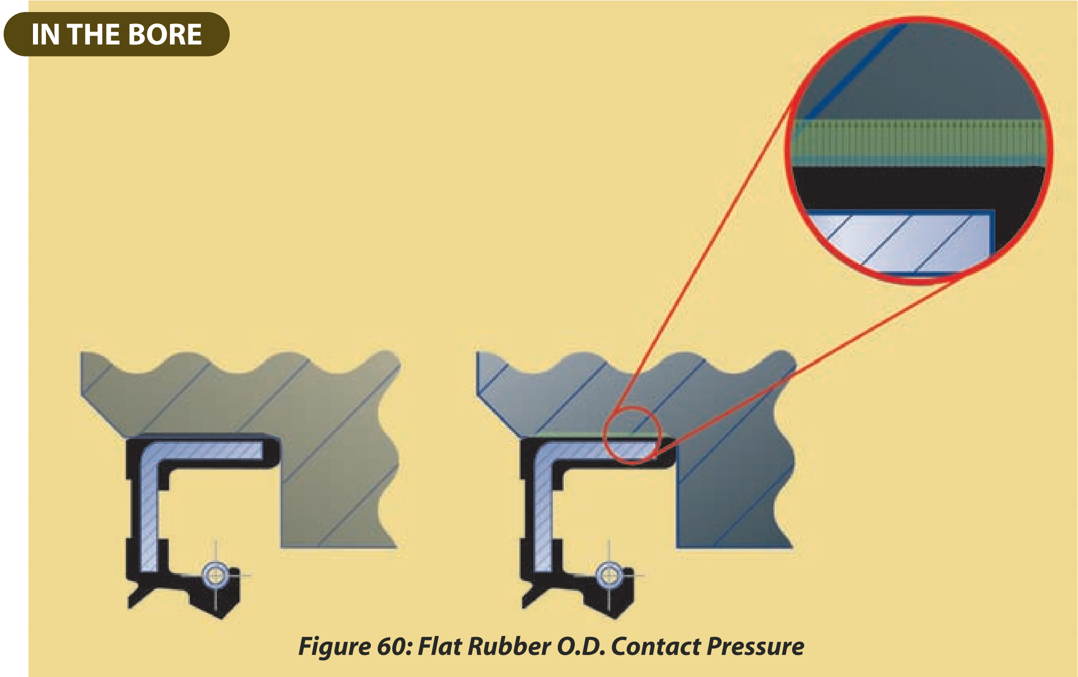


Figure 60: Flat Rubber O.D. Contact Pressure

RUBBER COVERED

Figure 61: Ribbed Rubber Coating on the Seal O.D.

slightly following installation due to shearing stresses between the rubber and the housing bore. An exaggerated example of springback is shown in ***Figure 63***.

Even if installation is perfect, excessive heat during service may cause the rubber coating to take a compression set, thus creating a leak path. In order to compensate for rubber's higher coefficient of thermal expansion (compared to metal) and for the greatly reduced stiffness of the rubber O.D. (again, compared to metal), greater initial interference between the seal and the bore is required than when using metal O.D. seals.

Metal and rubber O.D. seals may be needed for truly tough

IN THE BORE

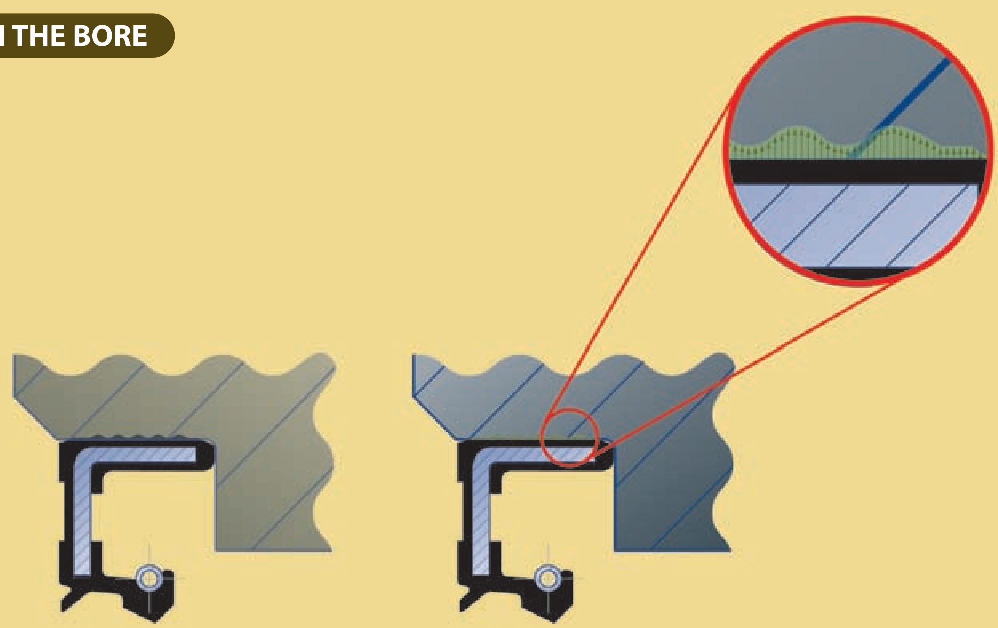


Figure 62: Ribbed Rubber O.D. Contact Pressure

applications. The metal provides retention while the rubber provides sealability. For example, the design shown in ***Figure 64*** features a metal O.D. with a beaded rubber "heel." The metal portion protects the rubber portion from installation damage. The metal also assists with accurate alignment in the bore and minimizes seal cocking and/or movement during use. The rubber element allows a tighter elastic fit into the bore than with metal alone.

Another possibility is the "nose" gasket shown in ***Figure 65*** (next page). Nose gasket seals are often used when the bottom of the housing bore is not finished properly. The presence of the rubber "nose" helps prevent leakage. Be aware that half-rubber, half-metal O.D. seals are more difficult to manufacture and thus more expensive than the other seal O.D. treatments discussed in this section.

The O.D. specifications for rubber O.D. seals are shown in

SPRINGBACK

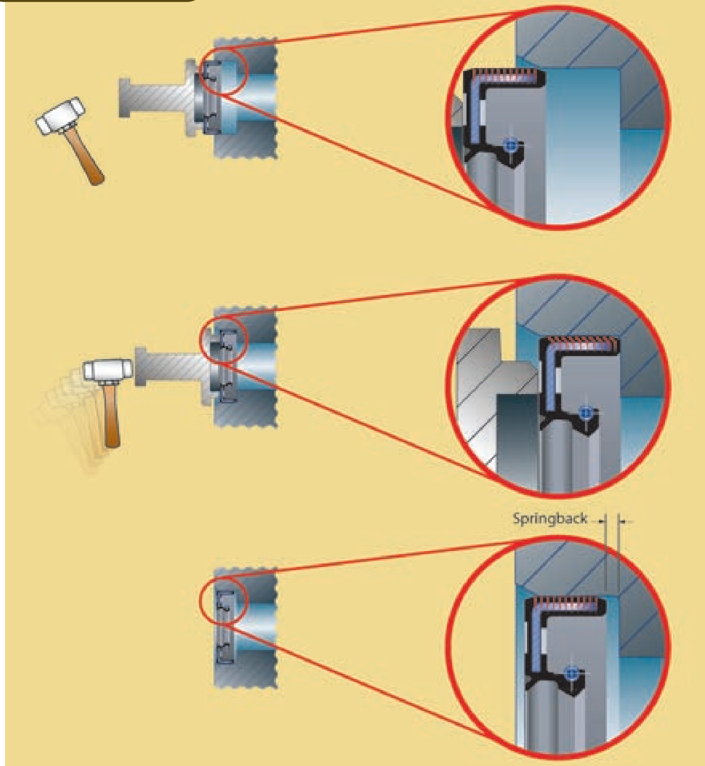


Figure 63: Seal Unseating Due To Shearing Stresses

METAL & RUBBER O.D.

Figure 64: Metal O.D. With Beaded Rubber Heel

designing the seal

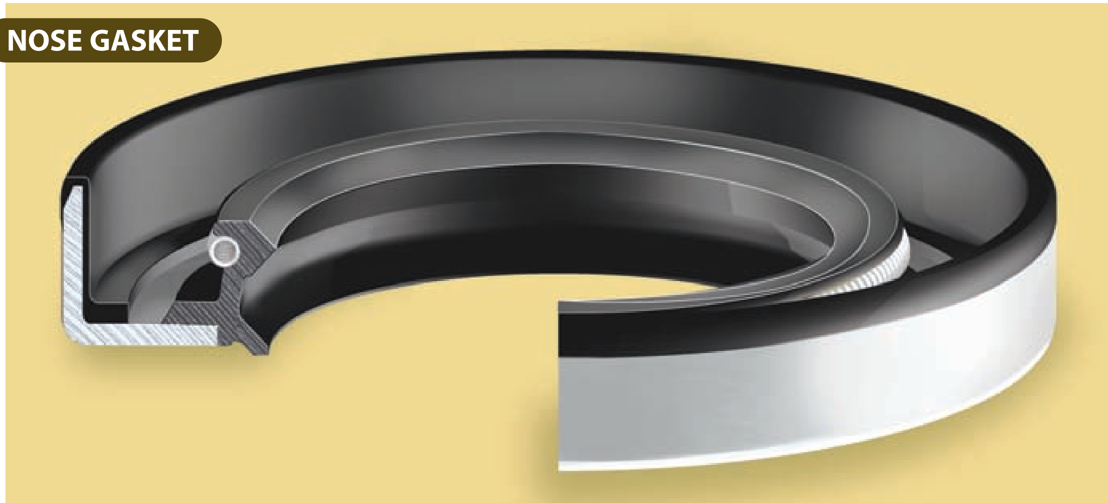


Figure 65: Metal O.D. With Rubber Nose

Table 26. As with ***Table 25***, nominal pressfit is the difference between the seal O.D. nominal dimension and the bore I.D. nominal dimension. Maximum out-of-round (O.O.R.) for the seal O.D. is the maximum deviation from a perfect circle.

There's one other important issue related to the seal case to consider: the *radial wall dimension* (RWD). The RWD of a seal is the radial distance between the seal O.D. and the lip I.D. (contact point) as measured on a complete but uninstalled seal (see ***Figure 66***). The extent to which this distance is not consistent is known as the *radial wall variation* (RWV). Excessive RWV will result in the contact force of the sealing lip on the shaft not being consistent, making leakage more

Seal O.D. (D_o) (mm)	Seal O.D. Tolerance (mm)	Nominal Pressfit (mm)	Diametrical Max. Out of Round (O.O.R.) (mm)	Seal O.D. (D_o) (inch)	Seal O.D. Tolerance (inch)	Nominal Pressfit (inch)	Diametrical Max. Out of Round (O.O.R.) (inch)
$D_o \leq 50$	+ 0.30 + 0.15	0.20	0.25	$D_o \leq 2$	± 0.003	0.008	0.010
$50 < D_o \leq 80$	+ 0.35 + 0.20	0.25	0.35	$2 < D_o \leq 3$	± 0.003	0.010	0.014
$80 < D_o \leq 120$	+ 0.35 + 0.20	0.25	0.50	$3 < D_o \leq 5$	± 0.003	0.0105	0.020
$120 < D_o \leq 180$	+ 0.45 + 0.25	0.32	0.65	$5 < D_o \leq 7$	± 0.004	0.012	0.026
$180 < D_o \leq 300$	+ 0.50 + 0.25	0.34	0.80	$7 < D_o \leq 12$	± 0.004	0.0125	0.031
$300 < D_o \leq 440$	+ 0.55 + 0.30	0.38	1.00	$12 < D_o \leq 20$	± 0.005	0.015	0.039
				$20 < D_o \leq 40$	± 0.006	0.018	0.045
				$40 < D_o \leq 60$	± 0.007	0.020	0.050

Table 26: O.D. Specifications For Rubber O.D. Seals

likely. The maximum allowable RWV values are shown in ***Table 27***.

Keep in mind that, though a single shaft seal design can be effective in a wide range of sizes, there does come a point at which the radial wall dimension becomes so small that the seal cannot be manufactured. Please consult R.L. Hudson & Company if you have any concerns about minimum radial wall dimensions.

As relates to the seal's interior (oil side), there are two main choices: metal or rubber. A rubber coating on the oil side interior is often used if the seal is to face corrosive media that would damage a metal interior.

RADIAL WALL DIMENSION

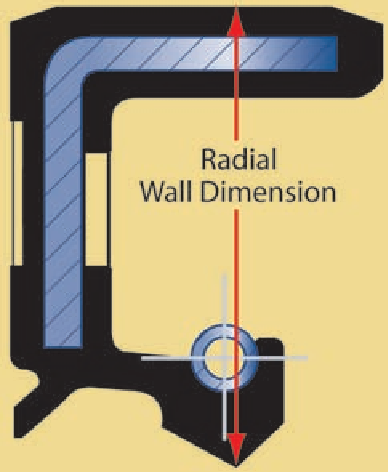


Figure 66: Radial Distance Between Seal O.D. & Seal Contact Point

Shaft Diameter (mm)	Maximum RWV (mm)	Shaft Diameter (inch)	Maximum RWV (inch)
$D_S \leq 75$	0.6	$D_S \leq 3$	0.025
$75 < D_S \leq 150$	0.8	$3 < D_S \leq 6$	0.030
$150 < D_S \leq 250$	1.0	$6 < D_S \leq 10$	0.040

Table 27: Maximum Radial Wall Variation Values

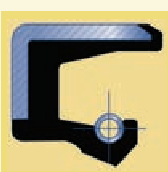
designing the seal

Standard Designs.

In some cases, finding the right shaft seal will simply require that you match your application requirements to one of the standard designs that have been developed to meet most situations. Very demanding applications, such as those requiring sealing in extreme environments, may necessitate use of a non-standard design. In some cases, you may benefit from developing an entirely new design with the help of our in-house engineering department.

Standard designs are basic lip and case configurations that have proven effective in various applications. Each design is designated by a two- or three- letter alphabetic code (such as "SB" or "SBR"). The first letter of this code always refers to the design of the sealing lip. For example, the "S" in both "SB" and "SBR" denotes a single lip seal with a garter spring. The second (or second and third) letter(s) always refer(s) to the design of the seal's case. For example, the "B" in "SB" denotes a metal O.D. with rubber on the fluid side for added protection against corrosion. The "BR" in "SBR" denotes a metal O.D. with rubber on the fluid side and a rubber nose for enhanced O.D. sealing.

"Standard designs are basic lip and case configurations that have proven effective in various applications."

Type	Description
	**SC** Spring-loaded single lip seal w/ flat rubber O.D.
	**SCW** Spring-loaded single lip seal w/ ribbed rubber O.D.
	**SBY** Spring-loaded single lip seal w/ metal O.D.
	**SB** Spring-loaded single lip seal w/ rubber-covered fluid side
	**SBR** Spring-loaded single lip seal w/ rubber-covered fluid side & rubber nose
	**SAX** Spring-loaded single lip seal w/ metal outer & inner cases

Parameters For All S-Lip Designs

Shaft Diameter	Max. Total Eccentricity	Max. Continuous Shaft Speed at 0 psi (NBR Lip)*
0.500″	0.004″	8000 RPM
1.000″	0.006″	7000 RPM
2.000″	0.010″	4500 RPM
3.000″	0.012″	3800 RPM
4.000″	0.016″	2700 RPM

* as pressure increases, maximum shaft speed decreases; maximum recommended pressure is 5 psi

Typical Lip Materials

Nitrile (NBR)	-25° to +225° F
Hydrogenated Nitrile (HNBR)	-25° to +300° F
Fluoroelastomer (FKM)	-15° to +300° F
Polyacrylate (ACM)	-25° to +275° F
Silicone (VMQ)	-65° to +300° F

Table 28: S-Lip Designs

S-LIP DESIGNS

Standard shaft seals can be separated into categories based on lip design. As already noted, the "S" designs are single lip seals with garter springs; they are intended for non-pressure fluid sealing and severe grease sealing applications. The operating parameters for six different standard S-lip designs are shown in ***Table 28.***

Type	Description
	**TC** Spring-loaded double lip seal w/ flat rubber O.D.
	**TCW** Spring-loaded double lip seal w/ ribbed rubber O.D.
	**TBY** Spring-loaded double lip seal w/ metal O.D.
	**TB** Spring-loaded double lip seal w/ rubber-covered fluid side
	**TBR** Spring-loaded double lip seal w/ rubber-covered fluid side & rubber nose
	**TAX** Spring-loaded double lip seal w/ metal outer & inner cases

Parameters For All T-Lip Designs

Shaft Diameter	Max. Total Eccentricity	Max. Continuous Shaft Speed at 0 psi (NBR Lip)*
0.500″	0.004″	8000 RPM
1.000″	0.006″	7000 RPM
2.000″	0.010″	4500 RPM
3.000″	0.012″	3800 RPM
4.000″	0.016″	2700 RPM

* as pressure increases, maximum shaft speed decreases; maximum recommended pressure is 5 psi

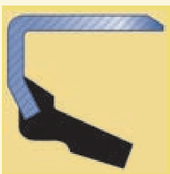
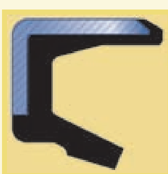
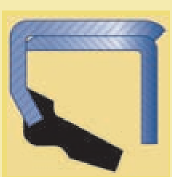
Typical Lip Materials

Nitrile (NBR)	-25° to +225° F
Hydrogenated Nitrile (HNBR)	-25° to +300° F
Fluoroelastomer (FKM)	-15° to +300° F
Polyacrylate (ACM)	-25° to +275° F
Silicone (VMQ)	-65° to +300° F

Table 29: T-Lip Designs

T-LIP DESIGNS

"T" designs are just like the "S" designs but with added secondary lips for light-duty contaminant exclusion. ***Table 29*** lists the operating parameters for six different standard T-lip designs.

Type	Description
	**VC** Non-spring-loaded single lip seal w/ flat rubber O.D.
	**VCW** Non-spring-loaded single lip seal w/ ribbed rubber O.D.
	**VBY** Non-spring-loaded single lip seal w/ metal O.D.
	**VB** Non-spring-loaded single lip seal w/ rubber-covered fluid side
	**VBR** Non-spring-loaded single lip seal w/ rubber-covered fluid side & rubber nose
	**VAX** Non-spring-loaded single lip seal w/ metal outer & inner cases

Parameters For All V-Lip Designs

Shaft Diameter	Max. Total Eccentricity	Max. Continuous Shaft Speed at 0 psi (NBR Lip)*
0.500″	0.003″	4000 RPM
1.000″	0.005″	3000 RPM
2.000″	0.006″	2300 RPM
3.000″	0.008″	1700 RPM
4.000″	0.010″	1400 RPM

* as pressure increases, maximum shaft speed decreases; maximum recommended pressure is 4 psi

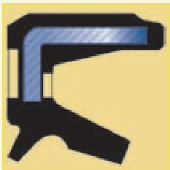
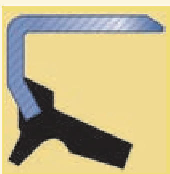
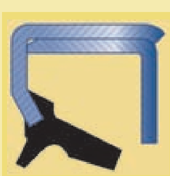
Typical Lip Materials

Nitrile (NBR)	-25° to +225° F
Hydrogenated Nitrile (HNBR)	-25° to +300° F
Fluoroelastomer (FKM)	-15° to +300° F

Table 30: V-Lip Designs

V-LIP DESIGNS

“V” designs are single lip seals (without garter springs) intended for grease or viscous fluid retention. The operating parameters for six different standard V-lip designs are shown in ***Table 30.***

Type	Description
	**KC** Non-spring-loaded double lip seal w/ flat rubber O.D.
	**KCW** Non-spring-loaded double lip seal w/ ribbed rubber O.D.
	**KBY** Non-spring-loaded double lip seal w/ metal O.D.
	**KB** Non-spring-loaded double lip seal w/ rubber-covered fluid side
	**KBR** Non-spring-loaded double lip seal w/ rubber-covered fluid side & rubber nose
	**KAX** Non-spring-loaded double lip seal w/ metal outer & inner cases

Parameters For All K-Lip Designs

Shaft Diameter	Max. Total Eccentricity	Max. Continuous Shaft Speed at 0 psi (NBR Lip)*
0.500″	0.003″	4000 RPM
1.000″	0.005″	3000 RPM
2.000″	0.006″	2300 RPM
3.000″	0.008″	1700 RPM
4.000″	0.010″	1400 RPM

* as pressure increases, maximum shaft speed decreases; maximum recommended pressure is 4 psi

Typical Lip Materials

Nitrile (NBR)	-25° to +225° F
Hydrogenated Nitrile (HNBR)	-25° to +300° F
Fluoroelastomer (FKM)	-15° to +300° F

Table 31: K-Lip Designs

K-LIP DESIGNS

"K" designs are the same as the "V" designs but with added secondary lips for light-duty contaminant exclusion. **Table 31** lists the operating parameters for six different standard K-lip designs.

Non-Standard Designs.

The savvy seal designer recognizes both the possible uses and the inherent limitations of a given product. Though the standard shaft seals shown previously are ideal for a wide variety of applications, they are not the best solution to every design problem.

Since trying to force a standard shaft seal to perform beyond its capabilities can only lead to failure, you will probably be faced from time to time with difficult or unusual situations that require you to consider other seal cross-sections. Or, perhaps you can save money or simplify installation while still satisfying an application's needs by using a different design.

What follows is information on other sealing options that operate in place of (or in addition to) standard shaft seals. Each alternative has both advantages and disadvantages, so the specific needs of your application will always dictate which one is best for you. Familiarity with your options, however, should make the choice somewhat easier.

"Though the standard shaft seals shown previously are ideal for a wide variety of applications, they are not the best solution to every design problem."

DOUBLE LIP SEALS

Whereas many shaft seals feature only one spring-loaded lip, it is possible to design a seal with two spring-loaded lips. The double-lip seal shown in ***Figure 67*** is one example. Double-lip seals are typically used to separate fluids. Because of their unusual design, double-lip seals require greater bore depth than standard shaft seals. As with standard shaft seals, it is also possible to design a double-lip seal with a rubber O.D. An example of this is shown in ***Figure 68***.

DOUBLE LIPS

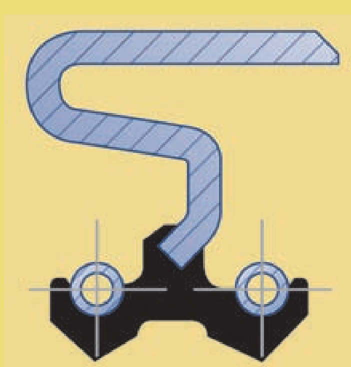


Figure 67: Metal O.D.

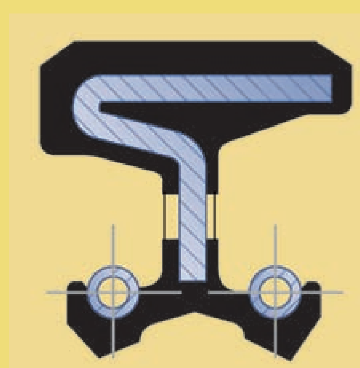


Figure 68: Rubber O.D.

HYDRODYNAMIC SEALS

It's important to note that pump rate increases as shaft speed increases, and this improves seal reliability. Pumping action can also be enhanced through the addition of artificial pumping aids molded onto the air side of the seal lip, resulting in what is known as a *hydrodynamic seal*. Helical ribs are molded on the air side of the seal and can be used only if the shaft rotates in one direction. The result is a uni-directional hydrodynamic seal, such as is shown in ***Figure 69.*** This is also known as a *helix seal*.

UNIDIRECTIONAL HELIX SEAL

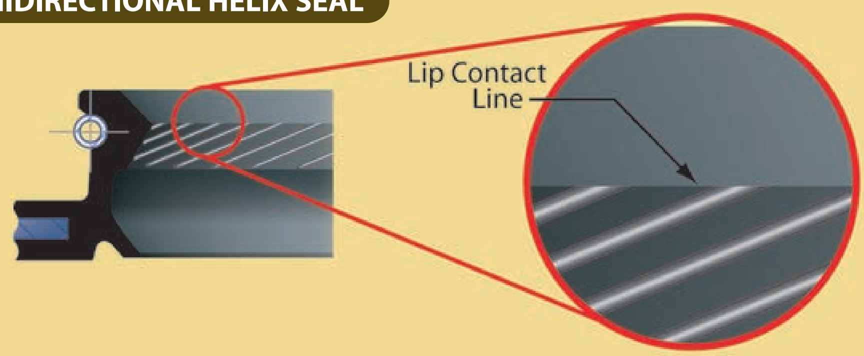


Figure 69: Helical Ribs Molded on the Air Side of the Seal Lip

These ribs accentuate the seal's pumping action in order to force fluid weepage back under the lip. Unfortunately, these ribs do have potential disadvantages. Contaminants may fall victim to the pumping action and be directed toward the contact point, thus increasing the chances of lip and shaft wear. Presence of a secondary lip for contaminant exclusion may help, but a secondary lip may also allow a vacuum to develop between the two sealing lips that would distort the primary lip and result in leakage. In some cases, a screw thread (rather than helical ribs) can be molded, coined, or machined onto the air side of the sealing lip; this creates

UNIDIRECTIONAL SPIRAL SEAL

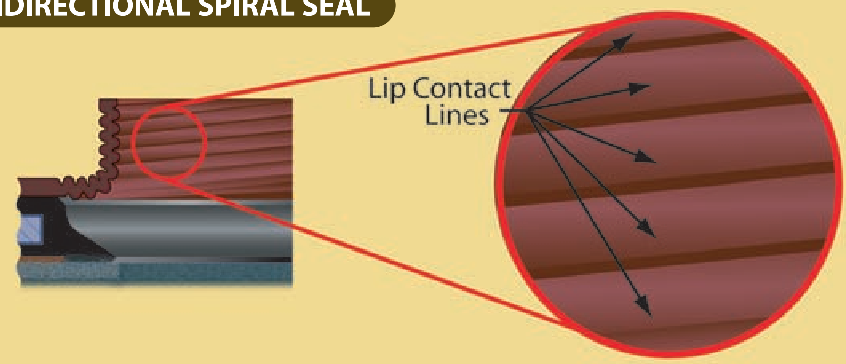


Figure 70: Screw Thread on the Air Side of the Seal Lip

BIDIRECTIONAL RIBS

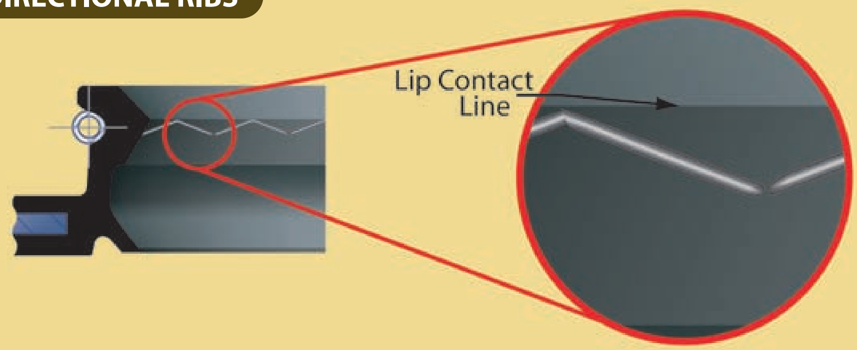


Figure 71: Bidirectional Ribs Molded on the Air Side of the Seal Lip

what is known as a *spiral seal*. The PTFE sealing lip shown in ***Figure 70*** is a good example of this.

When ordering either a helix or spiral seal, it is very important to specify the direction (clockwise or counter-clockwise) of the shaft rotation as viewed from the air side. Helical ribs or spiral threads are designed to function in only one direction, and a mismatch between their orientation and shaft rotation will cause the seal to pump liquid *out* rather than back in, resulting in leakage.

BIDIRECTIONAL TRIANGULAR PADS

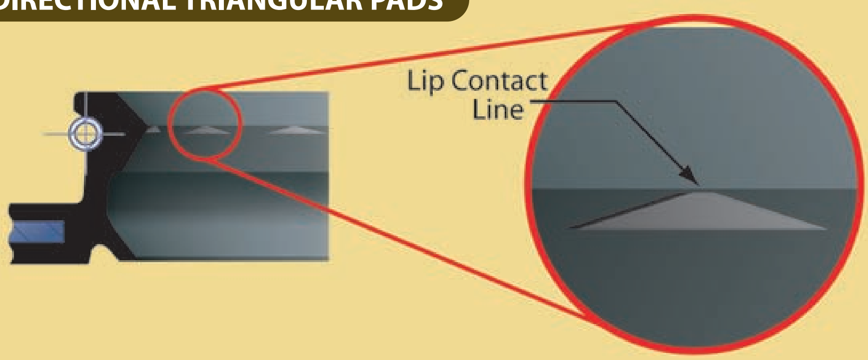


Figure 72: Triangular Pads Molded on the Air Side of the Seal Lip

Applications in which shaft rotation is bi-directional require different lip designs. One option is to mold bidirectional ribs onto the air side of the sealing lip. These ribs function similarly to unidirectional ribs, except that they facilitate hydrodynamic pumping in both directions. ***Figure 71*** shows what bidirectional ribs look like. Bidirectional pumping can also be facilitated through triangular pads molded onto the air side of the lip. ***Figure 72*** shows what these might look like.

Table 32 (next page) compares the measured pump rates for various types of hydrodynamic seals. Seals with triangular pads have fewer pumping elements (since pads take up

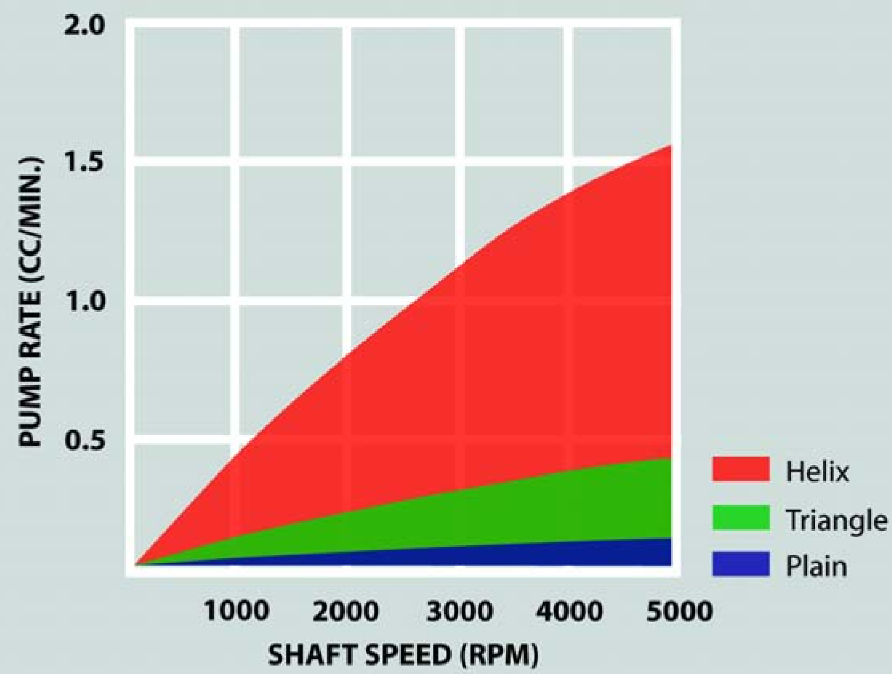


Table 32: Pump Rates for Hydrodynamic Seals

more room than ribs), so triangular pad seals are not as effective at pumping as unidirectional helix seals. Overall, helix seals pump best, followed by triangular pad seals and plain trimmed lip seals with no added pumping elements.

No matter what type of hydrodynamic seal you might use, it is extremely important that the hydrodynamic element (i.e. the ribs or pads) make proper contact with the shaft. It is recommended that the contact patterns be viewed through a transparent plastic shaft specifically designed to facilitate the viewing of helices footprints. An example of a non-centering plastic shaft is shown in ***Figure 73***.

NON-CENTERING PLASTIC SHAFT

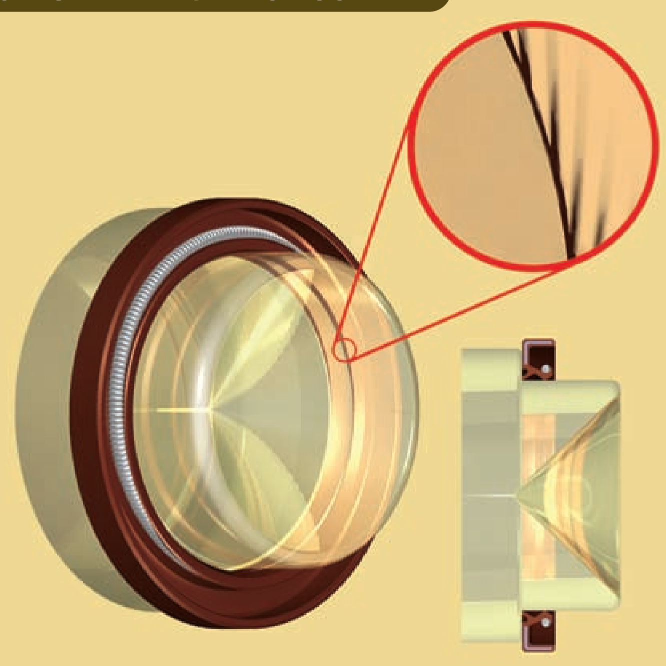


Figure 73: Facilitates Viewing of Helices Footprints

A self-centering plastic shaft is shown in ***Figure 74***. The difference between this fixture and the one shown in ***Figure 73*** is that the self-centering plastic shaft features an added O.D. shoulder. This shoulder allows the seal O.D. to seat as it would in a true housing bore, thus more fully replicating the actual service configuration.

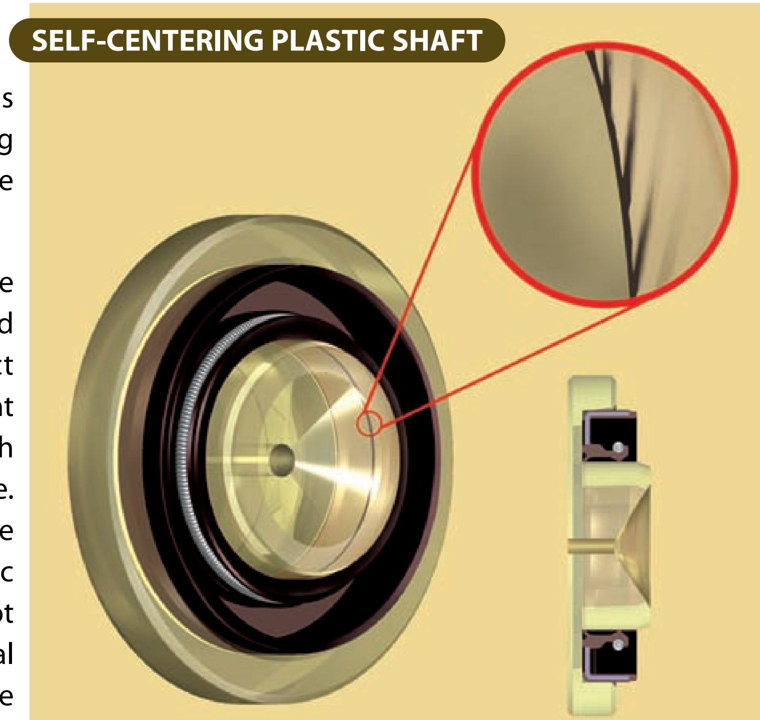


Figure 74: Facilitates Viewing of Helices Footprints

Table 33 shows some examples of both good and bad lip contact patterns as they might look if viewed through a plastic shaft fixture. It's important to note that hydrodynamic elements that do not touch the primary seal lip, are too high, or are too shallow will not result in the formation of an advantageous (in-pumping) contact pattern.

UNIDIRECTIONAL HELIX RIBS	Actual Lip	Contact Pattern on Fixture
No contact with primary lip.		
Helix ribs too high.		
Helix ribs too shallow.		
Good helix ribs and contact pattern.		
BIROTATIONAL PADS	**Actual Lip**	**Contact Pattern on Fixture**
No contact with primary lip.		
Pads too high.		
Pads too shallow.		
Good pads and contact pattern.		

Table 33: Lip Contact Patterns

FLANGE SEALS

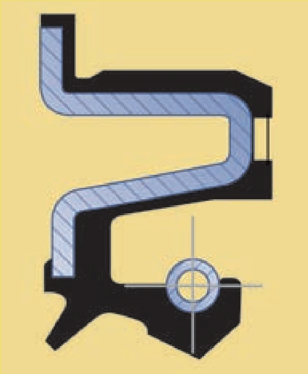


Figure 75: Reverse Channel w/Flange

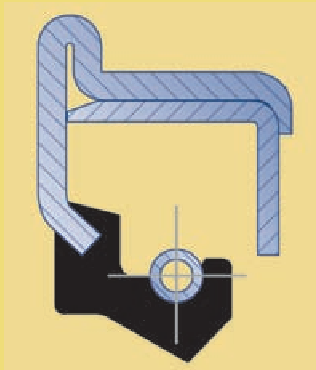


Figure 76: Double Case w/Flange

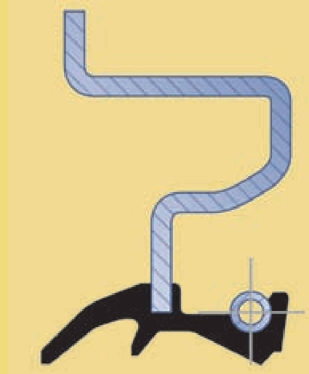


Figure 77: Dual Dirt Lips w/Flange

FLANGE SEALS

Flange seals are typically used for applications in which there are concerns about proper installation or removal. If the flange is present to help ensure proper seal alignment, the housing face must be machined to be perpendicular to the axis of the shaft. The installation procedure must ensure that the flange is touching the housing face completely around the circumference. Sometimes a semi-liquid bore sealant is applied to the seal O.D.; the flange will help keep the sealant on the O.D. until the seal is installed in the housing.

Additionally, the flange will increase the metal case strength and provide for easier removal when the seal needs to be replaced. ***Figure 75*** shows a design featuring both a reverse channel case and a flange. The design shown in ***Figure 76*** features a double case (for added rigidity) and a flange. ***Figure 77*** shows a seal with dual radial dirt lips, a reverse channel case, and a flange.

UNITIZED SEALS

Figure 78: Triple Dirt Lip w/ Flange

Figure 79: Ribbed Rubber O.D.

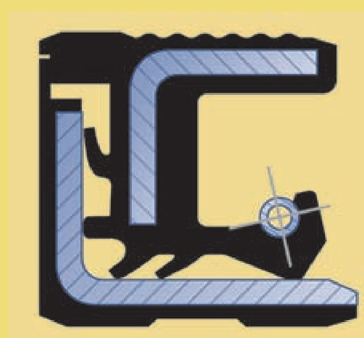


Figure 80: Quadruple Dirt Lips

UNITIZED (CASSETTE) SEALS

Whereas many shaft seals are designed to be installed onto a running surface, a *unitized seal* is a shaft seal that incorporates a running surface into the seal design. Also known as *cassette seals*, unitized seals are commonly used when the actual shaft surface lacks an acceptable finish (due, for example, to prior damage or improper metal finishing).

Some unitized seals include both a metal sleeve that serves as the contact surface for the primary sealing lip and a vertical flange against which an axial dirt lip can seal. Because unitized seals come with their own sealing surface(s), they effectively take the burden of providing these surfaces off of the end-user.

Figure 78 shows a unitized seal with a flange. ***Figure 79*** shows a unitized seal with a ribbed rubber O.D., and ***Figure 80*** shows a unitized seal with a ribbed rubber O.D. and a full rubber interior. All three designs feature a vertical element to facilitate axial sealing.

TRIPLE LIP SEALS

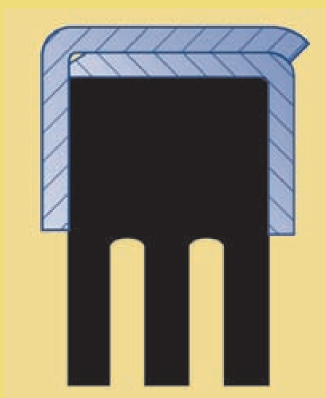


Figure 81: Internal Sealing

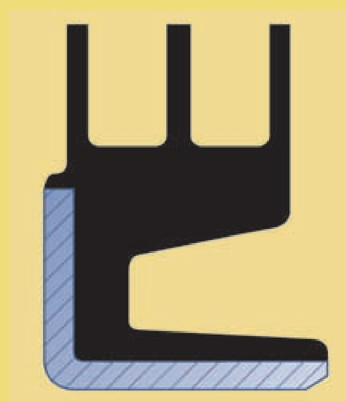


Figure 82: External Sealing

TRIPLE LIP (MUD) SEALS

Springless seals featuring multiple sealing lips are common in applications where heavy dirt and mud are encountered (such as in farming equipment). Grease is packed between the sealing lips. These designs are suitable for applications in which shaft speeds are less than 2.5 meters per second (500 feet per minute).

Figure 81 shows a triple-lip seal (with a double case) designed for internal sealing. ***Figure 82*** shows a triple lip seal (with a single case) for external sealing.

ROTATING BORE SEALS

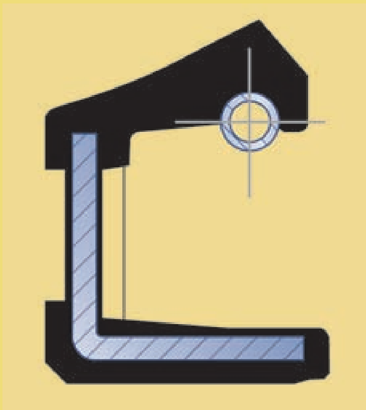


Figure 83: Single Lip

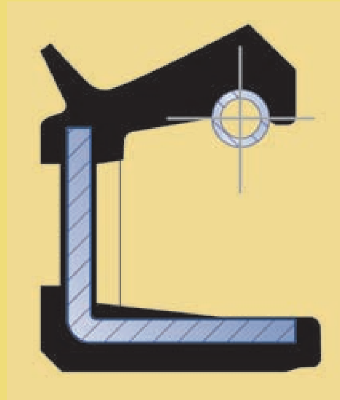


Figure 84: Dual Lip

ROTATING BORE SEALS

In most shaft seal applications, it is the shaft that is in motion. In some instances, however, the housing bore may be in rotation, meaning that sealing is necessary on the seal O.D. rather than the I.D. (which will be stationary).

Figure 83 shows a spring-loaded single lip design for O.D. sealing. ***Figure 84*** shows spring-loaded dual lip seal for O.D. sealing.

HIGH ECCENTRICITY SEALS

Standard shaft seals will typically fail in applications with excessive shaft vibrations or dynamic runout. Non-standard designs with convoluted lips can often overcome this problem by providing increased followability in the radial direction. An example of such a convoluted-lip design is shown in ***Figure 85***.

Keep in mind that, though high eccentricity seals can often withstand excessive runouts for short periods without leakage, they are not intended to function indefinitely. Attention should always be given to proactively reducing shaft eccentricities when possible as a means of precluding the need for a special seal design.

HIGH ECCENTRICITY SEAL

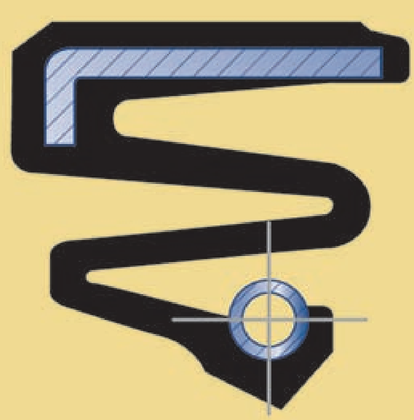


Figure 85: Convoluted Lip

PTFE SHAFT SEALS

Polytetrafluoroethylene (PTFE, trade name Teflon®) shaft seals are used in several specific types of applications, such as high pressure hydraulic pump and motor and diesel engine crankshaft applications. Some PTFE shaft seals feature PTFE lips clamped in place using spacers and rubber gaskets inside a metal case. This clamping can be necessary because it's tough to bond PTFE to rubber or metal. Other designs, however, do bond PTFE to rubber or metal.

Figure 86 shows the "EPT" design, a single lip seal for high pressure applications. The unitized "EHX" design shown in ***Figure 87*** features both a primary lip and a secondary, exclusion lip (both made of PTFE). The EHX design is typically used for diesel engine applications. A unitized seal with PTFE bonded to rubber, which is then bonded to metal, is shown in ***Figure 88***.

PTFE offers advantages over standard shaft seal materials. If your application requires a seal that can withstand high pressures or high temperatures, PTFE shaft seals may be the answer. PTFE seals can withstand temperatures ranging from -65° F to +325° F (-54° C to +163° C). PTFE seals can tolerate shaft speeds of up to 12,000 feet per minute (fpm). Because of PTFE's unique chemical structure, PTFE seals offer excellent resistance to most chemicals and fluids.

Keep in mind, however, that PTFE shaft seals are not without disadvantages. As with the aforementioned advantages, these disadvantages stem from the nature of the PTFE itself. Because PTFE is stiffer than traditional elastomeric lip materials, PTFE lips will not form the tiny pores (microasperities) that are integral to the pumping action seen in successful shaft seals. To compensate, a spiral groove must be machined or coined into the primary lip surface if it is intended to seal oil under low pressure conditions. This groove can simulate the pumping action by screwing fluid back into the sump, but the spiral groove limits seal usage to applications in which the shaft rotates in only one direction. Examples of PTFE lip seals featuring these spiral grooves are shown in ***Figures 87***, ***88***, and ***89*** (see next page).

PTFE SHAFT SEALS

Figure 86: EPT Seal

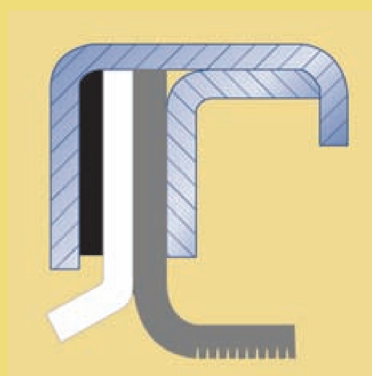


Figure 87: EHX Seal

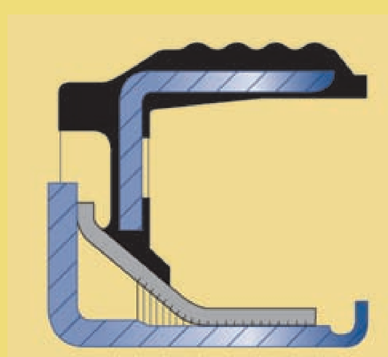


Figure 88: Bonded & Unitized

PTFE SHAFT SEAL

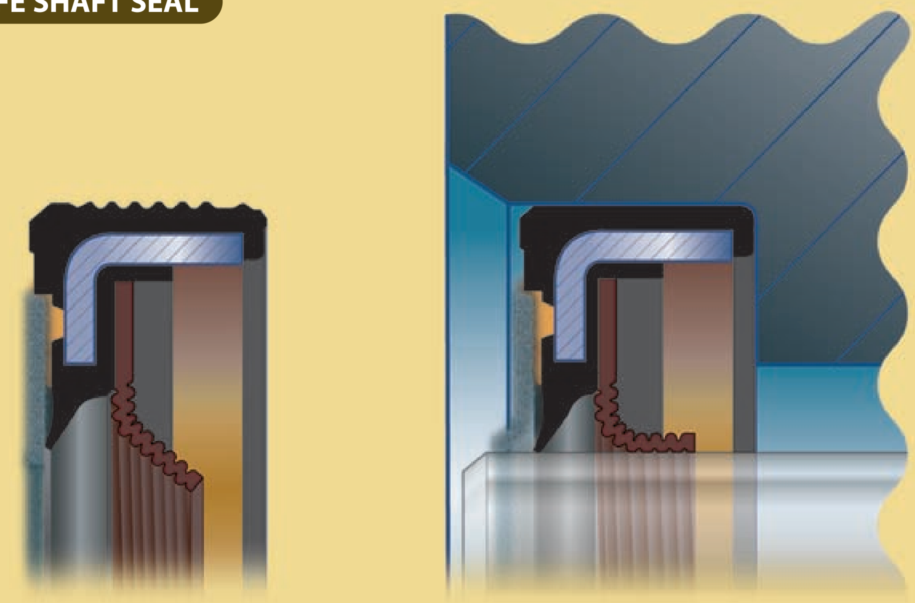


Figure 89: Coined Spiral Groove on Sealing Lip

PTFE also has much less memory (ability to regain its original shape following deformation) than traditional, elastomeric lip materials. This reduced memory makes the lip less able to maintain consistent contact with the shaft, particularly in the presence of shaft eccentricity. As a result, leakage becomes more likely.

Finally, PTFE lips are more delicate than traditional lips and can therefore be easily damaged during installation. For this reason, a PTFE seal may come pre-assembled onto a wear sleeve. This seal-sleeve combination can be slipped over the shaft with less chance of damaging the seal. Seals may also be shipped with disposable protective sleeves.

AXIAL DIRT LIP SEALS

Because of the potentially damaging effects of outside contamination on shaft seal lips, studies have been conducted to try to determine the most effective excluder lip designs. Compared with other designs (including single and dual radial dirt lips), axial dirt lips proved the most effective. The seal shown in ***Figure 90*** features an axial dirt lip in combination with dual radial dirt lips and a spring-loaded primary lip. ***Figure 91*** shows a seal with a axial dirt lip and a spring-loaded primary lip.

HIGH PRESSURE SEALS

Standard shaft seal designs are not adequate when subjected to pressures of 10 psi or higher. As shown in ***Figure 92***, higher

AXIAL DIRT LIP SEALS

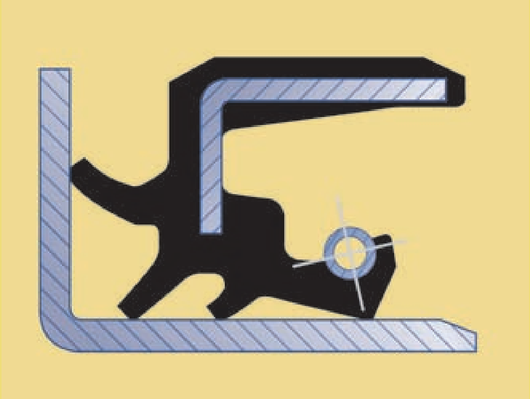


Figure 90: Axial & Radial Dirt Lips

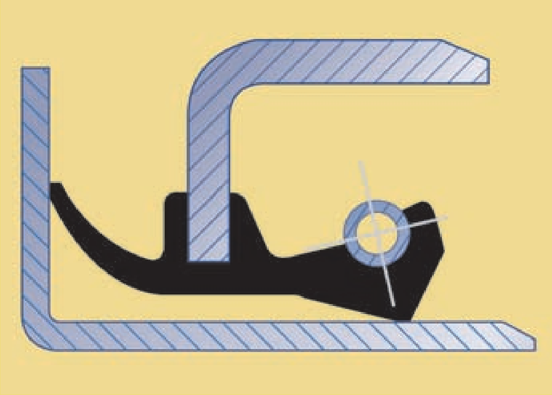


Figure 91: Axial Dirt Lip

pressures can distort the sealing lip, resulting in a greater-than-desired amount of contact between the air side surface of the lip and the shaft.

Under some conditions, the pressure will distort the seal such that the sealing tip leaves the shaft. This phenomenon, known as *bell-mouthing*, generates more friction and heat, and, as a result, greater wear. This increased wear is evident on the air side surface of the lip rather than at the lip contact point. This greater wear shortens seal life. In some extreme instances, higher pressures have even been known to force the seal out of the bore or to tear the elastomeric lip away from its metal case.

In response to the need for seals capable of withstanding higher pressures, designers have developed a variety of non-standard designs. These typically feature heavier-than-normal cross-sections (to minimize lip distortion) and a greater bonding area between the lip and the metal case (to lessen the chances that the lip will be torn away).

Figures 93, ***94***, ***95***, ***96***, and ***97*** (next page) show some examples of various high pressure shaft seal designs.

LIP DISTORTION

Figure 92: Pressure-Induced Lip Distortion

HIGH PRESSURE SEALS

Figure 93: TCV Design

Figure 94: TCN Design

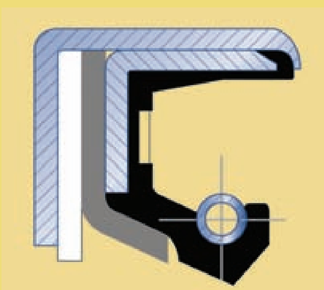


Figure 95: SAV Design

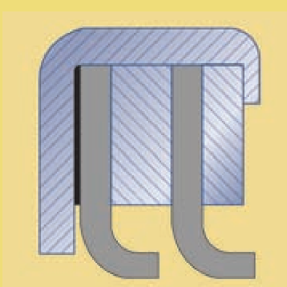


Figure 96: EP2 Design

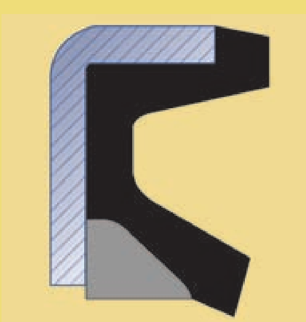


Figure 97: HP1 Design

The TCV seal shown in ***Figure 93*** is designed for applications up to 50 psi, though of course the actual pressure limit is influenced by shaft speed. The TCV features a flat rubber O.D. and rubber interior in conjunction with a heavy-duty spring-loaded primary sealing lip and a secondary contamination exclusion lip.

Figure 94 shows a TCN design. The TCN offers steel lip reinforcement for added pressure resistance, a heavy-duty spring-loaded primary sealing lip, a secondary contamination exclusion lip, and a flat rubber O.D.

The SAV seal shown in ***Figure 95*** incorporates a filled PTFE back-up for increased pressure resistance, a PTFE contamination exclusion lip, and a steel O.D.

Figure 96 shows the EP2 design. The EP2 features sealing lips made of PTFE in order to withstand high pressure, temperatures, and shaft speeds. The EP2 also boasts a double reinforced steel case.

The HP1 seal shown in ***Figure 97*** offers a filled PTFE backup to resist extrusion, a heavy-duty sealing lip, and a steel O.D. with rubber nose.

LINEAR (RECIPROCATING) SEALS

Though standard shaft seals can be effective in normal reciprocating applications, severe reciprocating applications in which oil or oil-water mixes must be retained typically require non-standard designs.

The TB4 design shown in ***Figure 98*** is one example of a reciprocating seal. Note that it features a stepped, spring-loaded primary sealing lip as well as a secondary exclusion lip. The TB4 can handle pressures up to 100 psi (7 kg/cm2).

Figure 99 shows a TC4 design, which is similar to the TB4 but with a rubber rather than metal O.D. The TC4 can also handle pressures up to 100 psi.

LINEAR (RECIPROCATING) SEALS

Figure 98: TB4 Design

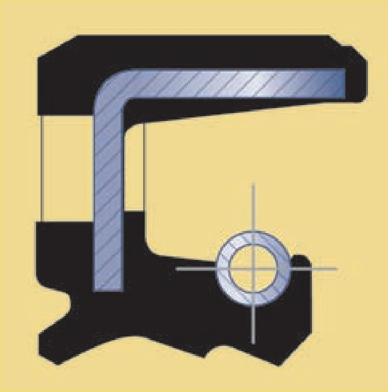


Figure 99: TC4 Design

designing the seal

Other Sealing Options.

As useful as both standard and non-standard shaft seal designs are, there are still some situations in which something other than a shaft seal will be needed or wanted.

What follows is an overview on three such shaft seal alternatives: V-rings, axial face seals, and bore plugs. Each alternative has pluses and minuses, so be sure to consider all aspects of your application needs during the design process.

"There are still some situations in which something other than a shaft seal will be needed or wanted."

V-RINGS

V-rings (also known as V-seals) are all-rubber seals that are installed directly onto a shaft to seal against a housing face, a bearing, or the back of a shaft seal. Offering simplicity and versatility, V-rings are chiefly used to protect bearings or other parts from contamination, as well as to retain grease. V-rings can also function effectively in dry applications with very low torque.

As shown in ***Figure 100***, a V-ring consists of three parts: a body, a conical self-adjusting lip, and a hinge. In service, the elastic body of the seal rotates with the shaft while the adjustable lip maintains a dynamic seal in axial contact with a perpendicular counterface. The seal shown in ***Figure 100*** is a "VA" style V-ring. VA style V-rings are the most common style and are used in appliances and conveyor rollers.

Widening the body of the VA style results in the "VS" style is shown in ***Figure 101***. The expanded body and tapered heel

V-RING STYLES

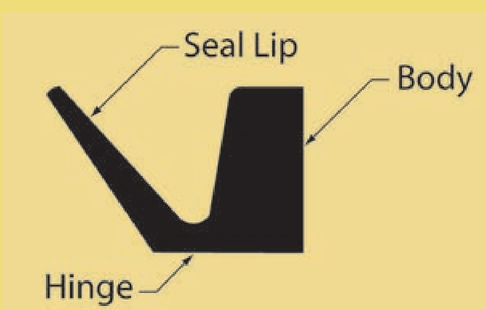


Figure 100: "VA" Style

Figure 101: "VS" Style

V-RING ADVANTAGES

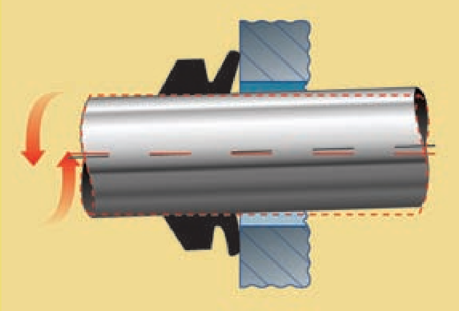


Figure 102: Tolerates Shaft Cocking

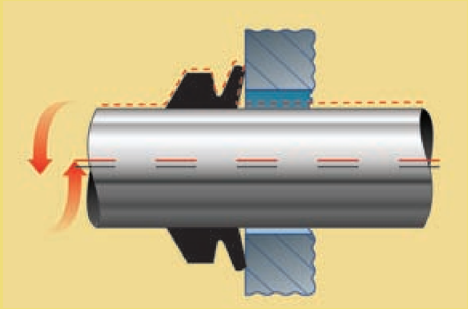


Figure 103: Tolerates Shaft-To-Bore MIsalignment

of the VS style hold the V-ring in place on the shaft. VS style V-rings are used in agricultural and automotive applications.

V-rings offer many advantages. As shown in ***Figure 102***, a V-ring tolerates greater radial variation (in this case, shaft cocking) than a traditional shaft seal. As shown in ***Figure 103,*** a V-ring also tolerates greater shaft-to-bore misalignment (STBM). A V-ring also tolerates greater dynamic runout (DRO) than a standard shaft seal. And, as shown in ***Figure 104***, a V-ring can function as both a seal and a slinger. V-rings are also less expensive than traditional shaft seals.

Another important advantage of the V-ring is its elasticity. A V-ring can be stretched during installation onto a shaft without disassembly of the unit, and one size V-ring can be used on a range of shaft sizes in either metric or English dimensions. A commonly used V-ring material is wear-resistant nitrile (-40° F to 225° F). V-rings made of fluoroelastomers are typical in applications with higher temperatures (-20° F to 300° F). As shown in ***Figure 105***, V-rings are often used as supplemental seals in conjunction with traditional shaft seals. Studies have shown that a V-ring

V-RING FUNCTIONS

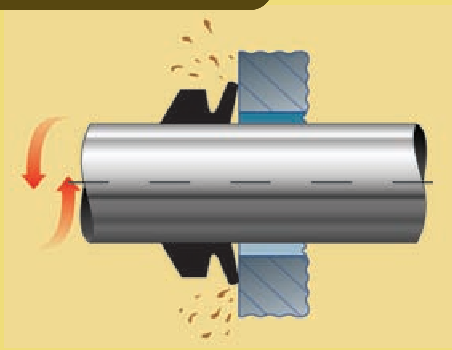


Figure 104: As Seal & Slinger

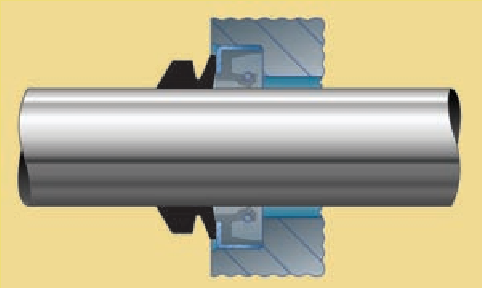


Figure 105: As Supplemental Seal

paired with a single lip shaft seal excludes contaminants better than a shaft seal with a conventional radial dirt lip.

Be aware that, because of their simplified design, V-rings are not intended to seal oils or other low viscosity fluids. V-rings are held in place by a hoop force generated when the rubber ring is installed on the shaft, but at high speeds, centrifugal force can cause the seal to lift off of the shaft. A garter spring or metal band can be employed to prevent this. Shaft motion can also cause the V-ring to slip axially on the shaft. A metal strap can be placed on the shaft to stop this.

AXIAL FACE SEALS

In certain highly contaminated applications, you may need the added protection offered by the AF1 and AF2 axial face seals. Combining the advantages of a rubber V-ring with the rigidity and contaminant protection of a metal case, these seals are well suited for use in rotary shaft applications in which high contamination reduces the life of radial seals and bearings. Such applications might include motors, gearboxes, speed reducers, saws, and lathes.

As typified by the AF1 design shown in ***Figure 106***, an axial face seal consists of a metal case and an elastomeric sealing element (available in nitrile and fluoroelastomer) that is stretch-fitted into the case. The metal case acts as a slinger to provide excellent protection from dirt, debris, and water. It also accommodates high rotating speeds without additional clamping hardware.

Different configurations are possible. ***Figure 107*** shows an AF2 design. The AF2 provides even better protection against contaminant entry than the AF1, but the AF2's extended case requires that a groove be machined into the housing.

AXIAL FACE SEAL DESIGNS

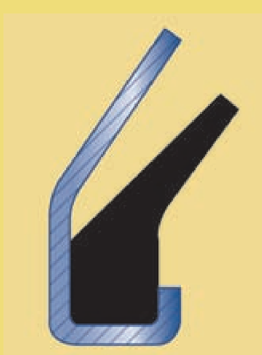


Figure 106: AF1 Design

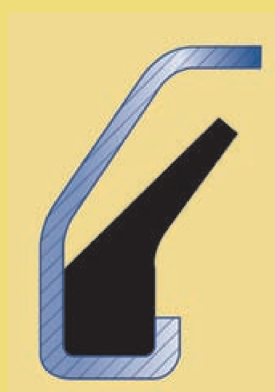


Figure 107: AF2 Design

AXIAL FACE SEAL INSTALLATIONS

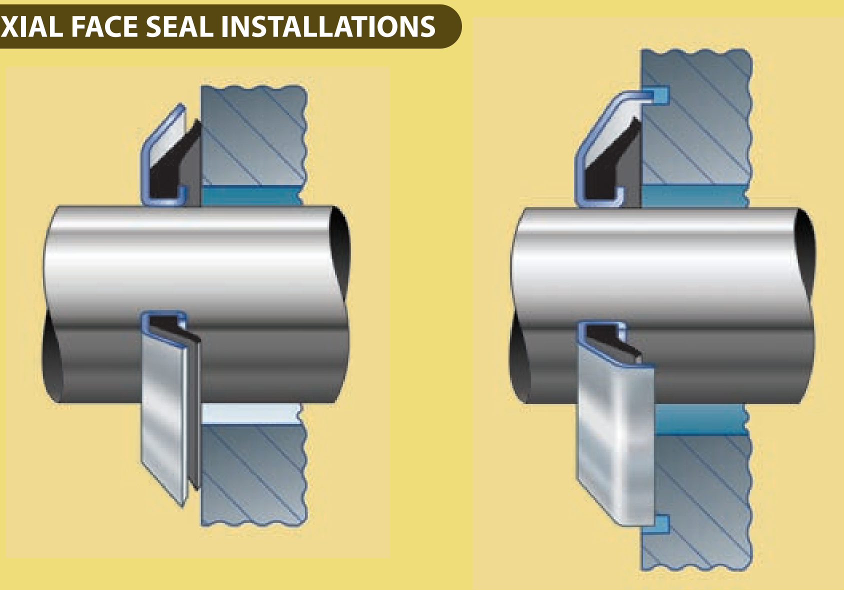


Figure 108: AF1 Installation *Figure 109: AF2 Installation*

Regardless of the design, an axial face seal is installed by being press-fitted onto the shaft. ***Figure 108*** shows an AF1 seal fitted onto a shaft, and ***Figure 109*** shows installation of an AF2 seal.

Axial face seals offer many advantages. For example, the metal case protects the elastomeric body and lip from damage and displacement by external debris, windings, and viscous media. The case also keeps the rubber lip in place even at high shaft speeds. The seal's compact design allows narrow installation widths. The seal also offers low frictional heat build-up and torque drag; as rotary speed increases, the axial face seal's lip moves away from the countersurface. Axial face seals are more expensive, however, than conventional V-rings.

BORE PLUGS

Bore plugs are completely static seals used to block off the end of a shaft. Molded from materials including nitrile and fluoroelastomer, most bore plugs are custom designed. ***Figure 110*** shows an example of a standard bore plug. Extended plugs for use on shafts that go beyond the housing bore are also possible.

BORE PLUG

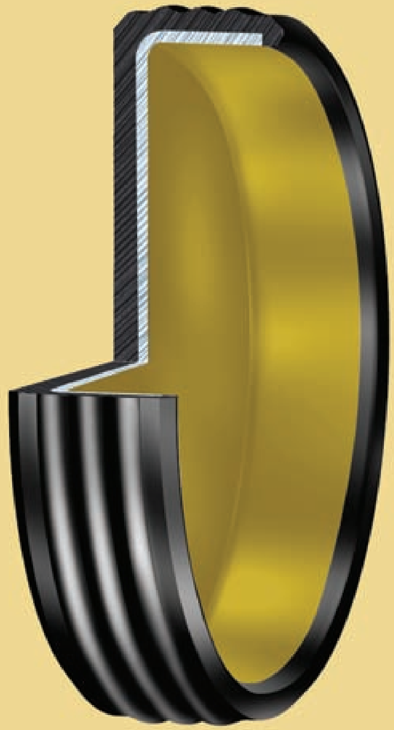


Figure 110: Standard Style Bore Plug

designing the seal

The Shaft.

"Shaft seal design is never complete without giving due consideration to the shaft on which the seal will be asked to function."

Shaft seal design is never complete without giving due consideration to the shaft on which the seal will be asked to function. There are several important shaft-related factors that determine a sealing lip's ability to establish and maintain a proper dynamic seal with the shaft.

These include the shaft material, surface finish, hardness, diameter, chamfer, and tolerances. Also critical are the shaft's motion (rotating, reciprocating, or oscillating) and speed, as well as the amount of eccentricity (shaft-to-bore misalignment and/or dynamic run-out) present.

MATERIAL

Most shafts for radial lip seal applications are made of mild steel, cast iron, or malleable iron. It is important that surface finish be carefully controlled to provide a proper sealing surface.

SURFACE FINISH SPECIFICATIONS

Shaft surface finish is defined by two main features: surface texture and machine lead. The first of these, surface texture, has a significant effect on the seal's elastomeric lip. As the Rubber Manufacturers Association (RMA) defines it in RMA OS 1-1, surface texture consists of three components: Ra, Rz, and Rpm.

The first step in determining Ra (roughness average) is to filter the profile obtained when a stylus moves across the surface of a shaft so that the areas of the profile above and below a certain line (called the *mean line*) are equal. The cutoff length (le) is the distance that the stylus travels across the shaft surface to obtain a set of readings. It is recommended that data from a total of five cutoff lengths (which equal one evaluation length, lm) be collected and analyzed after an appropriate pre-travel (lv) and post-travel (ln) of the stylus. The Ra is the average of the absolute value of the deviations from the mean line over the evaluation length (see ***Figure 111***). The RMA recommendation for Ra is 8-17 μin (0.20-0.43 μm) with a cutoff length of 0.010 inch.

RA

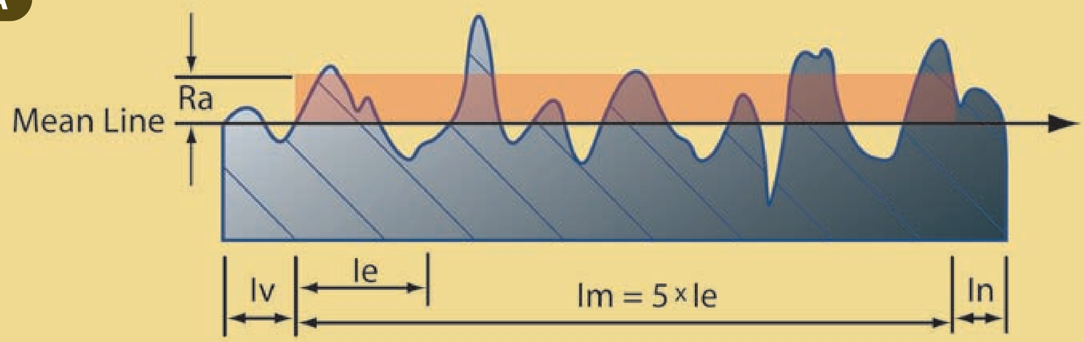


Figure 111: Roughness Average

But studies have shown that Ra numbers alone don't tell the whole story. In practice, a shaft surface finish that falls within recommended specifications can still cause seal failure. This is because, due to the averaging inherent to the Ra calculations, surfaces with widely varying profiles can all generate the same Ra value. An example of this phenomenon is illustrated in ***Figure 112***. Note the different capacities for seal lip abrasion represented by the three surface profiles. All have the same Ra value, yet each would affect a seal lip differently. Because Ra numbers alone may mislead, the RMA suggests using two additional parameters – Rz and Rpm – to more completely define surface finish.

As shown in ***Figure 113*** (next page), Rz is the average peak to valley height, or the average value of the greatest peak-to-

DIFFERENT PROFILES

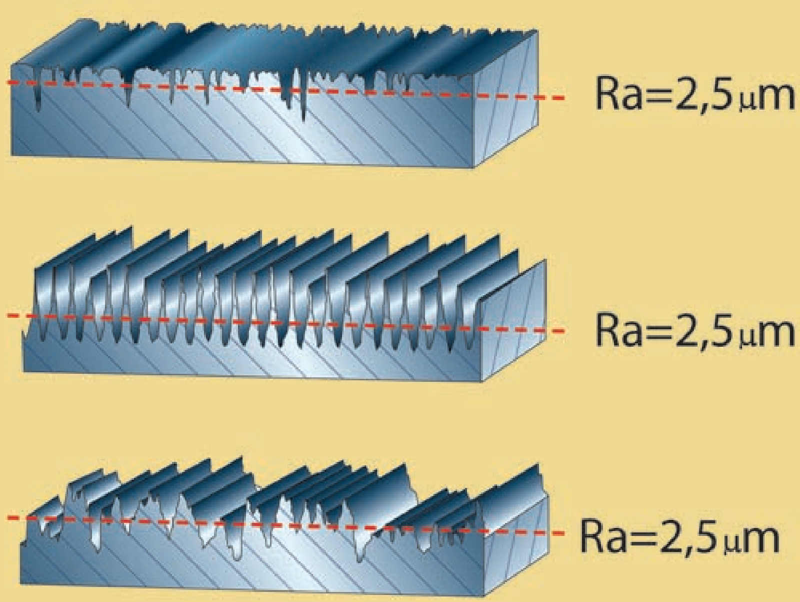


Figure 112: Varying Finishes, Same Ra

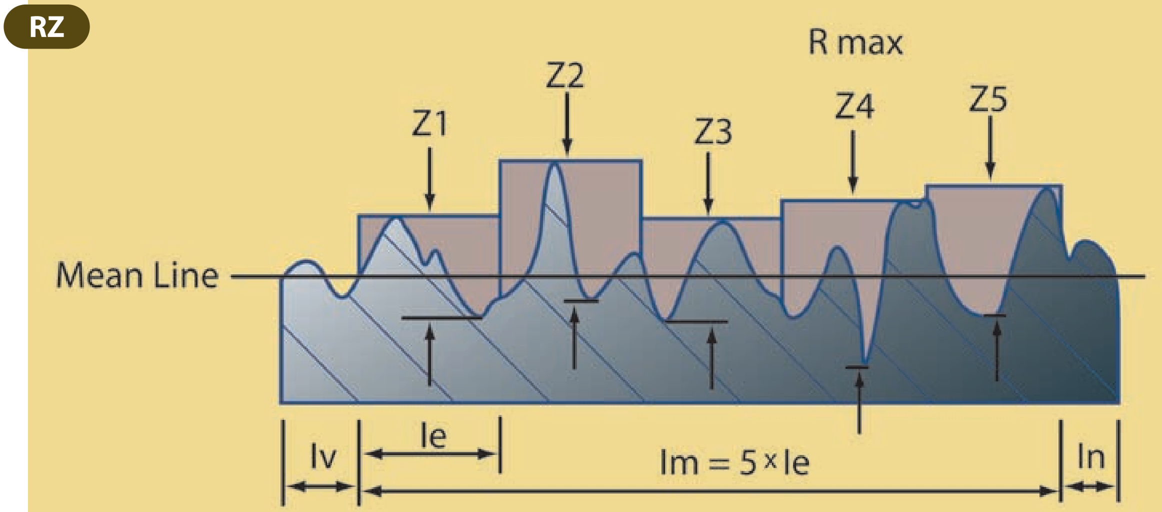


Figure 113: Average Peak-To-Valley Height

valley distances in five consecutive sample lengths (le) taken over the assessment length of the (filtered) profile.

As shown in ***Figure 114***, Rpm is the average peak to mean height, or the average value of the five highest peaks above the median in five consecutive sample lengths (le) taken over the assessment length of the (filtered) profile.

The RMA shaft surface texture recommendations are shown in ***Table 34***. These specifications are important because you want the shaft surface to be rough enough to hold pockets of oil to lubricate the seal lip without being so rough that lip damage will occur.

But proper numbers alone don't suffuce; even if the surface texture meets the specifications shown in ***Table 34***, the

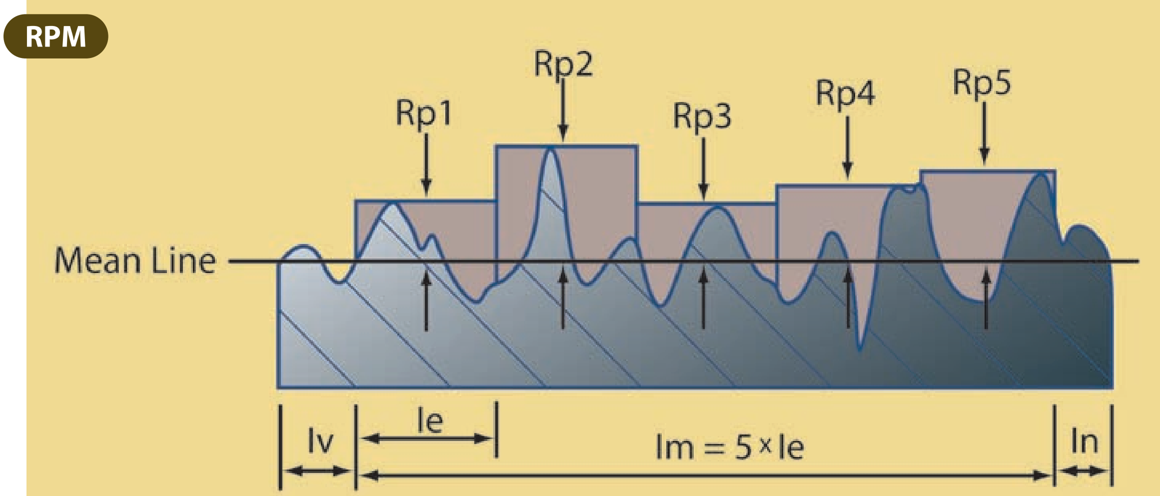


Figure 114: Average Peak-To-Mean Height

PARAMETER	RMA 0S-1 (1985) & OS-1-1 (1999)
Grinding Chatter	None allowed > 45 cycles (lobes)
(OOR) Out of Roundness	> 0.0050 mm (0.0002 in) at a max. of 2 lobes > 0.0025 mm (0.0001 in) at a max. of 7 lobes
Shaft Lead	Less than 0 +/- 0.05°
Ra (Surf. Roughness Avg.)	0.20-0.43 µm (8-17 µin)
Rz (Average Peak-to-Valley Height)	1.65-2.90 µm (65-115 µin)
Rpm (Average Peak-to-Mean Height)	0.50-1.25 µm (20-50 µin)
Instrument Parameters	0.25 mm (0.010 in) cutoff length 5µm (0.0002 in) 90° diamond stylus tip radius

Table 34: RMA Surface Texture Recommendations

presence of screw threads or spiral grooves on the shaft surface can create early leakage. Known as *shaft lead* (or *machine lead*), these threads or grooves result from relative axial movement of the finishing tool during the finishing operation. Improper shaft finishing can easily contribute to seal leakage and/or contaminant ingestion.

As detailed in RMA Handbook OS-1 (Shaft Finishing Techniques for Radial Lip Type Shaft Seals) and illustrated in ***Figure 115*** (next page), testing for the presence of shaft lead is a relatively simple process. With the shaft mounted in a holding chuck, the surface to be tested should be lightly coated with silicone oil. After ensuring that the shaft is level, a length of quilting thread (or, alternatively, unwaxed dental floss) is draped over the shaft. A one-ounce weight is suspended from the thread so as to create a string-to-shaft contact arc of 220 to 240 degrees.

The shaft is rotated at 60 RPM and the string is carefully observed to determine if axial thread movement exists. If movement is observed, the shaft is reversed to see if the string direction of motion is reversed. Thread movement combined with reversal when the shaft direction of rotation is reversed will betray the presence of lead; no movement means no lead is present. A full interpretation of thread movement is listed in ***Table 35*** (next page).

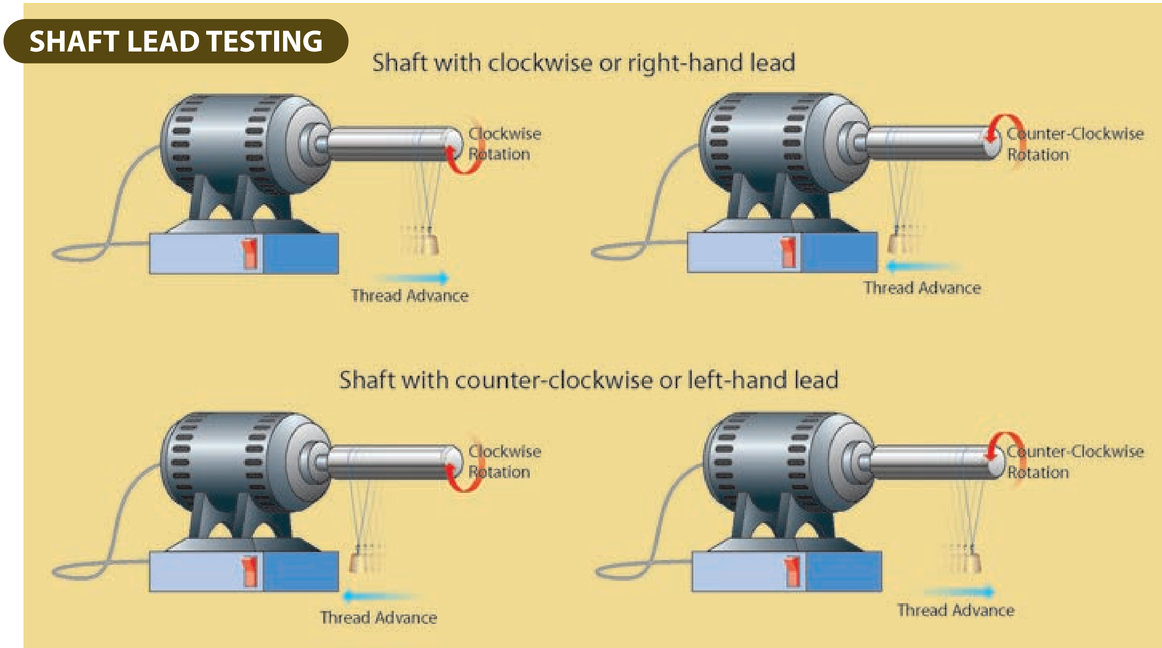


Figure 115: As Detailed in RMA Handbook OS-1

As important as surface texture is, the other shaft surface parameter – machine lead – is also critical. As previously shown in ***Table 34*** (page 119), the specification for shaft lead is 0 ± 0.05 degrees. There are a variety of shaft surface finishing methods, but keep in mind that many of these can

Observed Thread Movement **Clockwise Rotation**	**Observed Thread Movement** **Counter Clockwise Rotation**	**Interpretation**
Stationary	Stationary	No lead present.
Moves in same direction for both directions of shaft rotation. Remounting the shaft end-for-end reverses the direction of thread movement.		Tapered shaft. Unable to determine if shaft lead is present.
Moves in same direction for both directions of shaft rotation. Remounting the shaft end-for-end *does not* reverse the direction of thread movement.		Unlevel shaft. Unable to determine if shaft lead is present.
Thread moves away from the center for both directions of shaft rotation.		Crowned shaft. Unable to determine if shaft lead is present.
Thread moves toward the center for both directions of shaft rotation.		Cupped shaft. Unable to determine if shaft lead is present.
Thread moves from fixed end of shaft toward free end.	Thread moves from free end of shaft toward fixed end.	Right-hand (CW) lead present.
Thread moves from free end of shaft toward fixed end.	Thread moves from fixed end of shaft toward free end.	Left-hand (CCW) lead present.

Table 35: Shaft Lead Testing Interpretations

generate varying amounts of lead and are therefore problematic. Here is a closer look at many of these surface finishing methods, and the pitfalls to watch out for with each.

SURFACE FINISHING METHODS

Paper polishing by hand using emery cloth can generate lead if the paper is not held perpendicular to the axis of the shaft. Automated polishing can ensure the paper is perpendicular to the shaft, resulting in an acceptable, lead-free finish. An example of what a paper-polished surface would look like if viewed through a high-power microscope is shown in ***Figure 116***.

As shown in close-up in ***Figure 117***, honing can be problematic because it generates lead in the form of a crosshatched surface. This crosshatching creates channels through which fluid can escape.

Machine turning is unsuitable for shaft finishing because it always creates lead. If used, machine turning should always be coupled with an additional finishing operation that will eliminate lead. A close-up example of a machine-turned surface is shown in ***Figure 118***.

Unlike the preceding finishing methods, glass bead blasting doesn't generate shaft lead. Unfortunately, it doesn't remove it, either. Dimpling of the metal masks lead caused by machine turning without eliminating it. A close-up example of a glass bead blasted surface is shown in ***Figure 119***.

Similarly, metal peening also camouflages shaft lead caused by machine turning with dimples. Unlike with glass bead blasting, however, these dimples have tiny notches on one side. As shown in close up

SURFACE FINISHES

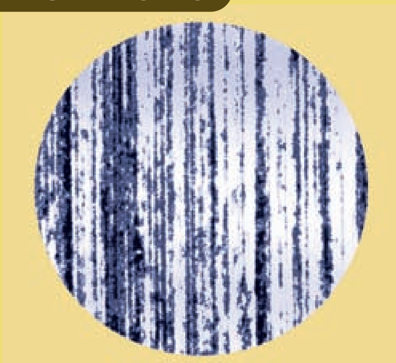


Figure 116: Paper Polishing

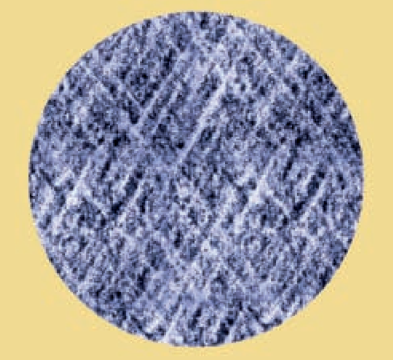


Figure 117: Honing

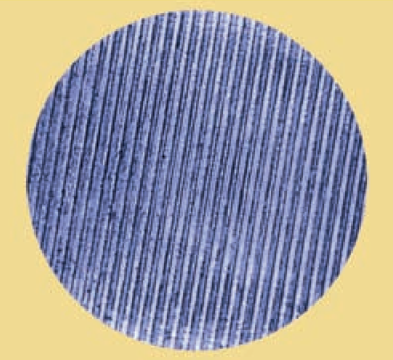


Figure 118: Machine Turning

Figure 119: Glass Bead Blasting

SURFACE FINISHES

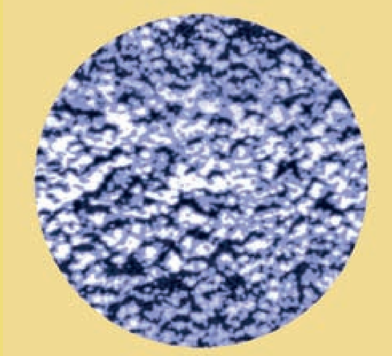


Figure 120: Metal Peening

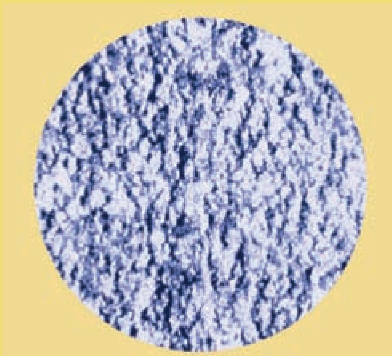


Figure 121: Stone Tumbling

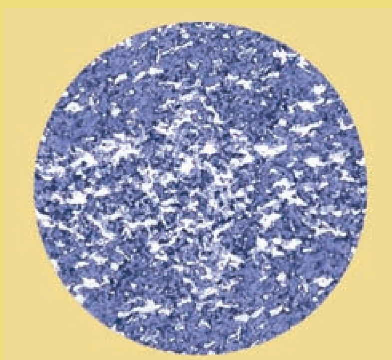


Figure 122: Roller Burnishing

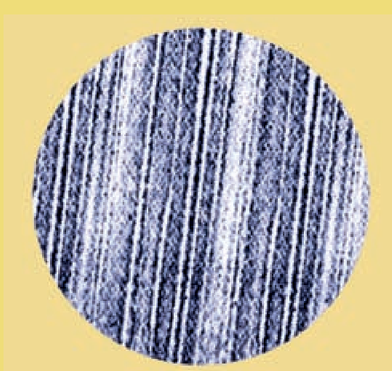


Figure 123: Through Feed Centerless Grinding

in ***Figure 120***, these notched dimples mimic lead.

Stone tumbling results in a uniform (rather than dimpled) shaft surface finish. However, stone tumbling still does not remove lead. A close-up example of a metal surface that has undergone stone tumbling can be seen in ***Figure 121***.

Roller burnishing doesn't generate lead, but it also doesn't remove it. Roller burnishing compresses – rather than eliminates – lead grooves. A close-up example of a roller-burnished surface is shown in ***Figure 122***.

As illustrated in ***Figure 123***, through-feed centerless grinding (also known as transverse grinding) can create shaft lead if the feeding process is too fast. Through-feed centerless grinding can produce *lobing* (out-of-round shafts).

Machine lapping involves the finishing of a shaft by rotating it between two rollers of varying speeds. One of the rollers utilizes an abrasive medium to wear the metal surface. The grit size of this medium can be chosen to give the proper surface texture, but machine lead can be created if the rollers are not properly aligned. In addition, machine lapping may not remove enough material to eliminate lead caused by turning. A close-up example of machine lapping is shown in ***Figure 124***.

Grit blasting is a process wherein abrasive particles (such as sand) are shot against metal to compress the surface and to leave behind tiny indentations capable of holding lubrication. Unlike glass bead blasting (which can only be used on unhardened shafts), grit blasting is possible with hardened surfaces. Applied correctly, grit blasting can eliminate mild to moderate (though not major) machine lead. A close-up view of a grit-blasted metal surface is illustrated in ***Figure 125***.

Sintering of compressed metal particles in a mold is sometimes used to produce porous shafts capable of holding lubrication. Though this lubrication-holding ability is helpful to the seal's lip, it comes at the expense of increased seepage and metal impurity, as well as reduced shaft strength. A close-up view of a sintered metal surface is shown in ***Figure 126***.

A drawn stamping involves use of a stamping die to form a metal sleeve. Many wear sleeves are produced in this way. Out-of-roundness is possible, and draw or work lines (such as are shown in ***Figure 127***) on the sleeve surfaces can cause leakage. It is recommended that the surface on which the seal rides be plunge ground or paper finished to form the proper surface texture without machine lead.

Plunge grinding has proven to be the most reliable finishing method for removing machine lead on rotating shafts. This is because plunge grinding eliminates any axial movement of the grinding wheel relative to the surface of the shaft. A mixed number (rather than whole number) RPM ratio (for example, 9.5 to 1) between the grinding wheel and the shaft (which should be rotating in opposite directions) is suggested to help prevent the introduction of spirals onto the shaft surface. Plunge grinding using mixed number ratios also greatly reduces the time required to achieve sparkout, the point where sparks are visible during the grinding operation. You must leave enough material on the shaft so that you can grind it to remove all traces of lead. If

SURFACE FINISHES

Figure 124: Machine Lapping

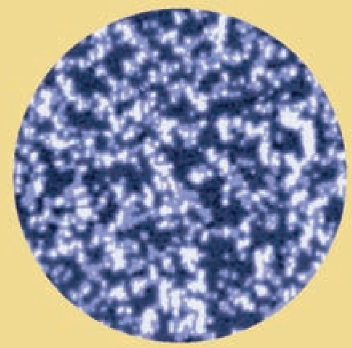


Figure 125: Grit Blasting

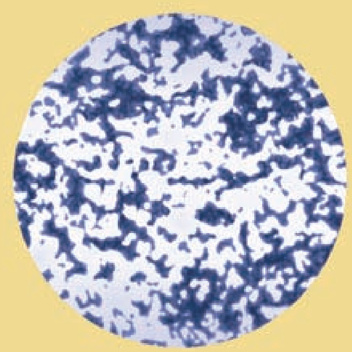


Figure 126: Sintering

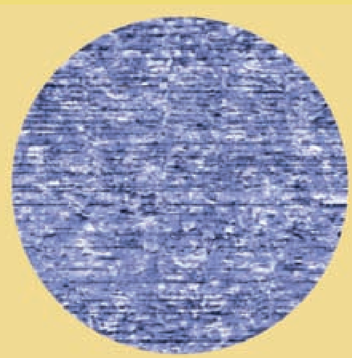


Figure 127: Drawn Stamping

all of these recommendations are followed, then the shaft surface should be free of lead. A grinding wheel with an 80-grit size will provide a surface finish of 8 to 17 µin. Ra per RMA recommendations. An example of a plunge-ground surface is shown in ***Figure 128***.

SURFACE FINISHES

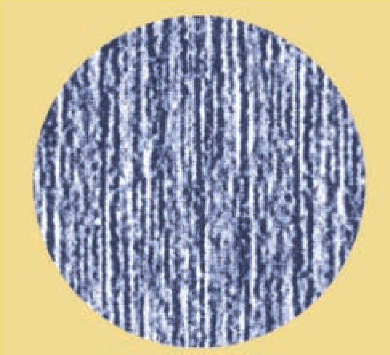


Figure 128: Plunge Grinding

As noted in ***Table 34***, there are two other interrelated concepts relating to shaft finishing. These are out-of-roundness and chatter. Out-of-roundness (OOR) pertains to the oval geometry of a lobed shaft. OOR can make it difficult for the sealing lip of a shaft seal to maintain proper contact with the shaft, particularly at elevated shaft speeds. OOR has two causes: deformation during assembly, and machining inconsistencies. When OOR becomes excessive (greater than 45 cycles or lobes as defined by the RMA) it is considered grinding chatter (also known as waviness). RMA specifications are that OOR be less than 0.0050 mm (0.0002 in.) at a maximum of 2 lobes, and less than 0.0025 mm (0.0001 in.) at a maximum of 7 lobes. ***Figure 129*** shows an example of a lobed, out-of-round shaft.

HARDNESS

The minimum hardness for the section of the shaft in contact with the seal is Rockwell C-30. A hardness of Rockwell C-45 may help prevent scratches, nicks, and handling damage.

DIAMETER

The diameter of the shaft has a tremendous impact on the performance of a shaft seal. At a constant speed, as shaft size increases, so does frictional drag on the seal's lip. This greater drag increases seal torque. ***Table 36*** shows how seal torque increases with increasing shaft diameter. Keep in mind that the data reflected in the following sections has been generated for a specific set of parameters to demonstrate general trends. Specific questions regarding suitability for a given application may necessitate further testing.

Greater drag also increases power consumption. ***Table 37***

OUT-OF-ROUNDNESS

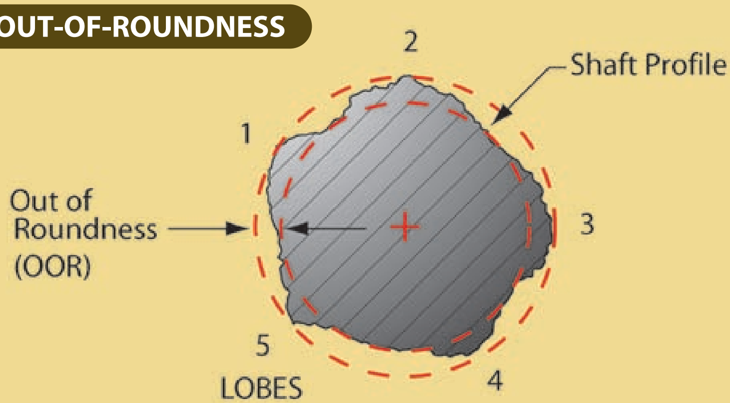


Figure 129: Lobed, Out-Of-Round Shaft

EFFECT OF SHAFT SIZE

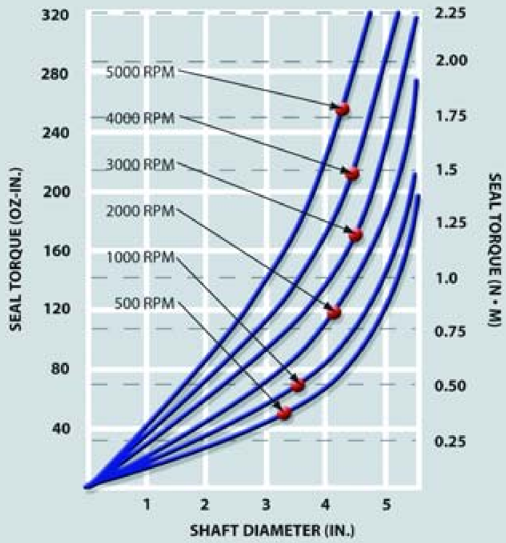


Table 36: Seal Torque vs. Shaft Diameter

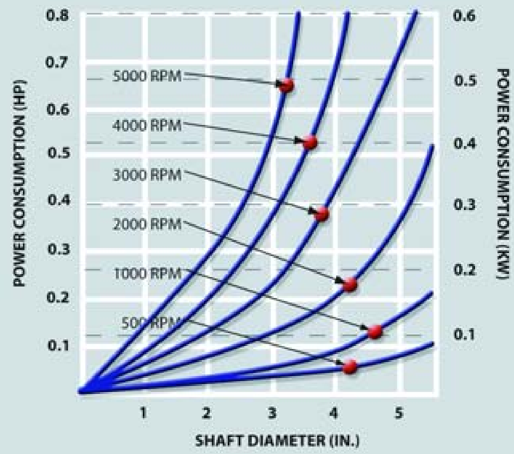


Table 37: Seal Power Consumption vs. Shaft Diameter

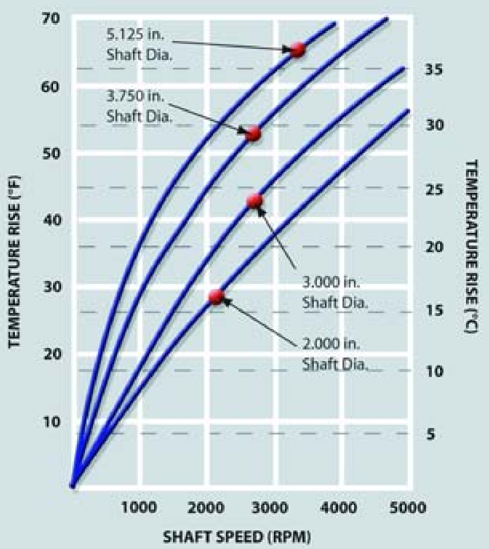


Table 38: Underlip Temperature vs. Shaft Diameter for Various Shaft Speeds

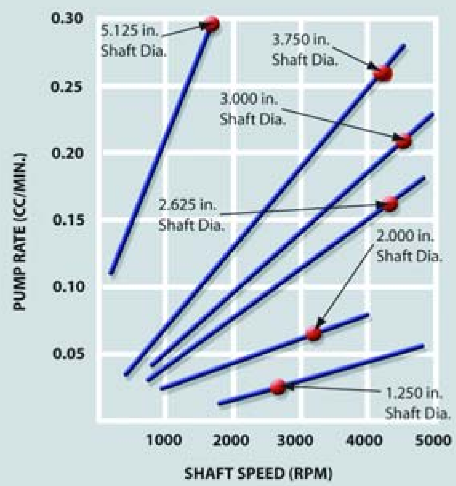
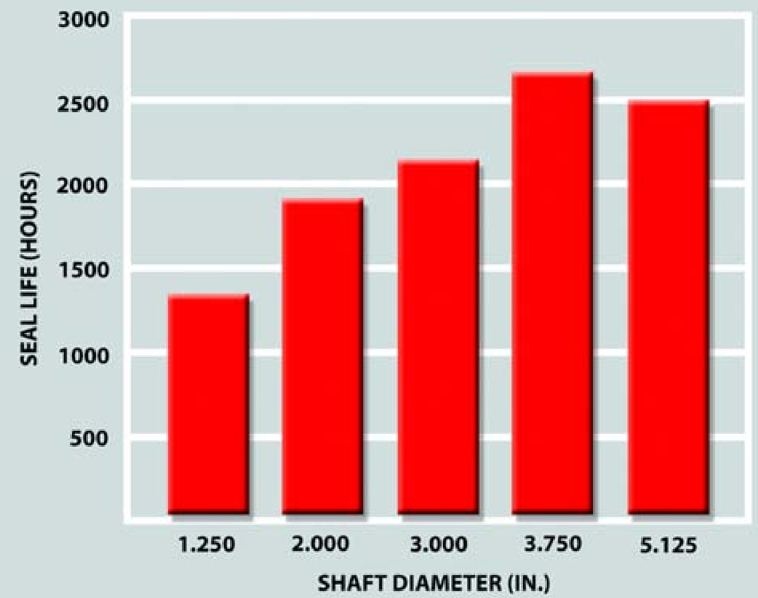


Table 39: Seal Pump Rate vs. Shaft Diameter for Various Shaft Speeds

Material: Nitrile (NBR); Shaft Size and Speed: Variable; Lubricant: SAE 30
DRO: 0.13 mm (0.005 inch); STBM: 0.13 mm (0.005 inch); Seal Cock: Zero
Pressure: Zero; Sump Level: Full; Sump Temperature: 93° C (200° F)

shows how seal power consumption increases with increasing shaft diameter. Increases in both seal torque and seal power consumption also increase a shaft seal's underlip temperature. ***Table 38*** shows how underlip temperature increases with increasing shaft diameter at various shaft speeds. Pump rate also increases as both shaft and seal size increase. ***Table 39*** shows how pump rate increases with increasing shaft diameter at various shaft speeds.



Material: Nitrile (NBR); Shaft Speed: 8.63 m/sec (1700 ft/min); Lubricant: SAE 30
DRO: 0.13 mm (0.005 inch); STBM: 0.13 mm (0.005 inch)
Seal Cock: Zero; Pressure: Zero; Sump Level: Centerline
Sump Temperature: 93° C (200° F); Cycle: 20 hours on, 4 off

Table 40: Seal Life vs. Shaft Diameter

Because increases in shaft diameter have such a profound impact on seal operating characteristics, the seal's cross-section is typically increased (as is lip-to-shaft interference level) as shaft size increases in order to ensure that seal life remains constant. ***Table 40*** shows how seal life can vary with increasing shaft diameter.

SHAFT CHAMFER

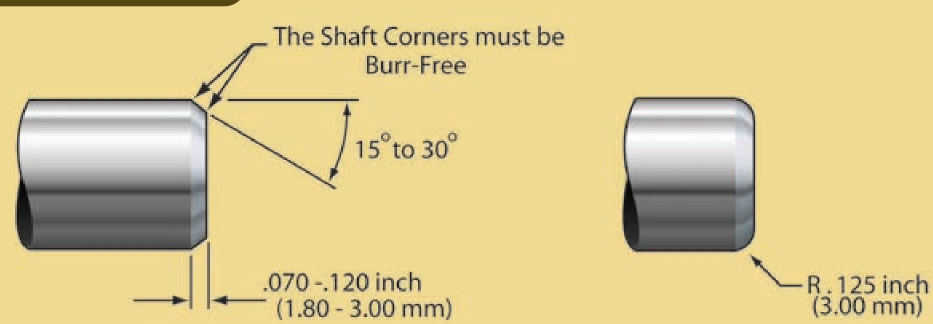


Figure 130: Recommended Chamfer Dimensions

CHAMFER

The shaft needs a burr- and nick-free chamfer to prevent damage to the sealing lip during installation. Recommended chamfer dimensions are shown in ***Figure 130***. In lieu of a chamfer, the lead-in corners of the shaft may also be radiused to 0.125 in. (3 mm).

Shaft Diameter (D_S) (mm)	Tolerance (mm)	Shaft Diameter (D_S) (inch)	Tolerance (inch)
$D_S \leq 100$	± 0.08	$D_S \leq 4.000$	± 0.003
$100 < D_S \leq 150$	± 0.10	$4.000 < D_S \leq 6.000$	± 0.004
$150 < D_S \leq 250$	± 0.13	$6.000 < D_S \leq 10.000$	± 0.005
		$D_S > 10.000$	± 0.010

Table 41: Shaft Tolerance Guidelines

TOLERANCES

Table 41 lists recommended shaft tolerance guidelines in both inches and millimeters.

MOTION

Shaft motion takes one of three forms: rotation, reciprocation, or oscillation. For rotating shaft applications (see ***Figure 131***), the direction of rotation is very important to seal design; a shaft may rotate only clockwise (CW), only counterclockwise (CCW), or be birotational (variously rotating both CW and CCW). It's also important to note whether the rotation is continuous or intermittent.

The speed at which a shaft rotates is noted in revolutions per minute, or RPM. When designing a shaft seal, it's important to know the shaft's normal RPM (the speed at which it rotates most of the time), as well as any temporary acceleration it may undergo (up to its maximum RPM).

The surface speed of the shaft is dependent on RPM and the shaft diameter. Surface speed is typically expressed in feet per minute (FPM) or meters per minute (MPM). ***Table 42*** (next page) can assist you in determining the relationships between shaft diameter, MPM, FPM, and RPM. To find either the MPM or FPM for a given shaft diameter and RPM, lay a straight edge over the known values on the Shaft Diameter and RPM lines. The straight edge crosses the MPM/FPM line at the corresponding speed. In

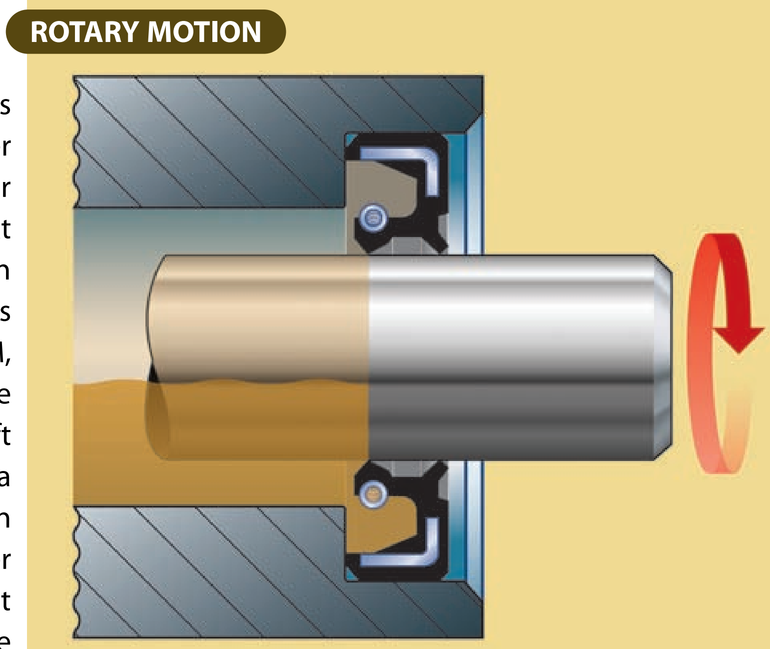


Figure 131: Shaft Seal in Rotary Application

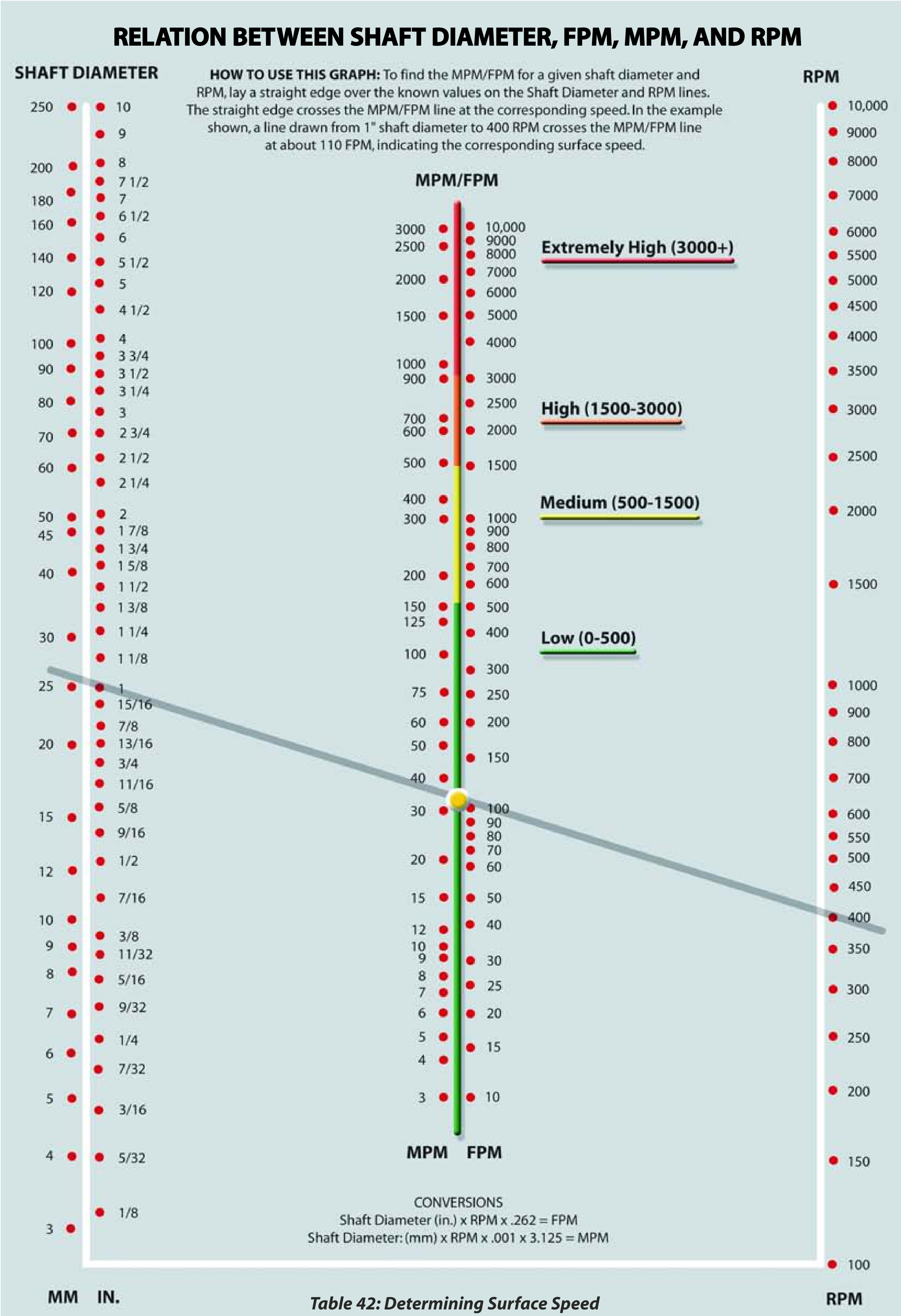


Table 42: Determining Surface Speed

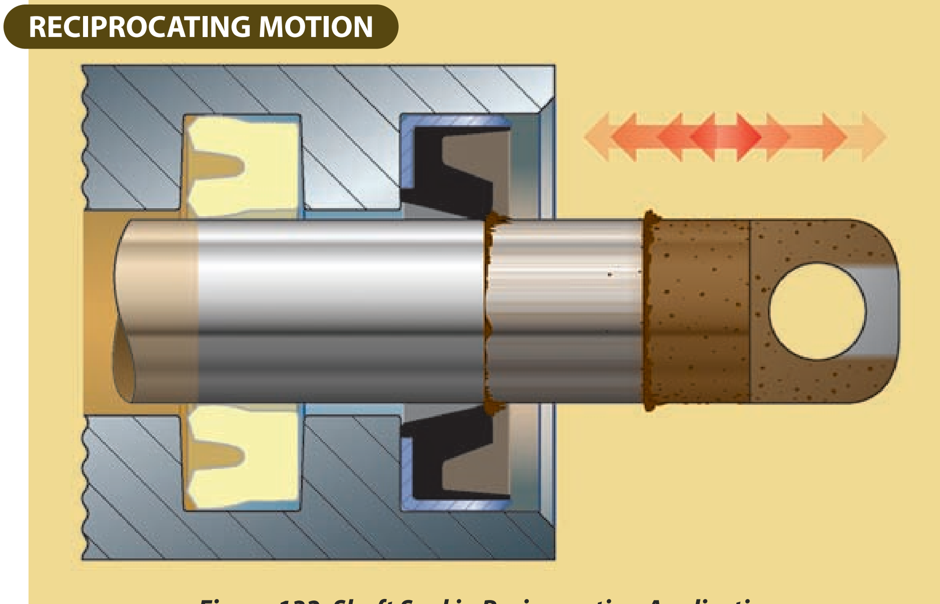


Figure 132: Shaft Seal in Reciprocating Application

the example shown, a line drawn from 1" shaft diameter to 400 RPM crosses the MPM/FPM line at about 110 FPM, thus indicating the corresponding surface speed of the shaft.

The motion of a reciprocating shaft (one that moves back and forth, see ***Figure 132***) is chiefly defined by two variables: 1) the length of each stroke (defined as the total movement in one direction and expressed in inches or millimeters), and 2) the number of cycles per minute (the number of times the shaft makes a "round trip," i.e. one stroke in and one stroke out).

An oscillating shaft rotates, but rather than going round and round endlessly (as with a true rotary shaft), an oscillating shaft rotates back and forth within an arc defined by the degrees of rotation (see ***Figure 133***). An oscillating shaft is defined by 1) the degrees of this arc, and 2) the cycles per minute (the number of times the shaft moves across this arc and returns to its beginning position).

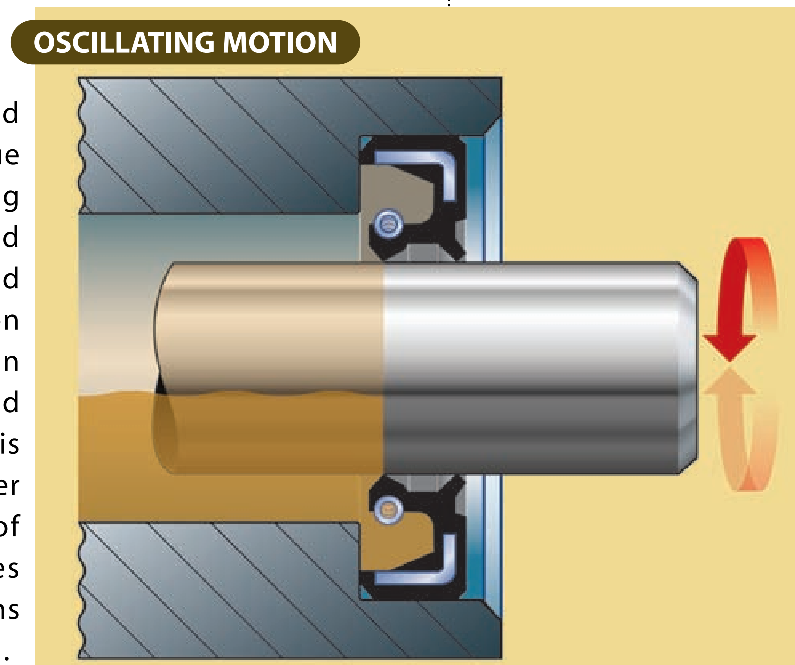


Figure 133: Shaft Seal in Oscillating Application

STBM

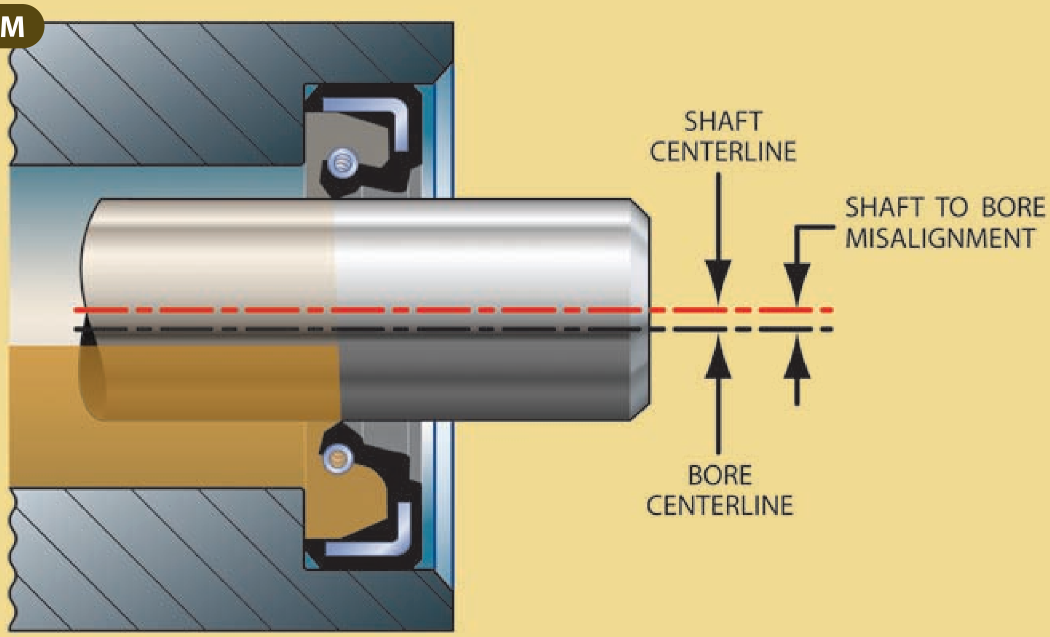


Figure 134: Shaft-To-Bore Misalignment

ECCENTRICITY

Good shaft seal design must also take into account any and all shaft eccentricity. Eccentricity may manifest itself as shaft-to-bore misalignment (STBM) and/or dynamic runout (DRO). As shown in ***Figure 134***, *shaft-to-bore misalignment* is the amount (expressed as a Total Indicator Reading, TIR, in inches or millimeters) that the shaft center is offset relative to the bore center. A static measurement taken with the shaft at

DRO

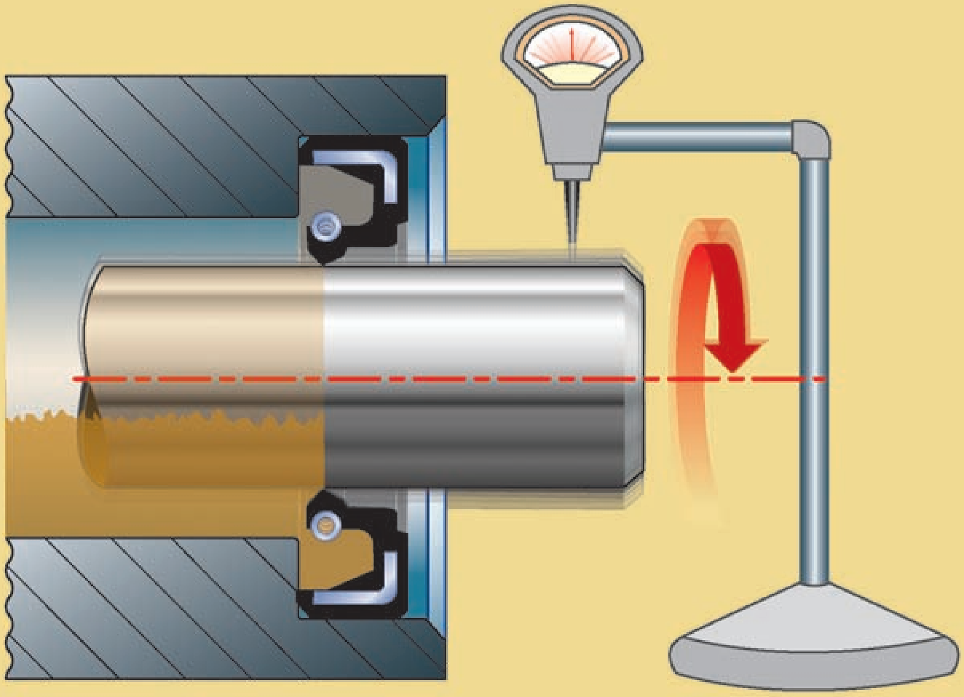


Figure 135: Dynamic Runout

rest, STBM almost always exists to some degree but should never be more than 0.010 in. (0.25 mm) TIR.

As ***Figure 135*** shows, *dynamic runout* is the amount (expressed as a TIR in inches or millimeters) that the shaft's sealing surface does not rotate around the true center. A dynamic measurement taken by applying an indicator to the side of the shaft as it slowly rotates, DRO should never exceed 0.010 in. (0.25 mm) TIR. As shown in ***Table 43***, seal life decreases as DRO increases. ***Table 44*** lists the maximum eccentricity values (STBM plus DRO) in both inches and millimeters for most common seals.

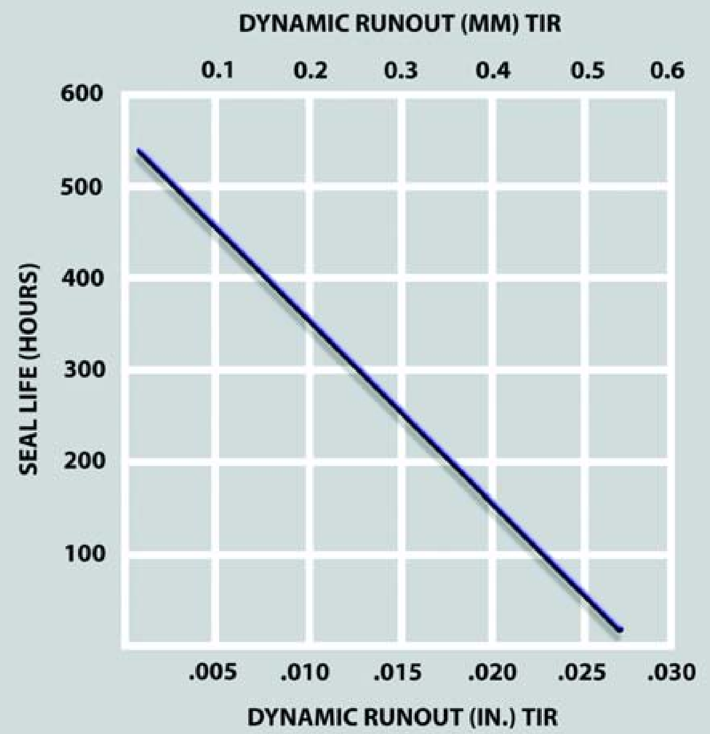


Table 43: Seal Life vs. Dynamic Runout

Maximum Shaft Speed (ft / min)	Maximum Shaft Speed (m / sec)	Maximum Total Eccentricity (inch)	Maximum Total Eccentricity (mm)
75	0.38	0.025	0.635
150	0.76	0.020	0.508
350	1.78	0.018	0.457
700	3.55	0.015	0.381
1000	5.08	0.013	0.330
1500	7.61	0.010	0.254
1800	9.14	0.009	0.228
2000	10.16	0.008	0.203
3000	15.23	0.007	0.178

Table 44: Maximum Total Eccentricity (STBM Plus DRO)

The Bore.

Good design doesn't stop with the shaft seal and the shaft. Due consideration must also be given to the housing bore. There are four important bore-related factors that can impact a shaft seal's ability to establish and maintain a static O.D. seal with the bore. These factors are bore material, surface finish, chamfer, and diameter tolerances.

MATERIAL

Most bores in contact with shaft seals are made of ferrous (iron-containing) materials such as steel or cast iron. Both steel and cast iron bore surfaces are compatible with either metal or rubber-covered O.D. seals. Bores made of softer alloys (such as aluminum) will typically require use of a rubber-covered O.D. because steel materials expand less than aluminum materials. For example, when subjected to high temperatures, an aluminum housing will expand more quickly than a shaft seal case made of carbon steel.

This disparity, known as *differential thermal expansion*, can loosen what was initially a tight fit between case and housing. As shown in ***Figure 136***, a looser fit can allow fluid to escape between the case and the bore (around the seal's

"Good design doesn't stop with the shaft seal and the shaft. Due consideration must also be given to the housing bore."

DIFFERENTIAL THERMAL EXPANSION

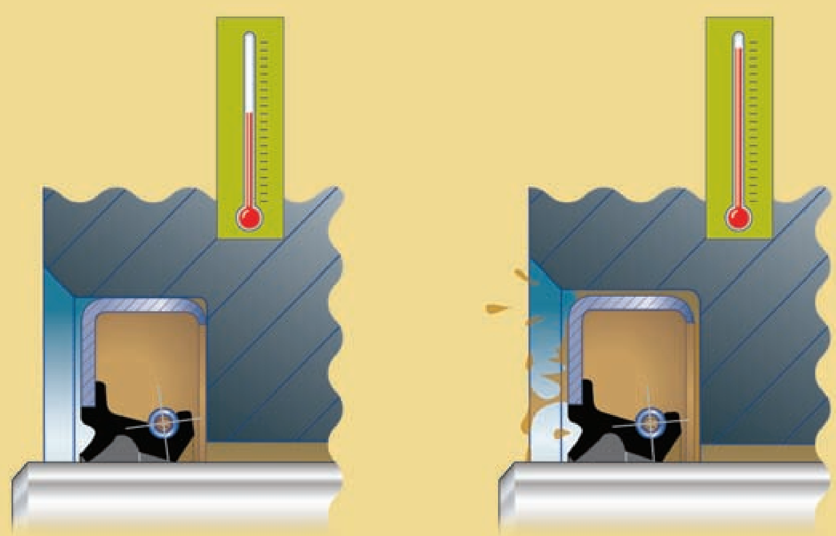


Figure 136: Development of a Leak Path

Material	Type	Coefficient of Thermal Expansion Fahrenheit	Celsius
Steel	Ferrous	7 µin / in-°F	12.6 µm / m-°C
Aluminum	Non-Ferrous	12.7 µin / in-°F	22.9 µm / m-°C
Nitrile	Non-Ferrous	62 µin / in-°F	111.6 µm / m-°C

Table 45: Coefficients of Thermal Expansion

O.D.), resulting in seal failure. In extreme cases, the seal case can actually turn in the bore. Because rubber has a higher coefficient of thermal expansion (expands more quickly) than ferrous materials, a rubber-covered O.D. will actually get tighter in the bore and more resistant to leaks as temperatures increase. Differential thermal expansion is also why plastic or nylon bores are not as common as metal bores; plastic and nylon expand much more quickly than metal, making it very difficult to keep a seal with a metal O.D. in place. And because a metal O.D. can easily damage plastic surfaces during installation, use of a plastic or nylon bore necessitates use of a shaft seal with a rubber-covered O.D. Metal O.D. seals can also damage aluminum housings during installation. The differing coefficients of thermal expansion for steel, aluminum, and nitrile rubber are shown in ***Table 45***.

Keep in mind that differential thermal expansion also applies in low temperatures, which cause different metals to contract at different rates. This differential contraction can also result in the creation of a leak path between the case and the bore. Designs in which both the housing bore and the seal case are made of the same metal can help eliminate concerns about expansion or contraction. When using the same metal isn't possible, a seal with a rubber covered O.D. may be required to maintain a proper fit. See ***page 87*** for more information on the various rubber O.D. designs.

SURFACE FINISH

The optimal surface finish for the bore largely depends on the type of shaft seal in use. Generally speaking, seals with a rubber-covered O.D. can handle a rougher surface finish than metal O.D. seals, but a surface that's too rough may damage the O.D. or prevent formation of a tight static seal. On the other hand, a surface that's too smooth may make it more difficult for a seal with a rubber-covered O.D. to stay in place.

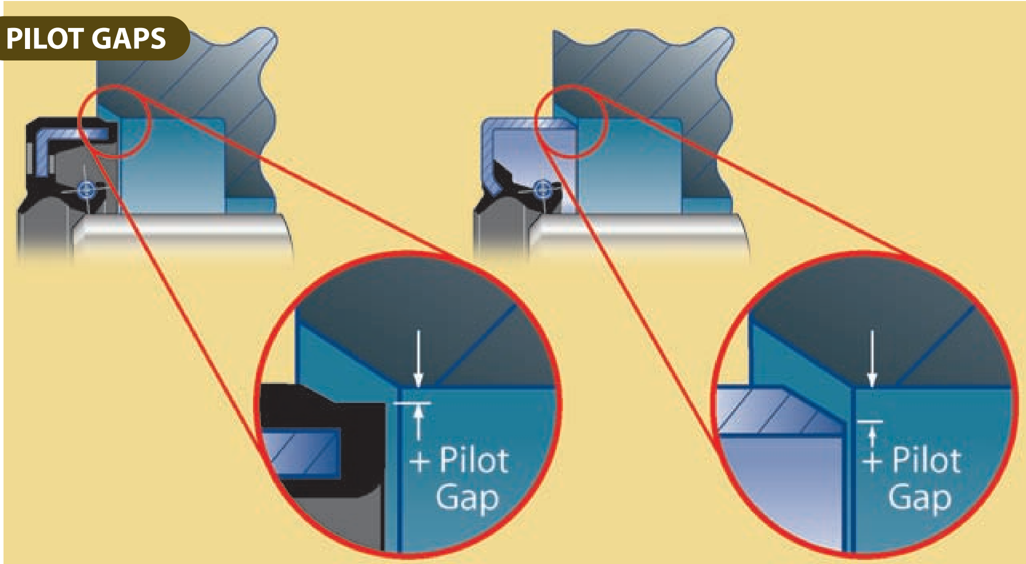


Figure 137: Installation Chamfers

While there is no accepted standard for minimum bore roughness, we recommend a roughness of at least 2.03 μm (80 μin.) Ra. If the bore exceeds 2.54 μm (100 μin.) Ra, then a rubber O.D. seal or use of O.D. sealant is recommended. Bore roughness should never exceed 3.175 μm (125 μin.) Ra.

CHAMFER

The bore needs a burr- and nick-free chamfer to prevent damage to the seal's outside diameter during installation. A proper chamfer creates what is known as a *pilot gap*. The pilot gap is one half the difference between the diameter of the leading edge of the seal O.D. chamfer and the housing bore I.D. A positive gap ensures that the seal O.D. makes proper

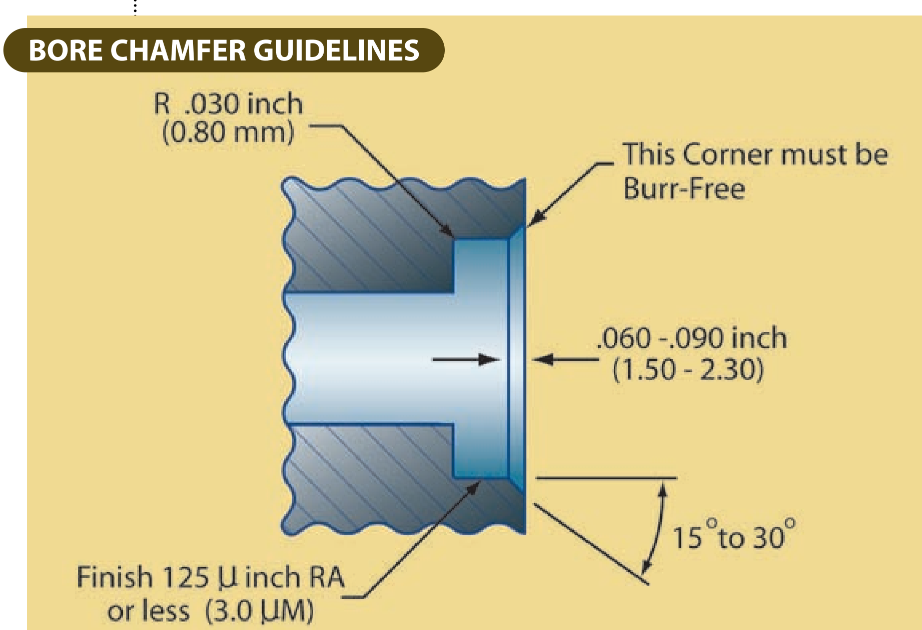


Figure 138: Recommended Angle and Depth

Standard	Seal Width (W)	Housing Bore Depth	Chamfer Length	Max. Bore Corner Radius	Surface Finish
ISO 6194/1	W ≤ 10 (mm)	W +0.9 (mm)	0.70-1.00 (mm)	0.50 (mm)	>3.2μm RA
	W > 10 (mm)	W +1.2 (mm)	1.20-1.50 (mm)	0.75 (mm)	
RMA	All values of W	W +0.016 (in)	0.06-0.09 (in)	0.047 (in)	>125μin RA

Table 46: ISO/RMA Bore Recommendations

contact with the bore chamfer prior to actually entering the bore, thus facilitating proper installation. A negative (or nonexistent) gap would create excessive interference and prevent the seal O.D. from aligning properly with the bore chamfer, thus hindering proper installation and likely causing damage to the seal O.D. Examples of positive pilot gaps are shown in ***Figure 137***.

Chamfering also reduces the force required for proper assembly, helps prevent seal misalignment (cocking), and minimizes case distortion. As shown in ***Figure 138***, the chamfer should be between 0.060 in. and 0.090 in. long and angled from 15° to 30°. Recessed bores should also have a 0.030 in. radius at the back of the bore. Both the ISO and RMA recommendations regarding bore depth, chamfer length, and corner radius are listed in ***Table 46***.

TYPE

There are two types of bores: straight bores and counter bores. Straight bores have no built-in shoulder to help seat the seal, making installation trickier. With straight bores, the installation tool needs a flange to position the seal at the

STRAIGHT BORES

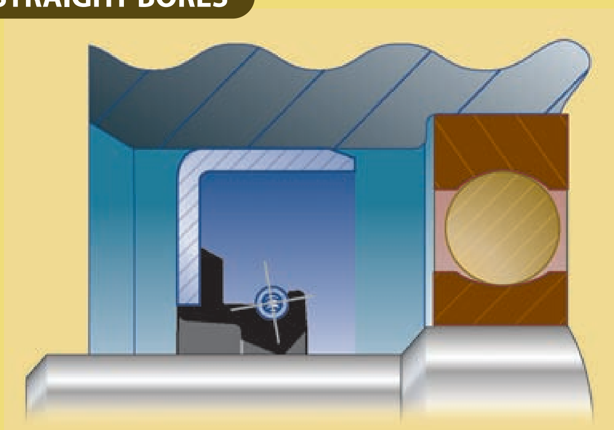


Figure 139: Non-Flanged Seal

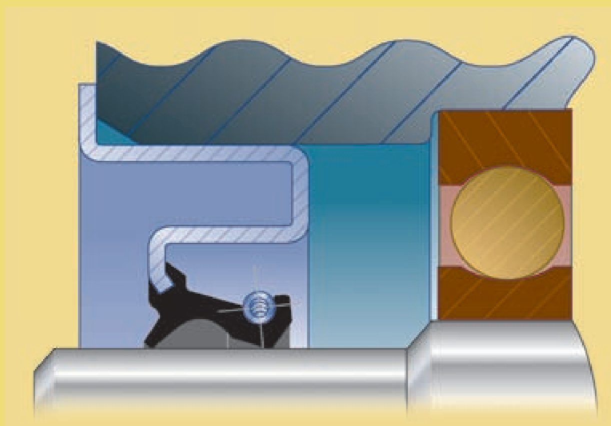


Figure 140: Flanged Seal

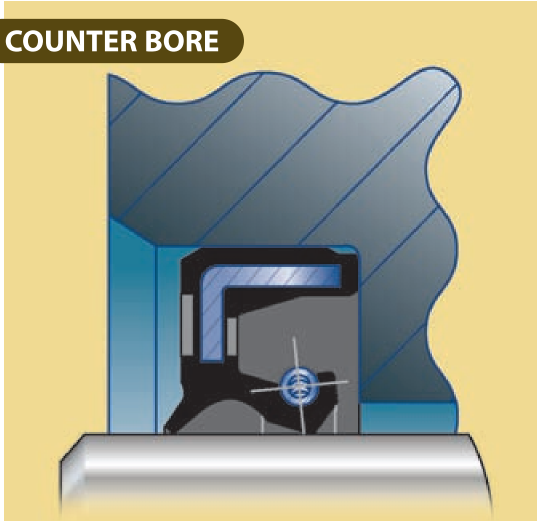


Figure 141: Nose Gasket

proper location. Alternatively, the seal itself might be designed with a flanged case to facilitate proper positioning. ***Figure 139*** shows a non-flanged seal being installed into a straight bore. ***Figure 140*** shows a flanged seal installed.

Counter bores have a shoulder against which the seal seats, making installation less difficult. By effectively stopping the seal at a particular point relative to the shaft, a counter bore controls the placement of the primary sealing lip on the shaft surface. Shaft seals with a rubber nose also allow the formation of a face seal at the counter bore. ***Figure 141*** shows an example of a nose gasket installed into a counter bore.

DIAMETER TOLERANCES

Table 47 lists the recommended bore inside diameter tolerance guidelines for ferrous materials in both inches and millimeters.

Bore Diameter (D_b) (mm)	Tolerance (mm)	Bore Diameter (D_b) (inch)	Tolerance (inch)
$D_b \leq 50$	+ 0.039 - 0.00	$D_b \leq 3.000$	± 0.001
$50 < D_b \leq 80$	+ 0.046 - 0.00	$3.000 < D_b \leq 7.000$	± 0.0015
$80 < D_b \leq 120$	+ 0.054 - 0.00	$7.000 < D_b \leq 12.000$	± 0.002
$120 < D_b \leq 180$	+ 0.063 - 0.00	$12.000 < D_b \leq 20.000$	± 0.003
$180 < D_b \leq 300$	+ 0.075 - 0.00	$20.000 < D_b \leq 40.000$	± 0.004
$300 < D_b \leq 440$	+ 0.084 - 0.00	$D_b > 40.000$	± 0.006

Table 47: Bore I.D. Tolerances

How a Shaft Seal Works.

As noted in "Anatomy of a Shaft Seal" *(page 77)*, shaft seals have a large number of variable features. The ways in which these variables work together to form a successful seal is not simple. On the contrary, a complete understanding of how a shaft seal functions is not arrived at easily. What follows, however, is an overview of the major principles at work in shaft sealing.

Once installed, a typical shaft seal is defined by two sealing surfaces. In order for the seal to perform successfully, both of these surfaces must function properly. The first is a tight static seal formed as a result of contact, or *interference*, between the seal's outside diameter (O.D.) and the housing bore. The seal O.D. is designed to be slightly larger than the bore, typically .004" to .008" larger for metal O.D. seals and .006" to .012" larger for rubber-covered O.D.s. (The exact amount of interference depends on the bore size.) This difference between seal size and bore size ensures a tight pressfit that leaves no room for leakage around the O.D. The tightness of this fit also keeps the seal retained in the bore.

"A complete understanding of how a shaft seal functions is not arrived at easily."

RADIAL FORCE

The second, dynamic sealing surface forms between the elastomeric lip and the rotating shaft. Use of a seal whose inner lip diameter is slightly smaller than the shaft diameter ensures that the sealing lip will be expanded (stretched outward) by the shaft upon installation. The interaction of 1) the lip's inherent *beam force* and 2) this outward stretching (*hoop force*) plus 3) the hoop force generated by the spring results in a total *radial force* (also known as *load*) between the lip and the shaft. As shown in ***Figure 142***, the radial force generated when the seal is installed is distributed on the shaft beneath the sealing lip.

RADIAL FORCE

Figure 142: Contact Pressure Distribution Pattern

The pressure distribution shown in ***Figure 142***—a greater pressure gradient on the oil side than on the air side—is a direct result of the steeper angle on the oil side of the lip. Tests have shown that this angular difference has a lot to do with the effectiveness of a seal. Here's how it all seems to work: The shaft surface is plunge ground to meet RMA standards. The ground shaft surface will abrade away a very thin layer of rubber from the seal tip that is contacting the rotating shaft. If the shaft finish is too smooth, then lip abrasion will not occur. If the shaft finish is too rough, then the seal lip will experience excessive wear. The seal lip material must be properly formulated to ensure the formation of microscopic pores (known as *microasperities*) on the seal lip's wear path. If the material has not been formulated properly, microasperities will not form and the seal wear path will appear relatively smooth when viewed with a high powered microscope (see ***Figure 143***).

MICROASPERITIES

Figure 143: Comparison of Lip Wear Paths

MICROASPERITIES

Once formed, microasperities are advantageous because they serve as reservoirs to hold lubrication that prevents further lip wear. They also contribute to an inherent pumping capability. ***Figure 144*** shows the microasperities as they might look if seen through a glass shaft at rest.

As microasperities form on the lip's wear path, the plunge ground surface on the shaft that is under the seal lip is worn smooth. A smooth path or wear band is created around the shaft in the circumferential direction. As the shaft rotates, the contact point of the lip is sheared in the circumferential direction. The microasperities are pulled so that they are directionally oriented at an angle to the shaft, and they are elongated, thus creating tiny helices.

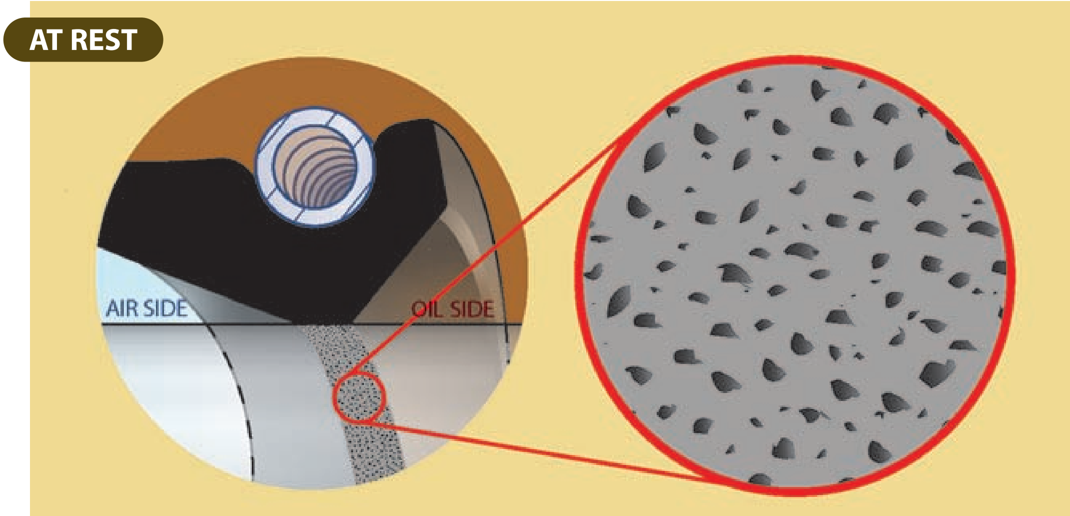


Figure 144: Microasperities As Seen Through a Stationary Glass Shaft

Because the oil side angle of the seal lip is larger than the air side angle, the helices on the oil side of the contact band are shorter with a larger helix angle and a larger pressure gradient than the helices on the air side of the contact band. Because of these geometrical features, the pumping activity of the helices on the air side is greater than the pumping activity of the helices on the oil side of the contact band. The net result is an in-pumping effect that prevents oil leakage from the oil reservoir (sump).

Figure 145 shows the directional orientation of the helices as they might look if seen through a rotating glass shaft. It's important to note that the pumping action of most shaft seals is very directional (e.g. back toward the seal's fluid side). It is therefore imperative that shaft seals be installed in the proper direction. Installing a seal backward, such that the air

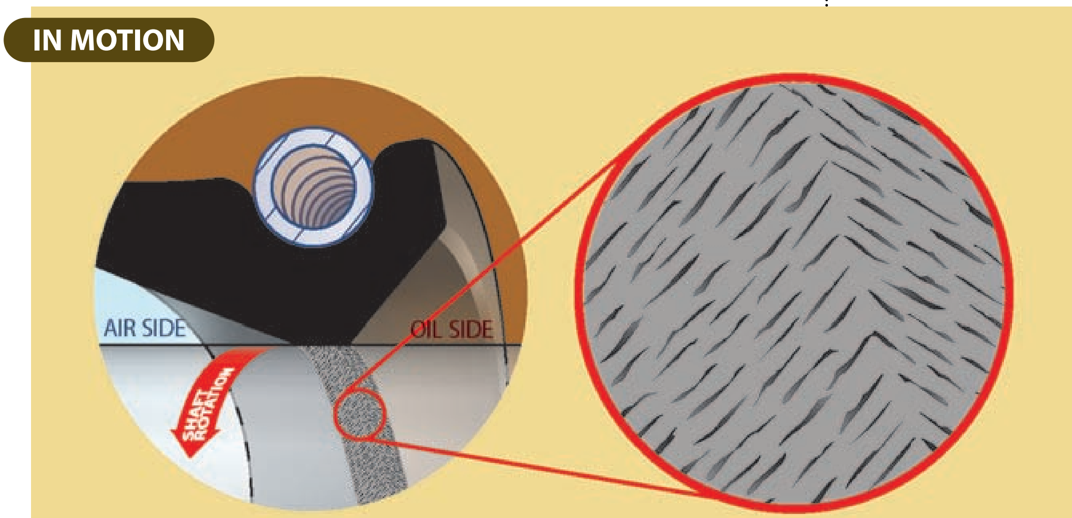


Figure 145: Microasperities Forming Helices As Seen Through a Rotating Glass Shaft

BENEATH THE LIP

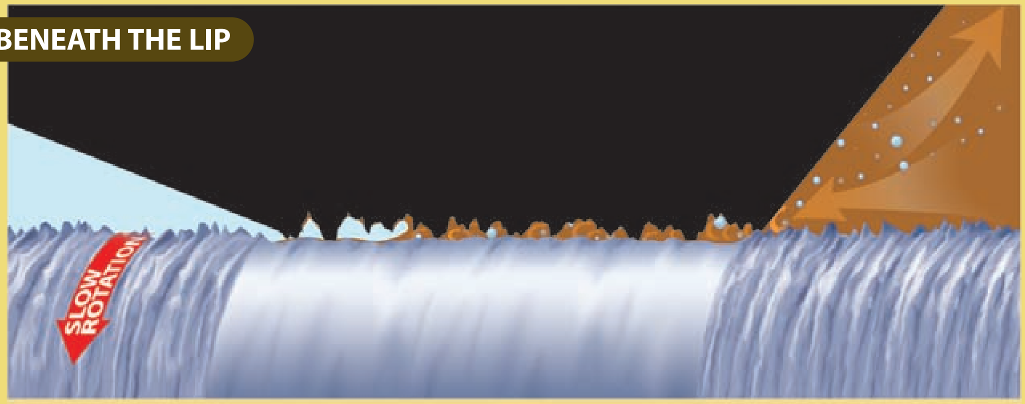


Figure 146: Microasperities Assist Underlip Fluid Pumping

side angle is facing the fluid, will result in immediate leakage due to the pumping of fluid out of the sump.

MENISCUS

As a result of the surface tension present between the air, the fluid, and the shaft, a curved meniscus develops at the meeting point between air and fluid (on the air side of the sealing lip). Studies have been conducted to attempt to numerically model the placement of this meniscus in various operating conditions. These studies indicate that the precise location of the meniscus can be a function of shaft speed. ***Figure 146*** offers an enlarged and exaggerated look at how lip microasperities affect fluid flow beneath the sealing lip and at the probable location of the meniscus.

Figure 147 shows the meniscus shifted toward the fluid side (*ingested*) as a result of increasing shaft speed. Hydrodynamic theory predicts that the underlip film thickness increases even as in-pumping continues. But even if shaft speed continued to increase, and the meniscus continued to move toward the fluid side, it would seem logical that an imbalance between air and fluid under the lip could develop that would

INGESTED MENISCUS

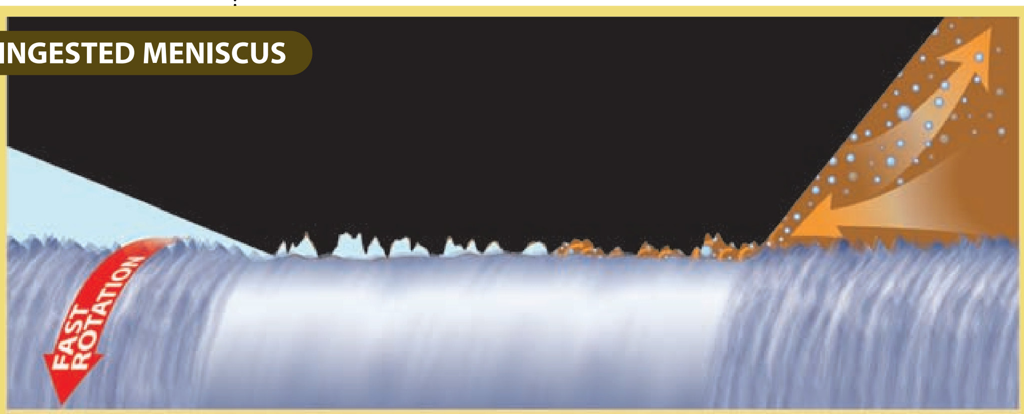
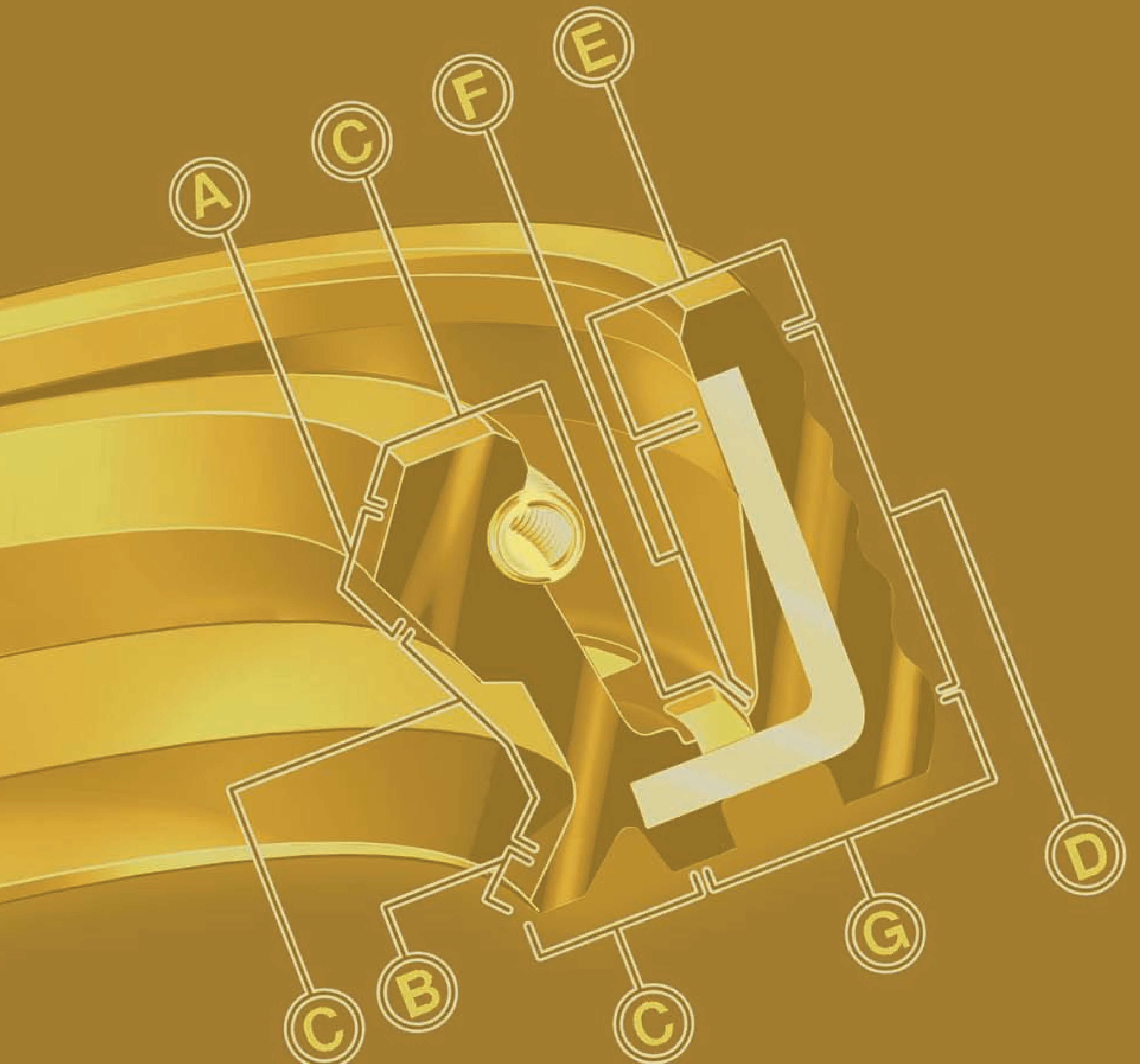


Figure 147: Meniscus Shifted Inward Due To Increasing Shaft Speed

result in the lip making direct (unlubricated) contact with the shaft. This would lead to unwanted lip wear and premature seal failure. Fortunately, however, oil is retained in the microasperities, and this ensures that the portions of the lip contacting the shaft remain lubricated.



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How a Shaft Seal Is Made.

A shaft seal's ability to function effectively is the end result of not only good design, but also meticulous manufacturing. Proper mold making, case stamping, and rubber mixing are necessary prerequisites to molding a shaft seal. The molding itself must be carefully monitored to ensure quality and consistency. After molding, a typical seal must still undergo trimming, spring insertion, and finishing before it's ready for shipment and use.

Figure 148 (pages 144 and 145) is a schematic illustration of the ways in which these individual manufacturing processes come together to produce a high quality shaft seal. Numbers in parentheses refer to specific steps along the way.

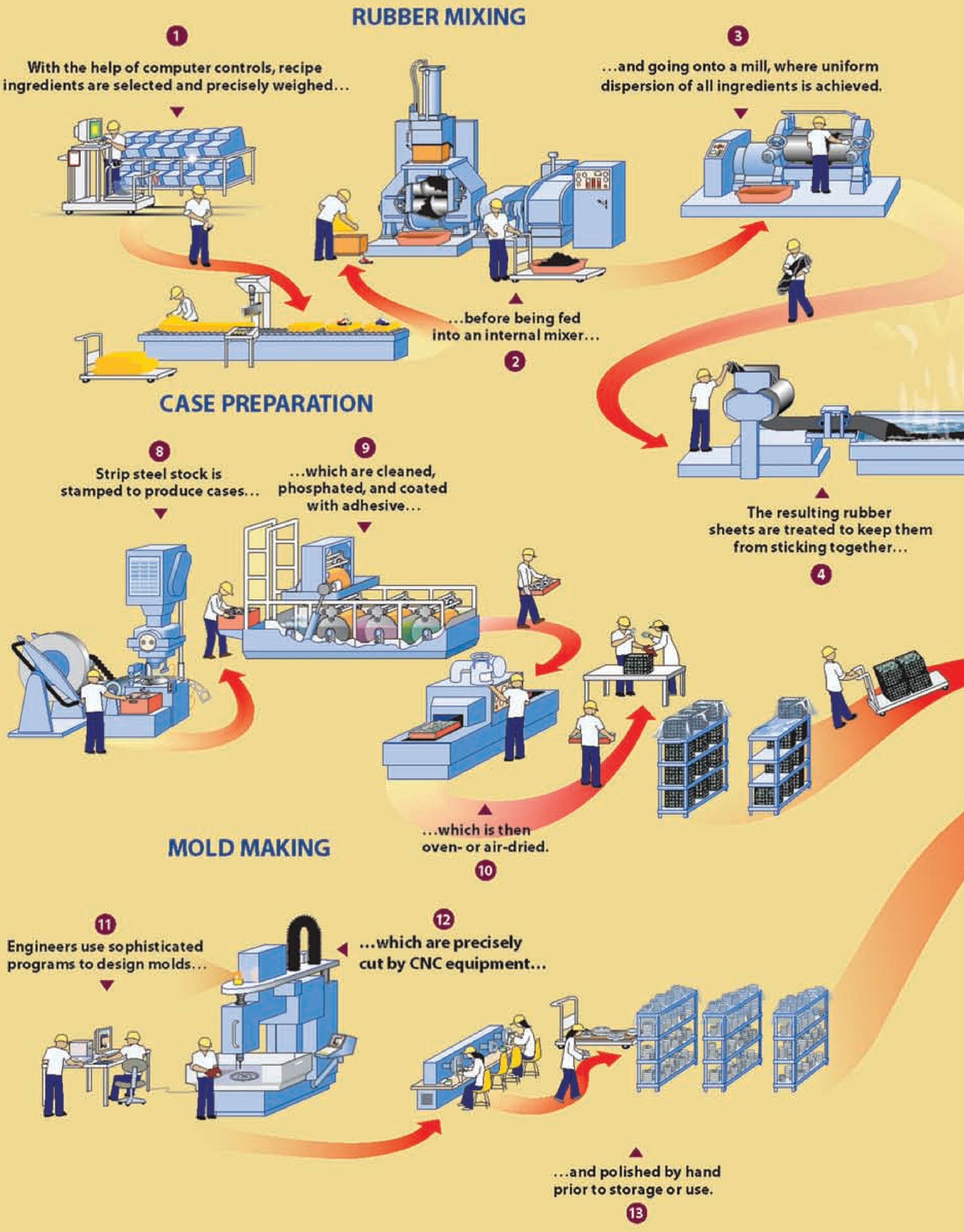
PREPARATION

Rubber recipe ingredients are carefully selected and precisely weighed (1). Thanks to sophisticated computer controls, the process cannot move forward unless the right amounts of the correct ingredients are chosen. Once assembled, the ingredients are fed into an internal mixer that uses rotors to heat and mix the compound (2). The rubber then goes onto a mill that further kneads it to ensure uniform dispersion of the ingredients (3).

The resulting rubber sheets are dipped in stearate to reduce stickiness (4), then air cooled to prevent premature curing, or *scorch* (5). Samples are taken for batch testing (6). If approved for production, the rubber is fed through an extruder to produce pre-forms (small pieces of rubber compound) for use in compression molding (7). If rubber is used in a transfer molding process, strips or pads are prepared for insertion into the transfer pot. Long strips of rubber are extruded for insertion into an injection press screw feeder.

Most shaft seals have metallic cases that have been press-stamped from strip steel (8). After stamping, the mild steel cases are placed in a rotating basket and submersed in a series of tanks designed to both clean the cases and apply a coat of zinc phosphate (9). This coat helps prevent rust, and it roughens the case's surface, making the metal more

"A shaft seal's ability to function effectively is the end result of not only good design, but also meticulous manufacturing."


RUBBER MIXING
1
With the help of computer controls, recipe ingredients are selected and precisely weighed...
2
...before being fed into an internal mixer...
3
...and going onto a mill, where uniform dispersion of all ingredients is achieved.
4
The resulting rubber sheets are treated to keep them from sticking together...
CASE PREPARATION
8
Strip steel stock is stamped to produce cases...
9
...which are cleaned, phosphated, and coated with adhesive...
10
...which is then oven- or air-dried.
MOLD MAKING
11
Engineers use sophisticated programs to design molds...
12
...which are precisely cut by CNC equipment...
13
...and polished by hand prior to storage or use.

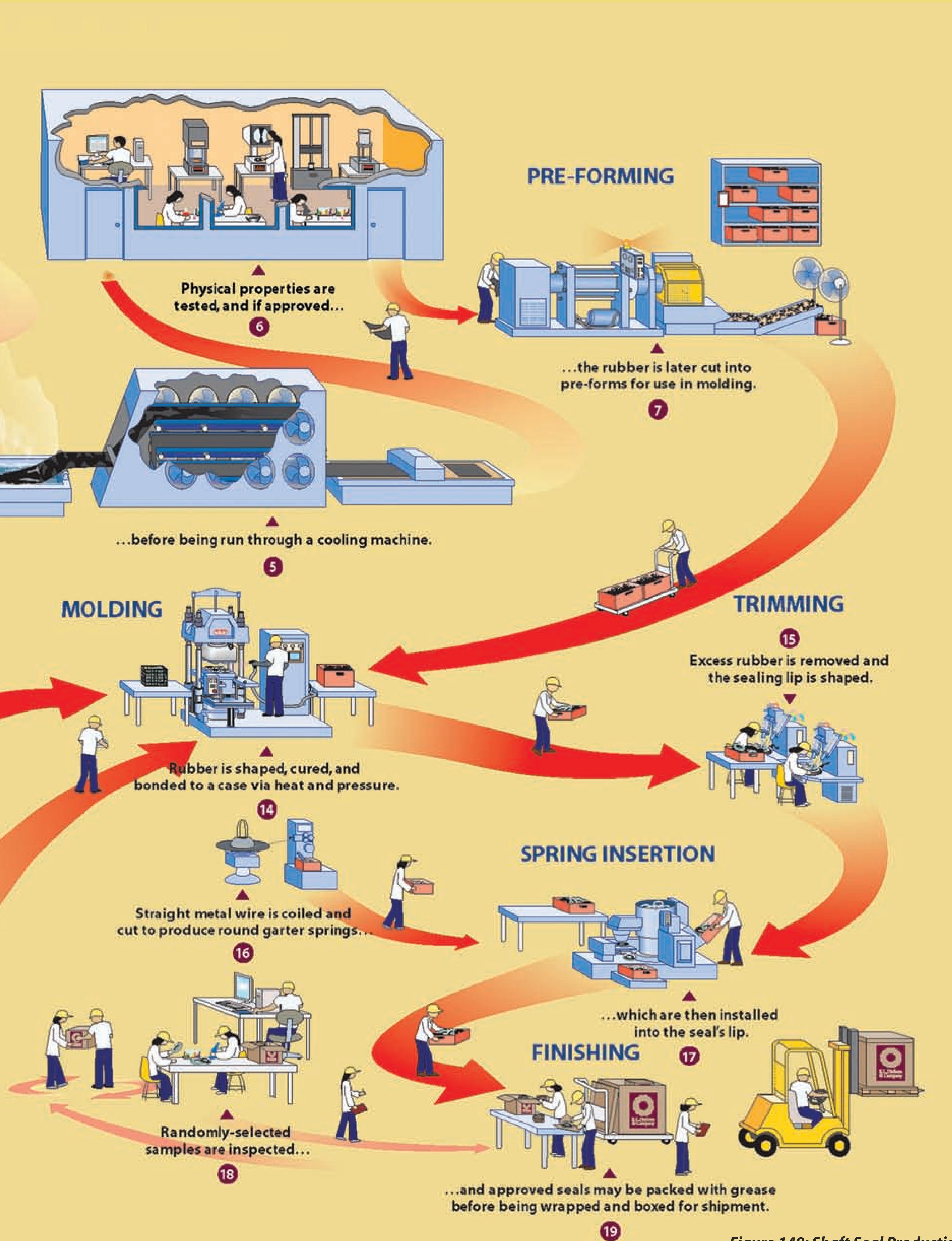


Figure 148: Shaft Seal Production

amenable to being bonded to rubber during molding. Adhesive is generally sprayed on the case and then oven- or air-dried as a last step before molding (10).

Mold making begins with a design generated by an engineer based on the desired final dimensions of the seal (11). Typically made of hardened tool steel, the mold is produced using computer numerical control (CNC) equipment (12). Once cut, the mold is hand-polished to a blemish-free finish, then cleaned prior to use or storage (13).

MOLDING

Molding is when rubber is shaped, cured, and bonded to a case via heat and pressure (14). There are three main methods: compression, transfer, and injection molding. Compression molding involves putting the uncured rubber compound into a heated, open mold cavity, then closing the mold under pressure (usually in a hydraulic press) to initiate vulcanization. In transfer molding, the uncured rubber compound is put in a transfer chamber (pot), heated, then squeezed down through a sprue, runner, and gate system leading into a closed mold cavity. With injection molding, the preheated rubber is injected under pressure through a runner system and into a closed, heated mold.

They differ in how the rubber enters the mold, but all three methods can be used to make shaft seals. Our illustration shows compression molding. Each cavity of the mold is loaded with a metal case and a rubber pre-form, then the mold is closed for simultaneous shaping and curing of the rubber, as well as bonding of the rubber to the case.

FINISHING

After the molded seals are taken from the cavities, any excess rubber (flash) is removed. The sealing lips may be knife-trimmed in an automated process (15). Automation may also be used to insert garter springs, which were produced previously in a separate preparatory stage (16), into the sealing lips. Spring insertion may be done by hand for low volume, unusual seal designs or for very large seals (17).

The seal lips are greased if the application requires lip lubrication. Though each step of the manufacturing process is carefully controlled and monitored to ensure product quality, a final audit inspection is conducted on a sample of the finished parts taken from a randomly selected box (18). All parts are shipped when this final inspection has been successfully completed (19).

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Quality Assurance.

We are proud to say that RL Hudson is a preferred supplier of fluid sealing devices and custom-molded rubber, plastic, and polyurethane products for a diverse group of manufacturers. As such, we have attained a worldwide reputation with leading original equipment manufacturers as an organization committed to quality and customer satisfaction.

Recognizing that quality is not only conformance to established acceptance standards, R.L. Hudson & Company is dedicated to being responsive to the ever-changing needs of our customers. Additionally, one of our prime objectives is to seek out continuous improvement so as to cost effectively provide our customers with a product line of the highest available quality.

In accordance with global Quality Assurance's movement toward one unified International Standard, it is our policy to assure process integrity by operating to a Quality System defined by QS 9000 and ISO 9001. As part of this system, we established our own in-house Quality Assurance department (see ***Figure 149***) whose sole purpose is to oversee the inspection of the products we offer, including shaft seals. We feel strongly that having our own Quality Assurance department is a wise investment in the ultimate success of our customers' applications.

"We have attained a worldwide reputation with leading original equipment manufacturers as an organization committed to quality and customer satisfaction."

OUR LAB

Figure 149: In-House QA Department

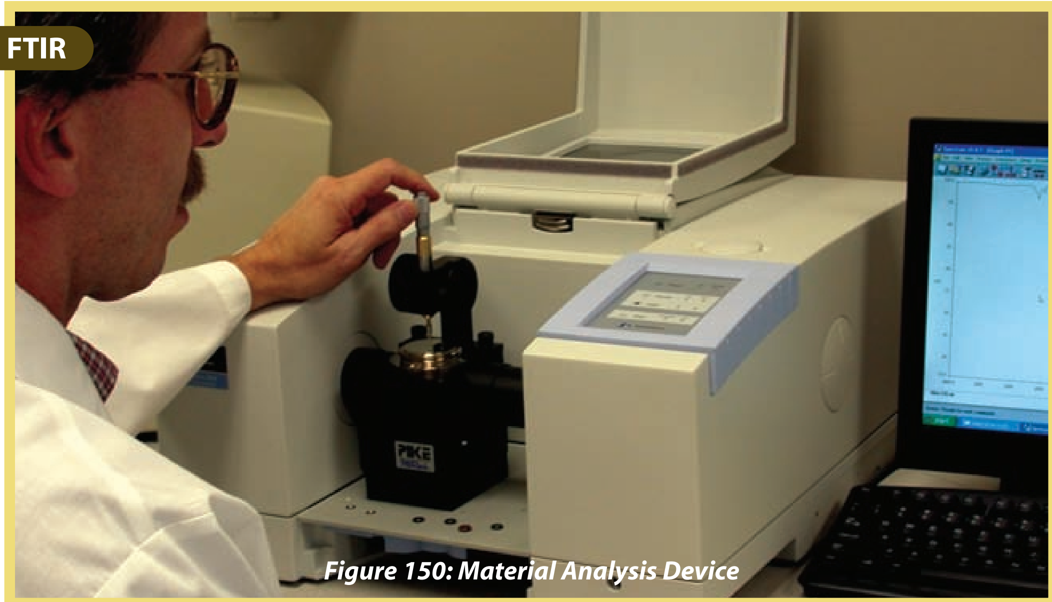

Figure 150: Material Analysis Device

MATERIAL ANALYSIS & VERIFICATION

One of the most sophisticated pieces of equipment in our quality lab is a Spectrum One™ FTIR (Fourier Transform InfraRed) spectrometer manufactured by Perkin Elmer. As shown in ***Figure 150***, this device allows us to analyze samples of rubber and plastic. ces are the exact same materials that we originally approved for a given application.

How does the FTIR work? Spectroscopy is a way to gauge the amount of radiation a particular material absorbs at various wavelengths. Subjecting a material sample to a beam of infrared radiation produces a recordable spectrum, a spectrum unique to that material. Study of this spectrum can yield useful information about the molecular makeup and chemical bonds of the material.

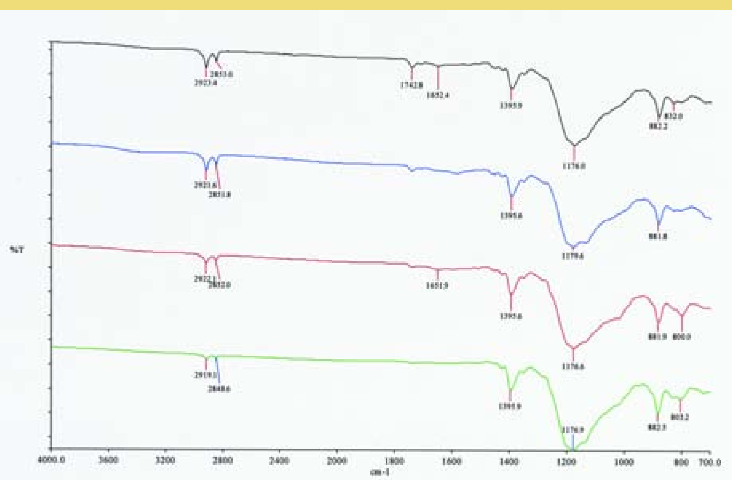

Table 48: FTIR Spectrum Graph

How does that help us (and our customers)? A sample taken from all first articles is analyzed using the FTIR, and an IR spectrum is generated. It's in the form of a graph such as the one shown in ***Table 48.*** This graph can be permanently stored in our computer database. A stored graph can serve as reference when analyzing samples from subsequent batches to make certain that the material from those batches matches the originally approved material. In other words, it's a high-tech way to absolutely verify material recipe consistency. The FTIR also affords us the ability to analyze any rubber or plastic material submitted to us so that we can determine (and, if need be, improve) the base material.

VIDEO IMAGING SYSTEM

Figure 151: Opti-Flex 4000 Series

VIDEO IMAGING

Since our QA lab's inception, we've utilized a Voyager 1000 video imaging system to assist with inspection of complex or miniature parts via a camera-computer interface. Regular upgrades to both the Voyager's hardware and the View Metrology Software (VMS) ensure that our images are as detailed as possible and that our measurement capabilities are as extensive as possible.

As shown in ***Figure 151***, we also have an Opti-Flex™ 4000 Series video imaging system. Manufactured by Flexbar®, the Opti-Flex gives us another precision measurement and inspection station. The Opti-Flex magnifies up to 276 times, and it's accurate to within .00015 of an inch.

ANNOTATED IMAGE

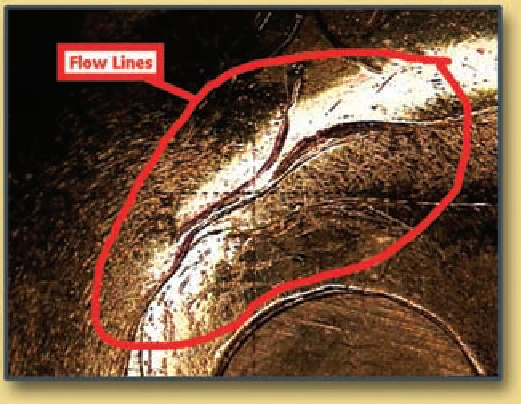

Figure 152: Opti-Flex 4000 Series

In many ways, the Opti-Flex is very much like the Voyager, with the main difference being that the Opti-Flex allows images to be directly annotated with dimensional data or other helpful labels. An example of an annotated image is shown in ***Figure 152***. These annotated images can then be stored for future retrieval or even sent as e-mail attachments. The e-mailing comes in particularly handy as an instantaneous way to communicate with our factories and our customers.

LIP FORCE TESTER

Figure 153: Split Shaft Device To Gauge Lip Force

RADIAL LIP FORCE TESTING

As a major supplier of radial lip shaft seals, we have a vested interest in verifying that the seals we provide will perform as expected in service. That is, that they will exert the right amount of radial force on the shaft onto which they are installed. Too little force means the sealing lip won't maintain proper contact with the shaft, allowing a leak path to develop. Too much force on the shaft will increase both friction and wear, shortening seal life.

As detailed in the Rubber Manufacturers Association (RMA) Handbook OS-6 ("Radial Force Measurement"), lip force can be gauged through use of a split-shaft testing device. One half of the shaft is held motionless, while the other half is free to move and therefore capable of reflecting how much force is being exerted on it by a shaft seal's lip. Depending on the device in use, measurement may be mechanical (as calibrated by a spring), pneumatic (using an air pressure gauge), or electronic (via a transducer). Studies conducted by the RMA have shown the electronic devices to be preferable. The electronic device we use in the R.L. Hudson & Company lab is shown in ***Figure 153***.

ENDURANCE TESTING

Because of the number of variables, and the ways in which these variables are always changing, it is very tough to accurately gauge shaft seal performance on equipment in the field. As a result, sophisticated devices have been developed that allow performance variables to be closely controlled and precisely gauged in laboratory settings.

The tester shown in ***Figure 154*** allows for the control of the test head temperature via either circulation of a heat transfer fluid or via electric heaters. In some cases, both methods are used together. Shaft speed is controlled, as is shaft-to-bore misalignment. The shaft surface is ground to specifications,

ENDURANCE TESTER

Figure 154: Variables Controlled & Gauged

and the amount of dynamic runout can be adjusted. Seals are typically run to the point of failure (as defined and agreed on by all concerned parties prior to testing). Statistical analysis can then be used to evaluate and compare various performance capabilities. This analysis should be augmented by a detailed inspection of the seal, the test fluid, the test shaft, and the housing so that the cause of failure is completely understood.

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Possible Manufacturing Defects.

A well-designed shaft seal is valuable only if manufactured well. What follows is an overview of the most common imperfections arising from manufacturing or handling variables. They do not reflect changes in a seal as a result of use.

These defects will typically be visible without magnification. Most may be seen in more than one part of a seal, so the importance of a defect depends on location and on the seal's purpose (e.g. oil or grease retention, or contaminant exclusion). A blister at the lip contact point of a seal used to retain oil is critical and can easily cause leakage; a blister on the seal's interior surface should not hurt sealability. These descriptions are general and may not be equally true for all applications.

Blisters.

A blister is an enclosed cavity that protrudes from, and thus deforms, a rubber surface (see Figure 155). Blisters can form on a shaft seal's lip due to inconsistencies during the manufacturing process or, in some cases, due to excessive heat during storage or transport.

A blister at the contact point on the primary lip of a shaft seal designed to retain oil or grease can be critical. This is because the blister can limit the lip's ability to maintain proper contact with the shaft. Without good followability between the lip and the shaft, a leak path can develop. A blister on an oil or grease seal's secondary (dirt) lip, flex section, O.D., or endface can also decrease sealabilty, though not typically to a critical extent.

"A blister is an enclosed cavity that protrudes from, and thus deforms, a rubber surface."

BLISTERS

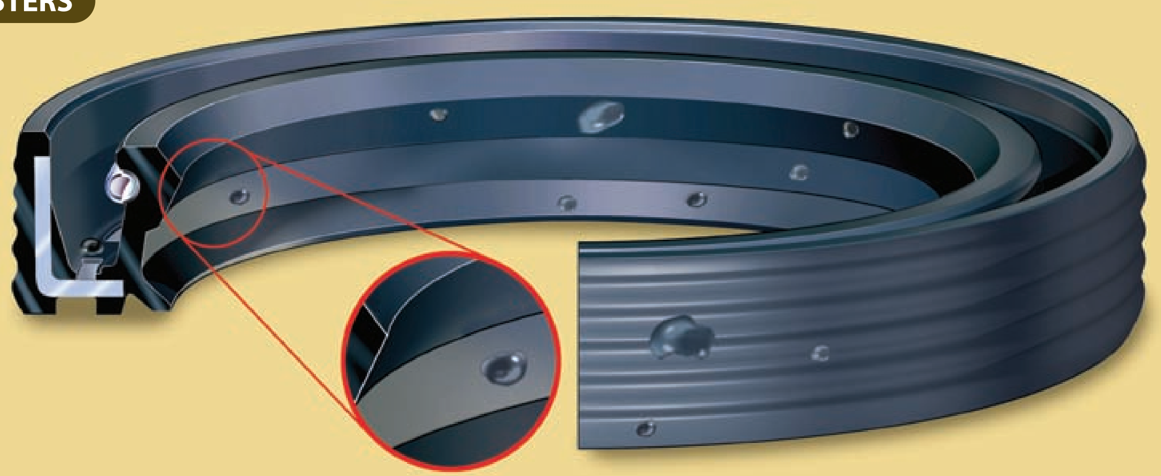

Figure 155: Enclosed Protrusions

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Cuts.

A crack is a sharp break or fissure in a rubber surface; a cut is a slice-like opening. Though they can look identical (see *Figure 156*), they typically have different causes. A crack can be caused by excessive strain and/or exposure to detrimental conditions, such as ozone, weather, or ultraviolet (UV) light; a cut is generally caused by unwanted contact between the surface and a sharp object.

A cut at the contact point on the primary lip of a shaft seal designed to retain oil or grease can be critical. This is because the cut compromises the integrity of the dam formed by the sealing lip, and leakage can easily result. A cut on an oil or grease seal's secondary (dirt) lip, flex section, O.D., or endface can also cause serious problems. Because neither the interior nor the exterior of the case contacts the shaft, a cut in either of these areas tends to be of minor importance for an oil or grease seal.

"A crack is a sharp break or fissure in a rubber surface; a cut is a slice-like opening."

CUTS

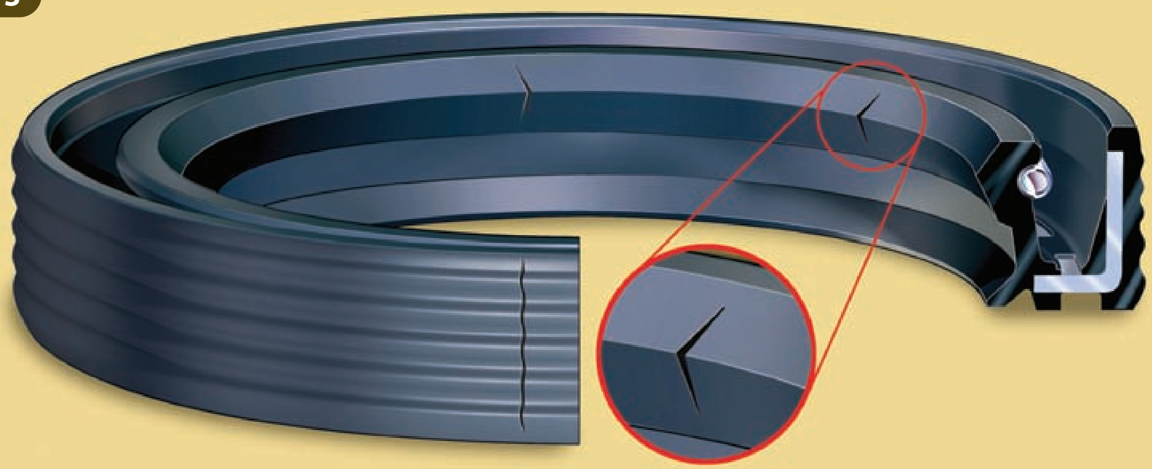

Figure 156: Slice-Like Opening

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Damaged Mold.

Damage to a mold (typically nicks in the metal surface) allows extra rubber to collect during the molding process. As a result, mold damage typically manifests itself as irregular protrusions along the rubber surfaces of a seal (see *Figure 157*). In order to minimize this problem, the mold should be hardened. In addition, seal manufacturers must be careful in handling molds and should retool if molds become damaged.

For trimmed lip shaft seals, the trimming process itself will prevent protrusions from being found on the oil side of the primary sealing lip. However, it is possible to find protrusions on the non-trimmed surfaces of the seal. If large enough, protrusions that encroach on the contact point from the air side of the primary lip may create holes in the lip/shaft contact area, resulting in leakage.

"Mold damage typically manifests itself as irregular protrusions along the rubber surfaces of a seal."

DAMAGED MOLD

Figure 157: Irregular Surface Protrusions

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Deformation.

Deformation is a change in the shape of a seal as a result of compression (see *Figure 158*). This compression might take the form of improper packaging or handling. Depending on the manufacturer and the needs of the customer, shaft seals can be packaged in a variety of ways, though bulk packing (with or without egg crate dividers) and roll packing (using paper or shrink wrap) are the most common methods. In some cases, seals are individually wrapped or boxed.

Once shipped and unpackaged, seals should be handled with care so as not to damage the metal cases or distort the rubber seal lips prior to installation. Seal lips are particularly susceptible to damage or distortion if suspended from nails, pegs, or other wall-mounted supports.

For shaft seals used to retain grease or oil, deformation can be a major issue if it occurs along the primary sealing lip, on a secondary lip, at the flex section, on the seal O.D., or on the endface. Deformation is not a major concern if it is confined to either the interior or exterior portions of the seal that do not contact the shaft or the housing.

"Deformation is a change in the shape of a seal as a result of compression."

DEFORMATION

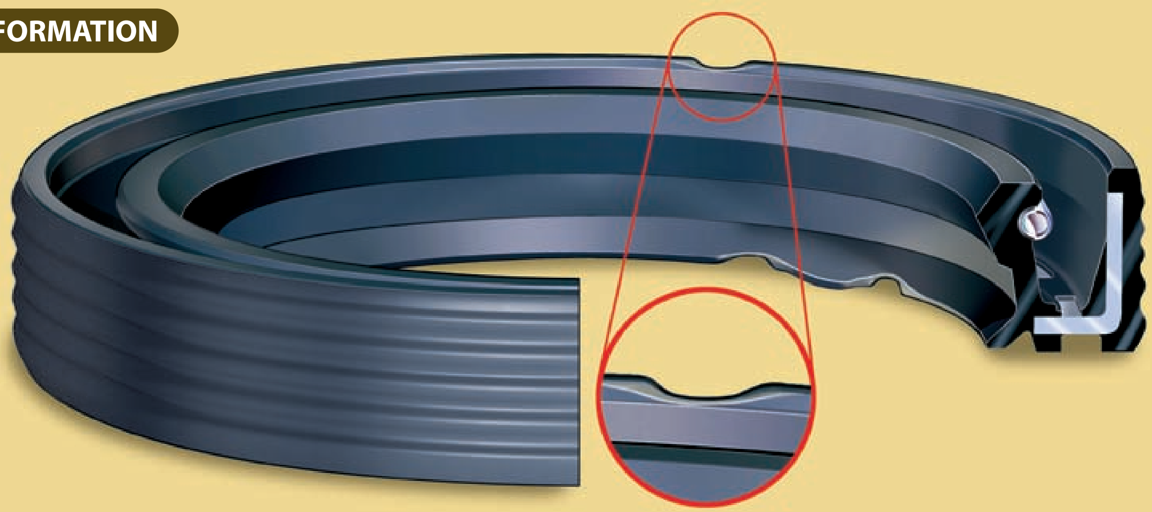

Figure 158: Shape Change Due To Compression

Dirty Mold.

A mold impression is an imperfection molded into the surface of a material (such as the rubber used to form a shaft seal's lip, see *Figure 159*). Mold impressions are typically caused by residue buildup within the mold cavity. In order to minimize instances of mold impression, seal manufacturers must be careful to maintain mold cleanliness.

For shaft seals used to retain grease or oil, mold impressions can be critical if they occur along the primary sealing lip. They can also be serious if they occur on a secondary lip or on the seal O.D. Mold impressions found in other places do not typically impair the seal's usability.

"A mold impression is an imperfection molded into the surface of a material."

DIRTY MOLD

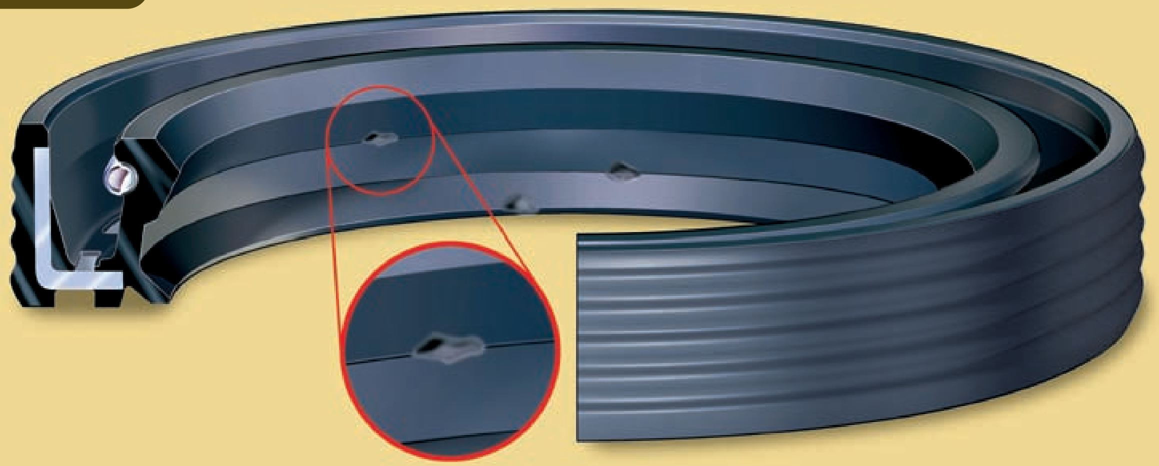

Figure 159: Surface Imperfections

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Flash.

Flash is excess rubber remaining on the parting line of a rubber product following molding. In the case of a shaft seal, excessive rubber flash may interfere with the sealing surfaces. Excessive flash can fold across the primary lip (as in Figure 160), a secondary lip, and/or the seal O.D.

Flash can be a major concern for seals designed to retain grease or oil if it occurs along the primary sealing lip, on a secondary lip, at the flex section, on the seal O.D., or on the endface. Flash is not usually a major concern if it is confined to either the interior or exterior portions of the seal that do not contact the shaft.

"Flash is excess rubber remaining on the parting line of a rubber product following molding."

FLASH

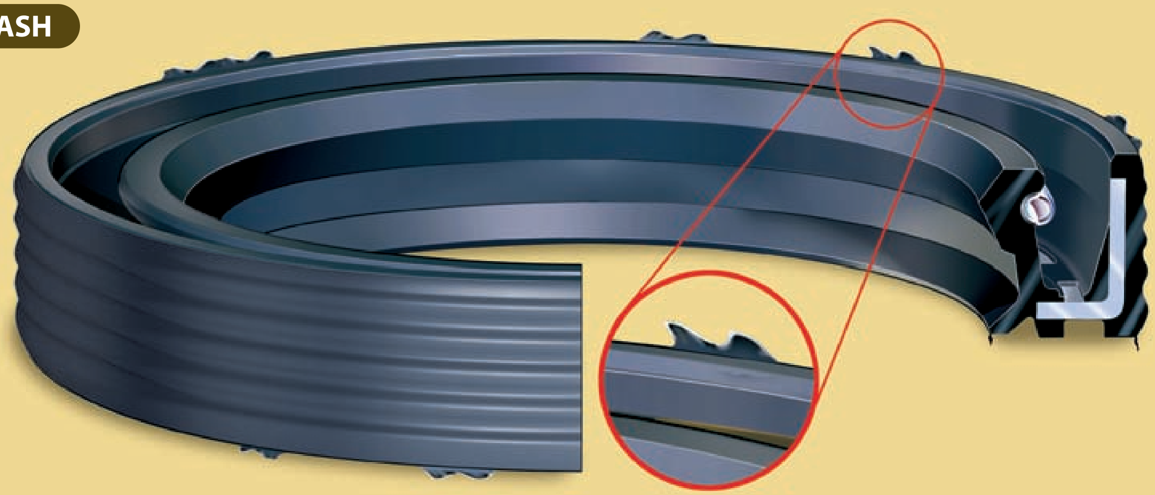

Figure 160: Excess Rubber

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Improper Trim.

Improper trim occurs when a seal has not been finished properly after molding. More specifically, it occurs when not all of the material that should have been trimmed from a surface (such as a primary sealing lip,as in *Figure 161*) was actually removed. Improper trim can also occur if the primary sealing lip is trimmed in the wrong direction (i.e. from the oil side to the air side, rather than from the air side to the oil side as it should be). Trimming in the wrong direction can leave a ridge of rubber at the contact point that will imprede proper lip function.

Improper trimming of a primary lip or secondary lip in shaft seals designed to retain grease or oil can be a major problem, especially if the untrimmed material prevents the seal from making or maintaining proper contact with the shaft. Improper trimming of the seal O.D. can prevent the seal from seating properly in the housing bore, and this increases the likelihood that a leak path will develop between the O.D. and the bore.

"Improper trim occurs when not all of the material that should have been trimmed from a surface was actually removed."

IMPROPER TRIM

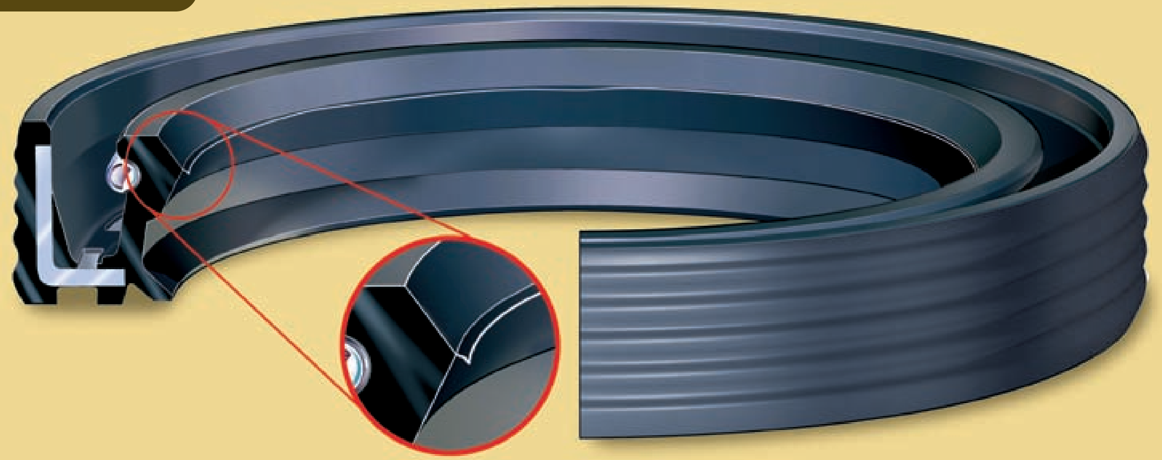

Figure 161: Incomplete Trim & Trim From Wrong Direction

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Knit Lines.

A knit line is an imperfection of the seal material due to premature curing (scorching) of the rubber during molding (see *Figure 162*). It is for this reason that seal manufacturers must be extremely diligent in controlling material temperature fluctuations throughout the production process. Mold temperatures must be especially monitored so as not to inadvertently harm the material.

In shaft seals designed to retain grease or oil, a knit line that occurs along the primary lip, the secondary lip, the flex section, the seal O.D., or the endface can be a major concern. Less problematic – though still potentially dangerous – are knit lines seen in the interior or on the exterior of the seal.

"A knit line is an imperfection of the seal material due to premature curing (scorching) of the rubber during molding."

KNIT LINES

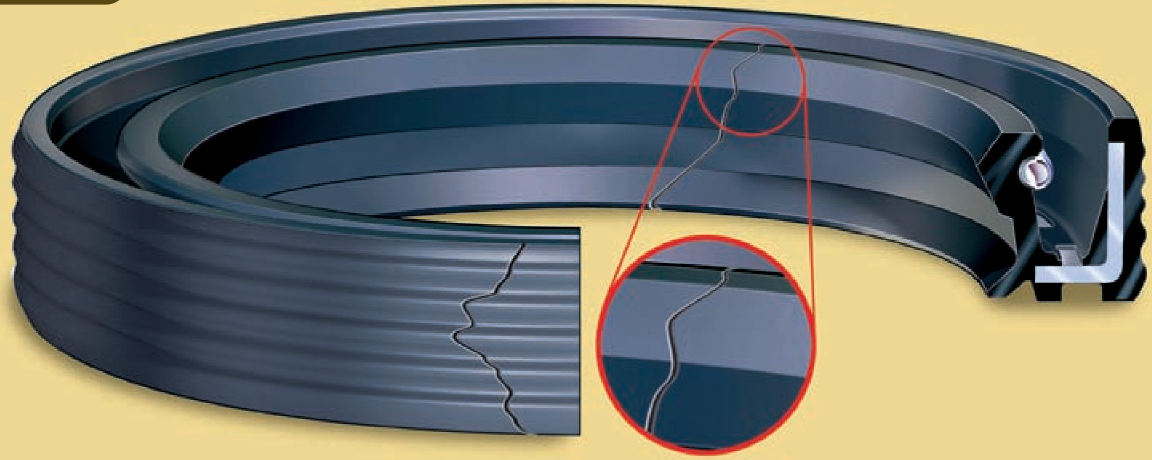

Figure 162: Imperfection Due To Premature Curing

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Material Contamination.

Material contamination occurs when unwanted, extraneous matter (such as dirt or other debris) is inadvertently included in the seal material. In a shaft seal, this can occur during molding of the rubber lip(s) (see *Figure 163*).

For seals designed to retain grease or oil, material contamination can be a critical problem if it occurs on the primary sealing lip. Presence of foreign matter (such as tiny bits of metal) can adversely impact the interface between lip and shaft, increasing friction or damaging the shaft itself such that a leak path develops. Material contamination along the secondary lip, at the flex section, or on the seal O.D. can also be serious. Foreign matter within the rubber of the endface, along the interior, or on the exterior is generally not a major issue.

"Material contamination can be a critical problem if it occurs on the primary sealing lip."

MATERIAL CONTAMINATION

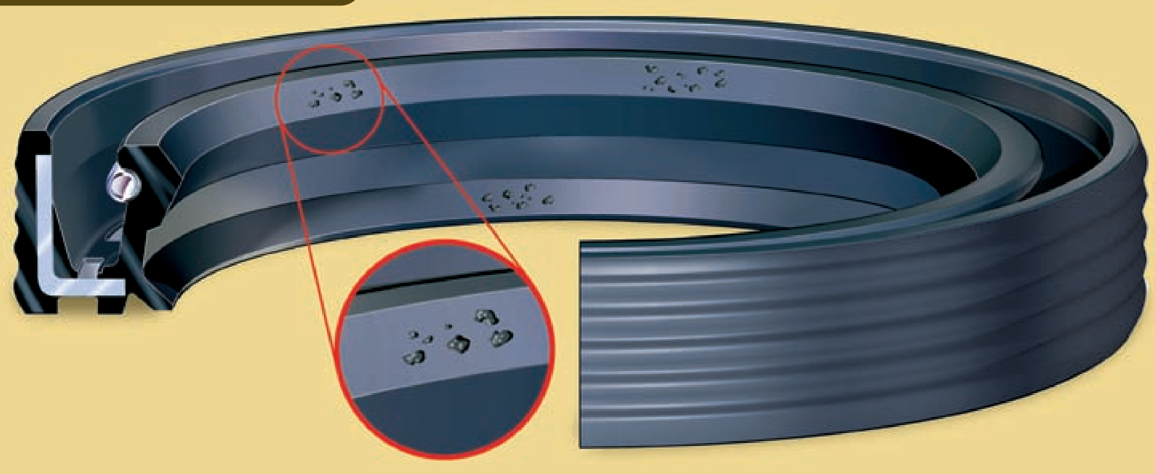

Figure 163: Unwanted Matter

Nicks.

Nicks are unwanted voids within the sealing material created after molding (see *Figure 164*). Nicks are most often caused during installation, but a nick could be caused at any point during the handling of a molded seal.

A nick on the primary sealing lip of a shaft seal designed to retain oil or grease is a critical problem. Because the profile of the lip is compromised, the lip does not make proper contact with the shaft. Nicks found in other areas — such as on a secondary lip, at the flex section, on the seal O.D., or on the endface — are also generally problematic. Nicks in either the interior or on the exterior of the seal generally do not cause great concern. Careful handling and proper installation can minimize the chances of inadvertently nicking the seal.

"Nicks are unwanted voids within the sealing material created after molding."

NICKS

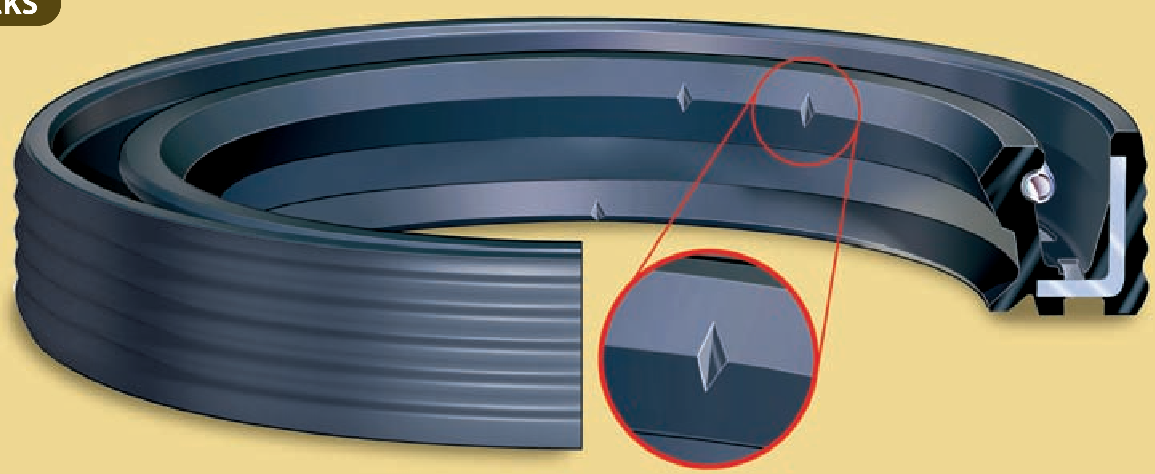

Figure 164: Unwanted Voids

Non-Fills.

Non-fills are unwanted voids within the sealing material (see *Figure 165*) created during molding (rather than after molding, as is the case with a nick). Non-fills are typically caused by improper material flow within the mold. Seal manufacturers must take great care in formulating compounds that will flow properly when the mold is correctly heated.

A non-fill on the primary sealing lip of a shaft seal designed to retain oil or grease is a critical problem. Because the profile of the lip is compromised, the lip does not make proper contact with the shaft. Non-fills on a secondary lip, at the flex section, on the seal O.D., or on the endface are also generally problematic. Non-fills in either the interior or on the exterior of the seal generally do not cause great concern.

"Non-fills are unwanted voids within the sealing material created during molding."

NON-FILLS

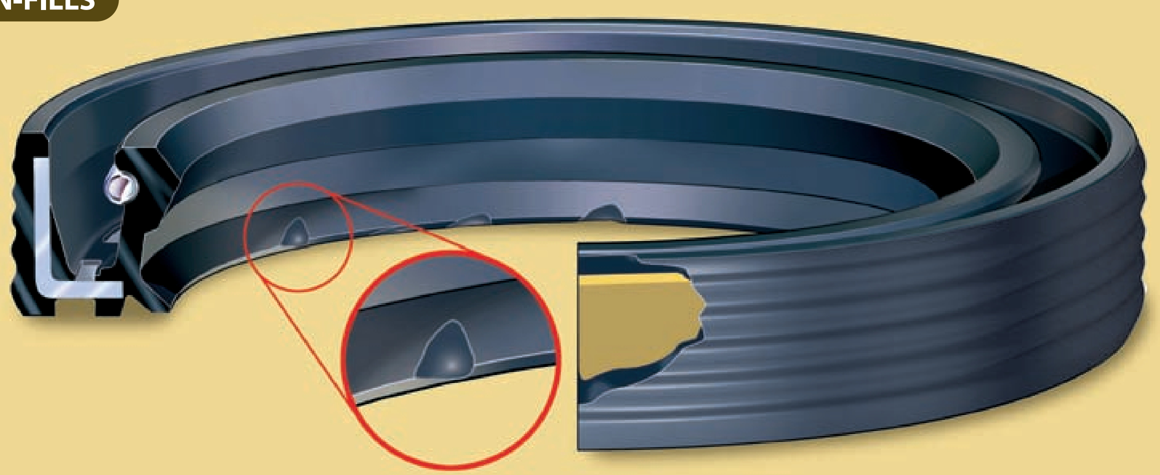

Figure 165: Unwanted Voids

Poor Bond.

Poor bonding means there is inadequate adhesion between two layers of material. In an elastomeric shaft seal, there is a rubber-to-metal bond between the elastomeric member and the metallic case (see *Figure 166*). PTFE components can also be bonded to rubber or to the metal case.

Though poor bonding is bad no matter where on the seal it might occur, it is most dangerous if it occurs at a shaft seal's flex section, the area in which the lip and the case are connected.

"Poor bonding means there is inadequate adhesion between two layers of material."

POOR BOND

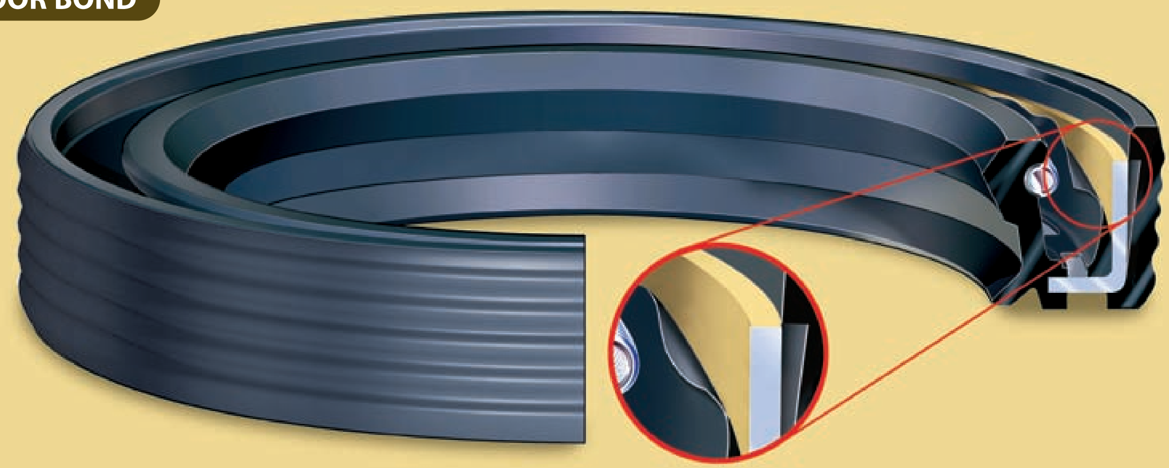

Figure 166: Inadequate Adhesion

manufacturing & quality assurance

Porosity.

Porosity is a condition in which a material (such as the rubber used to form a shaft seal) is full of numerous tiny bubbles (see *Figure 167*). The presence of these bubbles is a manufacturing defect due to air entrapment that, if present, should be spotted and corrected by the molder.

Porosity at the flex section can compromise the primary lip's ability to flex properly in response to shaft eccentricities. Porosity on a secondary lip can limit the lip's ability to exclude dust and dirt properly.

"Porosity is a condition in which a material is full of numerous tiny bubbles."

POROSITY

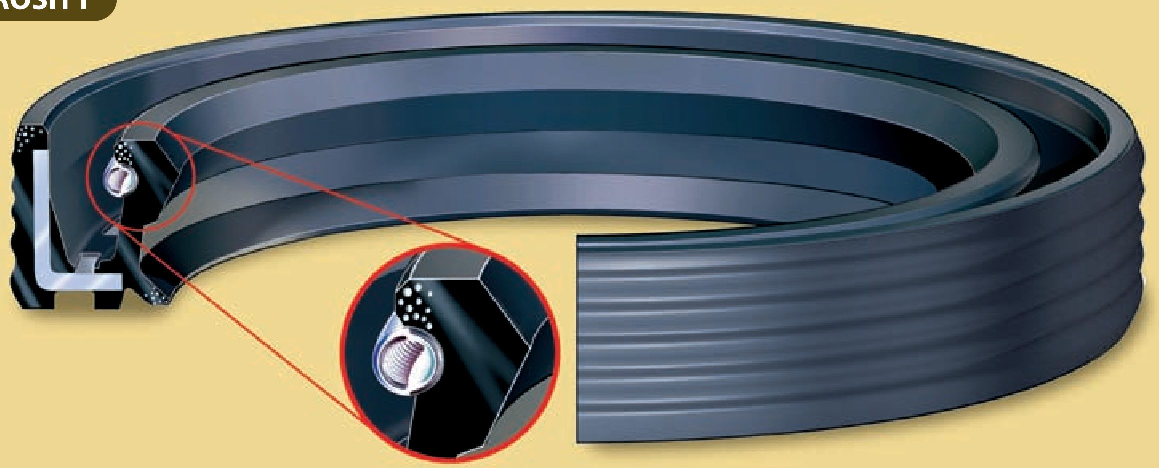

Figure 167: Numerous Tiny Bubbles

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Rough Trim.

Rough trim occurs when the trimming of a shaft seal's sealing surface leaves it with unwanted irregularities on both sides of the contact point (see *Figure 168*). This is a result of trimming in the wrong direction (i.e. from the oil side to the air side of the primary lip, rather than from the air side to the oil side as it should be). Extreme care must be taken during trimming to minimize instances of rough trim. Many shaft seal manufacturers now use automated trimming machines to help ensure accuracy and consistency.

Rough trim is a possibility on the seal lip(s), the outside diameter, and the leading edge. Depending on the application and the extent of the irregularities, rough trim can be a major obstacle to proper seal function.

"Rough trim occurs when the trimming of a shaft seal's sealing surface leaves it with unwanted irregularities."

ROUGH TRIM

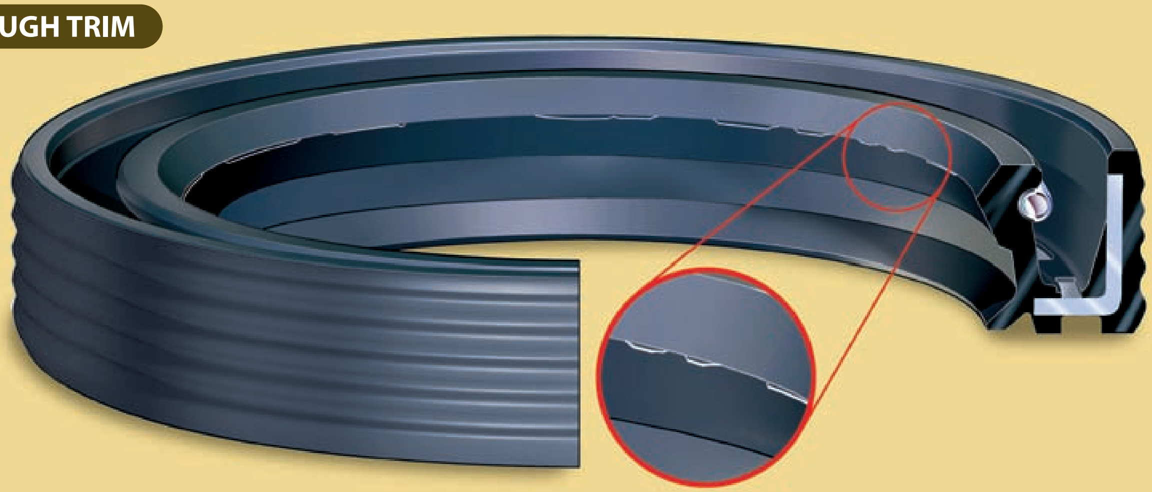

Figure 168: Unwanted Irregularities

Scoop Trim.

Scoop trim occurs when a seal surface is concave as a result of trimming (see *Figure 169*). Extreme care must be taken during trimming to minimize instances of scoop trim, which is caused when the seal lip is not properly supported (such as with a backup fixture) during trimming. Many shaft seal manufacturers now use automated trimming machines to help ensure accuracy and consistency.

The primary lip is the only place that lip trimming occurs, so this is the only lip that will encounter a scoop trim. Depending on the application and the extent of the concavity, scoop trim may be a major obstacle to proper seal function.

"Scoop trim occurs when a seal surface is concave as a result of trimming."

SCOOP TRIM

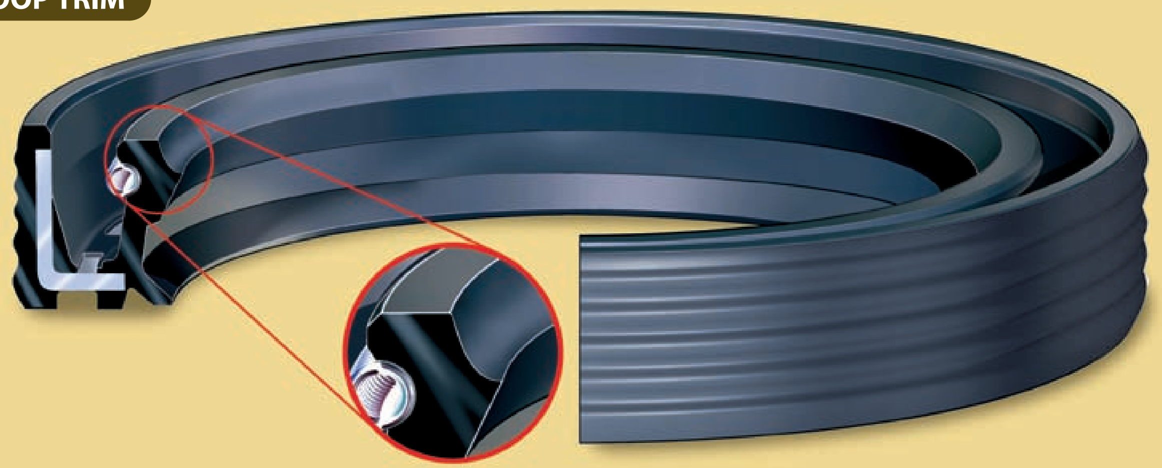

Figure 169: Concave Surface

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Scratches.

Scratches are blemishes on the surface of a sealing material due to abrasion (see *Figure 170*). Though typically superficial, scratches can be serious if they are large enough or deep enough. Scratches most often occur as a result of careless handling and improper installation.

A scratch along a shaft seal's primary lip, secondary lip, or at its flex section can be a major issue. A scratch on the primary lip can provide a leak path, whereas a scratch on a secondary lip can allow ingress of contamination. A scratch at the flex section could, if deep enough, substantially weaken the section and shorten seal life. Depending on the application and the severity of the scratching, scratches in other areas are typically not of major consequence.

"Scratches are blemishes on the surface of a sealing material due to abrasion."

SCRATCHES

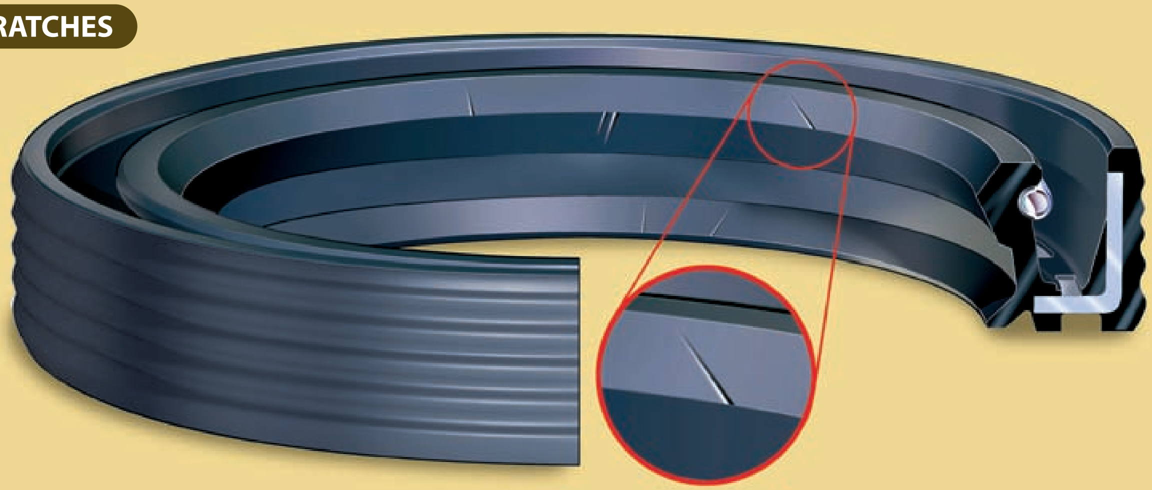


Figure 170: Surface Blemishes

manufacturing & quality assurance

Spiral Trim.

The trimming operation will leave a spiral pattern on the oil side of the seal's primary lip. Extreme care must be taken during the trimming operation to prevent the depth of the spiral groove from becoming excessive (as shown in *Figure 171*).

Many shaft seal manufacturers now use automated trimming machines to help ensure accuracy and consistency. A deep spiral trim can be a major obstacle to proper seal function.

"The trimming operation will leave a spiral pattern on the oil side of the seal."

DEEP SPIRAL TRIM

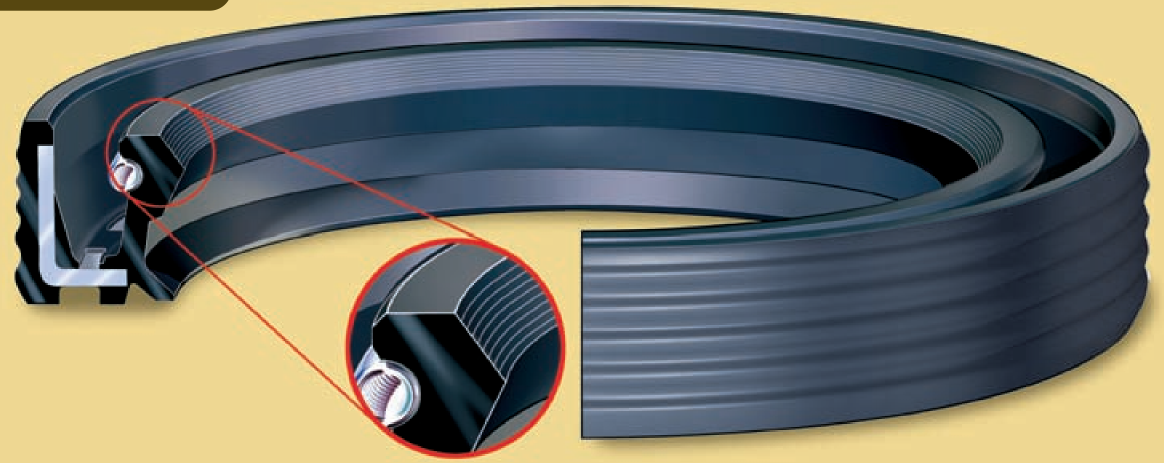


Figure 171: Spiral Pattern On Oil Side

manufacturing & quality assurance

Surface Contamination.

Surface contamination occurs when unwanted material (such as dust or dirt) is found on the surface of a seal (see *Figure 172*). Surface contamination can largely be eliminated through proper storage and handling of finished seals. Seals exposed to dirty environments (such as work benches) are most susceptible to contamination, especially if the seals have been pre-lubed. The lubrication can pick up and hold on to any grit, metal particles, or dirt the seal might contact.

Foreign material on a shaft seal's primary sealing lip is a big problem. This foreign material can damage both the lip and the shaft, with leakage likely. Foreign material on other surfaces – such as a secondary lip, at the flex section, on the O.D., or on the endface – can also cause problems.

Care must be taken in choosing a cleaning solution. Depending on the lip material, there are several commonly used solvents (such as high flash napthas or Stoddard solvent). Abrasive and chemical cleaners must not be used; they can irreparably damage the seal elements and/or the bonding.

"Surface contamination occurs when unwanted material is found on the surface of a seal."

SURFACE CONTAMINATION

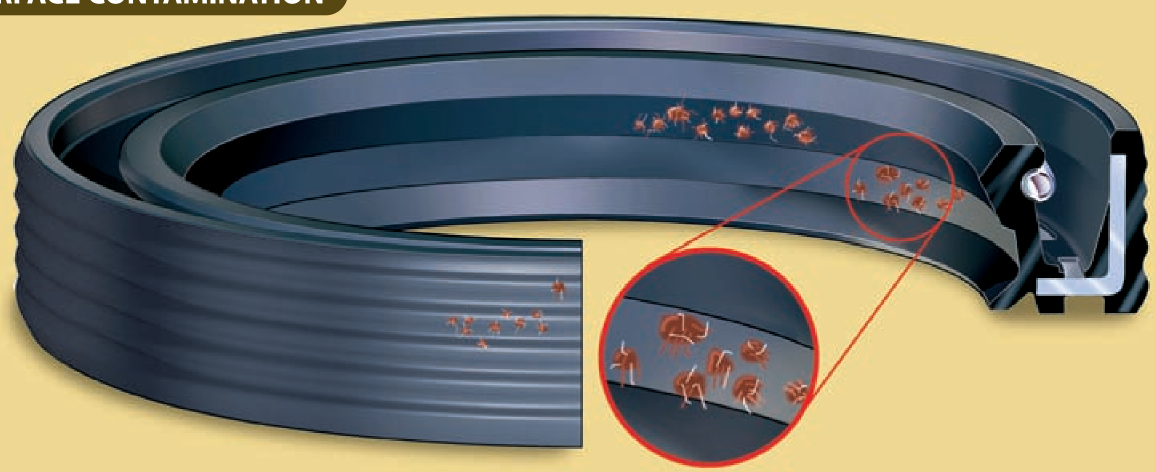


Figure 172: Unwanted Material On Surface

Tears.

Tears are instances of separation or pulling away of part of a sealing structure (see *Figure 173*). Tears typically occur due to careless handling.

A tear on a shaft seal's primary sealing lip will very likely alter its contact with the shaft and is thus a critical problem. Less serious, though still problematic, is a tear on a secondary lip, at the flex section, on the seal O.D., or on the endface. Tears to the rubber coating the interior or exterior of the seal normally don't prevent use.

"Tears are instances of separation or pulling away of part of a sealing structure."

TEARS

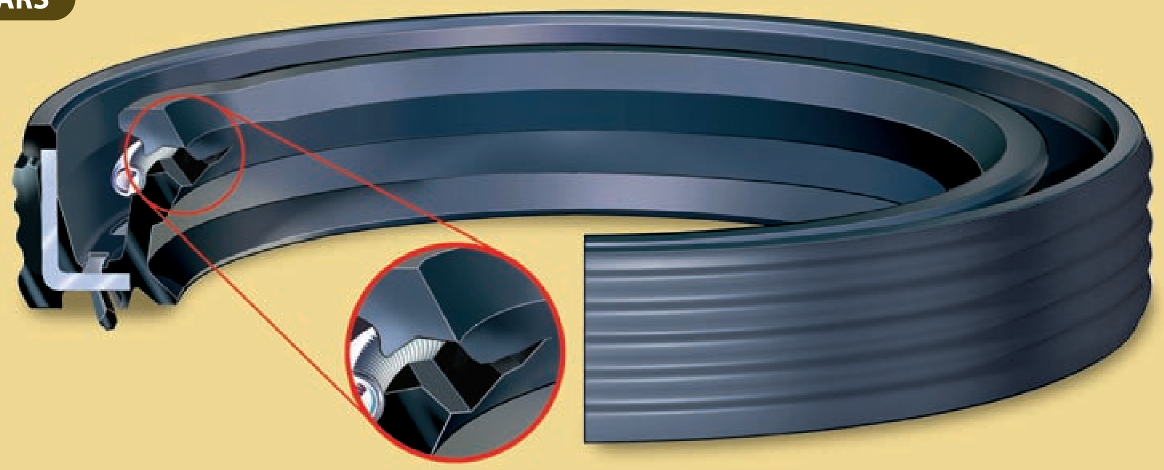


Figure 173: Separation or Pulling Away

manufacturing & quality assurance

Unbonded Flash.

Unbonded flash is loose rubber that has inadvertently adhered to the seal surface and may impair performance, or flash that does not properly bond to an intended mating material. As shown in Figure 174, the flap of rubber on the seal O.D. is flash that was not removed properly. It was not intended to bond to any surface on the case or seal. Conversely, the unbonded flash on the inside of the seal could in fact be expected to bond to the inside of the metal case. In this instance, however, it did not bond properly.

Unbonded flash can be caused by inconsistencies in the rubber compound and/or the material (such as metal) to which it is supposed to bond. Unbonded flash can also result from inconsistencies during molding (such as temperature or pressure fluctuations).

"Unbonded flash is loose rubber that has inadvertently adhered to the seal surface and may impair performance."

UNBONDED FLASH

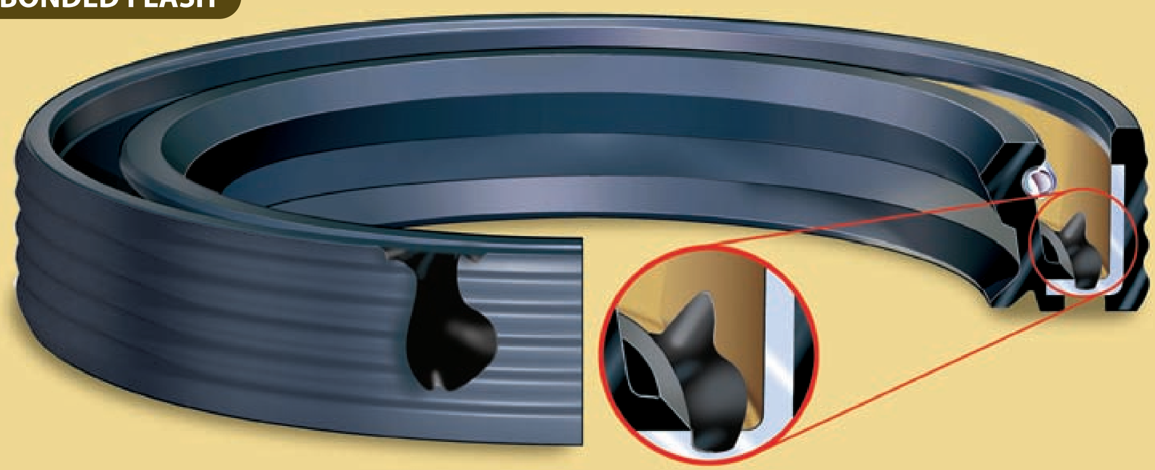


Figure 174: Improper Bonding

SEAL SECTORS

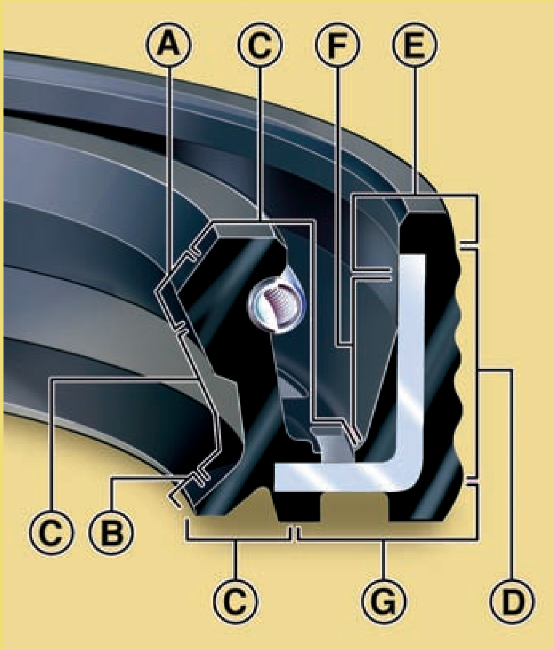


Figure 175: Dual-Lip, Rubber O.D. Seal

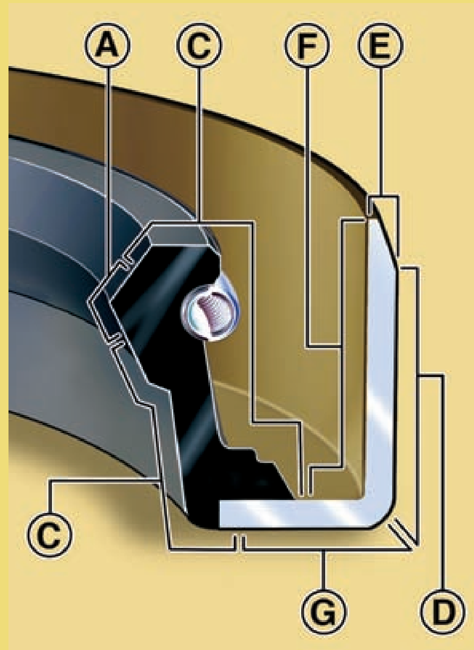


Figure 176: Single-Lip, Metal O.D. Seal

Primary Function	Defect	Critical	Major	Minor
Oil Retention	Blisters	A	B, C*, D*, E*	F*, G*
	Cuts	A	B, C*, D*, E*	F*, G*
	Damaged Mold	A, B*, C*	D*, E*	F*, G*
	Deformation	-	A, B, C*, D*, E*	F*, G*
	Dirty Mold	A	B, C*, D*	E*, F*, G*
	Flash	A, B, C*, E*	-	D*, F*, G*
	Improper Trim	A, B, C*	E*	-
	Knit Line	A	B, C*, D*, E*	F*, G*
	Mat. Contamination	A	B, C*, D*, E*	F*, G*
	Nicks	A, B, C*	D*, E*	F*, G*
	Non-Fills	A, B, C*	D*, E*	F*, G*
	Poor Bond	C, E*	D*	F*, G*
	Porosity	A	B, C*, D*, E*	F*, G*
	Rough Trim	A, B, C*	-	-
	Scoop Trim	-	A	B*, C*
	Scratches	A, B, C*	D*, E*	F*, G*
	Spiral Trim	A	-	B
	Surf. Contamination	A	B, C*, D*, E*	F*, G*
	Tears	A, B, C*	D*, E*	F*, G*
	Unbonded Flash	-	A, B, C*, D*, E*, F*	G*

LEGEND
A = Primary Lip
B = Secondary Lip
C = Flex Section
D = Seal O.D.
E = Endface
F = Interior
G = Exterior
* Figure 175 (rubber O.D.) only

Table 49: Summary of Manufacturing Defects

SEAL SECTORS

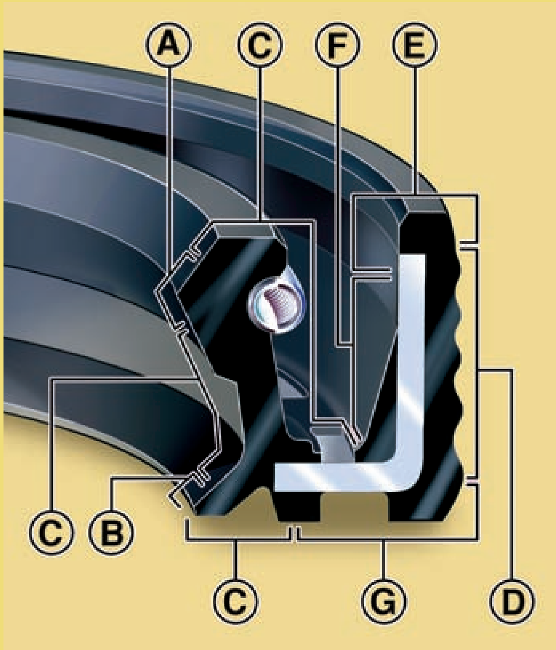


Figure 175: Dual-Lip, Rubber O.D. Seal

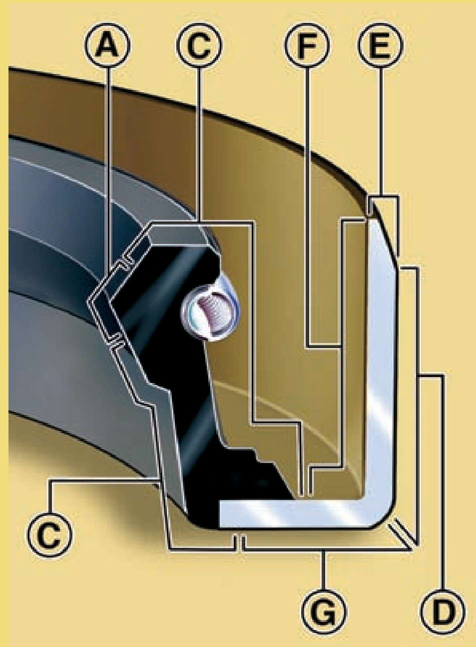


Figure 176: Single-Lip, Metal O.D. Seal

Primary Function	Defect	Critical	Major	Minor
Grease Retention	Blisters	A	B, C*, D*	E*, F*, G*
	Cuts	A	B, C*, D*, E*	F*, G*
	Damaged Mold	A	B, C*	D*, E*, F*, G*
	Deformation	-	A, B, C*, D*, E*	F*, G*
	Dirty Mold	A	B, C*	D*, E*, F*, G*
	Flash	-	A, B, C*, D*, E*	F*, G*
	Improper Trim	A, B, C*	E*	-
	Knit Line	-	A, B, C*	D*, E*, F*, G*
	Mat. Contamination	-	A, B, C*	D*, E*, F*, G*
	Nicks	A, B, C*	D*, E*	F*, G*
	Non-Fills	A, B, C*	D*, E*	F*, G*
	Poor Bond	C, E*	D*	F*, G*
	Porosity	A	B, C*	D*, E*, F*, G*
	Rough Trim	A	B, C*	-
	Scoop Trim	-	A	B, C*
	Scratches	A	B, C*, D*, E*	F*, G*
	Spiral Trim	-	A, B, C*	-
	Surf. Contamination	A	B, C*, D*, E*	F*, G*
	Tears	A	B, C*, D*, E*	F*, G*
	Unbonded Flash	-	A, B, C*, D*, E*	F*, G*

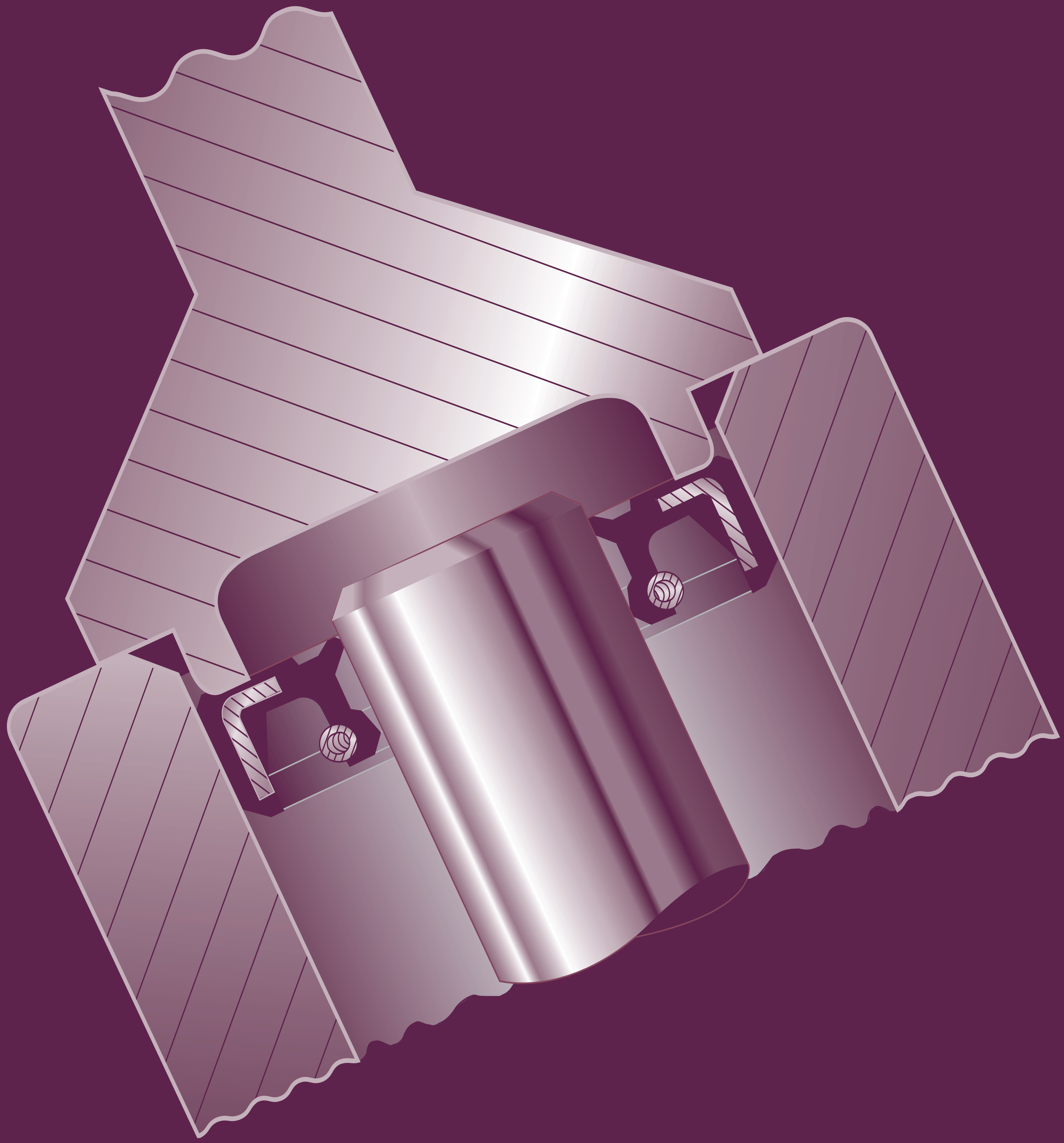
LEGEND
A = Primary Lip
B = Secondary Lip
C = Flex Section
D = Seal O.D.
E = Endface
F = Interior
G = Exterior
* Figure 175 (rubber O.D.) only

Table 49: Summary of Manufacturing Defects

Primary Function	Defect	Critical	Major	Minor
Dirt Exclusion	Blisters	-	A	B, C, D, E, F, G
(Figure 175 only)	Cuts	B, C	A	D, E, F, G
	Damaged Mold	-	A, B, C	D, E, F, G
	Deformation	-	A, B, C	D, E, F, G
	Dirty Mold	-	A, B, C	D, E, F, G
	Flash	-	A	B, C, D, E, F, G
	Improper Trim	B, C	A	E
	Knit Line	-	A, B, C	D, E, F, G
	Mat. Contamination	-	A, B, C	D, E, F, G
	Nicks	B, C	A	D, E, F, G
	Non-Fills	B, C	A	D, E, F, G
	Poor Bond	C	D, E	F, G
	Porosity	-	A, B, C	D, E, F, G
	Rough Trim	-	A	B, C
	Scoop Trim	-	A	B, C
	Scratches	-	A, B, C	D, E, F, G
	Spiral Trim	-	A	B, C
	Surf. Contamination	-	A, B, C	D, E, F, G
	Tears	B, C	A, D, E	F, G
	Unbonded Flash	-	A, B, C, E	D, F, G

LEGEND
A = Primary Lip
B = Secondary Lip
C = Flex Section
D = Seal O.D.
E = Endface
F = Interior
G = Exterior

Table 49: Summary of Manufacturing Defects



installation & performance

Installation.

Having given due consideration to all aspects of the shaft seal assembly (including the bore, the shaft, and the design of the seal itself), what remains is to successfully install the seal you've selected or designed. Simple as that may sound, proper installation is not always easy. As a matter of fact, it can be quite difficult, which explains why improper installation is the number one cause of shaft seal failure.

With that in mind, here are some things to be aware of as you install your shaft seal. Good installation practices include inspection of the seal and other components just prior to installation, use of the proper equipment during installation, and protection of the assembly after installation.

PRE-INSTALLATION INSPECTIONS

Because it is the most important part of the seal, the sealing lip should be closely inspected to make sure there are no nicks or tears at any point around its circumference. You should also be certain that the lip is not turned back. Either a torn or turned lip will quickly fail in service. If the seal design incorporates a garter spring, you should check to be sure that the spring hasn't been displaced out of its groove as a result of handling. The seal O.D. should also be free of damage such as cuts, dents, or scores. No matter where it is located, any damage should immediately disqualify a new seal from use. And because damage (especially the hard-to-see variety) can result from service, you should never reinstall a used seal. Since even the smallest amount of outside contamination can harm a sealing system, new seals that pass inspection should still always be wiped clean before installation.

Beyond the seal itself, the bore (and housing) should also be carefully examined. Housing edges must be free of burrs or other imperfections that can easily damage the O.D. of an incoming seal. The edges of the bore must be burr- and nick-free and follow standard guidelines as shown in ***Table 46*** (page 135). The chamfer of the bore should also be free nicks or burrs, and it should follow the guidelines shown in ***Figure 138*** (page 134).

"Improper installation is the number one cause of shaft seal failure."

The shaft should be inspected to ensure there are no nicks or burrs, and it should be finished to RMA standards as shown in ***Table 34*** (page 119). The chamfer of the shaft should also be free of nicks or burrs, and it should follow the guidelines shown in ***Figure 130*** (page 126). Above all, keep in mind that a new seal should never be run in the same shaft wear track as an old seal.

If grooving of the shaft surface exists from previous service, three options are available. A spacer can be placed within the bore (behind the seal) in order to make sure the seal contacts an ungrooved portion of the shaft. Alternatively, a metallic wear sleeve may be fitted over (and, if need be, adhered to) the damaged shaft to provide a more suitable sealing surface. Use of a thin-walled sleeve will normally make it possible to thus retrofit a damaged shaft surface without changing the seal dimensions or design. In some cases, it may be necessary to refinish or replace the shaft.

USE OF INSTALLATION EQUIPMENT

Because a shaft seal should never run without proper lubrication, both the seal lip and the shaft should be lubricated (typically with the same oil or grease being sealed) prior to installation of the seal. In addition to making the installation both easier *and* less potentially damaging to the seal, lubrication also helps protect the sealing element during the initial break-in period. Continued lubrication minimizes wear and maximizes service life. ***Figure 177*** shows a double lip seal packed with grease between the primary and secondary lips. When two seals are installed in tandem, the entire space between the two seals may be packed with grease. In some cases, seal suppliers will pre-lube seals upon request.

GREASE PACKING

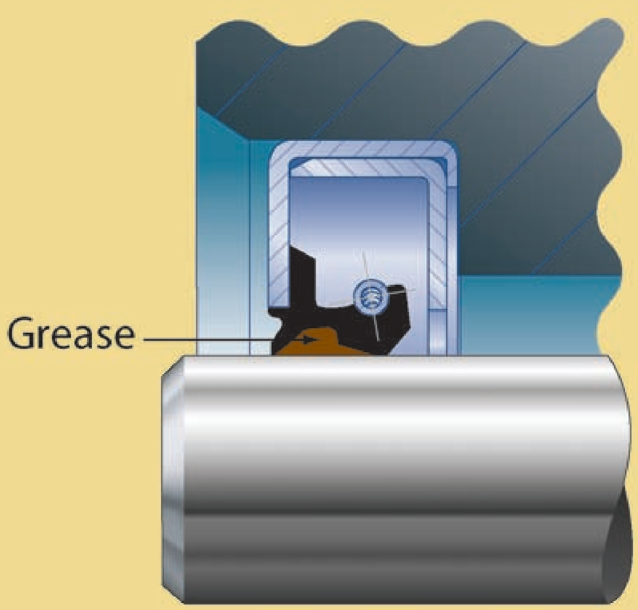


Figure 177: Provides Seal & Shaft Lubrication

As obvious as it may sound, care must be taken to install the seal in the right direction. If replacing a previously used seal, be sure to note the direction in which the primary lip of the old seal was facing, then ensure that the primary lip of the new seal faces the same way. Failure to orient the seal properly relative to the fluid being sealed will result in instantaneous leakage upon startup.

But even if it's facing the right direction, the seal must also be installed at a right angle (perpendicular) to the centerlines of both the shaft and the bore. Anything less than a right angle means the seal is angularly misaligned (cocked). Installing a standard shaft seal into a housing can be a problem if there is no counterbore to help align and seat the seal. Even if initial installation is perfect, the absence of a counterbore makes it easy for the seal to become cocked when the shaft is slipped into place (see ***Figure 178***). Seal cocking is most common in blind designs that prevent the field assembly team from seeing whether the seal is properly seated.

SEAL COCKING

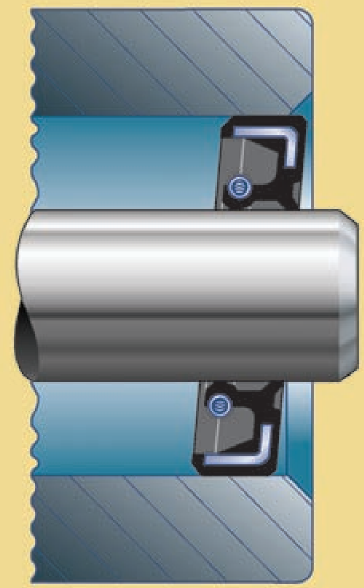
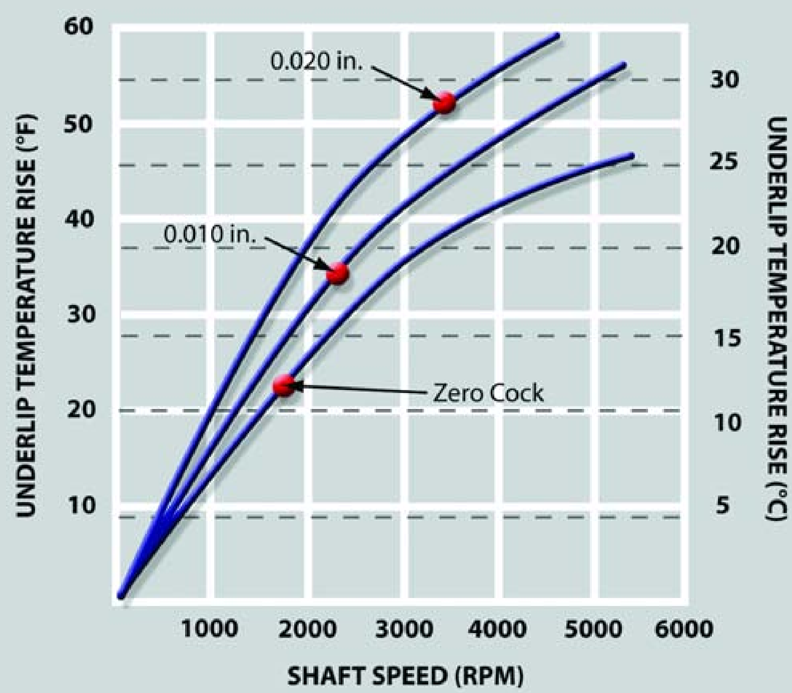


Figure 178: Misaligned (Cocked) Seal

Seal cocking is problematic for several reasons. For example, it can contribute to uneven wearing of the sealing lip. Cocking also increases the chances that any garter spring might become dislodged from its groove in the lip (a phenomenon known as *spring pop out*). Damage to the lip itself and/or to the seal O.D. is also more likely. In addition, seal cocking increases the temperature at the interface between the shaft and the seal lip. High temperature hastens hardening and cracking of the seal. ***Table 50*** shows how underlip temperature increases as a result of seal cocking.

Sometimes seal cocking can be prevented through use of special designs; the non-standard "TAY" shaft seal is a perfect example. The TAY design features a



Material: Nitrile (NBR); Shaft Size: 76.2 mm (3.000 inch); Lubricant: SAE 30
DRO: 0.13 mm (0.005 inch); STBM: 0.13 mm (0.005 inch)
Pressure: Zero; Sump Level: Full; Sump Temperature: 93° C (200° F)

Table 50: Underlip Temperature vs. Shaft Speed For Various Values of Seal Cocking

FLANGE SEAL

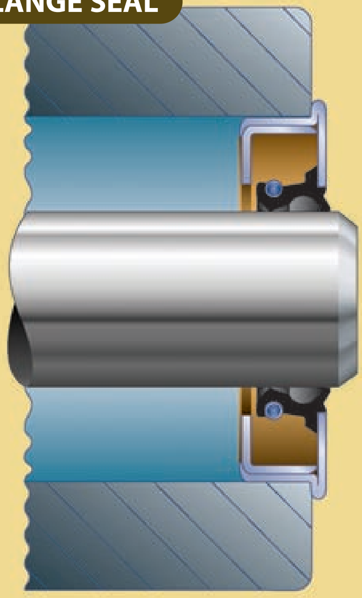


Figure 179: "TAY" Seal Installed

flange on the outside diameter of the seal. This flange helps to ensure that the seal seats properly against the housing face during initial installation (see ***Figure 179***). Because the presence of the flange also helps prevent the possibility of subsequent misalignment, seal cocking concerns can be prevented.

Seal cocking can also be prevented by the use of a properly designed or selected installation tool, and the right amount of force. Without the right tool, it's easy to damage or distort the seal lip or case. Installation tools (such as the one shown in ***Figure 180***) are generally made of steel and are designed to contact the seal near the O.D. (where the seal is most resistant to deformation). Pressing at the more vulnerable seal I.D. can distort the case and lead to leakage in service. The tool shown in ***Figure 180*** is also advantageous because it is designed to bottom out on the housing face, thus preventing seal cocking.

Depending on the specifics of the application, the tool may also be designed such that it can keep pressing until the seal bottoms out (as in a stepped housing, see ***Figure 181***) or until the tool bottoms out against the shaft face (see ***Figure 182***).

Depending on the application, the seal may be installed with the shaft already in place, or the shaft may be fitted into the assembly after the seal has been installed into the housing.

INSTALLATION TOOLS

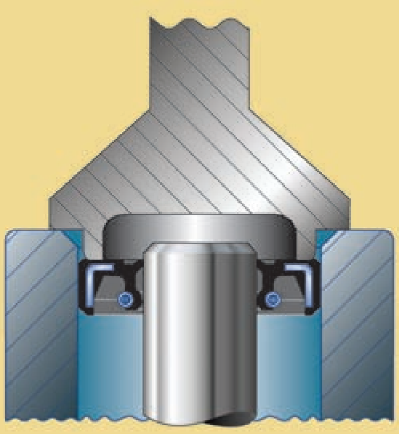


Figure 180: Bottoms on the Housing Face

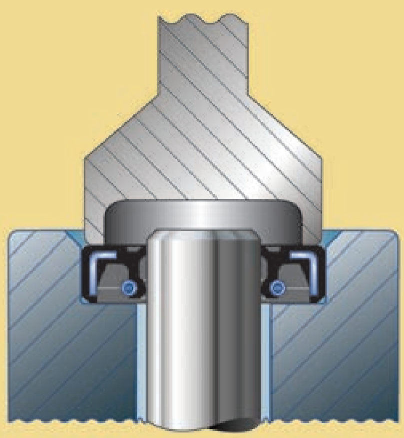


Figure 181: Bottoms Seal in a Stepped Housing

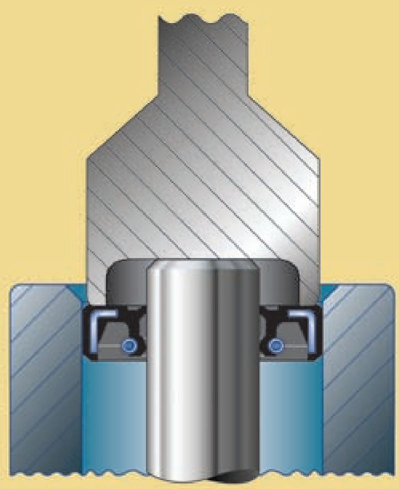


Figure 182: Bottoms on the Shaft Face

Either way, it is necessary to protect the sealing lip from splines, keyways, burrs on the shaft, and improperly finished chamfering areas. Use of a shield and/or lubrication can help. An assembly cone (acting as either a cap, as in ***Figure 183***, or a sleeve, as in ***Figure 184***) can be temporarily fitted onto or over the shaft to facilitate avoidance of potential hazards. If *lip inversion* (the turning over of the sealing lip due to friction during installation) is a concern, the cone can be oiled, or it can be made of a low-friction material such as PTFE. Assembly cones must be routinely inspected to make sure they have no burrs or scratches.

But even proper tools are no guarantee of good installation. Without the right amount of force, the seal will still not be installed properly. If installation is taking place in a factory, this force is often supplied by a hydraulic or pneumatic press. Use of such automated presses can eliminate guesswork by providing a constant force with which to push the seal into its housing. Because this force is closely controlled, the chances of inadvertently damaging the seal are greatly reduced. Installations done in the factory also tend to be cleaner due to the ability to more closely control the work environment.

ASSEMBLY CONES

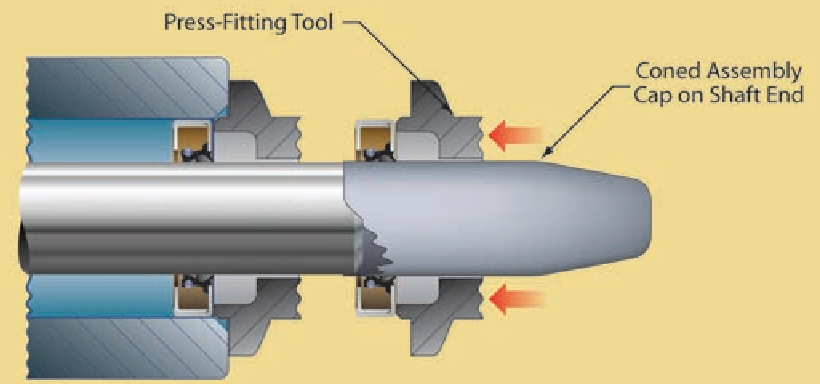


Figure 183: Cone As Cap

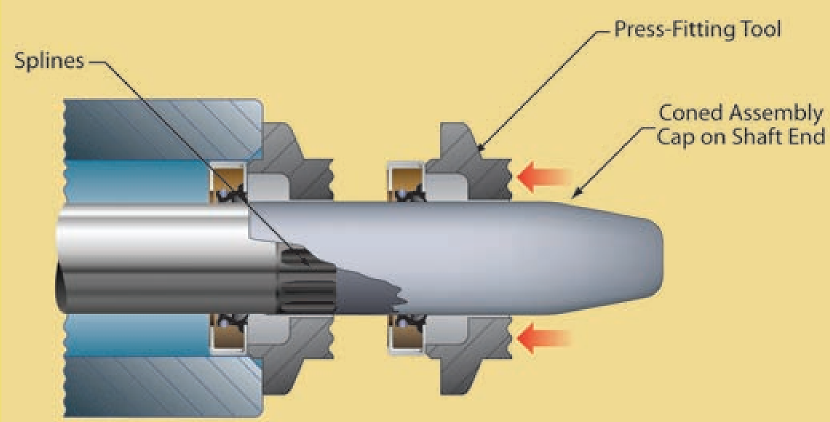
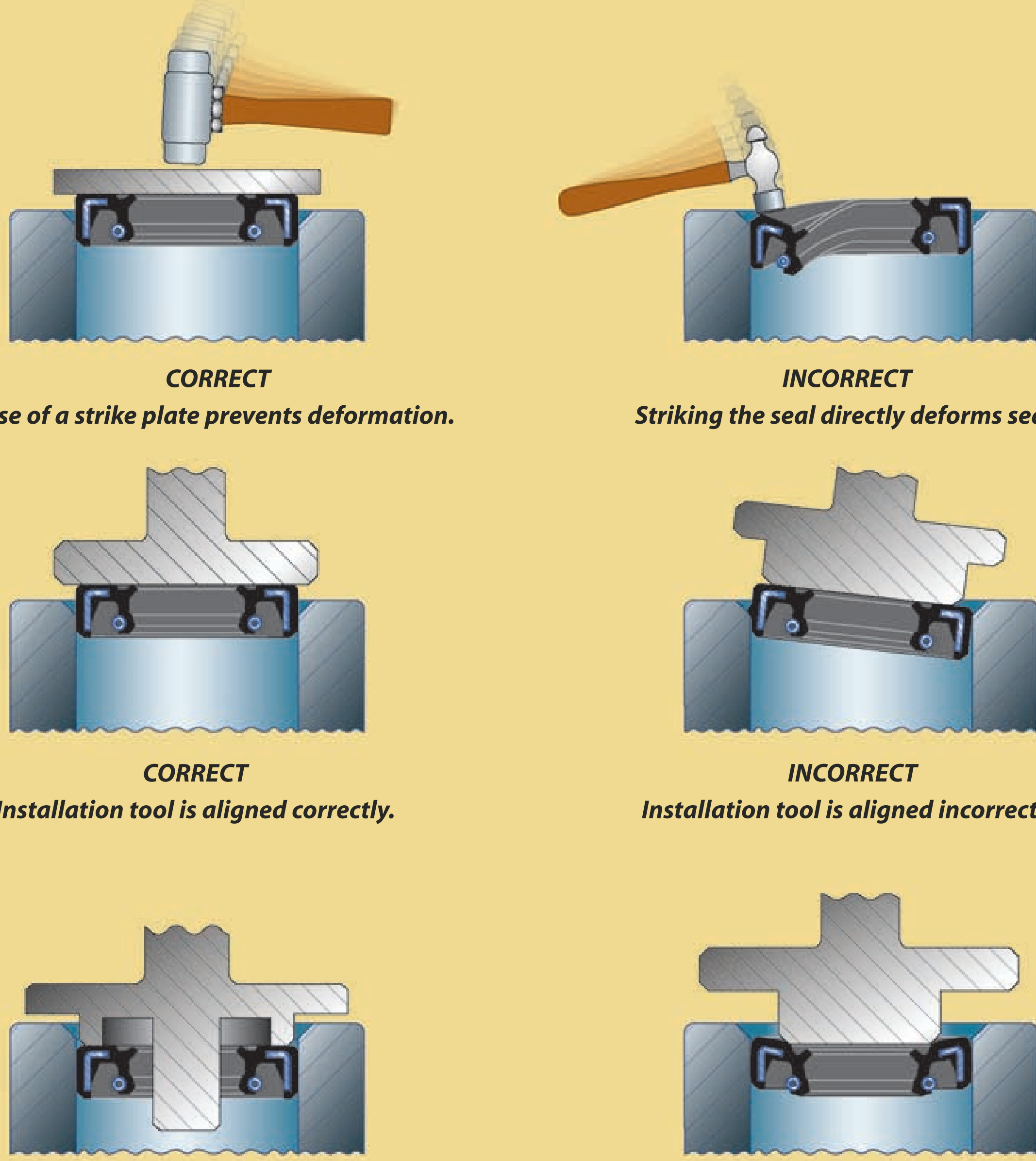


Figure 184: Cone As Sleeve

In contrast, installations done in the field tend to be both dirtier (due to reduced environmental control) and less precise (due to forced reliance on less reliable installation aids). Tools such as those used in factories aren't as common in the field. Installation force is often provided solely by hand-operated arbor presses or soft-faced mallets (used in conjunction with strike plates, see ***Figure 185***, next page). The results are almost always less consistent than with automated equipment. And because of

CORRECT VS. INCORRECT INSTALLATION



CORRECT
Use of a strike plate prevents deformation.

INCORRECT
Striking the seal directly deforms seal.

CORRECT
Installation tool is aligned correctly.

INCORRECT
Installation tool is aligned incorrectly.

CORRECT
Installation tool engages the seal case.

INCORRECT
Wrong size tool damages seal.

Figure 185: Proper Installation Is Critical

these inconsistencies, the chances of inadvertently damaging the seal are greater.

For metal O.D. seals, it may be helpful to apply a thin coat of bore sealant to the O.D. of the seal. This adhesive coating can help the seal stay in place (and form a more leak-proof seal) once it's installed in the housing. Be careful, however, that any sealant you may use does not contaminate other parts of the seal, particularly the lip, or the surface of the shaft. Such contamination can impair or inhibit the functioning of the seal by blocking the proper development of the lip-shaft interface.

With rubber O.D. seals, the rubber portion can be damaged during installation if proper lead-in chamfers are not built into the design. Care must also be taken due to a phenomenon known as *springback*. Springback is the tendency of a shaft seal with a rubber O.D. to unseat itself slightly following installation due to shearing stresses between the rubber and the housing bore. An exaggerated example of springback is shown in ***Figure 186***.

SPRINGBACK

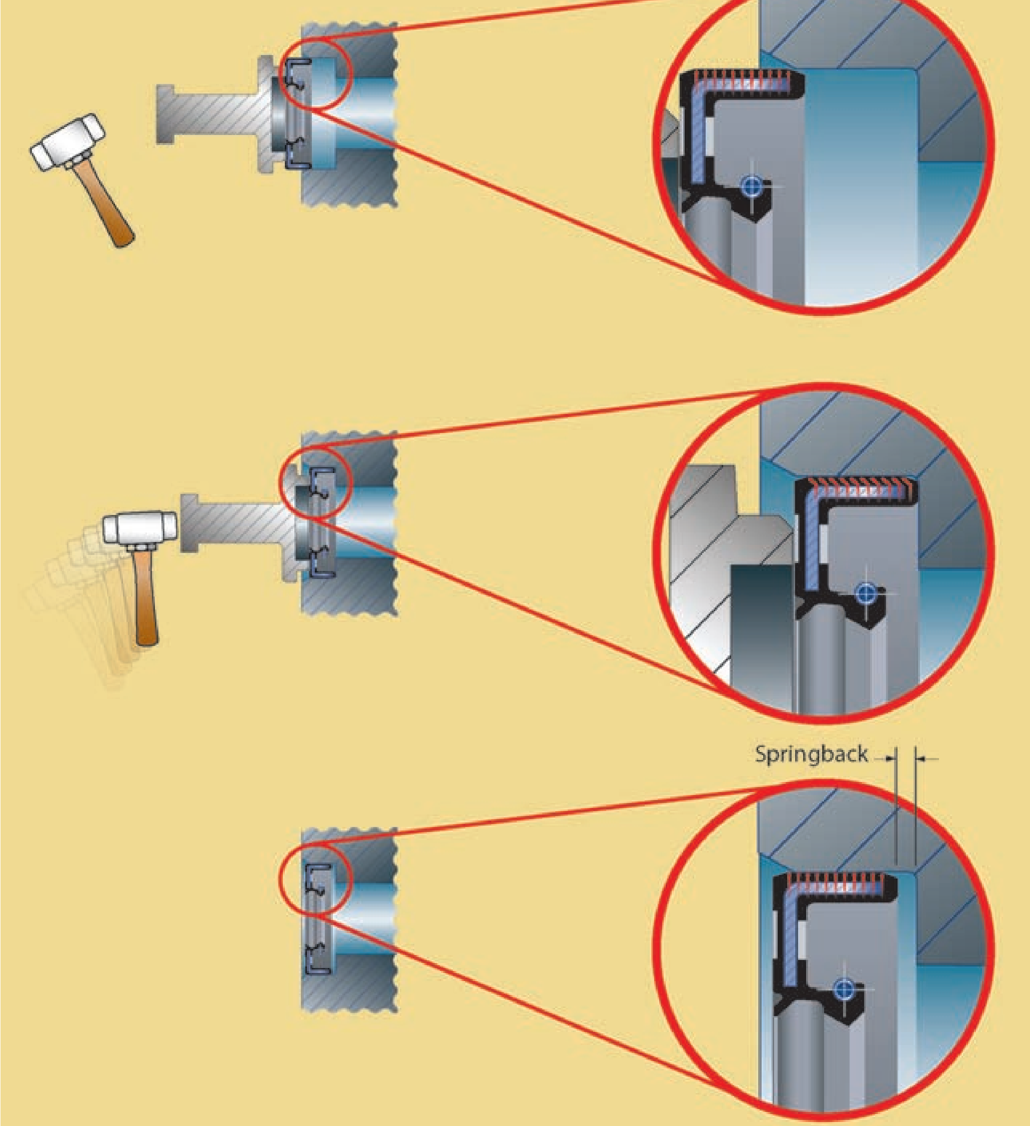


Figure 186: Seal Unseating Due To Shearing Stresses

POST-INSTALLATION CARE

Following installation, it's a good idea to double-check one last time that the proper functioning of the seal will not be impinged on by other parts of the assembly. Such impingement could lead to unplanned (and unwanted) friction, heat, and wear, all of which can contribute to premature seal failure.

You should also be cognizant of any treatments (such as painting or cleaning) to which the assembly in general (and the seal in particular) may be subjected. Unless proper precautions are taken to shield the seal, such treatments can impair its functionality and thus hasten its failure. For example, you should be careful to ensure that a painted assembly does not remain in the bake oven any longer than necessary to cure the paint; prolonged heat exposure can be very detrimental to the seal lip material.

Finally, the overall design of the assembly can hold hidden dangers that, if not addressed, can doom any shaft seal. For example, the assembly must provide adequate ventilation for the internal pressure within the seal area. Without proper ventilation, pressure can build to dangerous levels, even to the point of blowing the seal out of its housing. If a vent exists, make sure it is not clogged during painting. Clogged vents can cause excessive pressure to build up that could blow out the seal.

Table 51 can serve as a checklist to ensure that you have considered all of the important installation issues described in this section.

- ☐ Is the seal in good condition?
- ☐ Is the spring properly in place, or has it been displaced during handling?
- ☐ Have you carefully wiped the seal clean (so as not to damage it)?
- ☐ Have you made sure there are no nicks, scratches, or spiral grooves on the shaft surface?
- ☐ Have you pre-lubricated the seal's lip for initial break-in?
- ☐ Are you installing the seal with the lip facing in the right direction?
- ☐ Are you installing the seal at a right angle to the centerlines of the bore and shaft?
- ☐ Have you made sure that bore adhesives do not contaminate the shaft or seal lip?
- ☐ Have you taken measures to keep the lip from being damaged by passing over splines, threads, or burrs on the shaft?
- ☐ Have you inspected the bore to make sure there are no burrs or scratches?
- ☐ Have you ensured proper protection for the seal during painting or cleaning operations?
- ☐ Is adequate ventilation provided for internal pressure in the seal area?
- ☐ Have you made sure that assembly components do not rub and that any vents are not clogged?

Table 51: Seal Installation Checklist

Effects of Environment.

The environment in which a shaft seal operates has a huge influence on the design of the seal. The fluid being sealed and any external fluids that may contact the seal must be considered. Both the underlip and ambient (sump) temperatures are important. The speed of the shaft and the pressure under which the seal must operate are both critical factors, especially in high-pressure applications. The type and amount of lubrication must be fully understood. If necessary, provisions must be made for excluding contaminants. Keep in mind that the tabular data presented in this section is intended to show trends, not absolute results.

"The environment in which a shaft seal operates has a huge influence on the design of the seal."

FLUIDS

The fluid being sealed and any external fluids that may contact the seal must be carefully considered when selecting not only the lip material (for more on this, see our Chemical Compatibility Guide on ***pages 36*** and ***37***) but also when finalizing the overall design of the shaft seal. In many cases, the seal will be sealing lubricating liquids such as mineral oil or oil-based grease. Synthetic oil and synthetic grease are also commonly sealed with shaft seals. Other fluids include hydraulic oils, silicone oils, acids, alkalies, and solvents.

The specific fluid being sealed can determine some of the design characteristics of the seal. For example, a seal whose primary job is to seal grease will typically be designed differently than a seal retaining oil. Relative to oil, grease is a more viscous (thicker) fluid. Because of this increased viscosity, grease is less able to leak out of any small gaps that might develop between the sealing lip and the shaft. The development of such gaps could be prevented through use of a garter spring in the lip design; the spring would help maintain consistent positive interference between the lip and the shaft. But because grease is less able to flow (and therefore less likely to leak out if small gaps do develop), grease seals typically don't employ a spring. Oil, on the other hand, is less viscous (thinner) than grease and more able to leak out of a small gap. For this reason, oil seals typically do incorporate a garter spring in the lip design.

TEMPERATURE

In terms of shaft seal design, there are two important and interrelated temperature variables. The first of these is *sump temperature*, which is the temperature of the fluid—such as oil— within an assembly's sump (cavity containing fluid). At a given shaft speed, as sump temperature increases, the *underlip temperature* also increases. Underlip temperature is

EFFECT OF SUMP TEMPERATURE

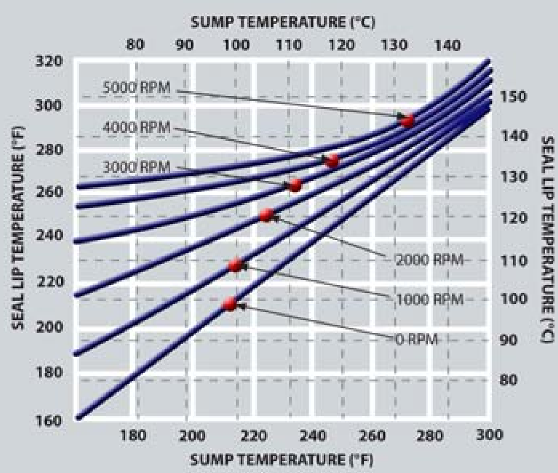


Table 52: Underlip Temperature vs. Shaft Speed For Various Sump Temperatures

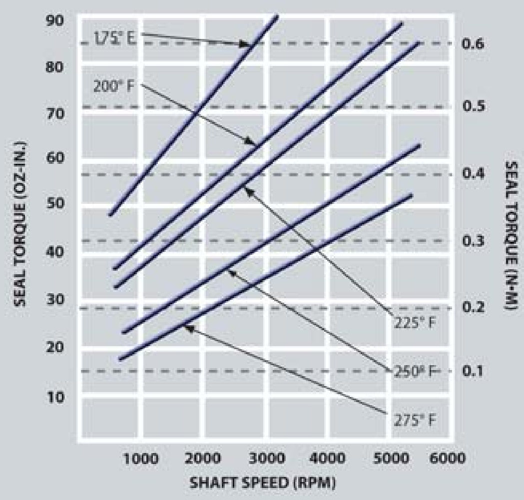


Table 53: Seal Torque vs. Shaft Speed For Various Sump Temperatures

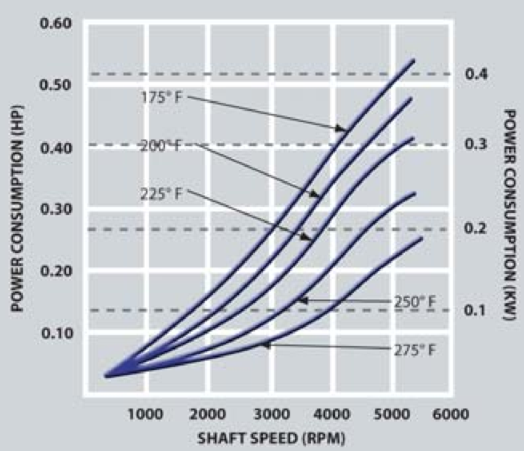


Table 54: Seal Power Consumption vs. Shaft Speed For Various Sump Temperatures

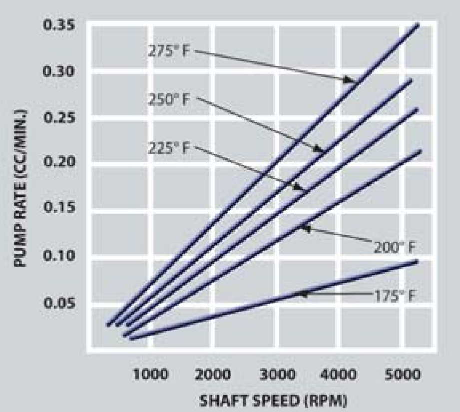


Table 55: Seal Pump Rate vs. Shaft Speed For Various Sump Temperatures

Material: Nitrile (NBR); Shaft Size: 76.2 mm (3.000 inch); Shaft Speed: Variable; Lubricant: SAE 30
DRO: 0.13 mm (0.005 inch); STBM: 0.13 mm (0.005 inch); Seal Cock: Zero
Pressure: Zero; Sump Level: Full; Sump Temperature: Variable

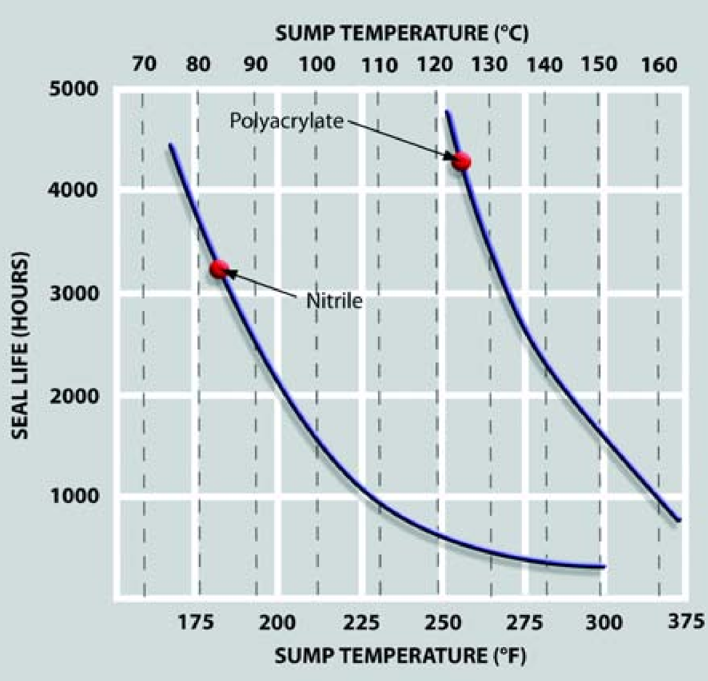
the temperature of the fluid between the shaft and the sealing lip at the contact point. ***Table 52*** (page 187) shows the relationship between underlip temperature and shaft speed at different sump temperatures.

If the shaft speed is held constant, both torque and power consumption decrease as the sump temperature increases. This is because fluid viscosity decreases with increasing temperature, and a less viscous fluid doesn't require as much energy to shear during shaft rotation. The relationship between seal torque and shaft speed at different sump temperatures is shown in ***Table 53*** (page 187).

Table 54 (page 187) shows the relationship between seal power consumption and shaft speed at different sump temperatures.

The inherent pumping ability (due to the presence of microasperities on the sealing lip) will increase as sump temperature increases at a constant shaft speed. The relationship between seal pump rate and shaft speed at different sump temperatures is shown in ***Table 55*** (page 187).

Though pump rate increases as shown in ***Table 53***, the overall result of increasing sump temperature is reduced seal life. As shown in ***Table 56***, the lifespan of a typical NBR seal, for example, is approximately cut in half for every 25° F (14° C) increase in sump temperature. This is because increasing sump temperature hastens hardening of the nitrile lip, making it less flexible and therefore less able to maintain proper contact with the shaft. But not all materials react as nitrile does. As can also be seen in ***Table 56***, polyacrylate polymer formulations have proven to be more resistant than nitrile to heat-induced lip hardening and therefore typically have longer seal life. Other materials (such as hydrogenated nitrile and fluoroelastomer) can also further extend seal life.



Material: Nitrile (NBR); Shaft Size: 76.2 mm (3.000 inch)
Shaft Speed: 2165 RPM; Lubricant: SAE 30
DRO: 0.13 mm (0.005 inch); STBM: 0.13 mm (0.005 inch)
Seal Cock: Zero; Pressure: Zero
Sump Level: Centerline; Cycle: 20 hours on, 4 off

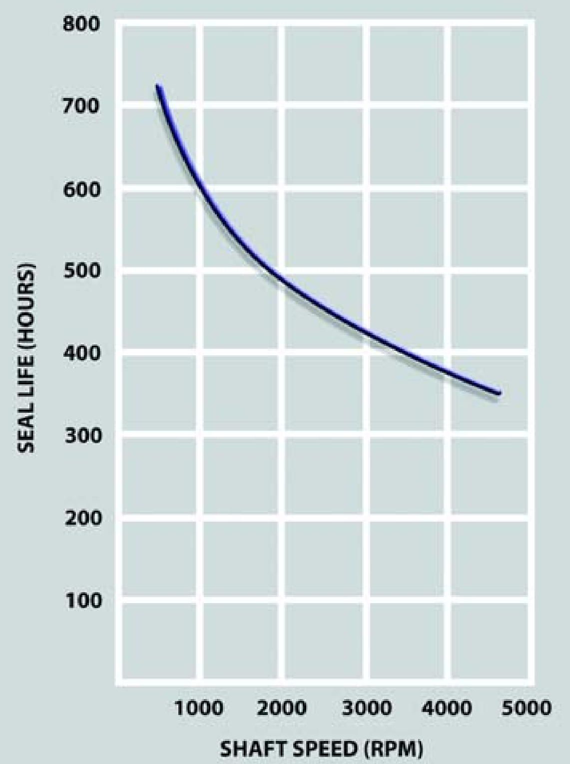
Table 56: Seal Life vs. Sump Temperature

SPEED

Seal life is reduced as shaft speed increases at a constant sump temperature. Generally speaking, seal life is cut in half if shaft speed increases ten times (for example, from 500 to 5000 rpm). A fuller example of the effect of shaft speed on seal life is shown in ***Table 57***.

PRESSURE

Just as increases in sump temperature and/or shaft speed can shorten seal life, an increase in pressure can also hasten seal failure. Standard shaft seal designs are typically effective in environments with maximum continuous pressures of 4 to 5 pounds per square inch (p.s.i.). Standard seals are not adequate when subjected to pressures greater than 7 p.s.i. ***Table 58*** shows some examples of how different seal cross-sections have been designed to handle increasing pressures.



Material: Nitrile (NBR); Shaft Size: 76.2 mm (3.000 inch)
Shaft Speed: Variable; Lubricant: SAE 30
DRO: 0.13 mm (0.005 inch)
STBM: 0.13 mm (0.005 inch)
Seal Cock: Zero; Pressure: Zero
Sump Level: Centerline; Cycle: 20 hours on, 4 off
Sump Temperature: 121° C (250° F)

Table 57: Seal Life vs. Shaft Speed

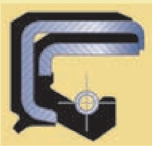
Pressure		Sample Seal Design	
(psi)	(bar)		
0 - 7	0 - 0.48		**TC**
7 - 50	0.48 - 3.45		**TCV**
50 - 150	3.45 - 10.3		**TCN**
150 +	10.3 +		**HP1**

Table 58: Pressure Classifications

Shaft Speed		Maximum Pressure	
(ft / min)	(m / sec)	(psi)	(kPa)
0 - 1000	0 - 5.076	7	48.3
1001 - 2000	5.081 - 10.152	5	34.5
2001+	10.157+	3	20.7

Table 59: Maximum Pressure Limits vs. Speed (Standard Shaft Seals)

The amount of pressure that a seal can safely withstand is dependent on the speed at which the shaft is moving. As shaft speed increases, the maximum pressure limit decreases. ***Table 59*** shows this relationship between shaft speed and the maximum pressure that standard shaft seals can withstand.

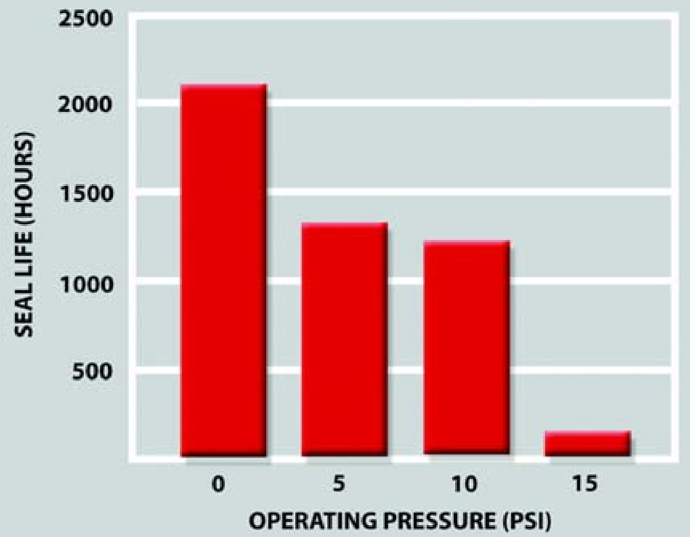
As shown in ***Figure 187***, higher pressures can distort the sealing lip, resulting in a greater-than-desired amount of contact between the air side surface of the lip and the shaft. Under some conditions, the pressure will distort the seal such that the sealing tip leaves the shaft. This phenomenon, known as *bell-mouthing*, generates more friction and heat, and, as a result, greater wear. This increased wear is evident on the air side surface of the lip rather than at the lip contact point. This greater wear shortens seal life. In some extreme instances, high pressures have even been known to force the seal out of the bore or to tear the elastomeric lip away from its metal case. The relationship between increasing operating pressures and seal life for standard seal designs is shown in ***Table 60***.

In response to the need for seals capable of withstanding higher pressures, designers have developed a variety of non-standard shaft seals. These non-standard designs typically feature shorter lip lengths and increased beam thickness in order to minimize lip distortion. These specialty designs also often incorporate a greater bonding area between the lip

LIP DISTORTION

Figure 187: Effects of Pressure on Standard Shaft Seal Lip

and the metal case in order to lessen the chances that the lip will be torn away. In many applications, a back-up ring made of filled PTFE or nylon is used to support the seal lip when high pressure is applied.



Material: Nitrile (NBR); Shaft Size: 76.2 mm (3.000 inch); Shaft Speed: Variable
Lubricant: SAE 30; DRO: 0.13 mm (0.005 inch)
STBM: 0.13 mm (0.005 inch); Seal Cock: Zero; Pressure: Variable
Sump Level: Centerline; Sump Temperature: 93° C (200° F)

Table 60: Seal Life vs. Operating Pressure (Standard Shaft Seals)

LUBRICATION

Because they are dynamic in nature, shaft seals typically require lubrication in order to function smoothly and to prolong seal life. This lubrication is needed at the point where the lip contacts the shaft. Without such lubrication, the lip will undergo accelerated and excessive wear to the point of seal failure. The fluid (such as oil) being sealed typically provides this vital lubrication.

The ability of oil to lubricate is closely tied to its viscosity, or its resistance to flow. The thicker an oil is, the more viscous it is, i.e. the less it flows. Several different methods exist for gauging an oil's viscosity, but it is often noted in centistokes (cSt) at 40° C or 100° C. The International Organization for Standardization (ISO) developed a viscosity classification system based on an oil's cSt at 40° C. Shown in ***Table 61***, this system assigns a viscosity grade number based on the median whole number within a given oil's viscosity grade range. The higher the grade number, the more viscous the oil.

Lubrication performs two vital roles in relation to rotary shaft sealing. By

ISO Viscosity Grade Number	Viscosity Grade Range (cSt at 40° C)	
	Minimum	Maximum
2	1.98	2.42
3	2.88	3.52
5	4.14	5.06
7	6.12	7.48
10	9.00	11.0
15	13.5	16.5
22	19.8	24.2
32	28.8	35.2
46	41.4	50.6
68	61.2	74.8
100	90.0	110
150	135	165
220	198	242
320	288	252
460	414	506
680	612	748
1000	900	1100
1500	1350	1650

Table 61: ISO Viscosity Classification System

forming a consistent film between the sealing lip and the shaft, lubrication reduces friction and minimizes wear. The presence of proper lubrication keeps the lip from undergoing what is known as *stick-slip*, irregular or jerky motion caused by alternating instances of adhesion and slipperiness between the lip and the shaft. In the absence of sufficient lubrication, stick-slip is often an issue for lips made from compounds lacking internal lubrication or from materials with a low modulus. In addition to reducing friction and wear, lubrication also dissipates heat. The sealed but circulating fluid draws heat away from the sealing lip.

The amount (or level) of fluid in the sump has a profound impact on seal life. A reduced sump level means a reduced ability to dissipate the heat generated at the lip-shaft interface. Less heat dissipation means a higher underlip temperature, and a higher underlip temperature translates into shortened seal life. ***Table 62*** shows the relationship between underlip temperature rise and shaft speed at different sump fill levels. An assembly in which lubrication remains low will typically necessitate use of a specialty compound (such as PTFE) to reduce friction and thus help compensate for reduced lubrication. ***Table 63*** shows the relationship between sump fill level and seal life for a standard nitrile material.

EFFECT OF SUMP FILL LEVEL

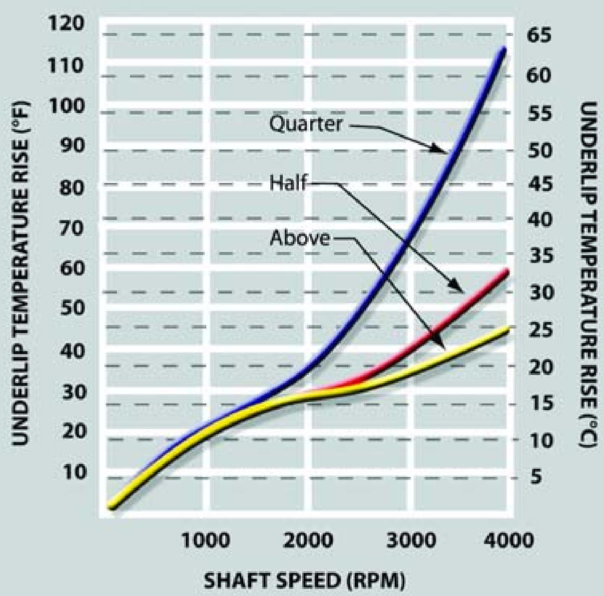


Table 62: Underlip Temperature Rise vs. Shaft Speed For Various Sump Fill Levels

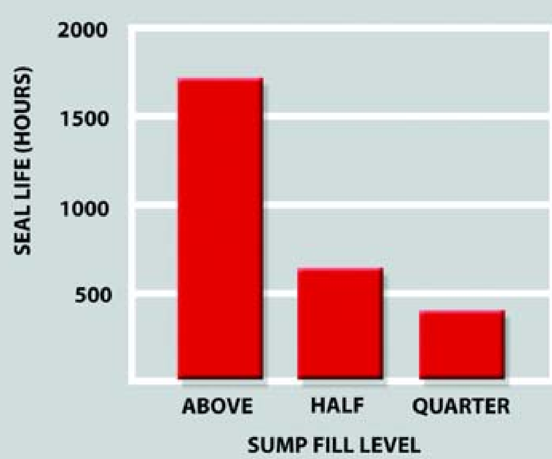


Table 63: Seal Life vs. Sump Fill Level (at 2165 RPM)

Material: Nitrile (NBR); Shaft Size: 76.2 mm (3.000 inch); Lubricant: SAE 30; DRO: Zero
STBM: Zero; Seal Cock: Zero; Pressure: Zero; Sump Level: Various; Sump Temperature: 93° C (200° F)

Many lubricants contain chemical additives that can cause lip degradation. ***Table 64*** lists many of these additives and their purpose. You should also note the necessity of pre-lubrication prior to installation of the seal. Pre-lubrication of the sealing lip using the same fluid as is being sealed is imperative to avoid dry-running of the seal upon the shaft at startup. Because of problems that may arise from fluid incompatibilities, it is not advisable to mix lubricants.

Type	Purpose	Typical Chemical Compounds
Antifoamants	Reduce foam in crankcase	Silicone polymers
Corrosion inhibitors	Neutralize acids, prevent corrosion from acid attack	Overbased (high pH) metallic sulfonates, phenates, fatty amines
Dispersants, detergents	Keep sludge, carbon, and other deposits suspended in the oil	Succinimides, neutral metallic sulfonates, phenates, polymeric detergents, amine compounds
Extreme pressure (EP) antiwear additives	Form protective film on engine parts, reduce wear, prevent scuffing and seizing	Zinc dialkylidithiophosphates; tricresyl phosphates, organic phosphates, chlorine and sulfur compounds
Friction modifiers	Reduce coefficient of static friction between metal surfaces	Long-chain polar compounds (amides, phosphates, phosphites, acids)
Metal deactivators	Form surface films so metal surfaces do not catalyze oil oxidation	Zinc dialkylidithiophosphates, metal phenates, organic nitrogen compounds
Oxidation inhibitors	Prevent or control oxidation of oil, formation of varnish, sludge, and corrosive compounds; limit viscosity increase	Zinc dialkylidithiophosphates, aromatic amines, sulfurized products, hindered phenols
Pour point depressants	Lower "freezing" point of oils, ensuring free flow at low temps	Low molecular weight methacrylate polymers
Rust inhibitors	Prevent rust formation on metal surfaces by formation of surface film or neutralization of acids	High base additives, sulfonates, phosphates, organic acids or esters, amines
Seal compatibility enhancers	Maintain softness and pliability of seal material	Phosphate esters, phenols, aromatic esters, sulfones or lactones
Viscosity index improvers	Reduce rate of viscosity change w/ temperature; reduce fuel consumption; maintain low oil consumption; allow easy cold starts	Polyisobutylene, methacrylate, acrylate polymers, olefin copolymers; may incorporate dispersant groups

Table 64: Lubricating Oil Additives

EXCLUDING CONTAMINANTS

Because external contaminants (such as dust, dirt, water, and mud) making their way into an assembly can damage both a shaft seal and its surroundings (including the shaft itself and the housing bore), seals in dirty environments often feature a secondary sealing lip. Whereas the function of the primary lip is to seal in fluid, the function of this secondary lip is to seal out contaminants. In some particularly dirty and demanding applications, more than one secondary lip may be utilized to help keep the seal environment clean.

A secondary lip (also known as a dirt lip, dust lip, or excluder lip) can take two forms: radial or axial. Though different, radial and axial dirt lips have characteristics in common. Both originate at the heel of the elastomeric member in order to seal on the air side (thus preventing harmful contaminants from reaching the primary lip-shaft interface). Unlike many primary lips, radial and axial dirt lips are *nonsprung*, meaning they do not incorporate a garter spring into their design.

The chief difference between a radial dirt lip and an axial dirt lip is in orientation relative to the shaft. A radial dirt lip extends radially down toward the shaft (see ***Figure 188***). Depending on the specific seal design, a radial dirt lip may or may not actually contact the shaft. Radial dirt lips find their widest use in low speed applications.

Rather than extending toward the shaft like a radial dirt lip, an axial dirt lip extends axially up away from the shaft (see ***Figure 189***). An axial lip makes contact with a vertical surface, one that is perpendicular to the centerline of the shaft. This surface is typically a wear sleeve, bearing surface, or housing. Studies have shown that axial dirt lips last longer and are thus preferable to

DIRT LIPS

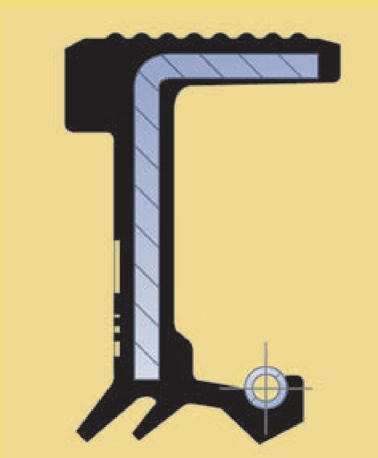


Figure 188: Radial Dirt Lips

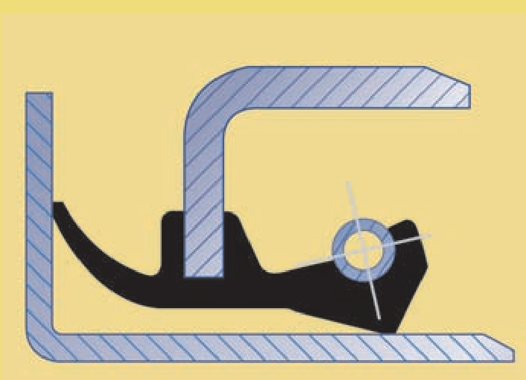


Figure 189: Axial Dirt Lip

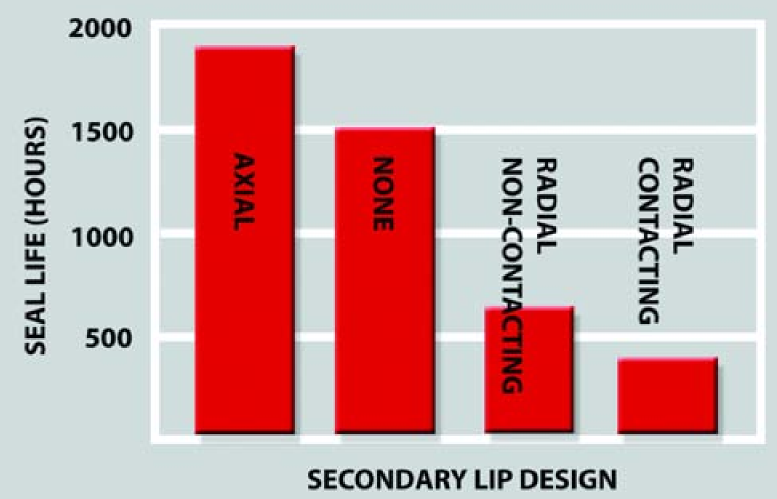
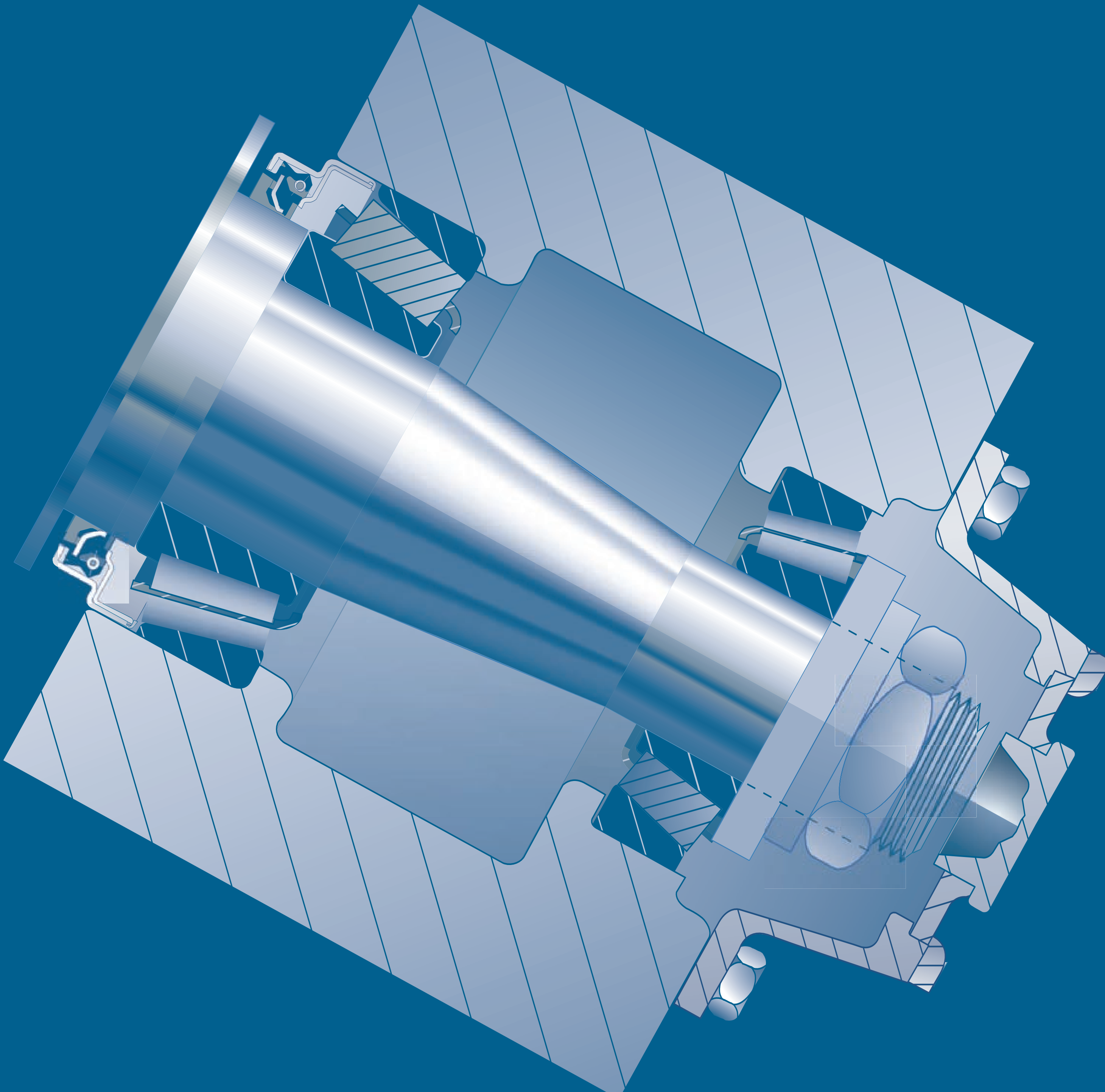


Table 65: Comparison of Secondary Lips

radial dirt lips for excluding dust and dirt in high-speed applications. Secondary lips are compared in ***Table 65***. Various levels of contamination are classified in ***Table 66***.

Contamination Level	Contamination Type	Mode of Contamination	Suggested Seal
None	None	None	SC
Light	Dust	Airborne	TC
Moderate	Dust, Dirt, Water	Splashing	Double lip w/ grease
Heavy	Dust, Dirt, Water, Mud	Partial Immersion	Axial Dirt Lip
Extreme	Dust, Dirt, Water, Mud Sand, Gravel, Abrasive Slurry	Full Immersion	QA / Mud block seal

Table 66: Contamination Classification



sample applications

Industrial.

Shaft seals are used in a variety of interesting industrial applications. For example, shaft seals are common in gearbox assemblies, hydraulic pumps and motors, reciprocating applications (as wiper seals), and in washing machine tubs. Here's a look at how shaft seals function in these various industrial environments.

GEARBOX SEALS

Shaft seals are commonly used in gearboxes, which convert high speed input from an electric motor into low speeds that drive various machines or conveyors. A cut-away view of a gearbox assembly is shown in ***Figure 190***.

Typical gearbox applications will have two shaft seal styles: one for the input seal and one for the output seal(s). The input seal is a higher speed (usually 1750 RPM) seal. Because both external and internal contamination can be a problem in gearbox assemblies, new generation, high-tech gearbox

"Shaft seals are used in a variety of interesting industrial applications."

GEARBOX ASSEMBLY

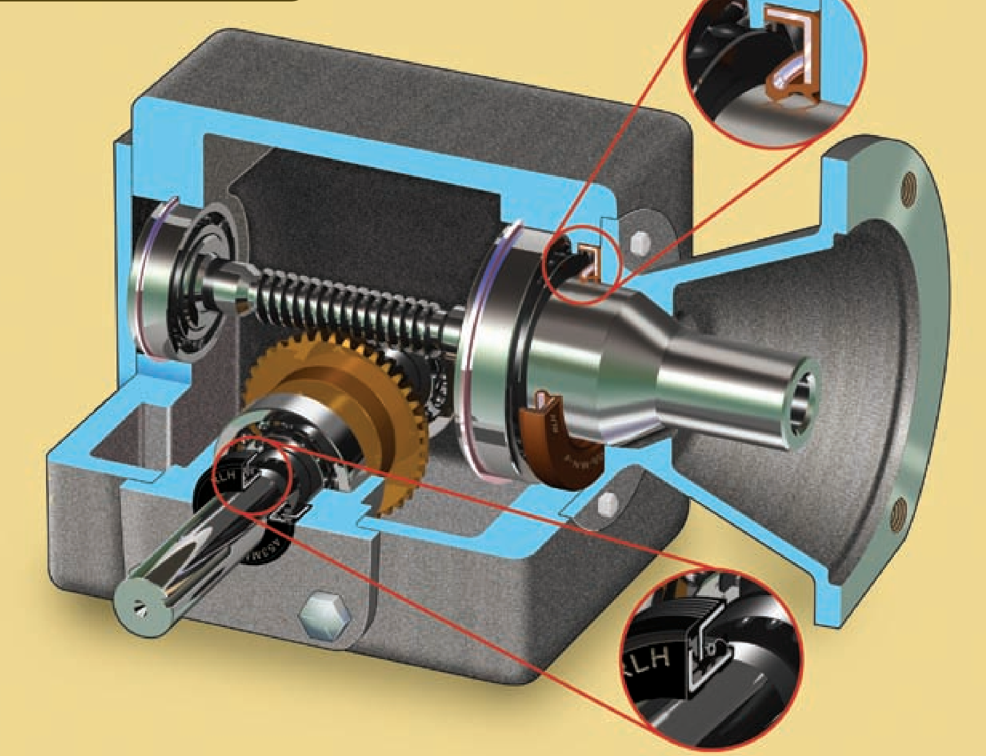


Figure 190: Input and Output Seals

GEARBOX SEALS

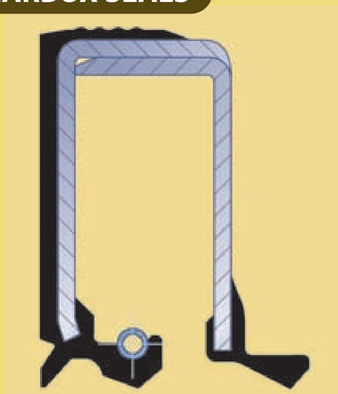


Figure 191: Input Seal

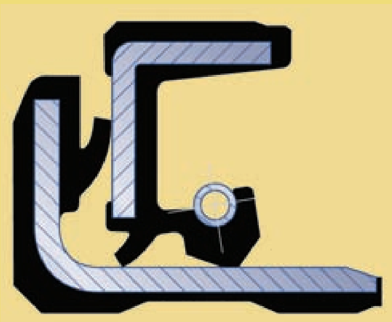


Figure 192: Output Seal

Figure 193: Output Seal

seals incorporate additional contamination exclusion, both internally and externally. The input seal is often a TC or TCW design made with fluoroelastomer (FKM). The input seal shown in ***Figure 191*** combines a TC design with an oil side contamination exclusion lip to prevent dirt and metal particles in the oil from reaching the primary sealing lip.

In addition to the input seal, there will also be one (or more) output shaft locations. The output seal usually operates at lower speeds than the input seal. (Actual output speed depends on the internal gear ratios set up in the gearbox.) But because the speeds are typically lower for output seals than for input seals, different cross-sections (designs that would generate too much friction and drag if used as input seals) can be used as output seals.

The cross-section of a typical output seal is shown in ***Figure 192***. Note that this is a QA-style design to exclude heavy contamination. Another possible QA-style output seal design is shown in ***Figure 193***; this design features an even greater number of contamination exclusion lips.

HYDRAULIC PUMP & MOTOR SEALS

Shaft seals used as hydraulic pump and motor seals can encounter both severe speeds (8000 RPM) and high pressures (1500 psi), so special designs are required. The seal in ***Figure 194*** is a crimped PTFE seal for use in high-pressure hydraulic motors. It features two PTFE lips held in place within the case by metal spacers. As shown in ***Figure 195***, a rubber seal with a PTFE back-up/support ring incorporated into the cross-section can also be used for high-pressure applications.

RECIPROCATING (WIPER) SEALS

Hydraulic and pneumatic applications often make use of special "WP" shaft seal designs as wiper seals. These designs utilize thickened sealing lips (without garter springs) intended for dust and dirt wiping (scraping) in reciprocating hydraulic and pneumatic cylinder applications. A wide variety of lip configurations are possible, as are both metal and rubber O.D.s. Six different WP-lip designs are shown in ***Table 67***. Wiper / scraper compounds are typically made from 90-durometer nitrile or polyurethane.

HYDRAULIC PUMP & MOTOR SEALS

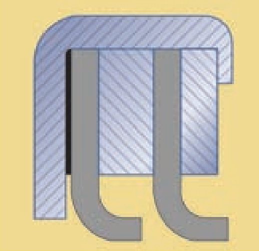


Figure 194: Crimped PTFE Seal

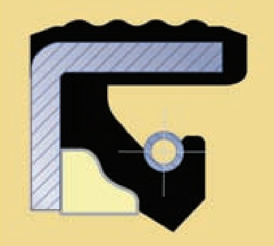
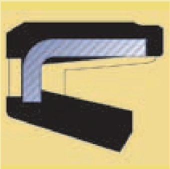
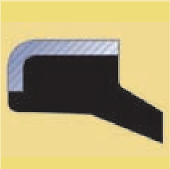
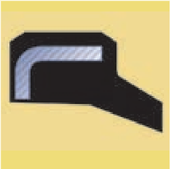


Figure 195: Rubber Seal w/PTFE Back-Up

Type	Description
	**WPB** Press-in, metal-encased rod wiper
	**WPC** Press-in rod wiper w/ flat rubber O.D.
	**WPK** Dual lip rod wiper / pressure seal w/ metal insert
	**WPM** Rod wiper w/ rubber-covered fluid side
	**WPR** Rod wiper w/ metal insert & rubber O.D.
	**WPV** Rod wiper w/ metal O.D.

Parameters For All WP-Lip Designs

Rod Diameter	Maximum Misalignment
0.500″	0.006″
1.000″	0.008″
2.000″	0.009″
3.000″	0.010″
4.000″	0.012″

Maximum recommended rod speed is 200 FPM.

Maximum recommended pressure is 4 psi.

Maximum recommended stroke length is 78″.

Table 67: WP-Lip Designs

NORMAL VS. TUB SEAL DESIGN

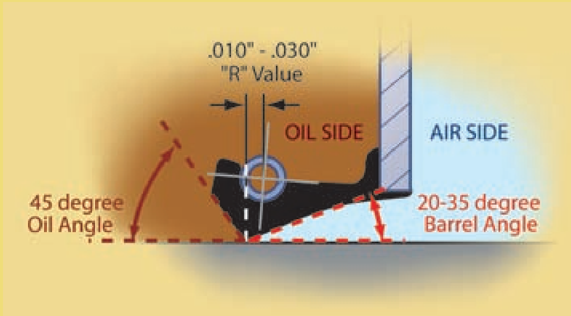


Figure 196: Normal Seal Design

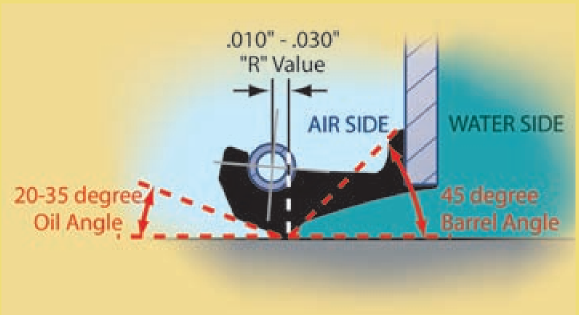


Figure 197: Tub Seal Design

WASHING MACHINE SEALS

One of the most interesting uses of shaft seals is in washing machines. Though it might at first glance seem simple, sealing a washing machine tub is a demanding application. Water must be contained, but the water, bleaches, detergents, and other washing products can easily corrode the metallic portions of the seal – the case and the spring.

What allows a tub seal to function effectively has everything to do with lip design. A comparison between the contact point of a normal seal and that of a tub seal is shown in ***Figures 196*** and ***197***. Note that the contact angles and the "R" value on the sprung lip of the tub seal are reversed compared to standard seal designs. This reversal is what allows a tub seal to prevent water and suds from getting into the machine's grease-filled bearings. And even as it keeps water out, the sprung lip also helps keep grease in. The reversed contact angles and "R" value also place the garter spring on the air side of the application, so water cannot reach it. This helps protect the garter spring from corrosion. ***Figures 198*** and ***199*** show two different tub seal designs.

WASHING MACHINE SEALS

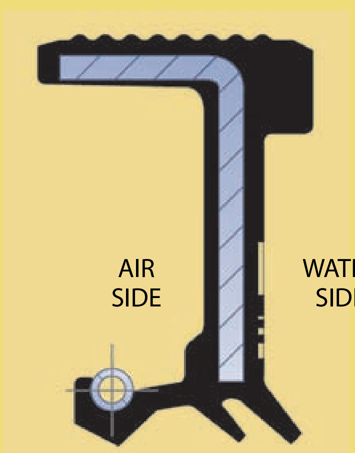


Figure 198: Single Case

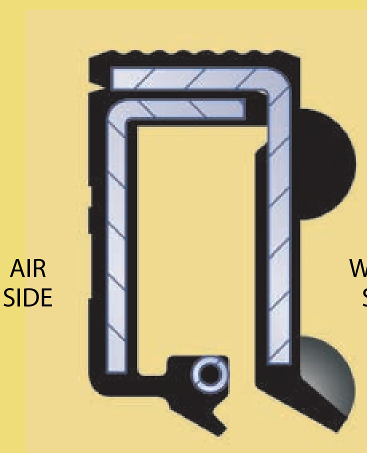


Figure 199: Double Case

sample applications

Automotive.

Because of the many ways in which shaft seals may be configured, they are common in a wide variety of automotive applications. These include usage as diesel and gasoline engine seals, valve stem seals, transmission seals, wheel (axle) seals, off-highway applications, heavy-duty pinion seals, power steering seals, shock absorber seals, and air conditioning seals. What follows is a closer look at the ways in which shaft seals function in these automotive environments.

DIESEL ENGINE SEALS

Diesel engine rear crankshaft applications (such as is shown in ***Figure 200***) can be particularly challenging for shaft seal designers, and for a number of reasons. High crankshaft speeds and large shaft diameters are common. This combination produces high shaft surface speed that results in high lip temperatures. The seal lip is also poorly lubricated because the area is splash lubricated. This causes the underlip temperature to be even higher. Large, random shaft deflections caused by piston slap can make it very difficult

"Because of the many ways in which shaft seals may be configured, they are common in a wide variety of automotive applications."

DIESEL ENGINE REAR CRANKSHAFT APPLICATION

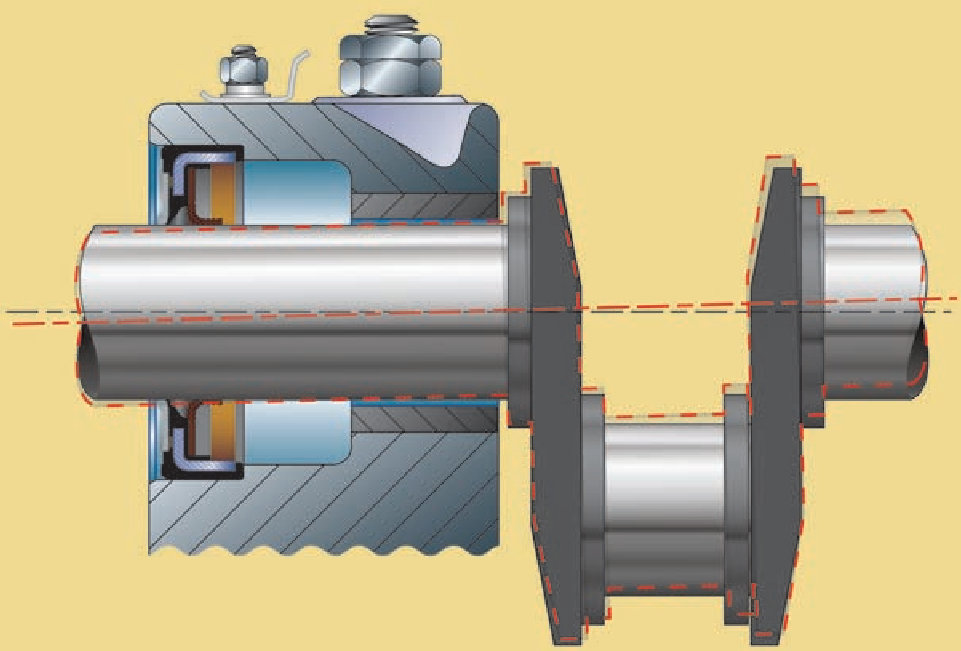


Figure 200: Challenging Environment

for the sealing lip to follow the shaft surface properly. Stick-slip or lip chatter can also occur, further increasing temperature. Oil degradation and coking can cause sludge to accumulate on the sealing lip. And as if all this weren't enough, diesel oils also contain additives that can degrade elastomers and hasten leakage.

Because of the high temperatures, early diesel engine crankshaft seals were made from either silicone or fluoroelastomers. Because of incompatibility with hydrocarbons and oil additives, silicone seals often became soft and disintegrated. FKM seals resisted chemical attack, but the lubricating properties of newer diesel oils were less than with older oils. Poor lubrication resulted and, in turn, led to lip chatter and stick-slip. Higher temperatures were generated, and seal damage was common. High temperatures also caused the oil to burn and sludge to build up on the sealing lip, resulting in leakage. Blisters would also often form on the air side of the seal lip. All things considered, diesel engine rear crankshaft applications are very tough.

Seal designers attempting to address these issues have found that the most effective seals for diesel engine crankshafts are those that feature a sealing lip made of a blend of PTFE (Teflon®) and fillers. The inherent slickness of the PTFE compensates for poor lubrication and eliminates stick-slip, which in turn helps to keep underlip temperature down. PTFE is also very resistant to chemical attack, making degradation of the lip by oil additives unlikely.

Conventional PTFE shaft seals are made by crimping one or more PTFE wafers into a steel case along with steel spacers and gaskets (which prevent leakage between the inside of the case and the PTFE lips). An example of this type of design

PTFE SEAL

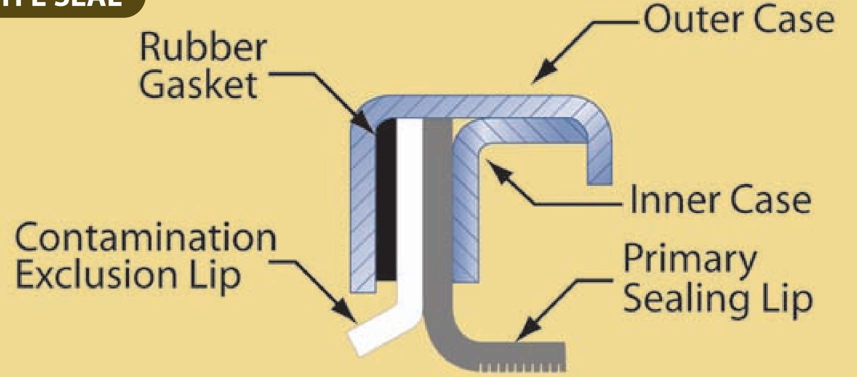


Figure 201: PTFE Wafers In Steel Case

PTFE SEAL

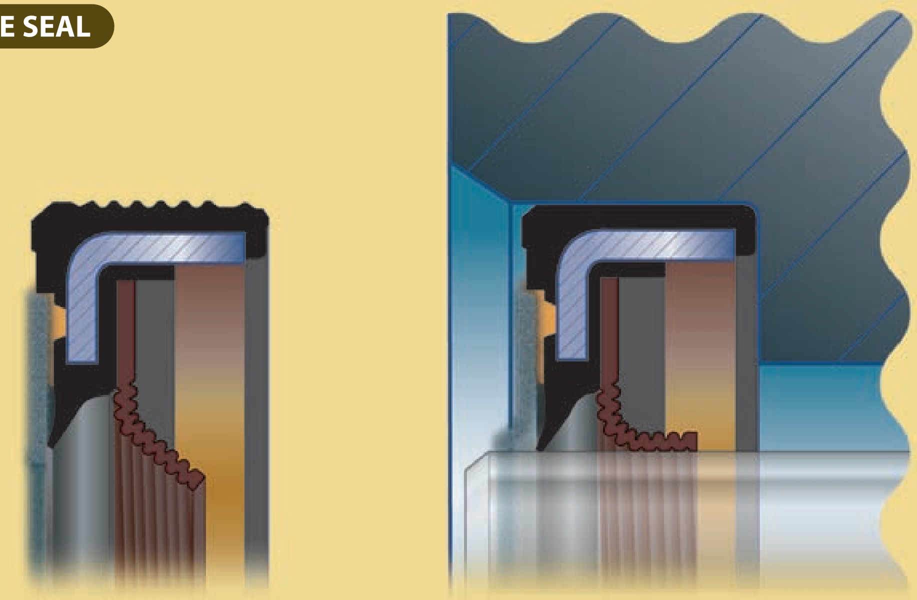


Figure 202: PTFE Lip With Dual Coined Spiral Pattern

is shown in ***Figure 201.*** The seal shown in this illustration has a radial dirt lip to exclude contamination and a rubber gasket between the inside of the metal case and the dirt lip to prevent internal leakage. Because PTFE is stiffer than traditional elastomers, it is not able to develop the microasperities vital to in-pumping of oil. To compensate for this, a spiral groove must be machined or coined into the surface of the primary sealing lip; this groove screws oil back into the sump. The seal is unidirectional and can be used only if the shaft always rotates in the same direction.

R.L. Hudson & Company is proud to offer an alternative design featuring a PTFE lip bonded to a rubber substrate, which is, in turn, bonded to a metal case. An example of this is shown in ***Figure 202***. Notice that the PTFE lip features a dual coined spiral pattern. The spiral on the air side pumps oil back to the oil side (unidirectionally). The coined spiral ridges on the oil side of the lip improve lip flexibility. Notice also that this design features rubber ribs molded on the seal O.D. to improve sealing between the housing bore and the seal O.D.

Diesel engine applications also include front crankshaft and camshaft seals. An auxiliary drive seal is common as well. These are all typically smaller than the rear crankshaft seal, so the speed and temperature difficulties inherent in rear crankshaft seals are not as pronounced. And though these seals also tend to be better lubricated than rear crankshaft seals, PTFE is still the lip material of choice for these diesel applications.

GASOLINE ENGINE SEALS

Though gasoline engines incorporate crankshaft and camshaft seals like diesel engines, gasoline engine seals tend to be different from their diesel engine counterparts. As gasoline engines have gotten both smaller and more powerful, the need for increasingly heat- and additive-resistant rear crankshaft seals has caused designers to turn away from silicone (which for years was the typical engine seal material) to fluoroelastomers. FKM seals are now the norm for rear and front crankshaft, camshaft, and auxiliary shaft seals.

HALF-RUBBER, HALF-METAL O.D.

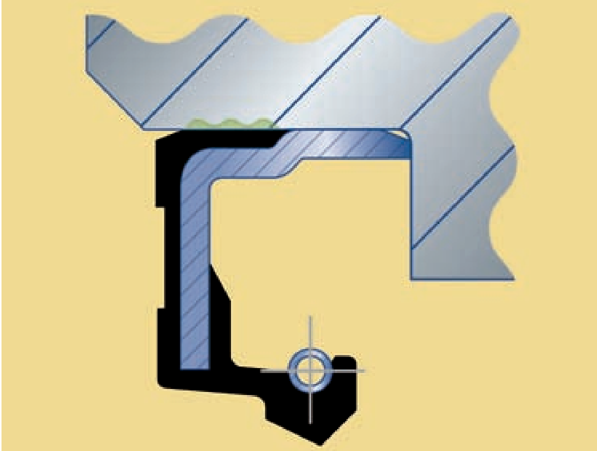


Figure 203: Excellent Retention & Sealability

VALVE STEM SEAL

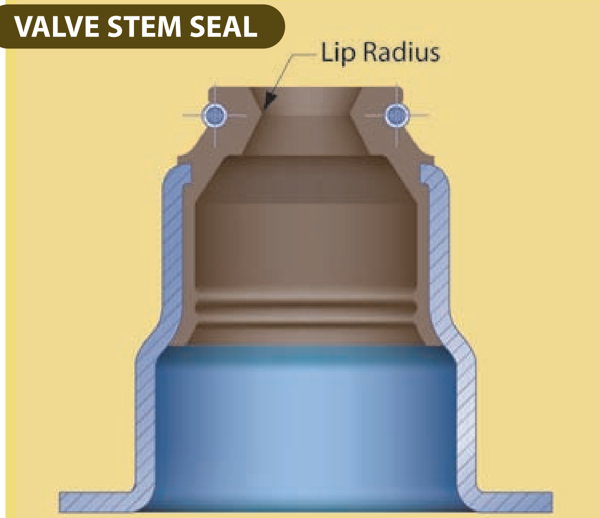


Figure 204: Lip Radius Meters Lubrication

Studies have shown that a rear crankshaft seal with a half-rubber, half-metal O.D. has advantages over either a full metal O.D. or a full rubber O.D. In particular, a seal with a half-rubber, half-metal O.D. may be more forgiving than a full rubber O.D. seal during installation. The metal portion of the half-and-half O.D. has a chamfer that helps guide the seal into the bore. The metal edge also eliminates the possibility that rubber will be sheared off of the leading edge of the O.D. during installation (as can sometimes happen with some full rubber O.D. designs). Because there is less rubber on the O.D., and a metal-to-metal pressfit, *springback* (in which a seal unseats itself after installation due to shearing stresses between the O.D. and the bore) is reduced.

Half-and-half seals are less prone to cocking in the bore, but they may require more force to install than a full-rubber O.D. shaft seal. As temperatures and pressures increase, half-and-half seals are also able to maintain higher retention force in aluminum bores. One thing to be aware of: half-and-half shaft seals are more difficult to manufacture than conventional seals, so it's imperative that extra care be given to procure quality seals. In addition, the initial cost of

VALVE STEM SEAL IN ASSEMBLY

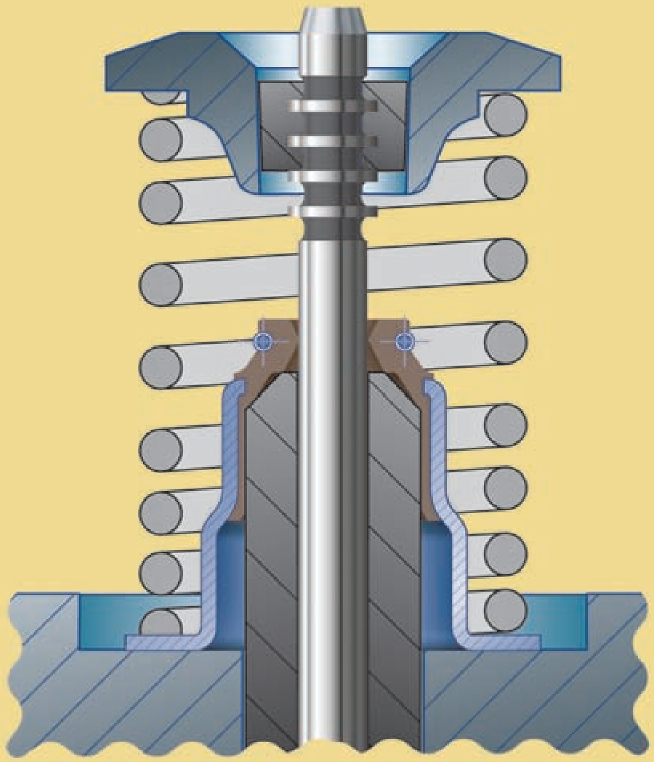


Figure 205: Stamping Flange On Seal O.D. Provides Seat For Valve Spring

the tooling is more expensive and also more expensive to maintain. The cost of a half rubber / half metal O.D. seal is often more expensive than a full rubber O.D. seal. An example of a half-rubber, half-metal O.D. seal is shown in ***Figure 203***.

VALVE STEM SEALS

Valve stem seals are sprung-lip seals pressfit over valve guides in internal combustion engines. The molded, radiused seal lip touches the valve stem, which is reciprocating within the valve guide. The magnitude of the lip radius meters (controls) the lubrication of the valve stem such that the stem receives neither too little nor too much oil. A stamping flange on the seal O.D. provides a seat for the valve spring to prevent damage to the surface of the valve seat area. A typical valve stem seal design is shown in ***Figure 204***. ***Figure 205*** shows this seal installed in an assembly.

TRANSMISSION SEALS

With their complex combination of mechanical, hydraulic, electrical, and computerized systems, automatic transmissions pose considerable challenges to seal designers. These inherent complexities in combination with

INPUT SEALS

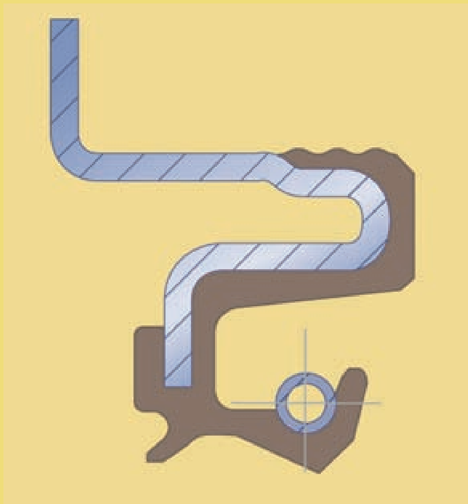


Figure 206: Single Spring-Loaded Lip, With Flange

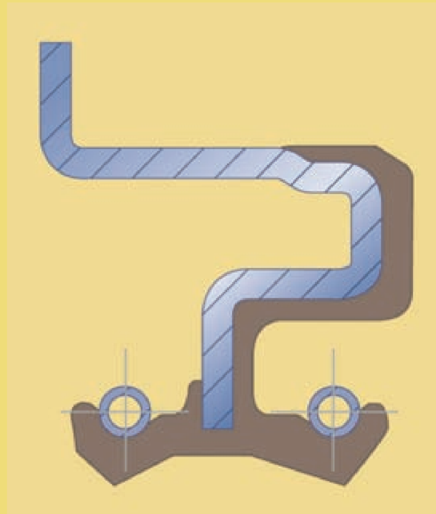


Figure 207: Dual Spring-Loaded Lip, With Flange

increasing warranty demands, higher shaft speeds, and higher temperatures make the design of shaft seals for use as transmission seals no easy feat.

Generally speaking, there are two main types of transmission designs: rear wheel drive and front wheel drive. Rear drive designs typically employ an input seal (also known as the front seal) to prevent leakage of transmission fluid at the interface between the torque converter and the transmission case. Rear drive designs also use an output seal (otherwise known as the rear seal) to prevent leakage past the output shaft (where the transmission connects to the driveshaft).

In front wheel drive designs, an input seal prevents leakage between the torque converter and the transaxle. Front drive designs employ two output seals, one at both of the

OUTPUT SEALS

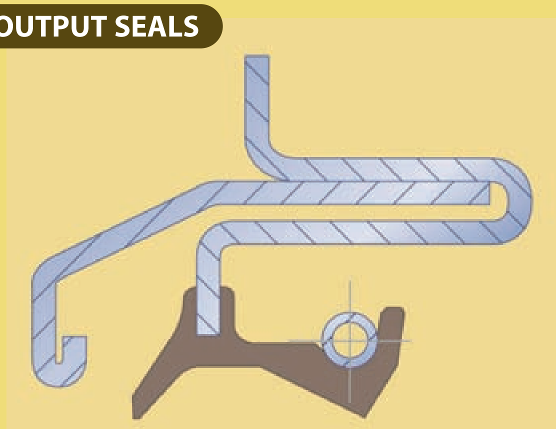


Figure 208: Non-Unitized, With Flange

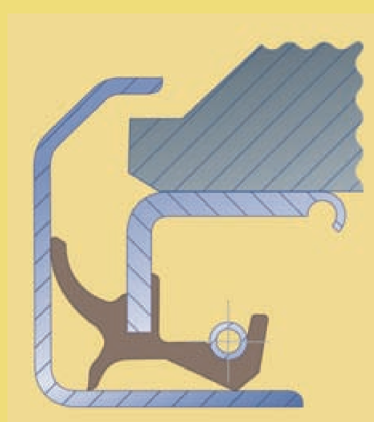


Figure 209: Unitized, With Axial Dirt Lip

opposing interfaces between the transaxle and the front drive axles.

Figures 206 and ***207*** show two examples of input seal designs. ***Figure 206*** shows a design incorporating a flange for easy removal. ***Figure 207*** shows a dual lip designed to facilitate fluid separation. This design also features a flange to aid removal.

Figures 208 and ***209*** show two examples of output seal designs. ***Figure 208*** shows a non-unitized design with shield and flange for easy removal. ***Figure 209*** shows a unitized design featuring an axial dirt lip.

WHEEL (AXLE) SEALS

Shaft seals are incorporated into automotive, truck, and off-road vehicle wheel assemblies to contain grease and to exclude dirt, mud, and other contaminants. The specific seal design typically depends on how and where the seal must function; for example, light duty versus heavy duty, and non-driven axles versus driven axles. The non-driven axles in light duty applications can typically be sealed effectively with a radial lip shaft seal such as the one shown in ***Figure 210***.

WHEEL SEAL APPLICATION

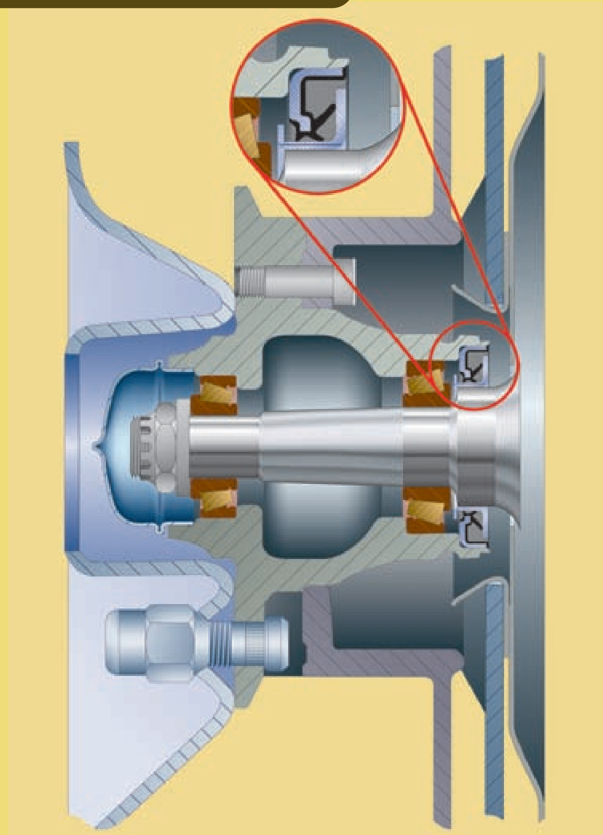


Figure 210: Non-Driven Auto & Light Truck Wheel Seal

WHEEL SEALS

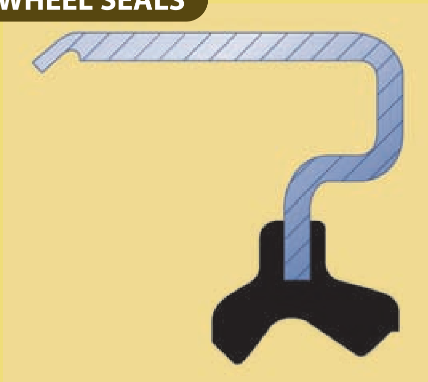


Figure 211: Non Spring-Loaded, Non-Unitized

Figure 212: Non Spring-Loaded, Unitized

Figure 213: Spring-Loaded, Unitized

Other designs are possible. ***Figure 211*** shows a simple non-spring-loaded and non-unitized design with a radial exclusion lip. Studies have shown that the most effective designs include an axial dirt lip (***Figures 212*** and ***213***) that contacts a vertical surface (e.g. a unitized wear sleeve) to exclude contaminants.

The driven axles in light duty applications are generally sealed with standard spring-loaded seals. Some wheel bearings are sealed and greased for life. These are called hub unit bearings. ***Figure 214*** shows a hub unit application for a driven axle. The spring-loaded seal prevents axle oil from entering the greased hub unit bearing.

HUB UNIT SEAL

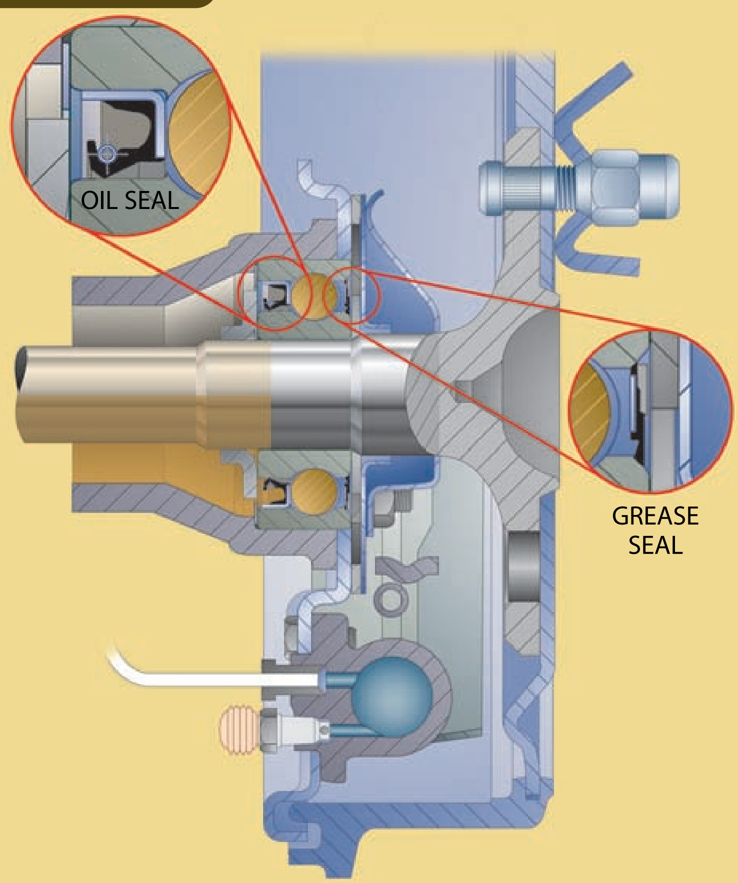


Figure 214: Driven Auto & Light Truck Wheel Seal Application

Heavy duty wheel end seals (also called oil bath seals) have become the focus of increasing demands of late. Warranty periods have been lengthened, temperature expectations raised, and contamination exclusion concerns elevated. As shown in ***Figures 215 and 216,*** both the non-driven and driven axles in heavy duty truck applications often feature oil-lubricated bearings that must be sealed. The hub

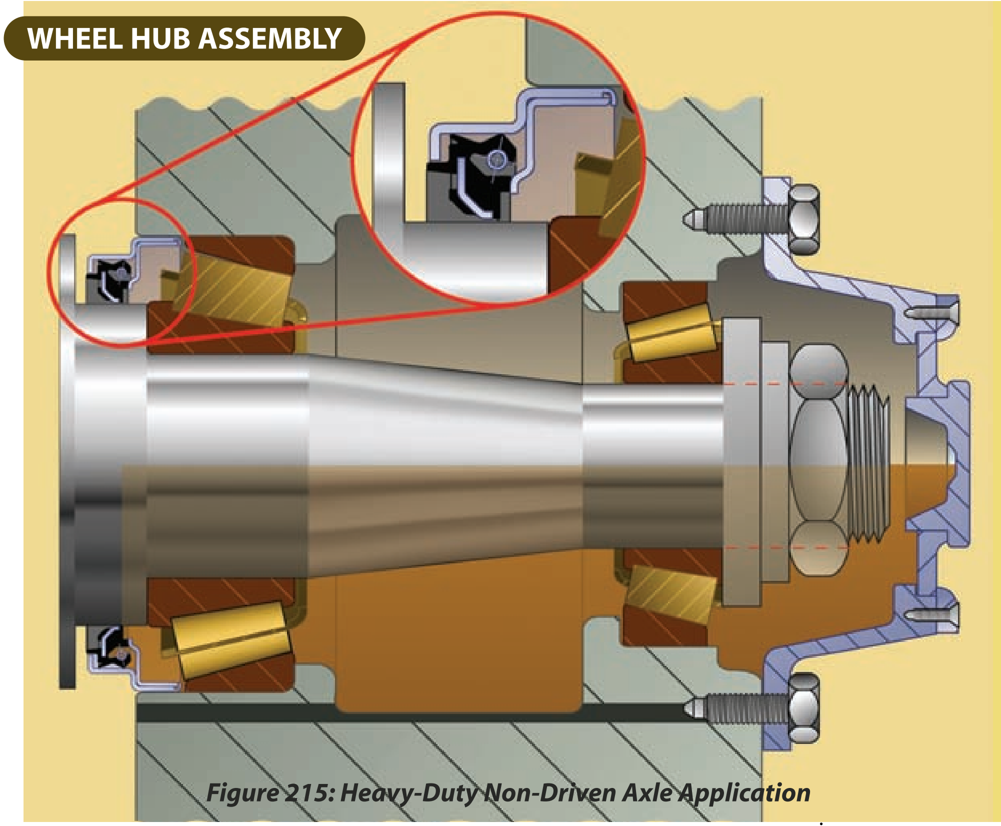


Figure 215: Heavy-Duty Non-Driven Axle Application

rotates while the spindle is stationary, and unitized seals are commonly used.

As shown in ***Figure 217*** (next page), the external seal design features a rotating seal sleeve in conjunction with a stationary sealing element. Because the O.D. of this seal is

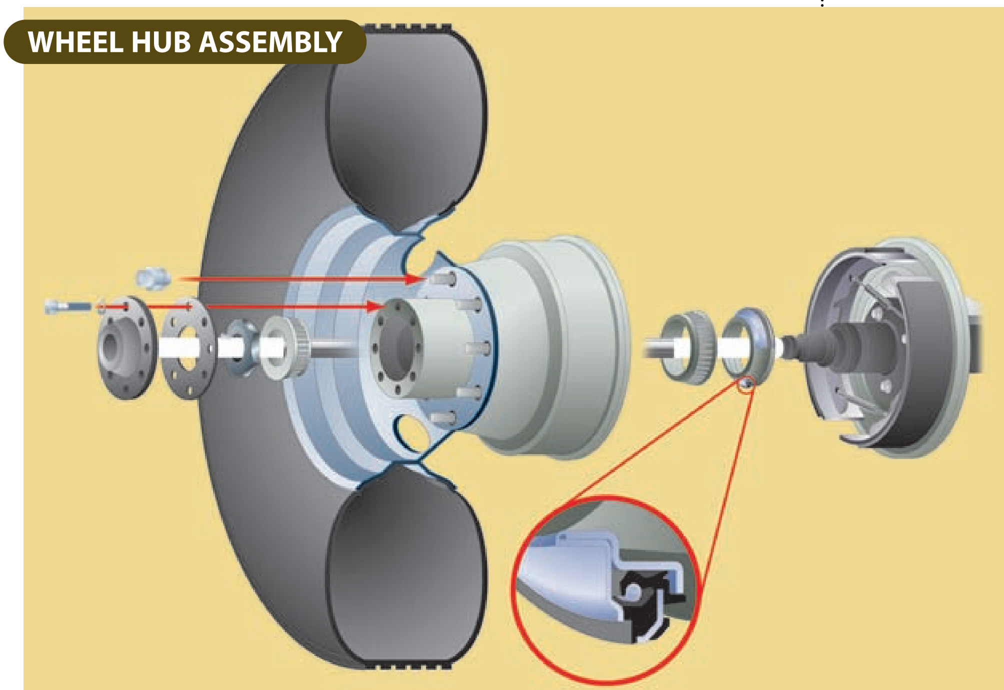


Figure 216: Heavy-Duty Driven Axle Application

WHEEL SEALS

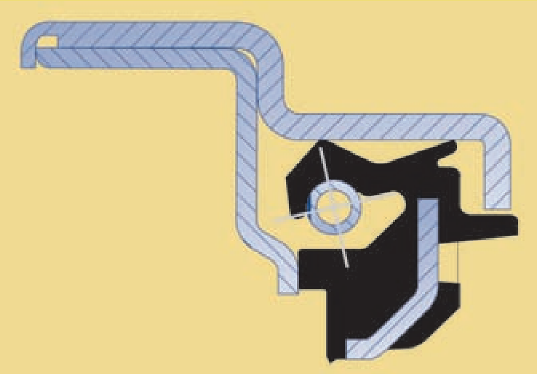


Figure 217: Metal O.D., Metal / Rubber I.D.

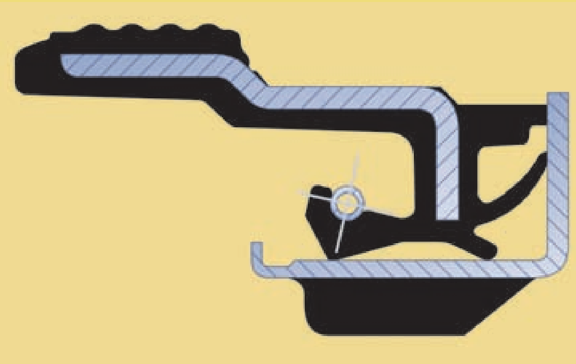


Figure 218: Rubber I.D. / O.D.

metal, tools are required for proper installation. ***Figure 218*** shows an internal seal design, which features a stationary sleeve and a rotating sealing element. With rubber on both the seal I.D. and O.D., this seal can be installed by hand without tools if necessary. Note that this internal design also features both an axial dirt lip and a flinger to maximize exclusion of contaminants.

Heavy duty wheel end applications lubricated with hypoid grease can also utilize the designs shown in ***Figures 217 and 218***. The spring-loaded, unitized design shown in ***Figure 213***

OFF-HIGHWAY MUD, DIRT & WATER WHEEL APPLICATION

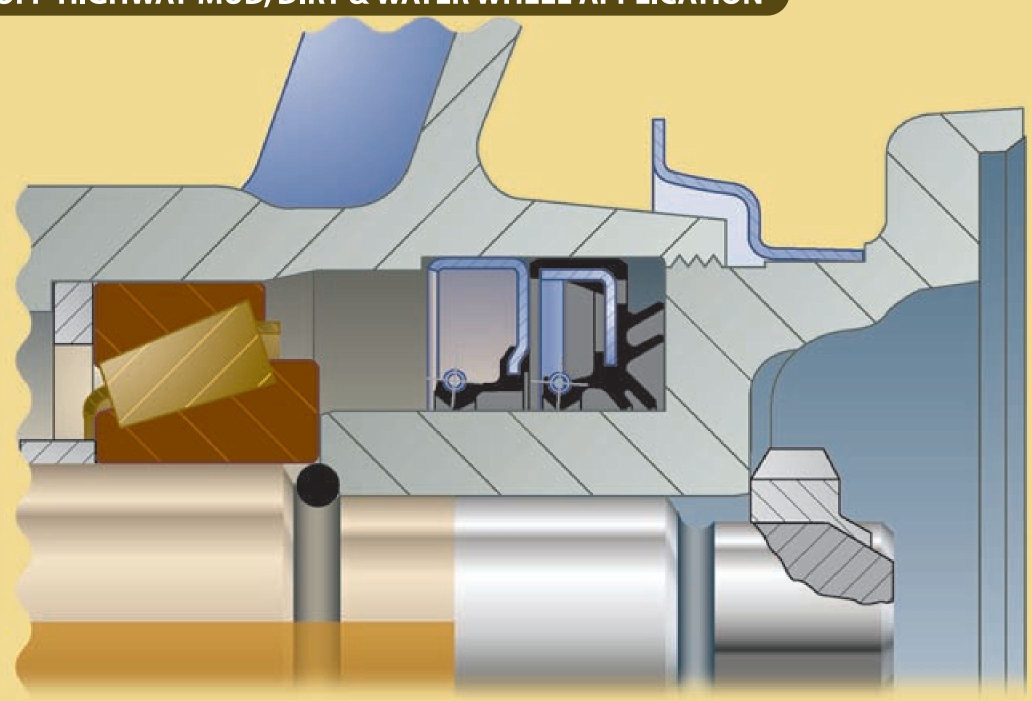


Figure 219: Dual Seal Design

(*page 208*) is also a possibility. Note that this design features both a metal I.D. and a metal O.D.

OFF-HIGHWAY MUD, DIRT, & WATER WHEEL SEALS

Exclusion of mud, dirt, and water in off-road wheel applications requires sophisticated seals. In some instances, dual seals are used, as in ***Figure 219***. Alternatively, non-standard single seals with rubber on both the I.D. and the O.D. may be used; an example of this is shown in ***Figure 220.*** This seal has its own running surface.

EXCLUDER SEAL

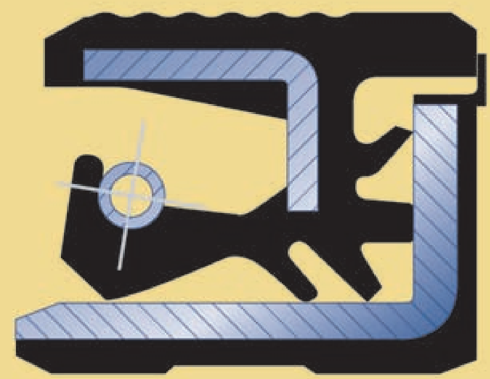


Figure 220: Rubber I.D/ O.D.

HEAVY-DUTY PINION SEALS

Like so many other seals, pinion seals – found in rear wheel drive transmissions where the driveshaft yoke enters the differential housing and so named because they're near the differential's pinion gear – are being subjected to increasingly stringent demands. Warranty periods are increasing, as are temperature requirements and the expectations for exclusion of contaminants. Pinion seals (also known as input seals) must also resist being broken down by the increasingly aggressive additives in gear oils. High shaft

PINION SEAL APPLICATION

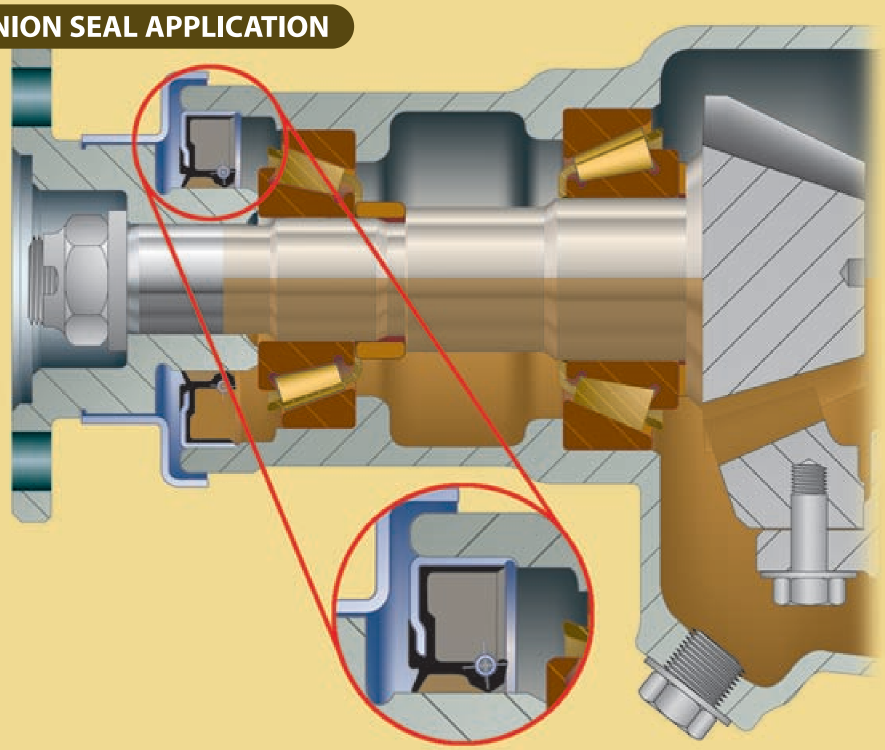


Figure 221: Heavy Duty Pinion Seal

PINION SEALS

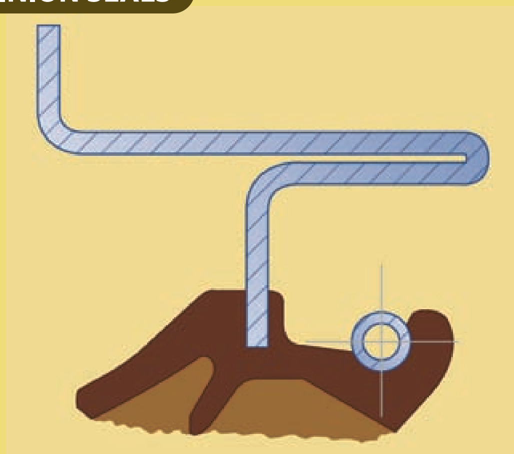


Figure 222: Dual Radial Dirt Lips

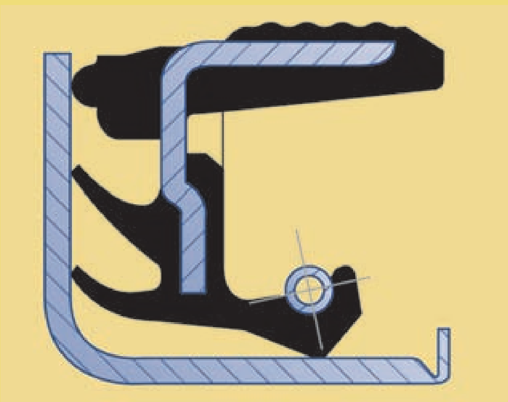


Figure 223: Unitized Seal w/ Dual Axial Dirt Lips & Thrust Bumper

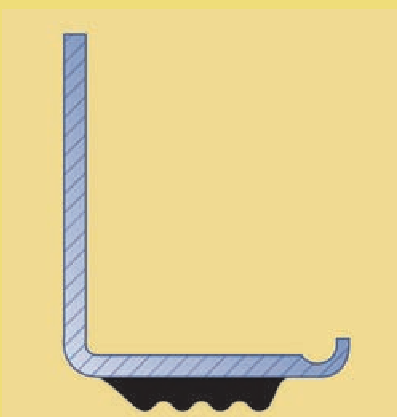


Figure 224: Rubber I.D.

speeds are common, and so are temperature extremes. The placement of pinion seals means they are constantly inundated by dust, water, and mud. Axial end play is typical. Typical pinion seal placement is shown in ***Figure 221*** (previous page). In particularly dirty environments, a flinger may also be installed on the yoke to help deflect contaminants.

A couple of different pinion seal configurations are common. As shown in ***Figure 222***, the first is a fluoroelastomer seal with a spring-loaded primary lip and dual dirt lips. The space between the primary lip and the dirt lips is packed with grease as a further barrier to contaminants. The seal is placed in the differential housing, and the yoke connecting the drive shaft to the pinion gear provides the sealing surface.

Contamination reaching the sealing lips can cause wear on both the lips and the yoke surface, so in some cases a unitized pinion seal design is better. As shown in ***Figure 223***, this design features two axial dirt lips, maximizing contaminant exclusion. A thrust bumper is molded on the seal to accurately locate the axial dirt lips. Sometimes a PTFE thrust bumper is utilized to minimize friction between the seal and the flange. This bumper further impedes contamination, as does the flange (which acts as a flinger). The wear sleeve provides the running surface for the seal, thus protecting the yoke.

As shown in ***Figure 224***, a seal to be installed in the differential housing bore may have rubber molded on the I.D. of the wear sleeve to facilitate yoke installation. The exception is high-speed applications; the presence of the rubber may cause unwanted heat build-up, which will, in turn, cause higher underlip temperature and shorter seal life. A seal without rubber on the I.D. should be installed onto the yoke (using a proper installation tool) rather than into the differential housing; the shaft-yoke combination can then be fitted over the pinion shaft.

POWER STEERING SEALS

Many power steering applications use high-pressure hydraulic systems. Seals for power steering applications (like the one shown in ***Figure 225***) include an input shaft seal (also known as a stub shaft seal) and a pinion seal. These seals are for shafts with slow oscillating rotation, and they typically have operating pressures that are 10 to 20 psi.

The outer rack seal and inner rack seal are reciprocating applications and can see pressures up to 1500 psi. Typical rack seals will have plastic or PTFE back-up rings to prevent seal lip blowout.

POWER STEERING SEALS

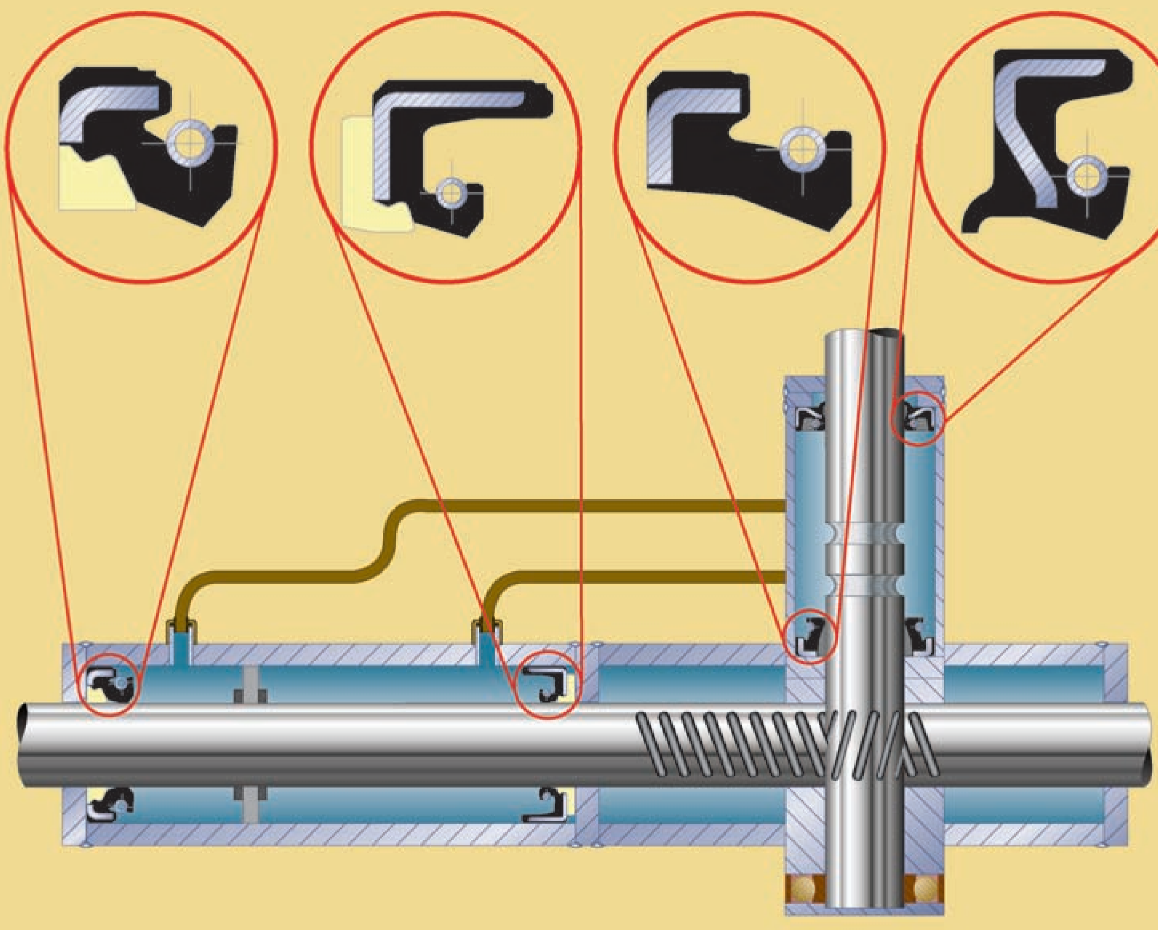


Figure 225: Typical Designs

SHOCK ABSORBER SEALS

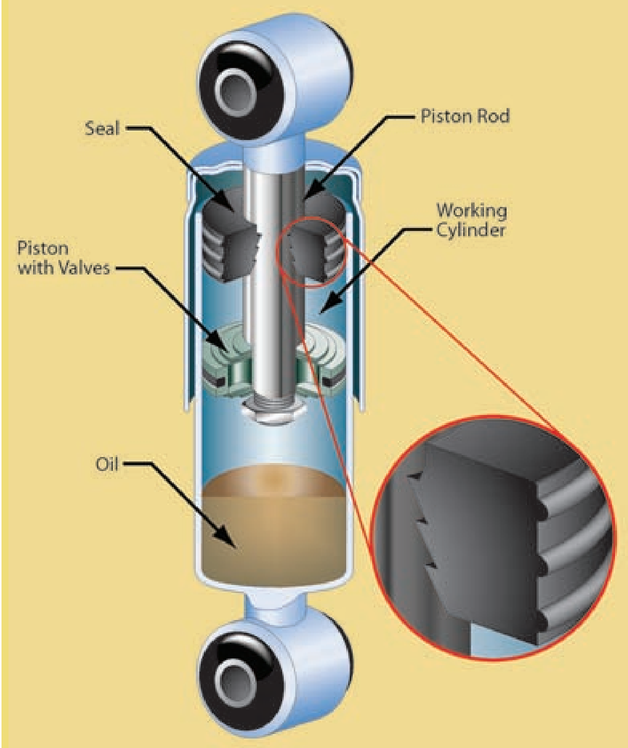


Figure 226: All Rubber Seal

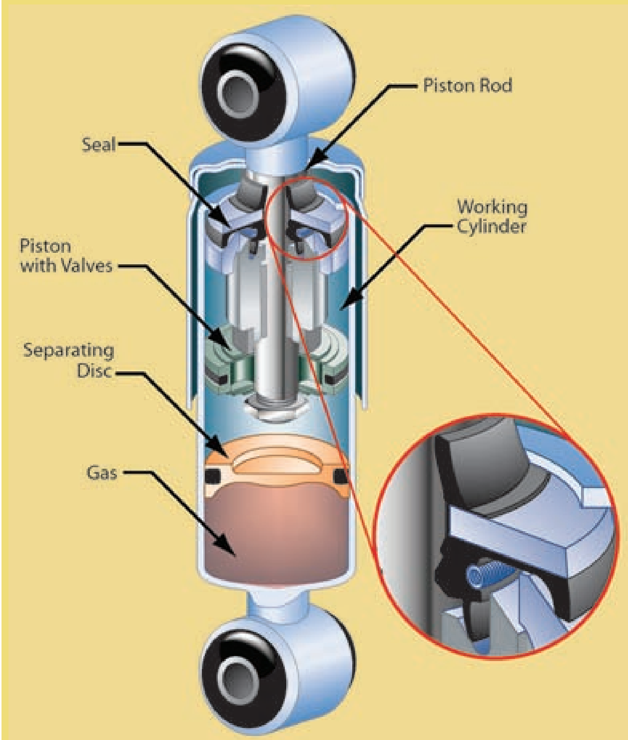


Figure 227: Seal & Check Valve

SHOCK ABSORBER SEALS

Shock absorber seals can be simple or complex. They can be all rubber seals such as the one shown in ***Figure 226***. Notice that this simple design features beads on the O.D. to aid retention. This type of design is generally used in shock absorbers filled with oil.

Many modern shock absorbers, however, use a combination of pressurized gas and oil. Seals for these applications tend to be more complicated. They typically include a primary seal lip, a contaminant exclusion lip, and a check valve to prevent excess pressure build-up as the shock absorber is cycled. An example of this more complex type of shock absorber seal is shown in ***Figure 227***.

AIR CONDITIONING SEALS

At one time, automotive air conditioning systems relied on R-12, a chlorofluorocarbon (CFC) refrigerant. But because studies have shown that CFC's contribute to ozone depletion in the atmosphere, a push was made in the 1990s to replace these CFC refrigerants with hydrofluorocarbon (HFC) refrigerants. R-12 was replaced by what is known as R-134a.

The advent of R-134a, however, also necessitated the use of different lubrication. This new lubrication, in combination with higher operating temperatures, forced seal designers to seek more resistant materials for air conditioning seals. Hydrogenated nitrile

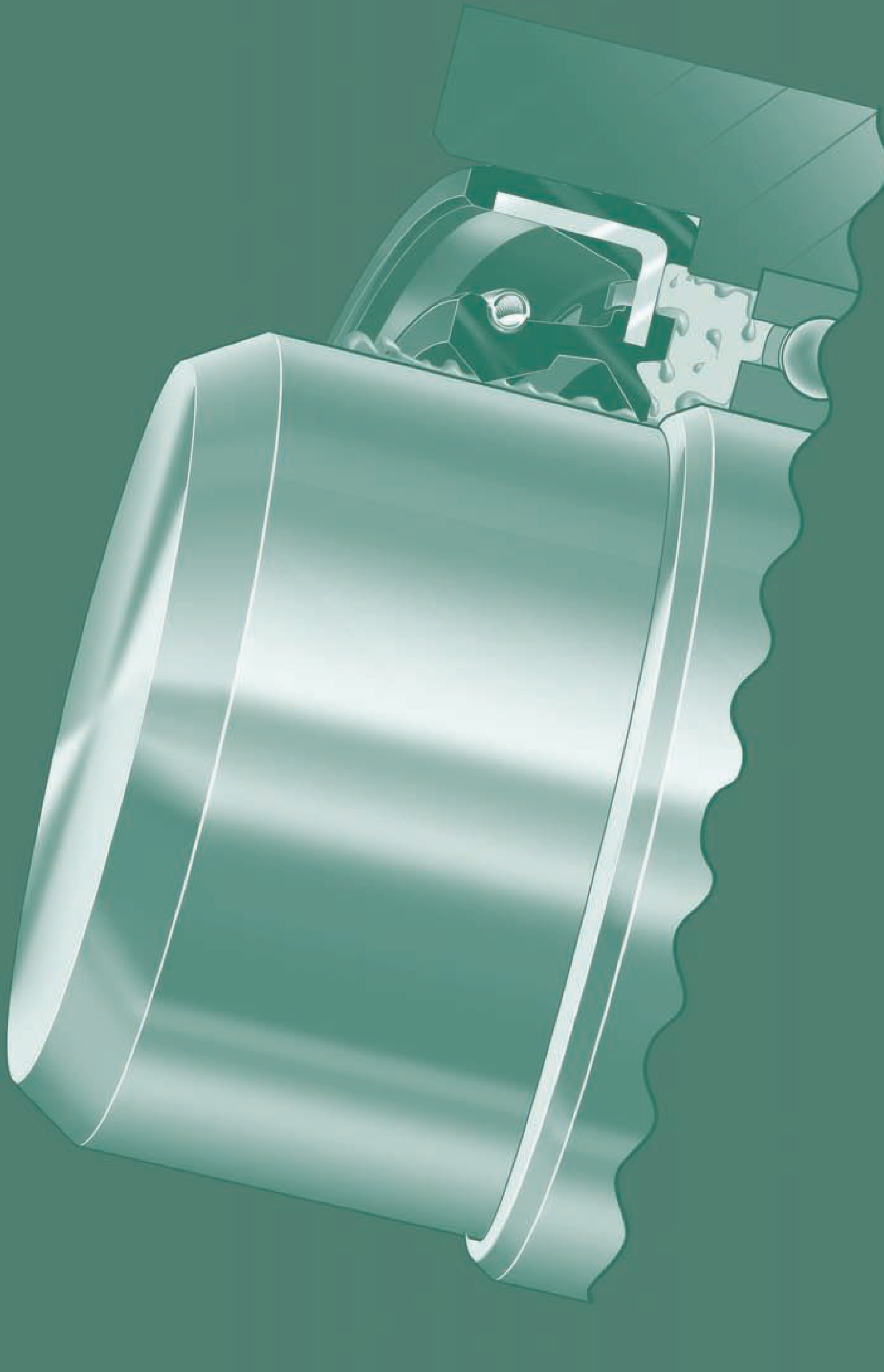
AIR CONDITIONING SEALS

Figure 228: HNBR Sealing Lip w/ PTFE Exclusion Lip & O-Ring

Figure 229: HNBR Sealing Lip w/ PTFE Exclusion Lip

(HNBR) has proven itself effective and has thus found wide use in automotive air conditioning seals.

Some older AC seals incorporated an O-ring for O.D. sealing. ***Figure 228*** shows an example of this. ***Figure 229*** shows a more modern design featuring a bonded rubber O.D. and fewer overall components. Both designs feature a primary lip made of HNBR and an exclusion lip made of polytetrafluoroethylene (PTFE).



troubleshooting

troubleshooting

Failure Analysis.

Designing a shaft seal and selecting the materials that will function well in a given environment are far from simple tasks. Even experienced seal designers are often met with unusual service requirements that test both their ingenuity and the capabilities of the seal. Whether your sealing needs are simple or complex, the factors to be considered are numerous enough to guarantee that not every seal will be successful in every application.

When shaft seals do leak, you can often determine why through careful inspection of all elements within the sealing system. This includes not only the seal, but also the shaft (or other running surface) and the housing bore. An examination of the lubricant should also be conducted. All of these elements should be studied both with the failed seal in place and following its removal.

In order to facilitate this inspection process, the Rubber Manufacturers Association (RMA) developed a series of four checklists. As shown in ***Table 68 (***next page), the first of these is designed to step you through examination of the sealing system and its environment prior to seal removal.

"Whether your sealing needs are simple or complex, the factors to be considered are numerous enough to guarantee that not every seal will be successful in every application."

With the seal ***still in place***,

1. Record:

Seal application: ....

Equipment identification: ....

Miles or hours of operation: ....

Type of complaint: ....

2. Record leakage amount:

- ☐ Slight
- ☐ Immediate area damp
- ☐ Heavy leakage

3. Record leakage source:

- ☐ Between shaft and seal lip
- ☐ Between seal O.D. and bore
- ☐ At retainer bolt holes
- ☐ At retainer gasket
- ☐ Between wear sleeve and shaft
- ☐ Through seal on assembled seals

4. Record condition of surrounding area:

- ☐ Seal area clean
- ☐ Mud or dust packed in seal area

5. Clean surrounding area and inspect:

- ☐ Nicks on bore chamfer
- ☐ Seal loose in bore
- ☐ Paint spray on seal lip
- ☐ Seal cocked in bore (amount) ....
- ☐ Seal installed incorrectly (backward)
- ☐ Seal case deformed
- ☐ Shaft-to-bore misalignment

6. Rotate shaft (if possible) and check for runout and end play:

- ☐ Excessive shaft end play (amount)...........
- ☐ Excessive shaft runout (amount) ....

7. Confirm leakage location:

If location of leakage cannot be confirmed at this point, either introduce ultra-violet dye into the sump or spray area with white powder, operate for 15 minutes, and check for leakage with ultraviolet or regular light.

8. Remove seal

When above analysis is complete, mark the seal at the 12 o'clock position and remove it carefully from the application. Record oil type and obtain used sample from application.

Table 68: RMA Checklist - Seal In Place

With the ***seal removed***,

1. Clean the seal in a mild solvent.

Do not attempt to scrape away carbon, etc. Inspect the seal using this checkllist.

2. Check primary lip area:

- ☐ Normal wear
- ☐ No wear
- ☐ Excessive wear
- ☐ Inverted lip due to poor installation
- ☐ Nicks, scratches, or cuts at lip contact area
- ☐ Hardened or cracked rubber
- ☐ Coked oil on lip
- ☐ Softening or swelling

3. Check seal outside diameter:

- ☐ Normal
- ☐ Severe axial scratches
- ☐ Peeled rubber
- ☐ Hardened rubber
- ☐ Nonfills or cuts

4. Check spring and spring groove area:

- ☐ Spring normal and in place
- ☐ Spring missing
- ☐ Spring corroded
- ☐ More than one spring
- ☐ Separated spring

5. Record the following measurements:

Primary lip inside diameter ....
Primary lip radial force ....
Seal outside diameter ....
Spring inside diameter ....
Spring tension ....
Primary lip wear band width:
Min ....
Max ....

Table 69: RMA Checklist - Seal Removed

As shown in ***Table 69***, the second checklist steps you through an examination of the seal following removal.

With the ***seal removed,***

1. Inspect the housing bore area:

- ☐ Measure bore diameter ....
- ☐ Bore chamfer damaged
- ☐ Flaws or voids in housing
- ☐ Tool withdrawal marks in bore
- ☐ Bore surface scratched or galled

2. Inspect the shaft wear track:

- ☐ Measure shaft diameter ....
- ☐ Shaft surface corroded
- ☐ Seal wear path in wrong direction
- ☐ Scratches or nicks at lip contact area
- ☐ Measure wear path width ....
- ☐ Discoloration on shaft surface
- ☐ Coked lubricant present
- ☐ Shaft chamfer damaged or missing
- ☐ Wear sleeve loose on shaft (if applicable)

3. Measure shaft characteristics (if possible):

- ☐ Measure shaft roughness ....Ra
- ☐ Measure depth of wear path ....
- ☐ Measure shaft lead ....Deg
- ☐ Measure shaft hardness ....Rc
- ☐ Check for proper shaft material

4. Inspect and compare properties of used application lubricant to new lubricant.

- ☐ Contaminants in filtered lube
- ☐ Color different from new lube
- ☐ Viscosity different from new lube
- ☐ Odor different from new lube

Table 70: RMA Checklist - Seal Removed

As shown in ***Table 70,*** the third checklist steps you through an examination of the housing bore, the shaft, and the lubricant in use following seal removal.

Because the use of these three checklists in their entirety may not be practical in field situations, the RMA also developed a shortened form designed to expedite inspection. That short form is shown in ***Table 71***.

Seal application: ....

Equipment identification: ....

Miles or hours of operation: ....

Type of complaint: ....

Step 1: Inspect the seal application before removal.

Amount of leakage:	☐ Slight	☐ Immediate area damp	☐ Heavy leakage
Condition of area:	☐ Clean	☐ Dusty	☐ Mud packed
Leakage source:	☐ Between lip and shaft	☐ Between OD and bore	
	☐ At retainer gasket	☐ Between elements of seal	
	☐ At retainer bolt holes	☐ Between wear sleeve and shaft	

Step 2: Wipe area clean and inspect. Check conditions found.

- ☐ Nicks on bore chamfer
- ☐ Seal loose in bore
- ☐ Seal cocked in bore
- ☐ Seal case deformed
- ☐ Paint spray on seal
- ☐ Seal installed improperly
- ☐ Shaft-to-bore misalign.
- ☐ Other ....

Step 3: Rotate shaft (if possible). Check conditions.

- ☐ Excessive end play
- ☐ Excessive runout

Step 4: Note location of leakage.

If location of leakage cannot be confirmed at this point, either introduce ultra-violet dye into the sump or spray area with white powder, operate for 15 minutes, and check for leakage with ultraviolet or regular light.

Step 5: Mark the seal at the 12 o'clock position and remove it carefully.

- ☐ Retain an oil sample

Step 6: Inspect the application with seal removed. Check conditions found.

- ☐ Rough bore surface
- ☐ Flaws or voids in bore
- ☐ Shaft clean
- ☐ Shaft corroded
- ☐ Coked lube on shaft
- ☐ Shaft discolored
- ☐ Shaft damaged

Step 7: Inspect the seal.

Primary Lip Wear	☐ Normal	☐ Excessive
	☐ Eccentric	☐ None
Primary Lip Condition	☐ Normal	☐ Damaged
	☐ Hardened (stiff)	☐ Soft (flexible)
Seal O.D.	☐ Normal	☐ Axial scratches
	☐ Damaged rubber	
Spring	☐ In place	☐ Missing
	☐ Separated	☐ Corroded

Comments:

Table 71: RMA Short Form

troubleshooting

Common Causes.

Careful shaft seal design can minimize the number of failures, but it is impossible to eliminate failure entirely. What follows is a look at the most common causes of shaft seal failure, along with suggestions on how these failures might be avoided. This is not a comprehensive list; rather, these are the causes you are most likely to see. Rule out all of these possibilities before assuming other causes.

"Rule out all of these possibilities before assuming other causes."

troubleshooting

Backward Installation.

Improper installation is the number one cause of shaft seal failure. As obvious as it may sound, care must be taken to install the seal in the right direction. If replacing a previously used seal, be sure to note the direction in which the primary lip of the old seal was facing, then ensure that the primary lip of the new seal faces the same way.

As shown in ***Figure 230***, failure to orient the seal properly relative to the fluid being sealed will result in instantaneous leakage upon startup. For more on good installation practices, see ***page 175***.

"Care must be taken to install the seal in the right direction."

BACKWARD INSTALLATION

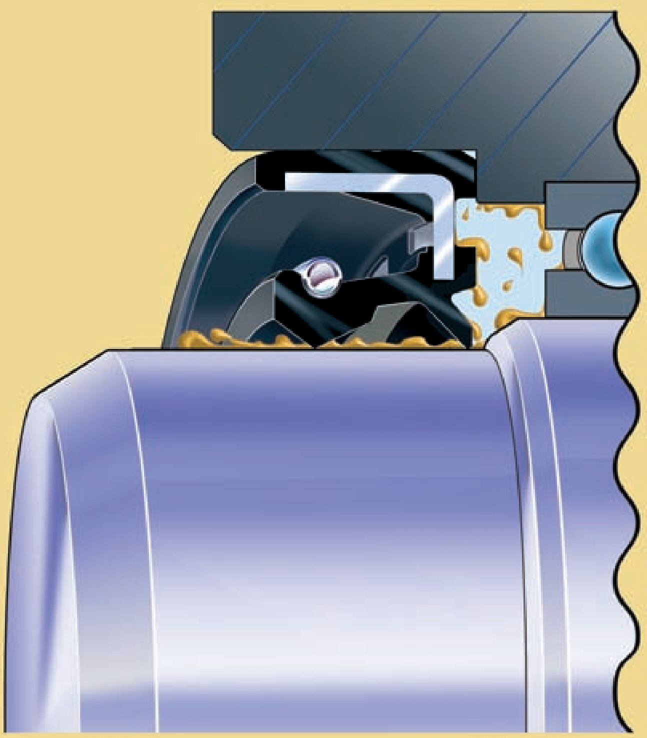


Figure 230: Immediate Leakage Results

troubleshooting

Blisters.

Excessive underlip temperature (heat beyond normal conditions) can cause blisters (such as those shown in *Figure 231*) to form on the air side of a shaft seal's elastomeric lip. Blisters can also form if the lip material is simply not suited to handle the normal operating temperature. Either way, blisters are problematic because they deform the sealing lip, causing it to lift away from the shaft, thus forming a gap for leakage.

Because blisters can dissipate in the absence of heat, blistering can be tough to diagnose once the assembly has cooled. If blistering is a possible cause of leakage, you should heat the seal and check to see if blisters reappear.

Reducing seal load and thus reducing the buildup of underlip heat can typically eliminate blistering. Alternatively, selecting a different, more suitable elastomer capable of handling higher temperatures may be the answer. For more on high temperature effects, see ***page 38***.

"The ability to withstand its environment is critical to a seal's success."

BLISTERS

Figure 231: Cause Lip To Lift Off Shaft

Broken Lip.

A shaft seal's elastomeric lip can break if flexed at temperatures lower than those for which the rubber is suited. Fracturing of the lip is most likely if the seal is exposed to low temperatures in combination with excessive dynamic runout of the shaft. *Figure 232* shows an example of a broken seal lip.

If lip breakage is a problem, check to be certain that the elastomer in use is suited to the temperatures of the application. For more on low temperature effects, see ***page 39***. You should also gauge the amount of dynamic runout and reduce it if possible. For more on shaft runout, see ***page 131***.

"A shaft seal's elastomeric lip can break if flexed at temperatures lower than those for which the rubber is suited."

BROKEN LIP

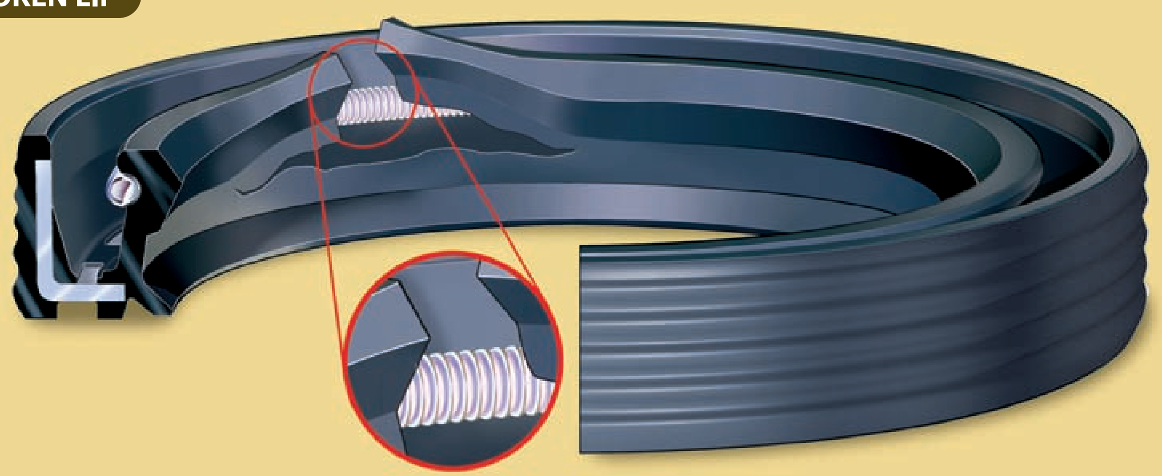


Figure 232: Fracture Caused By Low Temperatures

troubleshooting

Cocked Seal.

A shaft seal must be installed at a right angle (perpendicular) to the centerlines of both the shaft and bore. Anything less than a right angle means the seal is angularly misaligned (cocked). Installing a standard shaft seal into a housing can be tough if there is no counterbore to help align and seat the seal. Even if initial installation is perfect, the lack of a counterbore makes it easy for the seal to get cocked as the shaft is slipped in place (see *Figure 233*). Seal cocking is most common in blind designs that keep the field assembly team from seeing if the seal is properly seated.

"Seal cocking is most common in blind designs that keep the field assembly team from seeing if the seal is properly seated."

Cocking can contribute to uneven wearing of the sealing lip. Cocking also increases the chances that any garter spring might be dislodged from its groove in the lip (a situation known as s*pring pop out*). Damage to the lip itself and/or the seal O.D. is also more likely. In addition, seal cocking increases the temperature at the interface between the shaft and the seal lip, and this hastens hardening and cracking of the seal.

Cocking may be prevented through use of special designs. Some designs feature a flange on the seal O.D. to help seat the seal and hold it in place during service (see ***page 104***). Cocking can also be prevented by using a proper installation tool and the right amount of force (see ***page 175***).

COCKED SEAL

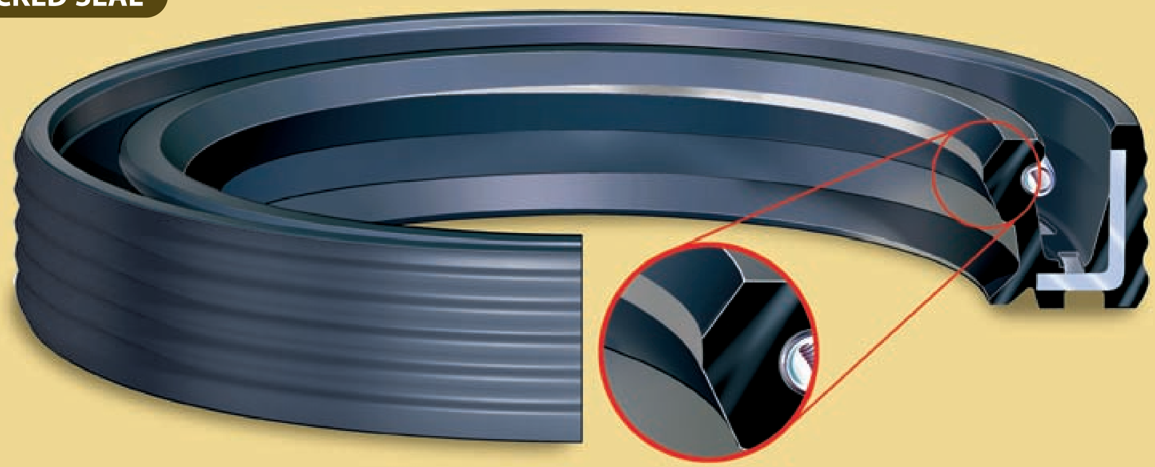


Figure 233: Angular Misalignment

Corroded Spring.

Many shaft seal designs incorporate a garter spring as part of the sealing lip. The garter springs helps maintain the right amount of radial load between the lip and the shaft. For seal designs incorporating a spring, the spring must be in good condition in order to function properly. A corroded spring (such as the one shown in *Figure 234)* loses tension and can contribute to leakage of the sealing lip.

Spring corrosion can result in several ways. For example, the spring may not have been properly coated with rust preventative during the seal manufacturing process. In some cases, the seal may have been exposed to an excessive amount of moisture prior to or during use. In other cases, the application may simply contain a highly corrosive fluid, or the wrong spring material may have been used.

Elimination of spring corrosion can take several forms. If the seal has been improperly manufactured, the seal supplier should be notified in order to initiate corrective action. Improper storage conditions must also be addressed so as to eliminate excessive moisture. Springs made of stainless steel are recommended for service in highly corrosive media.

"A corroded spring loses tension and can contribute to leakage of the sealing lip."

CORRODED SPRING

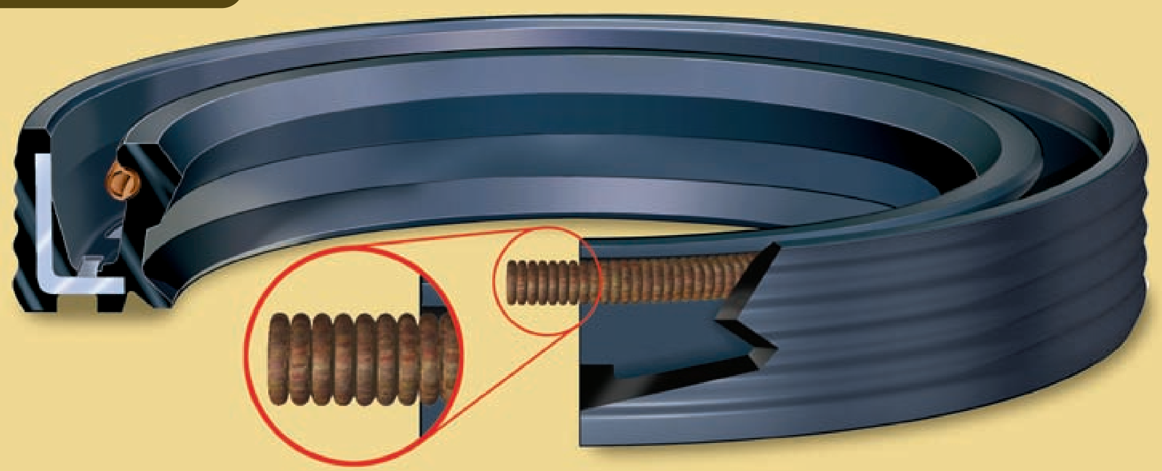


Figure 234: Causes Loss of Tension

troubleshooting

Cracked or Hardened Lip.

Cracking or hardening of the sealing lip is most common in shaft seals exposed to excessively high temperatures, especially if the exposure is lengthy. Deep cracks can act as leak paths, but even if cracks don't develop, a hardened lip is less able to follow shaft eccentricities at high speeds, and leakage becomes likely. Lip hardening is evident if flexing the lip results in the development of tiny cracks in the lip surface (known as crazing). *Figure 235* shows a hardened and cracked lip.

Correcting for hardening and/or cracking may mean making sure the elastomer in use for the lip is capable of handling the application's temperatures. For more on material temperature ranges, see the profiles beginning on ***page 41***. Localized increases in underlip temperature can result from inadequate lubrication, and these increases can hasten lip hardening. If such is the case, be sure you're using adequate lubrication and that the shaft is finished properly (so as to hold on to this lubrication). For more on shaft surface finish, see ***page 116***; for more on lubrication, see ***page 189.*** You should also be aware that cracking of the elastomeric lip in areas other than the wear track can result from ozone exposure. Be sure seals are not exposed to ozone-generating equipment (such as welders) before installation.

"Cracking or hardening of the sealing lip is most common in shaft seals exposed to excessively high temperatures."

CRACKED / HARDENED LIP

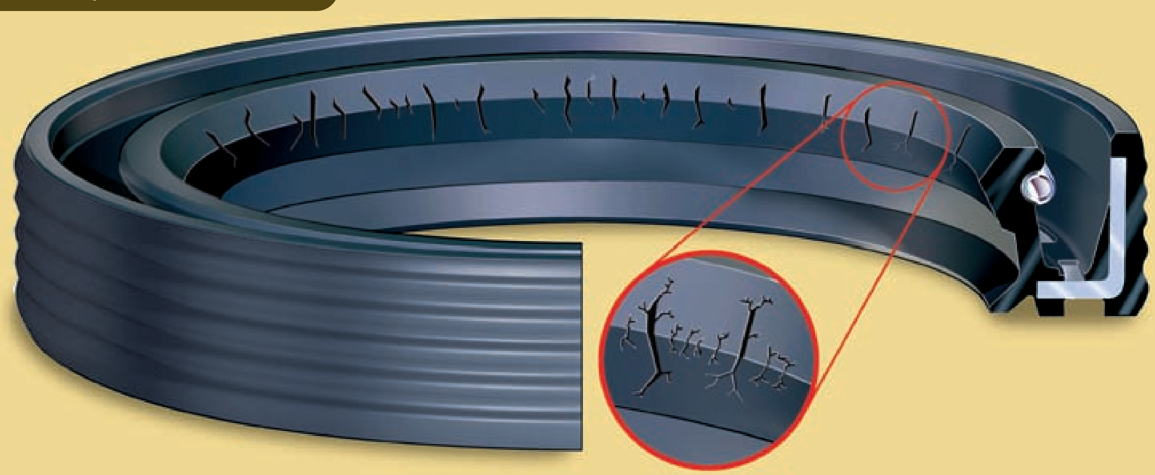


Figure 235: Cracks, Large & Small

Cut Lip.

Nicks, tears, or cuts on a shaft seal's elastomeric lip can result from unwanted and unprotected contact with splines or a keyway on the shaft. A cut lip cannot seal properly. A lip cut by splines may actually have a series of cuts that are the width of the splines. Cuts by a keyway may have two distinct cuts that are separated by the width of the keyway, or the lip may be sliced.

If splines or a keyway must be traversed, use of a shield and/or lubrication can help. An assembly cone can be temporarily fitted onto or over the shaft to facilitate avoidance of potential hazards. For more on use of installation tools, see ***page 176.***

Unlike a shaft seal lip cut by splines or a keyway on the shaft, a lip damaged by a defect on the shaft (such as a burr) or due to improper shaft chamfer usually only has one cut. ***Figure 236*** shows what single and multiple cuts might look like. In order to guard against shaft-induced lip damage, make sure all burrs are smoothed from the shaft surface. For more on recommended shaft surface finish, see ***page 116***. You should also follow the recommendations for shaft chamfer described on ***page 126***.

"A lip damaged by a defect on the shaft or due to improper shaft chamfer usually only has one cut."

CUT LIP

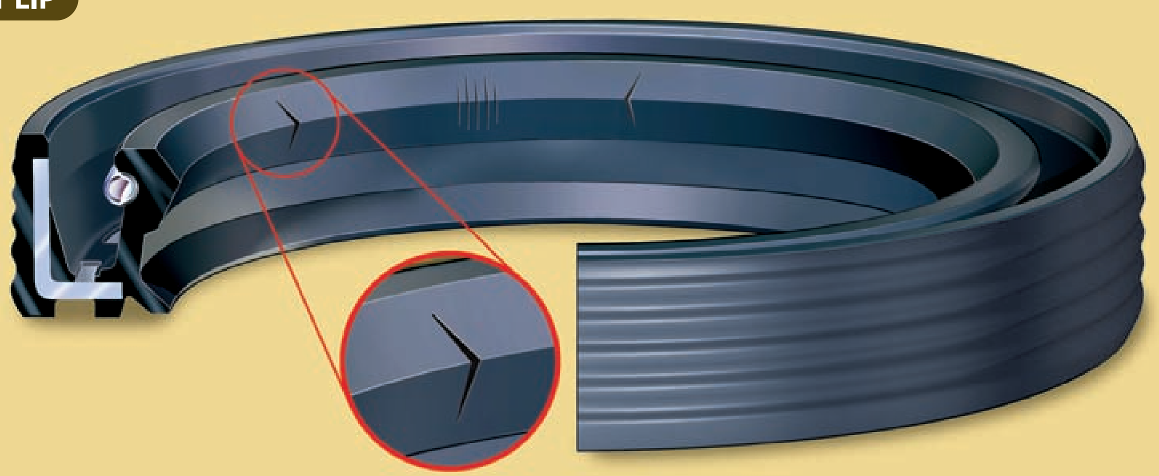


Figure 236: May Have One or More Distinct Cuts

troubleshooting

Damaged Case (Improper Bore Finish).

The scoring (scratching) of a shaft seal O.D. is most likely to be caused by improper machining of the housing bore. A housing surface that is too rough or that contains burrs will abrade the O.D. A scored O.D. can also result if the edges of the bore have not been chamfered properly. *Figure 237* shows an example of seal with a scored O.D.

The optimal surface finish for the bore largely depends on the type of shaft seal in use. Generally speaking, seals with a rubber-covered O.D. can handle a rougher surface finish than metal O.D. seals, but a surface that's too rough may damage the O.D. or prevent formation of a tight static seal. On the other hand, a surface that's too smooth may make it more difficult for a seal with a rubber-covered O.D. to stay in place. While there is no accepted standard for minimum bore roughness, we recommend a roughness of at least 2.03 µm (80 µin.) Ra. If the bore exceeds 2.54 µm (100 µin.) Ra, then a rubber O.D. seal or use of O.D. sealant is recommended. Bore roughness should never exceed 3.175 µm (125 µin.) Ra.

"The scoring (scratching) of a shaft seal O.D. is most likely to be caused by improper machining of the housing bore."

SCORED O.D.

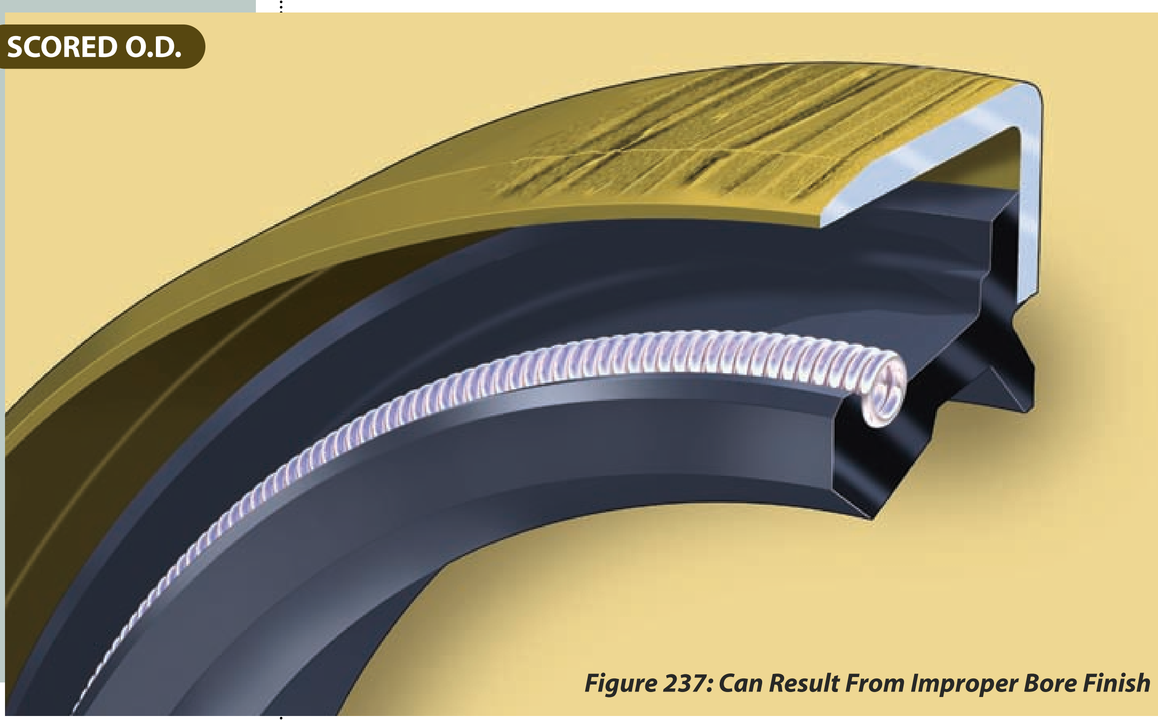


Figure 237: Can Result From Improper Bore Finish

Damaged Case (Poor Bore Chamfer).

Even if the proper installation tools and practices are used, a shaft seal's case can also be damaged if the housing bore into which it is being installed does not have a proper chamfer or if the bore is otherwise damaged. An improper chamfer or damaged bore can result in seal case deformation (such as is shown in *Figure 238*) on the outside of a seal with a metal O.D.

In order to avoid seal O.D. deformation, it is recommended that the bore have an appropriate lead-in chamfer. For more on bore chamfers, see ***page 134***.

"An improper chamfer or damaged bore can result in seal case deformation."

DAMAGED CASE

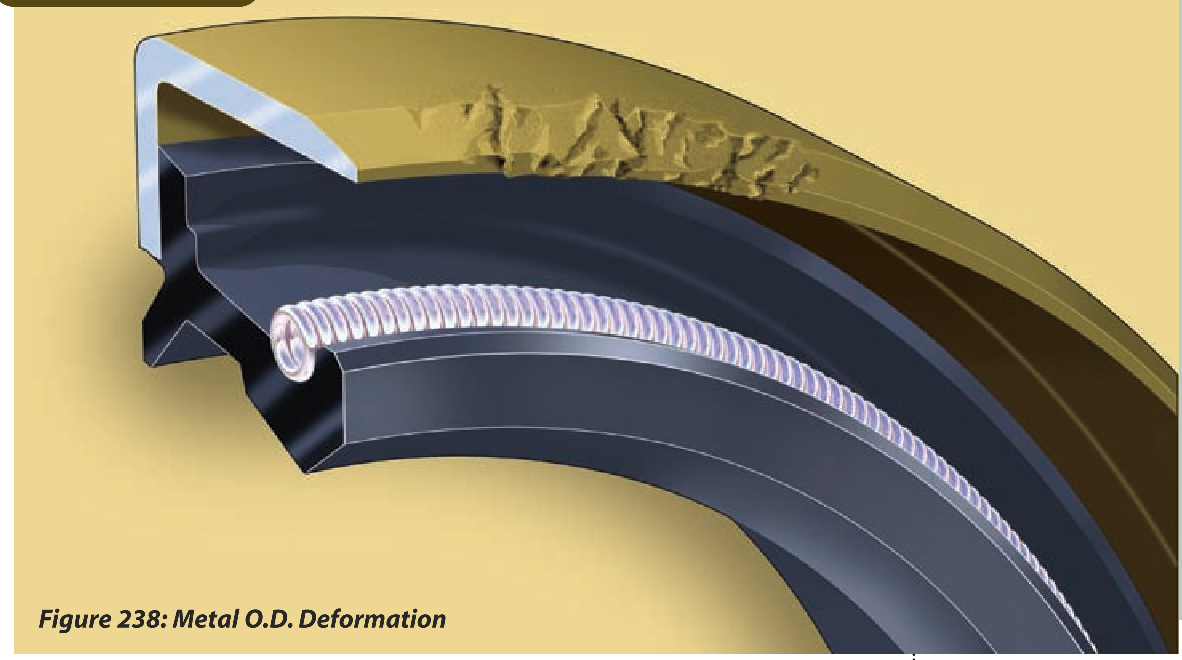


Figure 238: Metal O.D. Deformation

troubleshooting

Damaged Case (Poor Handling).

Deformation of the shaft seal case (such as the denting along the heel face shown in Figure 239) is most commonly caused by improper handling and/or installation practices. A dented case will not make proper contact with the housing bore. The elastomeric lips can also be deformed such that proper contact with the shaft is compromised. Use of a hammer as an installation tool is particularly dangerous for this reason.

Rather than a hammer, it is recommended that you use either the combination of a mallet and strike plate, or even better, a press (such as a hydraulic press) capable of applying consistent, even pressure. Alternatively, a specially designed installation tool can help guard against inadvertently damaging either the case or the sealing lip. For more on proper installation practices, see ***page 177***.

"Deformation of the shaft seal case is most commonly caused by improper handling and/or installation practices."

DENTED CASE

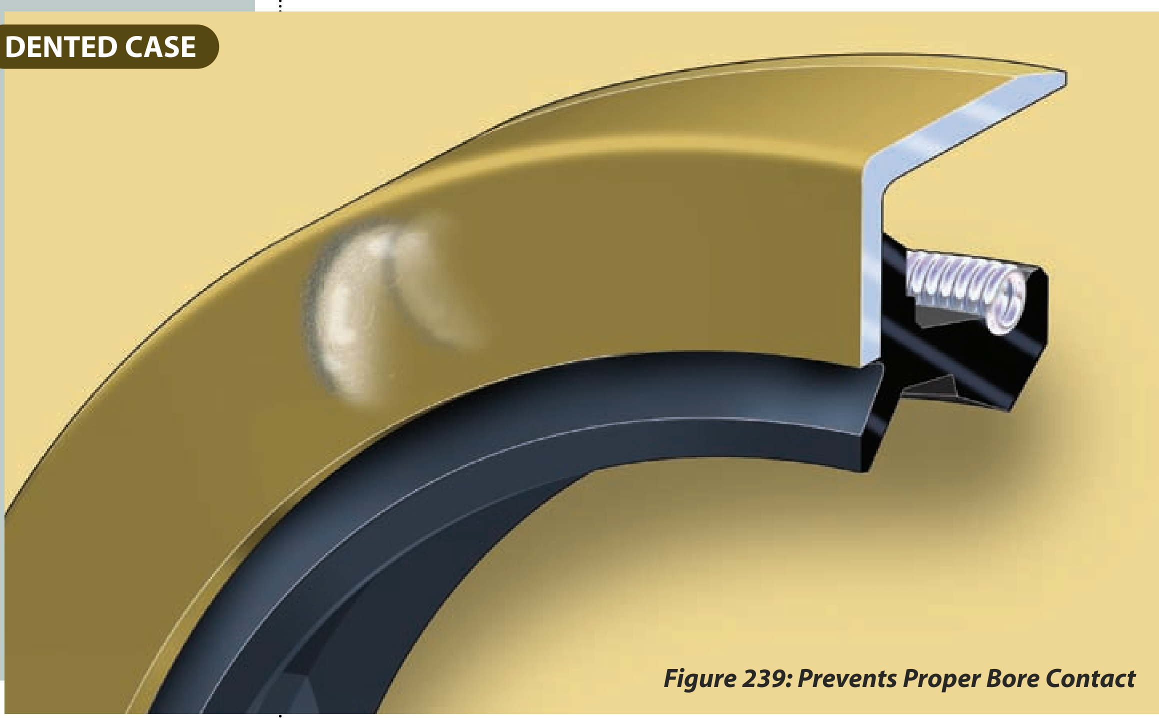


Figure 239: Prevents Proper Bore Contact

troubleshooting

Excessive Lip Wear.

Excessive lip wear (such as that shown in *Figure 240*) can be caused by too much lip load, improper shaft surface finish, and insufficient lip lubrication. Any one of these factors could cause excessive wearing of the seal lip; a combination of two or more of these factors would further hasten seal failure.

Excessive lip load can be the result of excessive system pressures, so make sure vents are not clogged. Be sure that the interference between lip and shaft is not too great, and that any garter spring in use is not inordinately increasing lip load. You should also check to be sure that the shaft surface finish falls within recommended parameters. A finish that's too rough will unduly abrade the sealing lip; a finish that's too smooth will not develop the microscopic pockets necessary to hold lubrication, meaning friction and wear will both increase. For more on shaft surface finish, see ***page 116***.

If insufficient lubrication is the culprit, it could also be that adequate lubrication was not provided prior to service. Be sure to provide proper lubrication on the air side of the seal or, for designs incorporating more than one lip, between the lips. For more on lubrication concerns, see ***page 191***.

"Excessive lip wear can be caused by too much lip load, improper shaft surface finish, and insufficient lip lubrication."

EXCESSIVE LIP WEAR

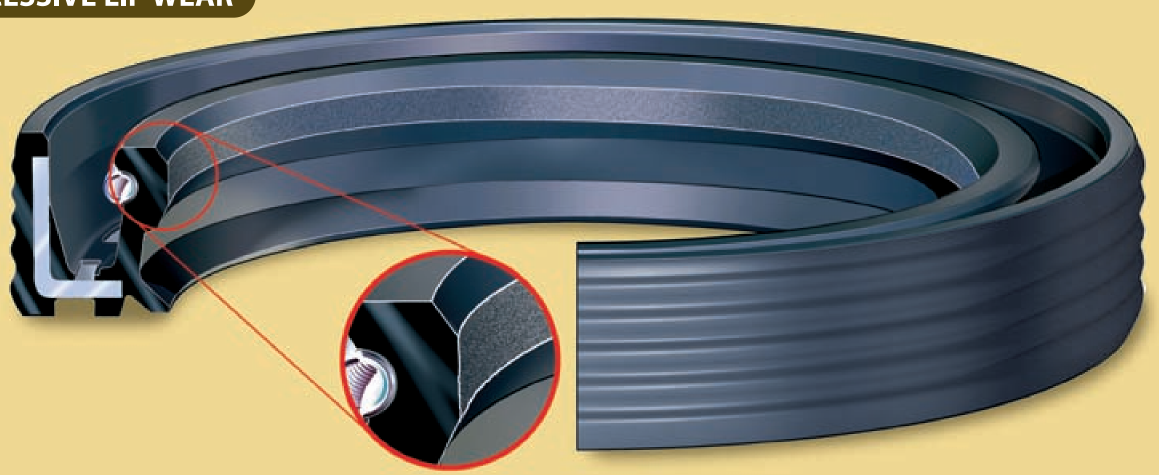


Figure 240: Hastens Seal Failure

troubleshooting

Excessive Material Swell.

Excessive swelling of a shaft seal's elastomeric lip is a good indicator that the lip material and the lubricant in use are not compatible. Material swell can be particularly problematic if some materials (such as silicone) come in contact with oils at high temperatures. Softening, swelling, and reversion of the material can occur under such conditions. An example of excessive lip swell can be seen in *Figure 241*.

If material swell is an issue in your application, check to be sure that the elastomer in use is compatible with the lubricant and any other fluids touching the seal, either during operation or cleaning. This includes any solvents that may be used during teardown. You should also check to be sure that system fluids are not being contaminated in some way; contamination could cause an otherwise acceptable seal material to swell or degrade.

"Material swell can be particularly problematic if some materials come in contact with oils at high temperatures."

EXCESSIVE MATERIAL SWELL

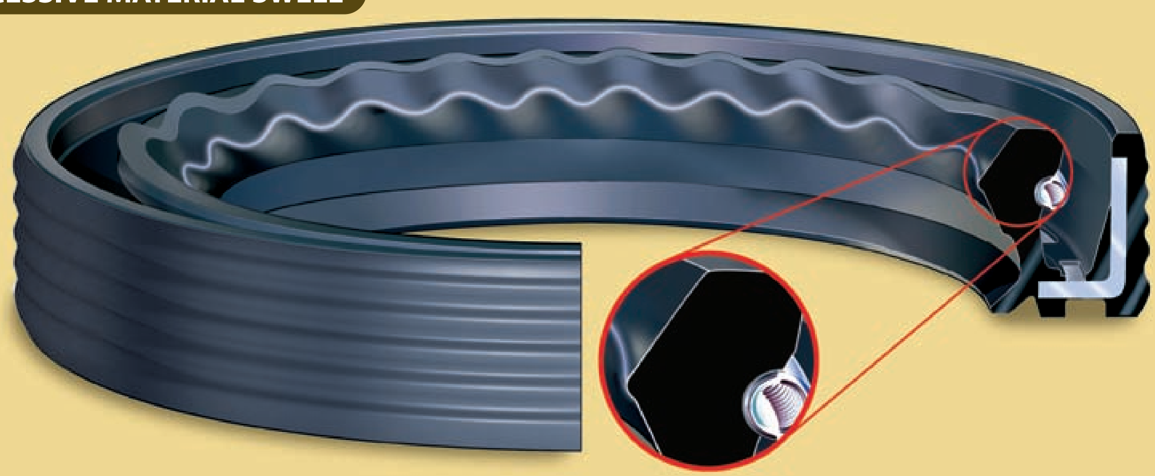


Figure 241: Indicator of Lip-Fluid Incompatibility

troubleshooting

Excessive STBM or Runout.

Even if a shaft seal is well designed and in good condition, excessive shaft-to-bore misalignment (STBM) can cause high-low lip wear (as shown in *Figure 242*). The seal lip wear pattern will be wide at one point and narrow at a point about 180 degrees away.

Excessive shaft-to-bore misalignment can result from poor initial alignment during assembly, in which case installation procedures should be reviewed. In some instances, seals manufactured with high radial wall variation will have seal lip wear patterns that are similar to those generated by shaft-to-bore misalignment. If such is the case, the seal manufacturing process should be adjusted to minimize this variation.

Excessive shaft runout may be the result of a bearing that has failed due to excessive bearing load capacity, or excessive wear or contamination. If so, replacing the failed bearing should solve the problem. Runout may also result from excessive shaft deflection, in which case you should balance the shaft and/or support the shaft better. It may also be that the shaft is not machined to tolerance and has lobes that generate shaft runout. Reviewing and adjusting the shaft production process may be needed. Excessive shaft runout creates a lip wear pattern of uniform width (see ***Figure 240***).

"Excessive shaft-to-bore misalignment (STBM) can cause high-low lip wear."

EXCESSIVE STBM

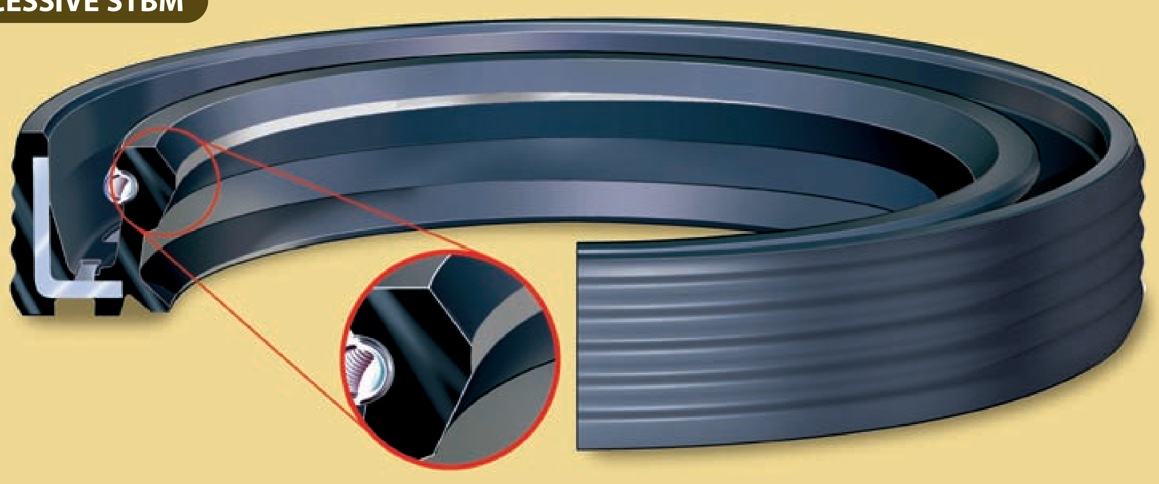


Figure 242: High-Low Lip Wear

troubleshooting

Excessive Shaft Wear.

There are two kinds of shaft wear. One type is wide and smooth with little depth; this does not usually cause seal leakage. The other type is due to contamination and results in wide, deep, rough grooves (such as the one shown in *Figure 243*). Deep shaft wear caused by contaminants will cause leakage.

Particulate matter such as dirt or sand can get trapped between the seal lip and the shaft, causing continual scraping of the surface. Precautions must be taken to minimize contaminant ingestion. Use of a seal design specifically engineered to prevent underlip contamination may be required. Contamination can also be contained in the application lubricant; frequent filter changes or use of a different lubricant may be advisable. Internal contamination can result from improper cleaning of castings to remove core sand and other debris. The seal user must ensure that castings are contaminant-free. Contaminants can also be in the rubber used to mold the seal lip. The seal supplier must use quality controls to prevent rubber contamination.

"Deep shaft wear caused by contaminants will cause leakage."

Too much lip to shaft interference can also cause a wide wear path on the shaft, so it is very important that lip interference is within desired limits. Excessive eccentricity and end play within the assembly will cause wide shaft wear. Be sure to check the condition and suitability of all bearings, as well as the alignment of all parts of the assembly.

EXCESSIVE SHAFT WEAR

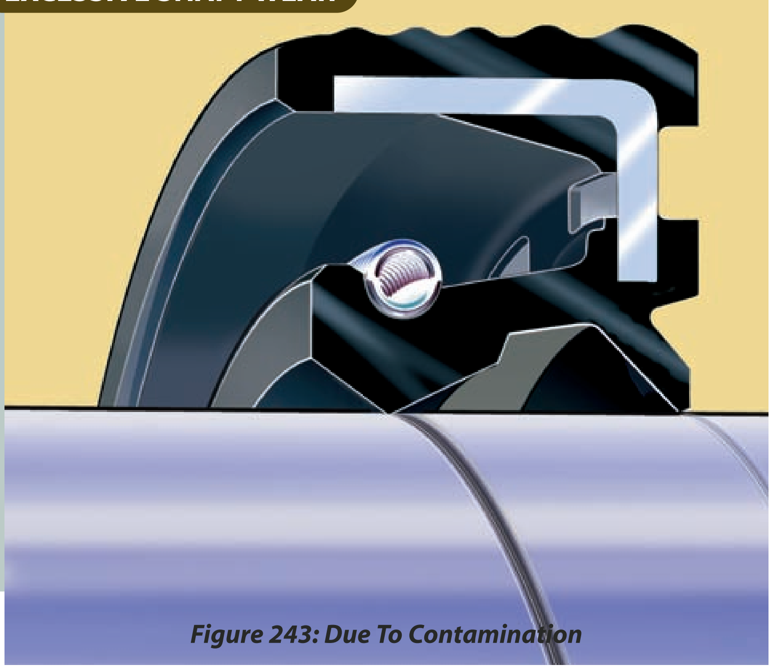


Figure 243: Due To Contamination

Inverted Lip.

Lip inversion (the turning under of a section of the sealing lip, see Figure 244) can easily occur during installation. Use of an improper seal design in a high-pressure application can also result in lip inversion. Once the lip is inverted, any spring is more likely to become dislodged, and leakage inevitably results.

Proper installation practices can greatly reduce the chances of lip inversion. For more on installation, see ***page 177***. Use of a shaft with a proper chamfer is of paramount importance. For more on shaft chamfering, see ***page 126***. If the shaft is not chamfered properly, use of an assembly sleeve becomes necessary.

"Once the lip is inverted, any spring is more likely to become dislodged, and leakage inevitably results."

INVERTED LIP

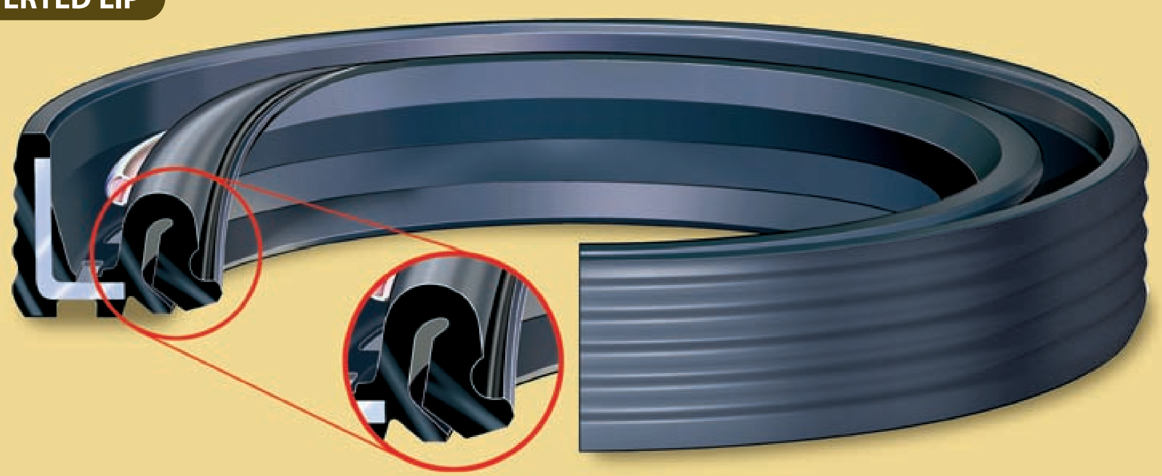


Figure 244: Can Easily Occur At Installation

troubleshooting

Material / Fluid Incompatibility

Incompatibility between a shaft seal's elastomeric lip material and the fluid being sealed is of primary concern to any seal designer. The best seal design in the world will still fail if the lip material loses its integrity in the face of the fluid it must seal.

Evidence of lip-fluid incompatibility can take many forms, but the most common manifestations are material swell and material degradation. ***Figure 245*** illustrates a seal lip that has been degraded as the result of incompatibility. A thorough researching of the compatibility between the lip material you plan to use and the fluid you'll be sealing is your best bet. For more on compatibility issues, see ***pages 36*** and ***37***.

"The best seal design in the world will still fail if the lip material loses its integrity in the face of the fluid it must seal."

MATERIAL / FLUID INCOMPATIBILITY

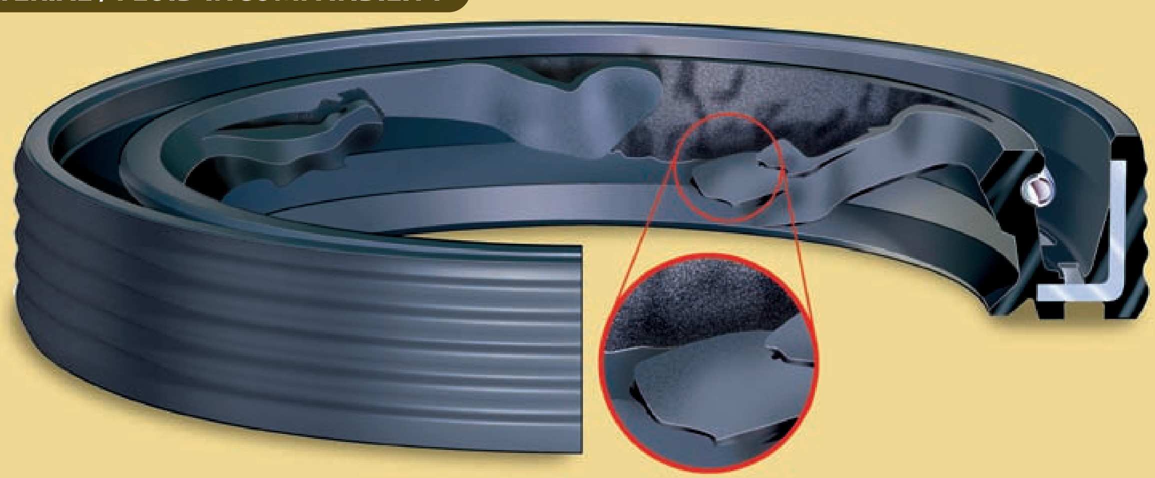


Figure 245: Material Degradation Due To Chain Scission

troubleshooting

Missing Spring.

Because garter springs help a shaft seal's lip maintain proper contact with the shaft, a missing spring (such as is illustrated in *Figure 246*) makes it much more likely that a leak path will develop between the lip and the shaft. A spring that is simply dislodged from its proper placement in the lip groove will be unable to function properly and is therefore problematic.

If your seal design incorporates a spring but the spring is missing, it's possible that the spring was never installed in the factory. Check to see if there are any spring coil marks within the spring groove and any light wear on the primary sealing lip. If so, the spring was there at some point; if not, the spring was never present, meaning a breakdown in the seal manufacturing process. It's possible that the spring was there initially but was dislodged during installation of the seal into the assembly. Checking your stock of unused seals to see if they have springs should help you determine which is most likely. If improper installation may have dislodged the spring, refine your installation procedures to prevent future reoccurrences. It's also possible that the spring joint itself separated. If the spring is available for review, see if the spring nib was formed properly. If not, notify the seal manufacturer.

"A missing spring makes it much more likely that a leak path will develop between the lip and the shaft."

MISSING SPRING

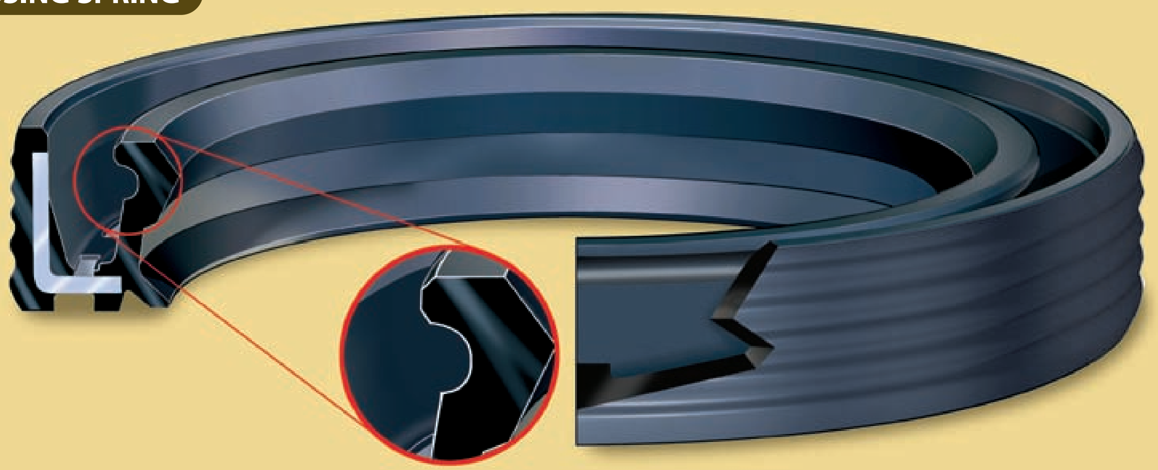


Figure 246: Contributes To Leak Path Development

troubleshooting

Oil Coking

Oil coking, the deposition of a hard layer of carbon on a shaft seal's lip, can result when oil comes in contact with high underlip temperatures. For example, the combination of high shaft speed and high sump temperature can cause excessive underlip temperature, which in turn burns the oil and deposits the layer of carbon on the lip. This carbon crust blocks the pumping ability of the lip, making leakage inevitable. *Figure 247* shows what an oil-coked lip looks like.

The only two ways to eliminate oil coking are to reduce the radial load of the lip, thus reducing wear and heat buildup, which in turn may bring the underlip temperature down enough to prevent the coking reaction with the lubricant. If this proves unfeasible, you may need to switch to a different lubricant, possibly a synthetic blend engineered to handle excessively high temperatures.

"Oil coking can result when oil comes in contact with high underlip temperatures."

OIL COKING

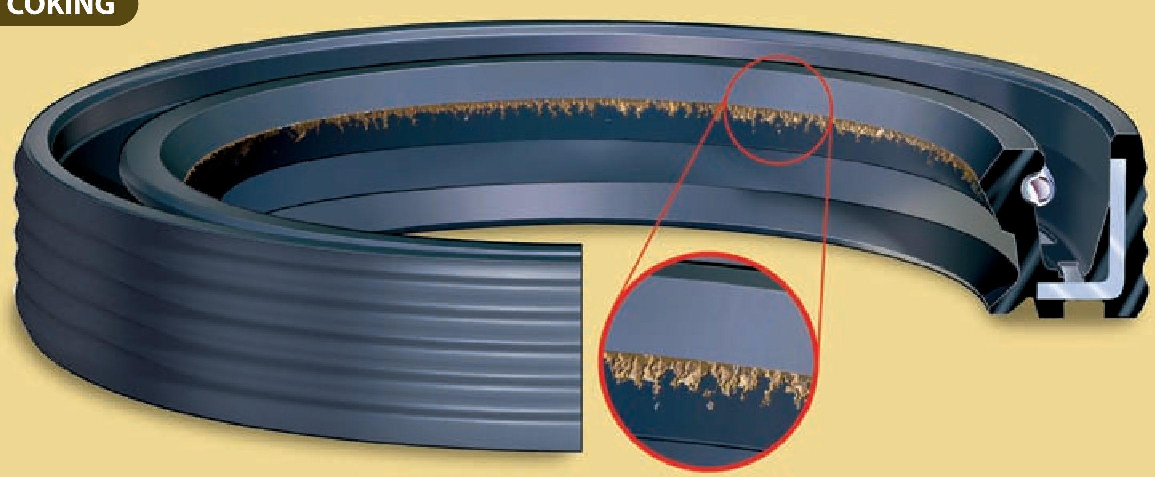


Figure 247: Blocks The Lip's Pumping Ability

section title

Paint Contamination.

In much the same way that oil coking interferes with the pumping action of the lip, paint on the underlip surface can also prevent the seal from functioning properly. As shown in *Figure 248*, paint on the seal lip can prevent the proper formation or functioning of the all-important microasperities that hold vital lubrication and contribute to the pumping of fluid back into the sump. Paint on the shaft itself in the region of lip contact can also cause problems and should be avoided.

Use of a paint mask can help prevent paint reaching unwanted areas. If assembly painting occurs in the field, be sure that those in charge of the process are aware of the dangers of getting paint on the lip surface, and specify use of a mask if necessary.

"Paint on the seal lip can prevent the proper formation or functioning of the all-important microasperities."

PAINT CONTAMINATION

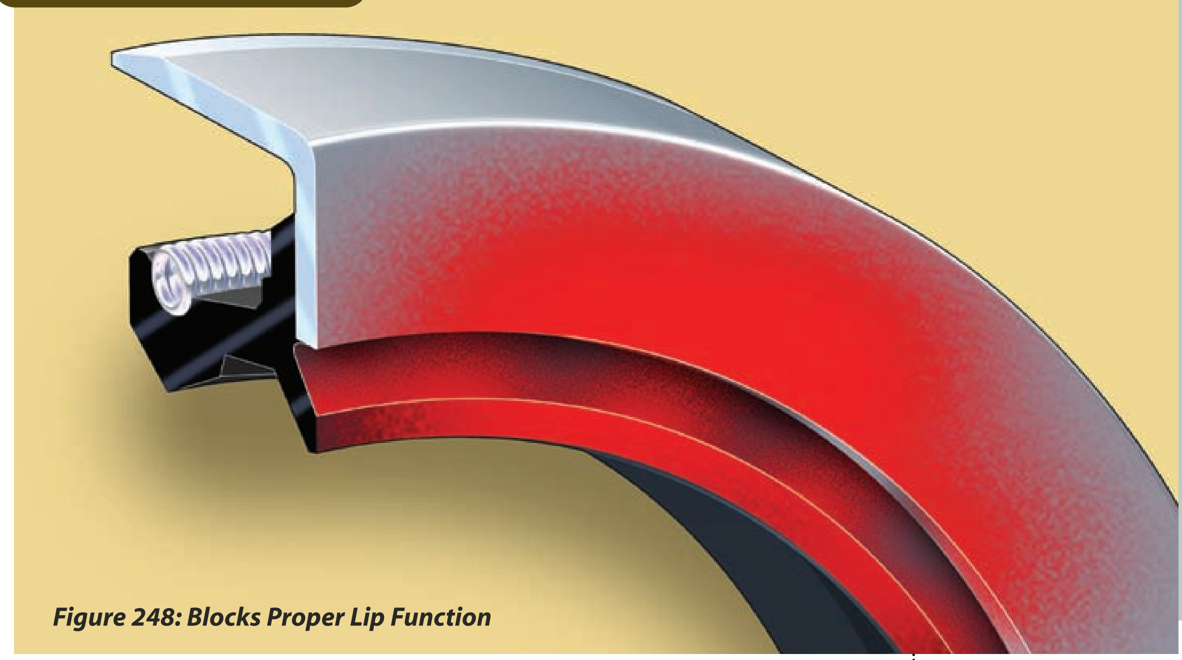


Figure 248: Blocks Proper Lip Function

troubleshooting

Scratched or Nicked Shaft.

A scratched or nicked shaft (such as is shown in *Figure 249*) is problematic because the scratches or nicks can prevent a shaft seal's lip from maintaining consistent contact with the shaft. Loss of lip-shaft contact can lead to leakage. Depending on its severity and placement, a damaged area of the shaft can also damage the lip, thereby hastening seal failure.

Scratching or nicking of the shaft can occur either during manufacture of the shaft or during assembly. Manufacturing processes should be reviewed to ensure material integrity. If necessary, nylon mesh sleeves can be used to protect the shaft during subsequent handling. You may also need to harden the shaft surface to make it more resistant to damage. A hardness of Rockwell C45 is recommended.

"Scratches or nicks can prevent a shaft seal's lip from maintaining consistent contact with the shaft."

SCRATCHED / NICKED SHAFT

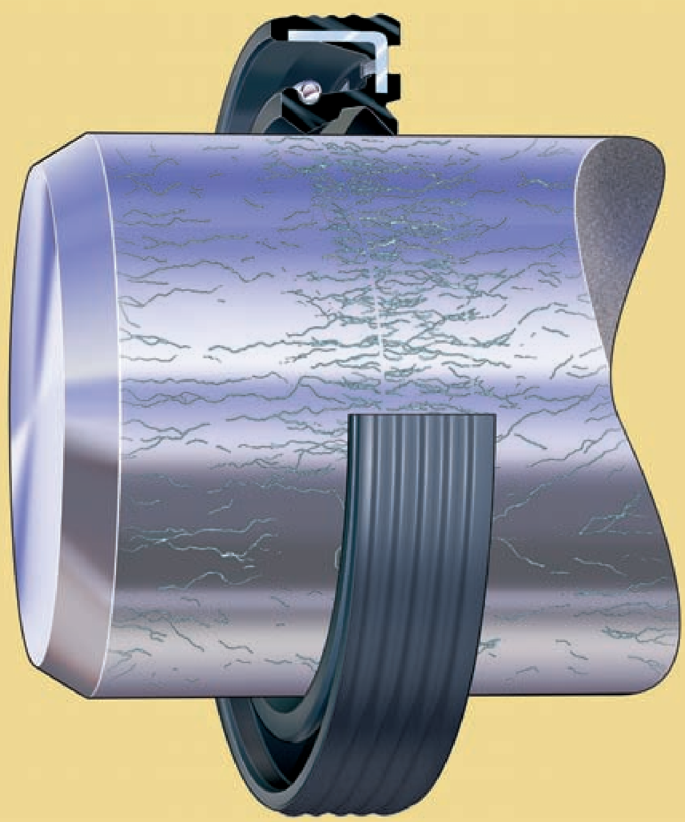


Figure 249: Can Cause Loss of Lip Contact

Sealant Contamination.

Sealant applied to the O.D. of a shaft seal to help hold it in place following installation is not uncommon. However, O.D. sealant can be problematic if it gets on the sealing lip or the shaft itself; in such instances it can impede the proper functioning of the underlip microasperities that hold lubrication and pump fluid back into the sump. *Figure 250* shows an example of what sealant contamination might look like.

Care should always be taken in applying O.D. sealant. If you're doing this yourself, be sure that the sealant is not inadvertently contacting other portions of the seal or assembly. You may also want to consider purchasing seals that are precoated as a way to minimize sealant contamination concerns.

"O.D. sealant can be problematic if it gets on the sealing lip or the shaft itself."

SEALANT CONTAMINATION

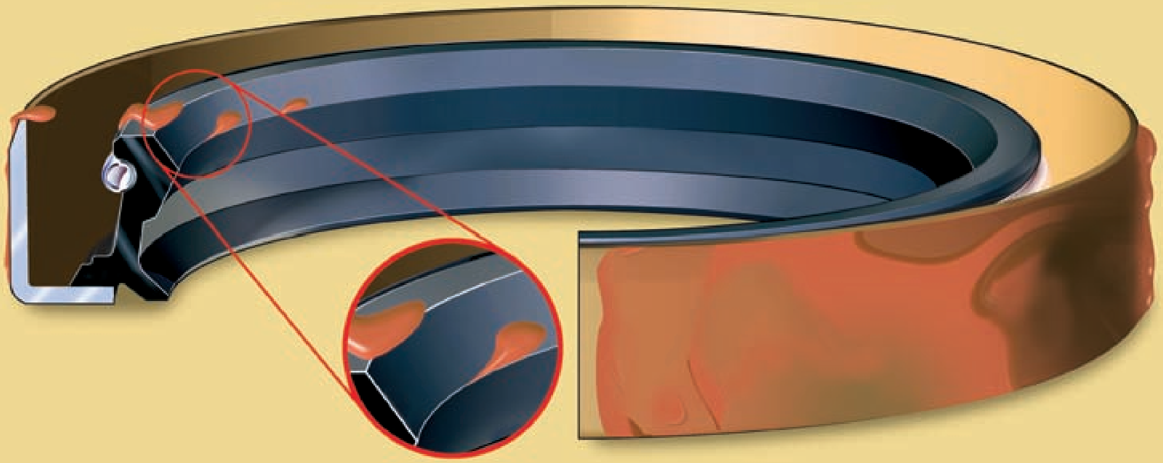


Figure 250: Can Impede Proper Lip Function

troubleshooting

Shaft Lead.

Screw threads or spiral grooves on the shaft surface can create leakage. Known as shaft lead (or machine lead), these threads or grooves result from relative axial movement of the finishing tool during the finishing operation. Improper shaft finishing can easily contribute to seal leakage and/or contaminant ingestion. *Figure 251* shows an example of shaft lead.

Plunge grinding has proven to be the most reliable finishing method for removing machine lead on rotating shafts. This is because plunge grinding eliminates any axial movement of the grinding wheel relative to the surface of the shaft. A mixed number (rather than whole number) RPM ratio (for example, 9.5 to 1) between the grinding wheel and the shaft (which should be rotating in opposite directions) is suggested to help prevent the introduction of spirals onto the shaft surface. Plunge grinding using mixed number ratios also greatly reduces the time required to achieve sparkout, the point where sparks are visible during the grinding operation. You must leave enough material on the shaft so that you can grind it to remove all traces of lead. If all of these recommendations are followed, the shaft surface should be free of lead. A grinding wheel with an 80-grit size will provide a surface finish of 8 to 17 µin. Ra per RMA recommendations.

"Screw threads or spiral grooves on the shaft surface can create leakage."

SHAFT LEAD

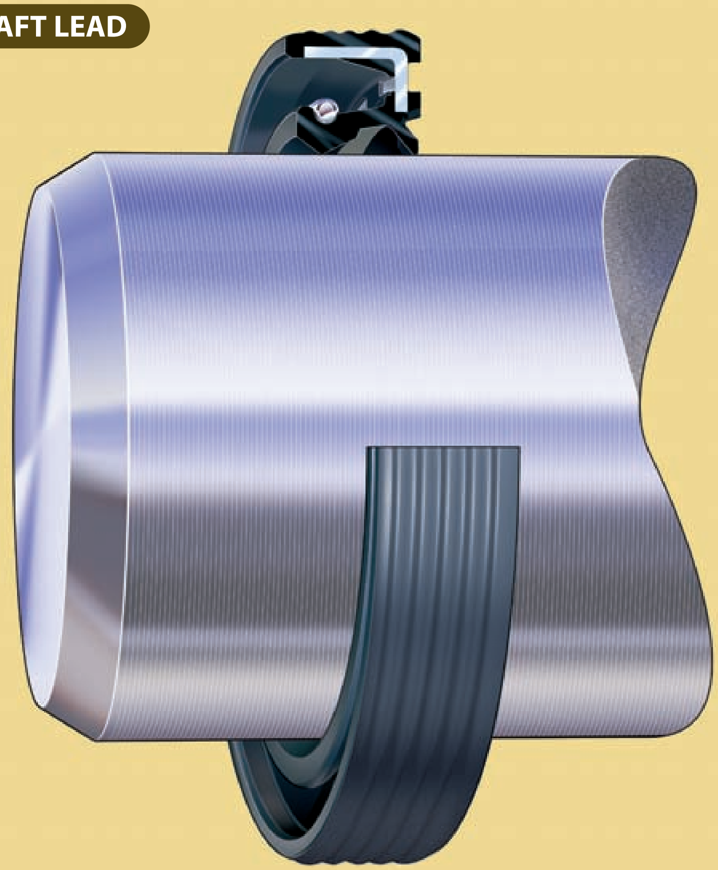


Figure 251: Can Contribute To Leakage & Contaminant Ingestion

troubleshooting

Stick-Slip.

Stick-slip is a phenomenon in which a shaft seal's sealing lip and the shaft surface alternate between adhesion and slipperiness. This alternating sticking and slipping is caused by insufficient lubrication. This insufficiency may be due to lack of proper initial lubrication, or it could be that high underlip temperatures have thinned the lubricant to the point that it can no longer consistently support the radial load of the seal.

Applications with splash lubrication at the seal area are prone to have stick-slip problems with the seal. As illustrated in ***Figure 252***, axial tears (known as chatter marks) in the seal contact pattern are indicative of stick-slip. Stick-slip often allows leakage and may, in extreme cases, even destroy the seal.

The key to preventing stick-slip is to make sure the sealing lip stays well lubricated. You might opt for a design with two lips, making sure that the area between the lips is grease-packed.

"This alternating sticking and slipping is caused by insufficient lubrication."

STICK-SLIP

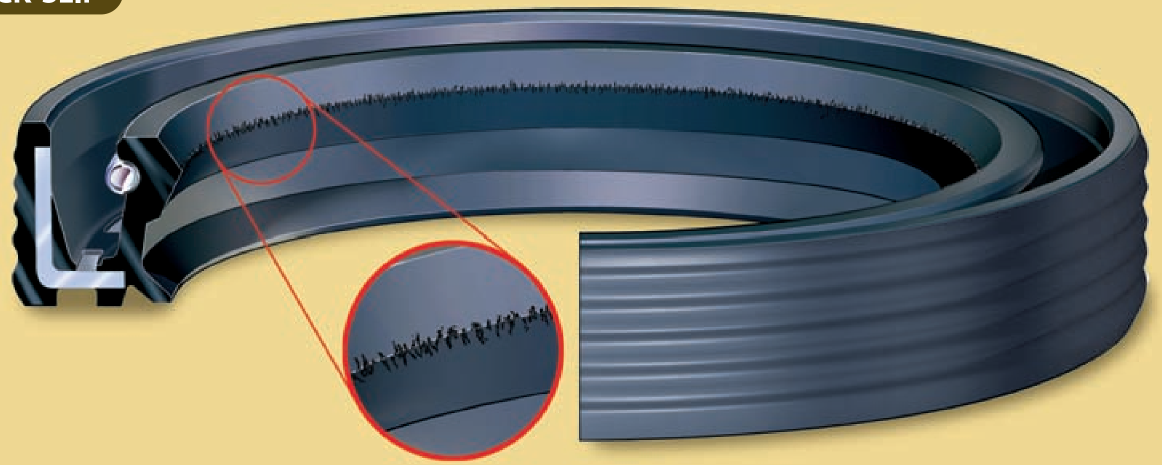
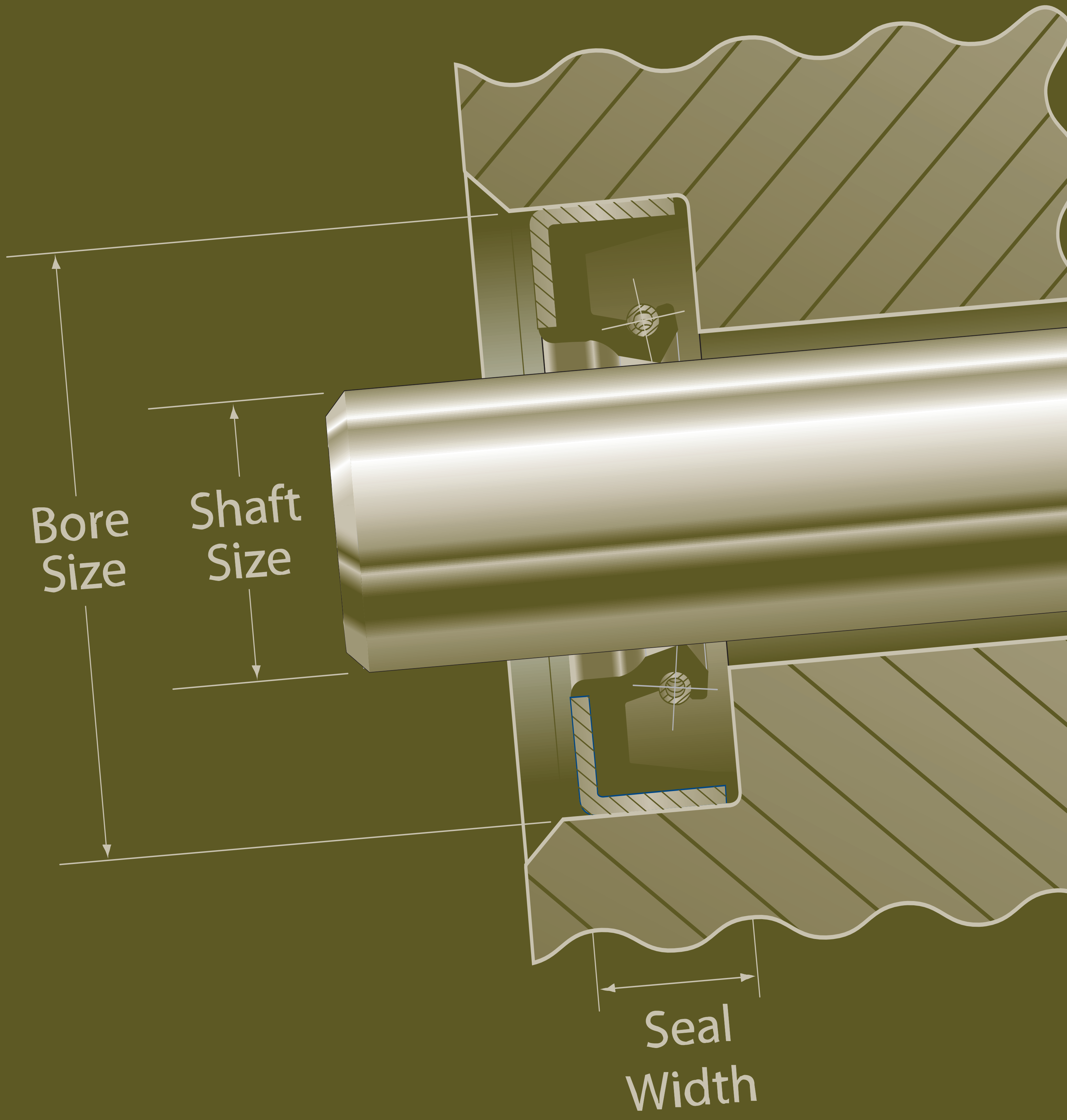


Figure 252: Alternating Adhesion & Slipperiness



ordering information

ordering information

Part Numbers & Descriptions.

Many shaft seal manufacturers use a legend-based part numbering system, which is intended to indicate the unique characteristics of any given seal. These characteristics typically include shaft size, bore size, seal width, seal design (style), O.D. treatment, and lip material.

R.L. Hudson & Company uses this legend-based system in our description, but *not* as our primary part number. There are many different applications requiring different seal designs and materials, some of which are minor, but important, modifications from the standard designation. It is possible for two seals to have the same legend-based part number, but have distinctly different seal designs (lip interference, for example) and material properties.

We individually design our seals for each unique application. We assign an R.L. Hudson part number, which is generated by our computer system. What follows is a description of the legend-based part numbering system, which we use as part of our description.

"We individually design our seals for each unique application."

SHAFT SIZE

The first two to four digits of a legend-based shaft seal part description indicate the shaft diameter (see ***Figure 253***) expressed in millimeters or thousandths of an inch.

BORE SIZE

The second two to four digits of the part description indicate the housing bore diameter (see ***Figure 253***) expressed in millimeters or thousandths of an inch.

SEAL WIDTH

Following a dash, the seal width (axial length of the seal, see ***Figure 253***) is expressed in millimeters or thousandths of an inch.

APPLICATION VARIABLES

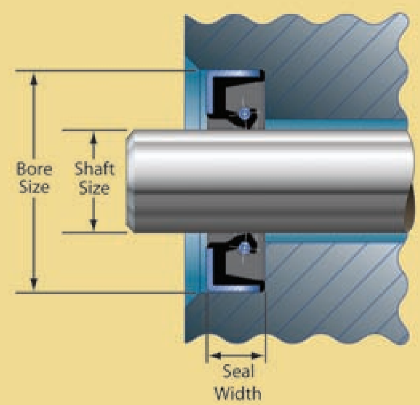


Figure 253: Pertinent Dimensions

SEAL DESIGN

Seal design (style) is specified by a two- or three-character alphabetic code, which designates the specific lip and case design of the seal. For more on standard seal designs, see ***page 94***; for more on non-standard designs, see ***page 99***.

TREATMENT	CODE
Rubber Covered	R
Precision Ground Metal	G
Metal w/Adhesive Coating	C
Metal w/Paint Coating	P

Table 72: O.D. Treatments

LIP MATERIAL	OLD CODE	OUR CODE
Nitrile	2	NBR
Polyacrylate	PA	ACM
PTFE	T	FEP
Fluoroelastomer	V	FKM
Silicone	S	VMQ

Table 73: Lip Materials

O.D. TREATMENT

A single letter code indicates the type of treatment required on the outside diameter (O.D.) of the seal. ***Table 72*** lists the four main options in this area. For more on O.D. treatments, see ***page 86***.

LIP MATERIAL

The last portion of a legend-based part description is a numeric or alphabetic code for the generalized base polymer used in the seal lip. ***Table 73*** lists the variety of choices available. Note that R.L. Hudson & Company uses a coding system based on actual ASTM D1418 material designations. For more on materials, see the profiles beginning on ***page 41***.

SAMPLE PART DESCRIPTIONS

A sample part description (in inch units) might read as follows: **10001500 – 375 SBY C NBR**

This indicates a seal for a 1.000″ shaft (1000) in a 1.500″ bore (1500), a .375″ seal width (375) in the "SBY" design (SBY) with an adhesive coating on the O.D. (C) and an elastomeric lip made of nitrile (NBR).

A sample part description (in millimeters) might read as follows: **30 60 7 SC R FKM**

This indicates a seal for a 30 mm shaft (30) in a 60 mm bore (60), a 7 mm seal width (7) in the "SC" design (SC) with a rubber-covered O.D. (R) and an elastomeric lip made of fluoroelastomer (FKM).

SPECIFYING YOUR SEAL

The "Shaft Seal Specification" form on ***page 249*** can help you address all of the questions that typically must be answered in order to design or select the proper shaft seal for a given application. Each variable on this form has been discussed in a preceding section of this book.

SHAFT SEAL ENGINEERING INFORMATION

DATE

COMPANY

LOCATION

CONTACT PERSON | PHONE

FAX | EMAIL

PART NUMBER

TYPE OF EQUIPMENT

DESCRIBE THE APPLICATION

SEAL PRINTS
❏ YES ❏ NO

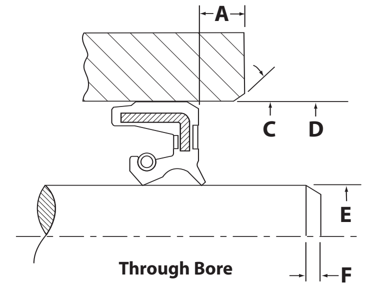
APPLICATION PRINTS
❏ YES ❏ NO

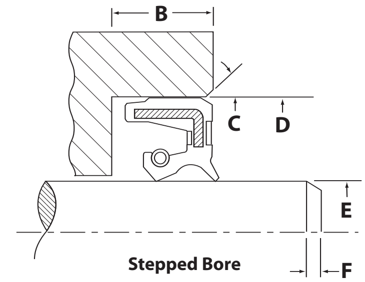
INSTALLATION TOOL PRINTS
❏ YES ❏ NO

INST. PROCEDURES ATTACHED
❏ YES ❏ NO

PRODUCTION DATE

EST. ANN. USAGE | PEAK MO. USAGE





BORE TYPE: ❏ THROUGH ❏ STEPPED

A. POSITION OF SEAL IN THROUGH BORE ______________________

B. STEPPED BORE DEPTH ______________________

C. BORE CHAMFER AND ANGLE ______________________

D. BORE DIAMETER ______________________

E. SHAFT DIAMETER ______________________

F. SHAFT CHAMFER AND ANGLE ______________________

G. SHAFT SEAL STYLE (SEE HANDBOOK, P. 94) ______________________

SHAFT

❏ HORIZONTAL
❏ VERTICAL

IF VERTICAL:
❏ TOP ❏ BOTTOM ❏ BOTH

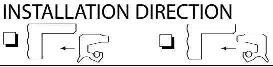
HOLES | SPLINES | KEYWAY

MATERIAL | FINISH RA | RPM | RZ | SHAFT LEAD SPECIFICATION | HARDNESS

BORE

MATERIAL | FINISH | HARDNESS

INSTALLATION DIRECTION
❏ ❏



MOTION

❏ ROTATING

RPM:
NORMAL MAXIMUM

DIRECTION OF SHAFT ROTATION FROM AIR SIDE
❏ CW ❏ CCW ❏ BI-ROTATIONAL

SHAFT RUNOUT/TIR | STB MISALIGNMENT

FREQUENCY OF ROTATION
❏ CONTINUOUS ❏ INTERMITTENT

❏ RECIPROCATING | STROKE LENGTH | CYCLES PER MIN.

❏ OSCILLATING | DEGREES OF ARC | CYCLES PER MIN.

FLUID MEDIUM

❏ INTERNAL | TYPE | LEVEL

❏ EXTERNAL | TYPE | LEVEL

TEMP.

❏ °C
❏ °F

MIN. TEMP. | NORM. TEMP. | MAX. TEMP.

PRESS.

MIN. PSI | NORMAL PSI | MAX. PSI

LUBE

❏ DRY ❏ MIST ❏ FLOODED

BEARING

❏ BALL OR ROLLER BEARING ❏ BUSHING

ADDITIONAL REQUIREMENTS

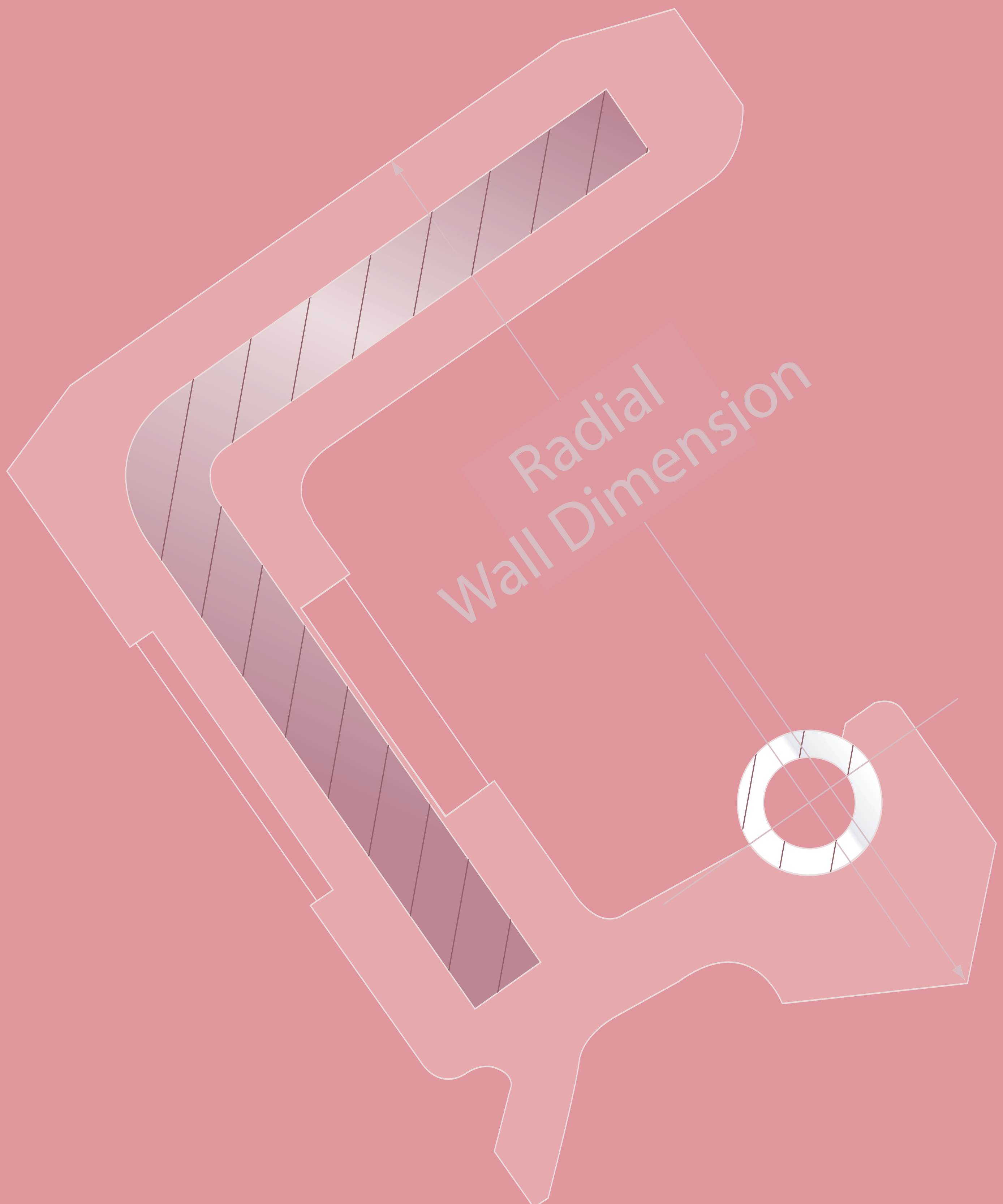
COMPOUND REQUIREMENT:

❏ NITRILE
❏ HIGH-TEMP NITRILE
❏ HNBR
❏ FLUOROELASTOMER
❏ SILICONE
❏ POLYACRYLATE
❏ PTFE
❏ OTHER ______________

OD TREATMENT:

❏ RUBBER COVERED
❏ BORE SEALANT
❏ GROUND METAL
❏ COATED METAL
❏ OTHER ______________

SPECIAL INSTRUCTIONS



reference

reference

Glossary

A

ABRASION - progressive wearing away of a surface in service (like the sealing lip of a shaft seal) by mechanical action such as scraping, rubbing, or erosion.

ABRASION RESISTANCE - resistance of a rubber compound to wearing away when in dynamic contact with an abrasive surface.

ADDITIVE - material added to an elastomeric compound to alter its properties, e.g. a reinforcing agent to improve strength or a plasticizer to aid flexibility and processibility.

ADHERE - (a) to cling or stick together; or (b) to cause two surfaces to stick together.

ADHESION - tendency of rubber or other material to stick to a contact surface; may result from chemical or physical interlocking.

ADHESIVE - substance used to hold materials together.

AIR SIDE - side of the seal facing away from the fluid being sealed.

AIR SIDE ANGLE - angle between the air side surface of the primary lip of a shaft seal and the shaft; also known as the barrel angle.

ASPERITIES - microscopic pores that develop on a shaft seal's elastomeric sealing lip at the point of contact with the shaft as a result of wear; asperities can be beneficial in that they help hold lubrication for the lip and facilitate a micropumping action that prevents leakage; also known as microasperities.

ASSEMBLY CONE - installation aid that fits over part or all of a shaft and decreases the chances of damaging a shaft seal's lip on potential hazards such as keyways or splines.

AUXILIARY LIP - non-spring-loaded, optional lip which, if present, extends axially or radially from the heel of the primary sealing lip on the air side of a shaft seal and prevents

"This glossary contains a wide variety of terms frequently used in the sealing industry. Familiarity with these terms will be beneficial as you select or design shaft seals."

contaminants from reaching the contact point; also known as a dirt lip, dust lip, or secondary lip.

AXIAL CLEARANCE - space between the end of a shaft seal's head section and an inner case; also known as lip clearance.

AXIAL DIRT LIP - non-spring-loaded lip that extends axially from the heel of the primary sealing lip on the air side of a shaft seal; impinges on a radial flange to prevent contaminants from reaching the contact point.

B

BARREL ANGLE - angle between the air side surface of the primary lip of a shaft seal and the shaft; also known as the air side angle.

BEAM LENGTH - axial distance from the thinnest portion of a shaft seal's flexible lip to the point at which the lip contacts the shaft.

BEDDING-IN - period of initial operation during which wear to a shaft seal's lip is most pronounced and the contact surface develops; also known as break-in or run-in.

BELL-MOUTHING - condition in which a shaft seal's elastomeric lip contacts the shaft on the seal's air side rather than on the tip.

BI-DIRECTIONAL SEAL - seal designed for use with a shaft that rotates in both a clockwise and counterclockwise direction; also known as a birotational seal.

BLISTER - an enclosed cavity that protrudes from, and thus deforms, the sealing surface.

BOND - adhesion between a shaft seal's elastomeric sealing lip and the metal case.

BONDED SEAL - type of shaft seal with an elastomeric sealing element bonded to a case during molding.

BORE - cylindrical surface machined into the housing to mate with the outside diameter of a shaft seal; also known as the housing bore.

BREAK-IN - period of initial operation during which wear to a shaft seal's lip is most pronounced and the contact surface develops; also known as bedding-in or run-in.

C

CAP - portion of a shaft seal's head section purposely removed during knife trimming or demolding; sometimes referred to as the "maidenhead."

CASE - rigid member (typically steel) to which the sealing lip is bonded during molding; protects the lip and provides a surface to be press-fitted into the housing; double case shaft seals feature both an inner case and an outer case; also known as a shell.

CASE WIDTH - axial width of the seal case.

CAVITY - hollow space within a mold in which uncured rubber is shaped and vulcanized; also known as mold cavity.

CAVITIES

Figure 254: Mold Cavities

CHAMFER - beveled edge in a component to facilitate assembly of a seal onto a rod or shaft, or into a cylinder or housing; also known as a lead-in chamfer.

CHATTER MARKS - axial tears in a shaft seal's contact pattern; indicative of stick-slip.

CHECKING - cracking or crazing of an elastomeric surface, such as the lip of a shaft seal.

CIRCUMFERENTIAL SPEED - speed of the moving shaft expressed in feet per second (fps).

CLEARANCE - gap between two mating surfaces, such as the necessary gap between a moving shaft and the housing in which it moves; a shaft seal can block this gap to prevent lubricant leakage.

COCKING - misalignment of a shaft seal such that it is not perpendicular to the bore in which it is supposed to fit and the shaft it is supposed to seal; may be caused by incorrect installation or improper design; also known as seal cocking.

reference

Figure 255: Garter Spring Coils

COIL - a single turn of a coiled wire garter spring.

COINING - process whereby patterns (such as helical ribs) are transferred onto a surface (such as a PTFE sealing lip) during a molding operation.

CONTACT POINT - point at which a shaft seal's sealing lip and the shaft touch; also known as the interface.

CRACK - sharp break or fissure in a rubber surface caused by excessive strain and/or exposure to detrimental environmental conditions, such as ozone, weather, or ultraviolet (UV) light.

CURE - heat-induced process whereby the long chains of the rubber molecules become cross-linked by a vulcanizing agent to form three-dimensional elastic structures. This reaction transforms soft, weak, non-crosslinked materials into strong elastic products; also known as vulcanization.

CUT - slice-like opening in a rubber surface caused by unwanted contact between the surface and a sharp object.

D

DEFORMATION - change in the shape of a seal as a result of compression.

DIFFERENTIAL THERMAL EXPANSION - variance in the heat-induced rates of expansion for two different materials (such as the metal of a housing bore and the metal case of a shaft seal); this variance may lead to the formation of a gap between the case and the housing, and this gap may allow leakage.

DIRT LIP - non-spring-loaded, optional lip which, if present, extends axially or radially from the heel of the primary sealing lip on the air side of a shaft seal and prevents contaminants from reaching the contact point; also known as an auxiliary lip, dust lip, or secondary lip.

DOUBLE LIP - describes a shaft seal with both a primary sealing lip and a secondary (contaminant exclusion) lip, or that has two sprung lips to separate fluids.

DRY RUNNING - shaft seal operation without lubrication at the contact point between the seal and the shaft; will contribute to the buildup of friction and heat, hastening seal failure.

DUROMETER - (a) an instrument that measures the hardness of rubber by its resistance to surface penetration of an indentor point; and (b) the numerical scale indicating the hardness of rubber.

DUST LIP - non-spring-loaded, optional lip which, if present, extends axially or radially from the heel of the primary sealing lip on the air side of a shaft seal and prevents contaminants from reaching the contact point; also known as an auxiliary lip, dirt lip, or secondary lip.

DYNAMIC RUN-OUT (DRO) - amount (in inches or millimeters) that the shaft's sealing surface does not rotate around the true center; taken by applying an indicator to the side of the shaft as it slowly rotates; should not exceed 0.010 inch (0.25 mm) TIR; also known as shaft run-out.

DYNAMIC SEAL - seal functioning in an environment in which there is relative motion (e.g. rotary, reciprocating, or oscillating) between the mating surfaces being sealed.

E

ELASTOMER - any natural or synthetic material meeting the following requirements: (a) it must not break when stretched 100%; and (b) after being held at 100% stretch for five minutes then released, it must return to within 10% of its original length within five minutes.

F

FATIGUE RESISTANCE - capable of withstanding fatigue caused by repeated bending, extension, or compression; also known as flex resistance.

FILM THICKNESS - in a shaft seal, the tiny distance between the primary sealing lip and the shaft that is typically occupied by a thin film of lubricant.

FLASH - excess rubber remaining on the parting line of a molded rubber product.

FLEX RESISTANCE - capable of withstanding fatigue caused by repeated bending, extension, or compression; also known as fatigue resistance.

FLEX SECTION - part of a shaft seal, typically the area between the head section (sealing lip) and the heel section (point of connection between the lip and the case); thickness of the flex section impacts the seal's ability to maintain optimal interference with the shaft.

FLEX THICKNESS - thickness of the area on a shaft seal between the head (sealing lip) and the heel (point of connection between the lip and the case); flex thickness should be thick enough to prevent unwanted lip distortion but not so thick as to compromise followability.

FLINGER - washer-like device designed to lend radial momentum to a liquid so as to keep it away from the sealing lip; may be incorporated into a wear sleeve; also known as a slinger.

FLUID SIDE - side of the seal facing the fluid being sealed; also known as the oil side.

FOLLOWABILITY - ability of the sealing lip to maintain contact with the shaft despite vibrations or dynamic runout.

G

GARTER SPRING - helically coiled spring, typically made of carbon steel or stainless steel wire, formed into a ring and used in a shaft seal to help maintain contact between the sealing lip and the shaft.

GRINDING CHATTER - excessive out-of-roundness of a shaft; defined by the RMA as greater than 45 cycles or lobes; also known as waviness.

H

HARDNESS - measure of a rubber's relative resistance to an indentor point on a testing device. Shore A durometers gauge soft to medium-hard rubber. Shore D durometers are more accurate on samples harder than 90 Shore A.

HEAD SECTION - part of a shaft seal, typically the air and fluid side surfaces of the sealing lip and spring groove (if present).

HEAD THICKNESS - radial distance between the bottom of a shaft seal's spring groove and the contact point between lip and shaft.

HEAT RESISTANCE - rubber compound's capacity to undergo exposure to some specified level of elevated temperature and retain a high level of its original properties.

HEEL GASKET - shaft seal featuring a rubber coating on the outside of the case (air side) to help prevent bore leakage and corrosion of the case by external contaminants.

HEEL SECTION - part of a shaft seal, typically the area of attachment for the sealing lip and the (outer) metal case.

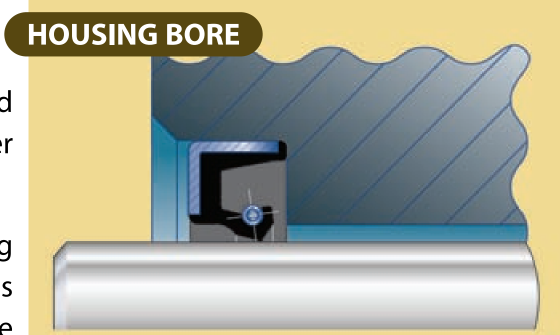


Figure 256: Contacts Seal O.D.

HOUSING BORE - cylindrical surface machined into the housing to mate with the outside diameter of a shaft seal; also known simply as the bore.

HYDRODYNAMIC SEAL - shaft seal utilizing helical ribs, pads, grooves, or sinusoidal patterns molded into the sealing lip on the air side to force fluid weepage back under the lip; correct orientation of the patterns (clockwise, counter-clockwise, or bi-directional) depends on the direction of shaft rotation as seen from the air side.

I

I.D. - inside diameter of a seal or component.

INCOMPLETE TRIM - instance in which not all of the material that should have been trimmed from a surface was actually removed.

INITIAL TENSION - the "preload" created in a spring by backwinding the coils during the manufacturing process such that force is required to pull the coils apart.

INNER CASE - in shaft seals featuring a double case, the innermost of the two cases; typically supplies increased structural rigidity for demanding applications.

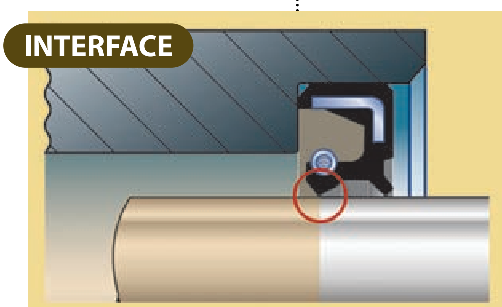


Figure 257: Lip-Shaft Contact

INTERFACE - point at which a shaft seal's sealing lip and the shaft touch; also known as the contact point.

INTERFERENCE - difference between the diameter of a shaft seal's sealing lip and the diameter of the shaft to be sealed; interference is designed in so that the lip diameter is smaller than the shaft diameter, thus ensuring the formation (and maintenance) of a contact point between the lip and the shaft.

INVERSION - reversal or turning over of a sealing lip that will lead to leakage; can occur during installation.

K

KNIT LINE - imperfection of the seal material due to premature curing (scorching) of the rubber compound.

L

LEAD-IN (CHAMFER) - beveled edge in a component to facilitate assembly of a seal onto a rod or shaft, or into a cylinder or housing; also known simply as a chamfer.

LEAK RATE - rate at which a fluid (liquid or gas) passes a seal or barrier.

LIP CLEARANCE - space between the end of a shaft seal's head section and an inner case; also known as axial clearance.

LIP DIAMETER - inside diameter of a shaft seal's primary lip measured with the garter spring (if used) installed.

LIP LENGTH - axial distance from the thinnest portion of a shaft seal's flexible lip to the point at which the lip contacts the shaft; also known as beam length.

LIP OPENING PRESSURE (LOP) - measure of the pressure required to flow air at 10,000 cm^3/minute between a shaft seal's contact point and a shaft-sized test mandrel; the seal case OD must be concentric with the mandrel and air must be applied to the outside lip surface.

LIP SEAL - device utilizing the planned interference between an elastomeric lip and a mating surface (such as a shaft) to prevent leakage.

LOAD - actual pressure at a sealing face; in the case of a shaft seal, the sum of the elastomeric lip's inherent beam force, the hoop force (as a result of lip stretch upon installation), and the garter spring tension, all of which contribute to shaft loading at the contact point.

LOW TEMPERATURE FLEXIBILITY - ability of an elastomeric product (such as the sealing lip of a shaft seal) to resist cracking or breaking when flexed or bent at low temperatures.

M

MACHINE LEAD - screw threads or spiral grooves seen on a shaft due to improper lathe machining; plunge grinding is recommended to eliminate machine lead.

MEMORY - an elastomer's ability to regain its original size and shape following deformation.

MENISCUS - curved boundary at the meeting point of air and fluid between the sealing lip of a shaft seal and the shaft.

MICROASPERITIES - microscopic pores that develop on a shaft seal's elastomeric sealing lip at the point of contact with the shaft as a result of wear; microasperities can be beneficial in that they help hold lubrication for the lip and facilitate a micropumping action that prevents leakage; also known simply as asperities.

MODULUS - the force in psi (stress) required to produce a certain elongation (strain), usually 100%, in a material sample; a good indication of toughness; also known as tensile modulus or tensile stress.

MOLDED LIP SEAL - shaft seal with a sealing lip formed by molding rather than by trimming with a knife.

MOLD IMPRESSION - imperfection molded into the surface of a material; typically due to nicks and other blemishes on the surface of the mold cavity.

N

NIB JOINT - point at which the two ends of an extension spring are joined to form a circular garter spring.

NICK - unwanted void within the sealing material created after molding.

NONFILL - unwanted void within the sealing material created during molding; typically due to improper material flow within the mold.

NON-SPRING LOADED - used to describe a shaft seal without a garter spring as part of its sealing lip.

NOSE GASKET - shaft seal featuring a rubber coating on the fluid side used to prevent leakage due to improper finishing of the bottom of the bore; the rubber coating also helps prevent corrosion of the case by the sealed fluid.

O

O.D. - outside diameter of a seal or component.

OFFSET - amount (in inches or millimeters) that the shaft center is offset relative to the bore center; taken without the shaft moving; almost always exists to some degree but should be no more than 0.010 inch (0.25 mm) TIR; also known as shaft-to-bore misalignment (STBM).

OIL SEAL - specific type of shaft seal designed to retain oil.

OIL SIDE - side of the seal facing the fluid being sealed; also known as the fluid side.

OIL SIDE ANGLE - angle between the fluid side surface and the shaft; also known as the scraper angle.

OSCILLATING SHAFT - rotating shaft with limited, reversing travel, as in an on/off valve.

OUTER CASE - in shaft seals featuring a double case, the outermost of the two cases; typically contains the inner case and provides the point of attachment for the sealing lip.

OUT-OF-ROUNDNESS (OOR) - extent to which a shaft's cross-section, a seal O.D., or the housing I.D. deviates from a true circle.

OUT-OF-ROUNDNESS

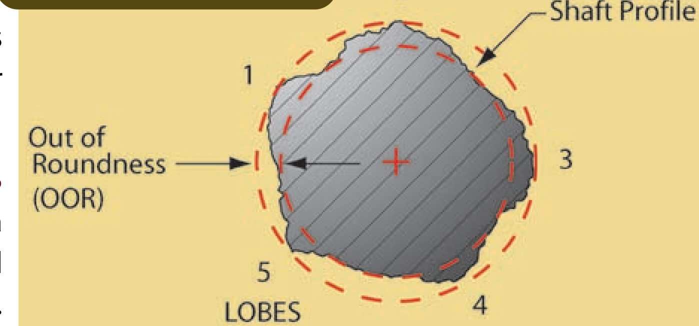


Figure 258: Deviation From A True Circle

P

PASSIVATION - process of reducing the chemical reactivity of a metallic surface through exposure to a nitric acid solution; sometimes used with stainless steel wire springs to accentuate corrosion resistance.

PLUNGE GRINDING - preferred method for finishing the surface of a shaft so as not to leave machine lead that can later contribute to seal leakage.

POOR BOND - inadequate adhesion between two layers of material; can occur between two rubber layers or in a rubber-to-metal bond.

POROSITY - instance in which a material is full of numerous tiny openings.

PRIMARY LIP - typically spring-loaded elastomeric lip of a shaft seal that prevents unwanted movement of fluid by maintaining contact (interference) with the moving shaft.

reference

Q

QS 9000 - Quality System developed by the automotive industry to supplement the ISO 9000 standard.

R

RADIAL DIRT LIP - non-spring-loaded lip that extends radially from the heel of the primary sealing lip on the air side of a shaft seal; impinges on the shaft to prevent contaminants from reaching the contact point.

RADIAL LOAD - sum of all forces (such as seal interference and garter spring tension) that maintain contact between a shaft seal's lip and the shaft; expressed in ounces per inch of shaft circumference; should be kept to just enough to seal without generating unnecessary friction and seal wear.

RADIAL WALL DIMENSION

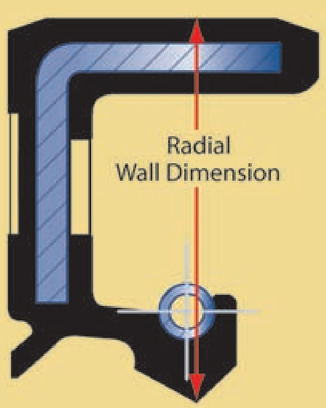


Figure 259: Radial Distance Between Seal O.D. & Seal Contact Point

RADIAL WALL DIMENSION (RWD) - radial distance between the shaft seal O.D. and the lip I.D. (contact point) as measured on a complete but uninstalled seal.

RADIAL WALL VARIATION (RWV) - extent to which the radial wall dimension of a shaft seal is not consistent; excessive variation can prevent a seal from seating properly in the bore.

RECIPROCATING SEAL - dynamic seal used to seal pistons or rods that are in linear motion.

RHEOMETER - cure meter which determines and plots a cure curve illustrating the state of cure for a given time and temperature; typically either an Oscillating Disk Rheometer (ODR) or a Moving Die Rheometer (MDR).

ROTARY SHAFT - shaft that rotates clockwise (CW), counterclockwise (CCW), or in both directions (variously CW and CCW).

ROUGH TRIM - instance in which the trimming of a sealing surface leaves it with unwanted irregularities on both sides of the contact point.

RUN-IN - period of initial operation during which wear to a shaft seal's lip is most pronounced and the contact surface develops; also known as bedding-in or break-in.

RUNOUT (SHAFT) - phenomenon which occurs when the shaft's axis and the axis of rotation are different, causing the shaft to wobble or gyrate; expressed in inches followed by the abbreviation "TIR" (Total Indicator Reading).

R VALUE - axial distance between the centerline of the garter spring and the contact point; a positive R value means the spring is located toward the air side relative to the contact point, and this is desirable; a negative R value means the spring is located toward the fluid side, which will result in immediate leakage; also known as spring position.

S

SCOOP TRIM - instance in which a seal surface is concave as a result of trimming.

SCORCHING - premature curing of rubber during storage or processing, usually caused by excessive heat.

SCORING - grooving of the shaft's surface (as by the sealing lip), or scratching of a shaft seal's O.D. (as by a rough housing bore).

SCRAPER ANGLE - angle between the fluid side surface and the shaft; also known as the oil side angle.

SCRATCH - superficial blemish on the surface of a seal or shaft due to abrasion.

SCUFFING - damage to a shaft seal's metal surface due to adhesive wear.

SEAL COCKING - misalignment of a shaft seal such that it is not perpendicular to the bore in which it is supposed to fit and the shaft it is supposed to seal; may be caused by incorrect installation or improper design; also known simply as cocking.

SEAL WIDTH - total axial measurement of a seal (including the case, if present).

SECONDARY LIP - non-spring-loaded, optional lip which, if present, extends down from the heel of the primary sealing lip on the air side of a shaft seal and prevents contaminants from reaching the contact point; also known as an auxiliary lip, dirt lip, or dust lip.

SHAFT - rotating, reciprocating, or oscillating component that operates within a cylinder or housing.

SHAFT DIAMETER - diameter of the shaft expressed in inches or millimeters.

SHAFT FINISH - usually meant to be the surface roughness measured in microinches or micrometers Ra; a low finish number is indicative of a smoother surface than a high finish number; rotating shafts need to be finished in accordance with the RMA specifications that also define Rz and Rpm; also known as surface finish.

SHAFT RUN-OUT - amount (in inches or millimeters) that the shaft's sealing surface does not rotate around the true center; taken by applying an indicator to the side of the shaft as it slowly rotates; should not exceed 0.010 inch (0.25 mm) TIR; also known as dynamic run-out.

SHAFT SEAL - dynamic seal designed to retain or contain fluids and/or exclude foreign materials through the exertion of radial pressure (due to interference) on a moving shaft.

SHAFT SPEED - speed of a moving shaft expressed in rotations per minute (rpm).

SHAFT-TO-BORE MISALIGNMENT (STBM) - amount (in inches or millimeters) that the shaft center is offset relative to the bore center; taken without the shaft moving; almost always exists to some degree but should be no more than 0.010 inch (0.25 mm) TIR; also known as offset.

SHELL - rigid member (typically steel) to which the sealing lip is bonded during molding; protects the lip and provides a surface to be press-fitted into the housing; double shell shaft seals feature both an inner shell and an outer shell; also known as a case.

SINGLE LIP - used to describe a shaft seal with one sealing lip.

SLINGER - washer-like device designed to lend radial momentum to a liquid so as to keep it away from the sealing lip; may be incorporated into a wear sleeve; also known as a flinger.

SPARKOUT - point where sparks are visible during a grinding operation.

SPIRAL TRIM - instance in which the trimming of a sealing surface leaves an undesirable deep spiral groove.

SPRINGBACK - tendency of a shaft seal with a rubber-covered O.D. to unseat itself slightly following installation due to shearing stresses between the rubber and the housing bore.

SPRING CLEARANCE - radial distance between the top of a shaft seal's garter spring and the innermost portion (whether metal or rubber) of the seal's case.

SPRING GROOVE - radiused opening molded into a shaft seal's elastomeric sealing lip to hold the garter spring.

SPRING LOAD - total tension generated by a garter spring when stretched to the designed deflected length; calculated as the combination of the spring's initial tension and its spring rate.

SPRING LOADED - used to describe a shaft seal with a garter spring as part of its sealing lip.

SPRING LOADED

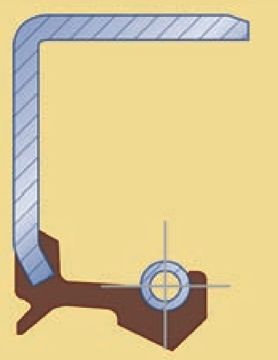


Figure 260: Garter Spring In Primary Lip

SPRING POSITION - axial distance between the centerline of the garter spring and the contact point; also known as R value; a positive R value means the spring is located toward the air side relative to the contact point, and this is desirable; a negative R value means the spring is located toward the fluid side, which will result in immediate leakage.

SPRING RATE - the additional force required to stretch a spring from position A to position B divided by the amount of stretch (N/mm).

SPRING WIND-UP - tendency of an assembled (but uninstalled) garter spring to deform rather than remain flat; for example, to twist into a figure 8.

SPRUNG INTERFERENCE - amount of interference between a sprung lip of a shaft seal and the shaft; calculated as the difference between the shaft diameter and the sprung lip diameter.

SPRUNG LIP - a shaft seal lip with a garter spring in place.

SPRUNG LIP DIAMETER - inside diameter of a shaft seal's primary lip measured with the garter spring installed.

STATIC SEAL - seal functioning in an environment in which there is no relative motion between the mating surfaces being sealed.

STICK-SLIP - phenomenon in which a shaft seal's sealing lip and the shaft surface alternate between adhesion and slipperiness due to insufficient lubrication; may allow leakage or even destroy the seal.

STRESS RELIEVING - process of relieving stresses in an unassembled coiled spring through exposure to heat; intended to help ensure that the spring force will not be adversely affected by heat during actual service.

SUMP - cavity or reservoir within which the fluids of a system are contained.

SUMP TEMPERATURE - temperature of the fluid within an assembly's sump.

SURFACE CONTAMINATION - unwanted material (such as dust or dirt) on the surface of a seal or shaft.

SURFACE FINISH - usually meant to be the surface roughness measured in microinches or micrometers Ra; a low finish number is indicative of a smoother surface than a high finish number; rotating shafts need to be finished in accordance with the RMA specifications that also define Rz and Rpm; also known as shaft finish.

SURFACE SPEED - speed of a moving shaft's surface expressed in meters per minute (mpm) or feet per minute (fpm).

T

TEAR - a separation or pulling away of part of a sealing structure.

TEAR RESISTANCE - resistance to the growth of a nick or cut in a rubber specimen when tension is applied.

TENSILE STRENGTH - force in pounds per square inch (psi) required to break a rubber specimen.

TIR - Total Indicator Reading; total range of a dial indicator reading when gauging misalignment.

TRIBOLOGICAL - of or relating to friction or frictional build-up due to motion and wear.

TRIMMED LIP SEAL - shaft seal with a sealing lip formed by a knife cut rather than by molding.

TRIMMING - removal of excess material and/or shaping of rubber following vulcanization.

TRIMMING

Figure 261: Automated Lip Trimming

U

UNBONDED FLASH - loose rubber that has inadvertently adhered to the seal surface and may impair performance, or flash that does not properly bond to an intended mating material.

UNDERLIP TEMPERATURE - temperature of the oil between a rotating shaft and a shaft seal's lip at the contact point; measured in test situations using an infrared camera.

UNIDIRECTIONAL SEAL - seal designed for use with a shaft that rotates in only one direction, either clockwise or counter-clockwise; also known as a unirotational seal.

UNITIZED SEAL - shaft seal that incorporates a running surface into the seal design; used when the actual shaft surface lacks an acceptable finish.

UNSPRUNG INTERFERENCE - amount of interference between an unsprung lip of a shaft seal and the shaft; calculated as the difference between the shaft diameter and the unsprung lip diameter.

UNSPRUNG LIP - a shaft seal lip without a garter spring.

UNSPRUNG LIP DIAMETER - inside diameter of a shaft seal's primary lip measured without the garter spring (if used) installed.

V

VISCOMETER - shearing disk device used to gauge the viscosity of a rubber sample under heat and pressure. Often referred to as the Mooney Viscometer, this device was once the most common tool for determining processing characteristics but has now largely been replaced by the rheometer.

VISCOSITY - resistance to flow; the thicker the substance (such as a liquid), the more viscous it is, i.e. the less it flows.

VULCANIZATION - heat-induced process whereby the long chains of the rubber molecules become cross-linked by a vulcanizing agent to form three-dimensional elastic structures. This reaction transforms soft, weak, non-crosslinked materials into strong elastic products; also known as cure.

W

WAVINESS - excessive out-of-roundness of a shaft; defined by the RMA as greater than 45 cycles or lobes; also known as grinding chatter.

WEAR SLEEVE - replaceable mild steel sheath drawn over a damaged shaft to provide an improved sealing surface or over a shaft made of a soft material (such as cast iron) to provide a harder, more wear-resistant sealing surface; also known as a wear ring.

WEEPAGE - very small amount of seal leakage; may or may not be enough to necessitate replacement of the seal or redesign of the assembly.

WETTING - application of, or formation of, a continuous liquid film on a surface.

Abbreviations.

ACM - polyacrylate rubber

ACN - acrylonitrile; component in nitrile rubber

AISI - American Iron and Steel Institute

ASTM - American Society for Testing and Materials

DIA - diameter

DRO - dynamic run-out

FDA - Food and Drug Administration

FEPM - tetrafluoroethylene-propylene rubber

FKM - fluoroelastomer

FPM - feet per minute; used as a measure of surface speed at the contact point between a seal and the shaft

HNBR - hydrogenated nitrile rubber

HSN - highly saturated nitrile; alternative name for HNBR

ID - inside diameter

IN. - inch

IRHD - International Rubber Hardness degrees

IRM - Industry Reference Material, as in IRM 903 oil

ISO - International Organization for Standardization

KN/M - kilonewtons per meter; SI equivalent of pli; sometimes a unit of measure in ASTM D 2000 line call-outs

LOP - lip opening pressure

MAX - maximum

MDR - moving die rheometer

MIN - minimum

MPA - megapascal; SI equivalent of psi; sometimes a unit of measure in ASTM D 2000 line call-outs

NBR - nitrile butadiene rubber (Buna N); copolymer of acrylonitrile and butadiene

"This list includes the abbreviations you will encounter most often in the sealing industry. Familiarity with them will save time as you select or design shaft seals."

NBS - National Bureau of Standards

NSF - National Sanitation Foundation

OD - outside diameter

ODR - oscillating disk rheometer

OOR - out-of-roundness

OSHA - Occupational Safety and Health Administration

PLI - pounds per linear inch

PSI - pounds per square inch

PTFE - polytetrafluoroethylene; a fluoroplastic

RA - roughness arithmetic average in surface measurements

RMA - Rubber Manufacturers Association

RPA - rubber process analyzer

RPM - revolutions per minute; also the average peak to mean height in surface measurements

RWD - radial wall dimension

RWV - radial wall variation

RZ - average peak to valley height in surface measurements

SAE - Society of Automotive Engineers

SG - specific gravity

SI - denotes The International System of Units (the modern metric system); taken from the French, “Le Système International d’Unités”

SPEC - specification

STBM - shaft-to-bore misalignment; phenomenon in which the shaft and the bore do not share a common center

TIR - Total Indicator Reading

UV - ultraviolet light

VMQ - vinyl methyl silicone rubber

reference

Temperature Scales.

° CENTIGRADE (CELSIUS)	° FAHRENHEIT	
100	212	(Boiling Point of Water)
95	203	
90	194	
85	185	
80	176	
75	167	
70	158	
65	149	
60	140	
55	131	
50	122	
45	113	
40	104	
35	95	
30	86	
25	77	
20	68	
15	59	
10	50	
5	41	
0	32	(Freezing Point of Water)
-10	14	
-20	-4	
-30	-22	
-40	-40	
-50	-58	
-100	-148	
-150	-238	
-200	-328	
-250	-418	
-273.1	-459.6	(Absolute Zero*)

**Absolute zero is, in theory, the lowest possible temperature. It is considered to be the point at which all molecular motion stops.*

"To convert Centigrade to Fahrenheit, multiply by 9/5 (1.8), then add 32."

"To convert Fahrenheit to Centigrade, subtract 32, then multiply by 5/9 (0.555)."

For temperature and other unit conversions, be sure to make use of our handy unit converter at www.rlhudson.com.

reference

English to Metric.

	To Convert From	To	Multiply By
Area	sq. in. (in^2)	sq. mm (mm^2)	645.16
	sq. in. (in^2)	sq. cm (cm^2)	6.4516
	sq. ft. (ft^2)	sq. meters (m^2)	0.0929
Density	pounds/cubic ft (lb/ft^3)	kilograms/cubic meter (kg/m^3)	16.02
Energy	British thermal units (Btu) (1 J = Ws = 0.2388 cal)	joules (J)	1055
Force	pounds - force (lbf) (1 N = 0.102 kgf)	newtons (N)	4.448
Length	inches (in)	millimeters (mm)	25.4
	feet (ft)	meters (m)	0.3048
	miles (mi)	kilometers (km)	1.609
Mass (Weight)	ounces (oz)	grams (g)	28.35
	pounds-mass (lb)	kilograms (kg)	0.4536
	short tons (2000 lb) (tn)	metric tons (1000 kg) (t)	0.9072
Power	horsepower (550 ft. lb/s) (hp)	kilowatts (kW)	0.7457
Pressure	pounds/square inch (psi)	kilograms (f)/square cm (kg (f)/cm^2)	0.0703
	pounds/square inch (psi)	kilopascals (kPa)	6.8948
	pounds/square inch (psi)	bars (100 kPa)	0.06895
Stress	pounds/square inch (psi) (1 N/mm^2 = 1 MPa)	megapascals (MPa)	0.006895
Torque or Bending Moment	pounds-force-foot (lb-ft)	Newtons-meter (Nm)	1.3567
	pounds-force-inch (lb-in)	Newtons-meter (Nm)	0.113
Velocity	feet/second (ft/s)	meters/second (m/s)	0.3048
Viscosity	dynamic (centipoise)	pascal-second (Pas)	0.001
	kenematic-foot^2/sec (ft^2/s)	meter^2/sec (m^2/s)	0.0929
Volume	cubic inch (in^3)	cubic centimeter (cm^3) (milliliter)	16.3871
	quarts (qt)	liters (1000 cm^3)	0.9464
	gallons (gal)	liters	3.7854

Table 75: English to Metric Conversion Factors

reference

Metric to English.

	To Convert From	To	Multiply By
Area	square millimeters (mm^2)	square inches (in^2)	0.00155
Density	kilograms/cubic meter (kg/m^3)	pounds/cubic ft (lb/ft^3)	0.0624
Energy	joules (J) (1 J = Ws = 0.2388 cal)	British thermal units (Btu)	0.000947
Force	newtons (N) (1 N = 0.102 kgf)	pounds - force (lbf)	0.2248
Length	millimeters (mm)	inches (in)	0.03937
	meters (m)	feet (ft)	3.281
	kilometers (km)	miles (mi)	0.621
Mass (Weight)	grams (g)	ounces (oz)	0.035
	kilograms (kg)	pounds-mass (lb)	2.205
	metric tons (1000 kg) (t)	short tons (2000 lb) (tn)	1.102
Power	kilowatts (kW)	horsepower (550 ft. lb/s) (hp)	1.341
Pressure	kilograms (f)/square cm (kg (f)/cm^2)	pounds/square inch (psi)	14.22
	kilopascals (kPa)	pounds/square inch (psi)	0.145
	bars (100 kPa)	pounds/square inch (psi)	14.503
Stress	megapascals (MPa) (1 N/mm^2 = 1 MPa)	pounds/square inch (psi)	145.039
Torque or Bending Moment	Newtons-meter (Nm)	pounds-force-foot (lb-ft)	0.737
	Newtons-meter (Nm)	pounds-force-inch (lb-in)	8.85
Velocity	meters/second (m/s)	feet/second (ft/s)	3.2808
Viscosity	pascal-second (Pas)	dynamic (centipoise)	1000
	$meter^2$/sec (m^2/s)	$foot^2$/sec (ft^2/s)	10.7643
Volume	cubic centimeter (cm^3) (milliliter)	cubic inch (in^3)	0.061
	liters (1000 cm^3)	quarts (qt)	1.057
	liters	gallons (gal)	0.2642

Table 76: Metric to English Conversion Factors

reference

Index.

"No book of this scope would be complete without a comprehensive index to help you quickly locate key terms and concepts."

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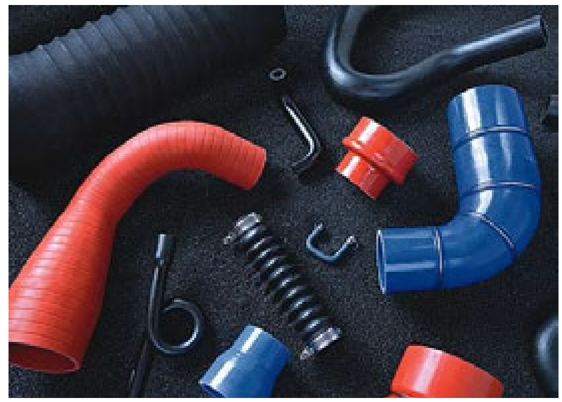
Product Range.

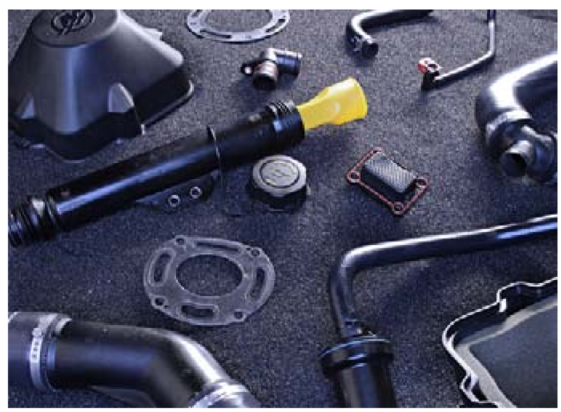
More than simply a supplier of shaft seals, RL Hudson is your source for a wide variety of top-quality sealing devices. Plus, we also specialize in designing custom-molded rubber, plastic, and polyurethane products. Our engineering team is eager to assist you with the development of a new part or the improvement of an existing design. Here's just a sampling of the many ways in which we can supply you with ... *solutions.*

Molded Rubber



Molded, Wrapped and Formed Hoses

Molded Plastic



Assemblies

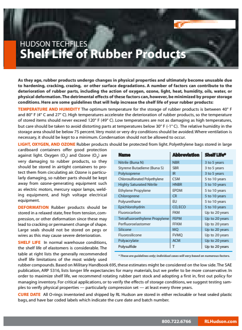
Sealing Products

Other Publications.

For over three decades, RL Hudson has maintained a commitment to both furthering our own knowledge of sealing solutions, and to sharing that knowledge with our customers. In addition to this *Shaft Seal Design & Materials Guide*, we also offer a wide array of other informational publications, all of which are available by contacting your account or territory manager.

O-RING DESIGN & MATERIALS GUIDE

Our 328-page guide explains how to design an effective O-ring seal for a variety of applications. This full-color guide includes detailed profiles of the most-used O-ring materials, as well as extensive chemical compatibility listings and dimensional tables.



HUDSON TECHFILES

Each of our informational bulletins presents a clear, concise look at a technical issue of interest to sealing professionals. Topics range from NSF-61 compounds to custom-molded rubber parts to quality rubber hoses.

SOLUTIONS

Our full-color newsmagazine provides quarterly company updates, employee profiles, technical articles, and humor. Ask your account or territory manager for a free subscription, or submit your name and address at www.rlhudson.com.

About This Book.

A NOTE FROM THE PUBLISHER

Having previously published the *O-Ring Design & Materials Guide* in early, I decided the next book in our technical series should be on the subject of Shaft Seal Design. As with our O-ring book, this book was a team effort from the start, and there are many people who contributed to the success of this project. I would like to take this opportunity to say thank you.

First and foremost, my special thanks go to Dr. Les Horve. As the industry's foremost authority on shaft seal design and performance, Les' technical contributions to the rubber and sealing industries are innumerable and immeasurable. His professional expertise and personal friendship served as the foundation for this book.

I am also grateful to the members of my staff who spent thousands of hours researching, writing, illustrating, and designing this book. In particular I want to acknowledge the work of Jim Morgan and Chris Owen. Without their dedication and determination this book would not be what it is.

I would also like to gratefully acknowledge the kind assistance of the following companies and individuals:

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Rick Hudson
CEO
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R.L. HUDSON & COMPANY *SHAFT SEAL DESIGN & MATERIALS GUIDE*

Publisher	*Rick Hudson*
Editor	*Jim Morgan*
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Illustrators	*Tom Short, Benny Alford*
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