Combined
Wet and Dry Gas Seal
Training Course

Course Notes

bringing technology to life

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Mechanical Seals

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Mechanical Seals

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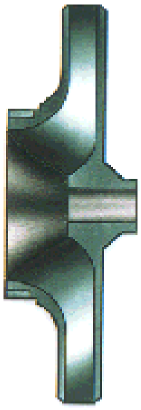
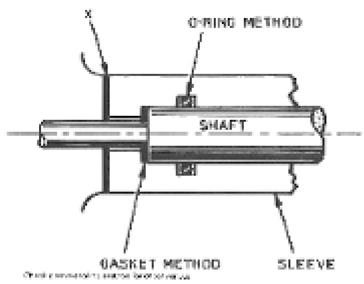
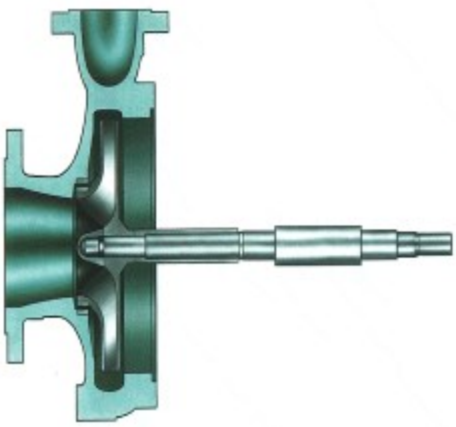
Pump Build-Up



The impeller is the heart of the pump. It provides the motion to move the liquid from one point to another. The impeller is rotated by a power source such as a turbine, engine or electric motor. This rotation causes the fluid surrounding the impeller to be rotated by the radial vanes and it is thrown outwards by centrifugal force from the centre of rotation (eye) to the outer extremity (tips of vanes). The pumping action is achieved by the rotating action of the impeller creating fluid momentum. The fluid motion through the pump causes a reduction in pressure at the inlet producing a suction effect.



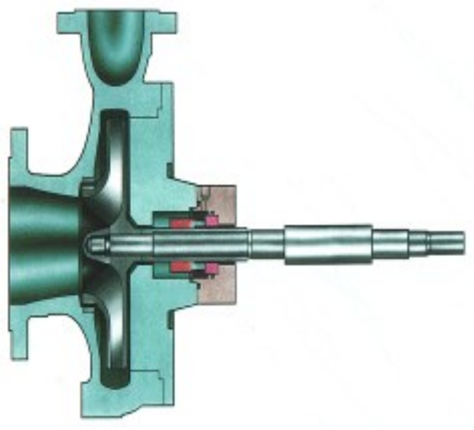
The impeller is attached to the shaft by means of a key or thread. It will often be locked to the shaft by a hub nut or screw. The diagram opposite also shows how to trap the sleeve between the shaft step and rear face of the impeller and the position of the sleeve gasket or o-ring. If the gasket used to seal the sleeve to the shaft is at position 'X' , ensure that there is a seal under the impeller nut or that a 'blind' impeller is used. The shaft is used to support the impeller and also to transmit power. Its size is determined by the need to be strong enough to transmit the full drive power and to ensure that any radial deflection is kept to a minimum. Sleeves are beneficial, as any damage to the shaft - be it physical or corrosive - is very expensive to replace or make good. Also it is at times necessary to have various finishes and hardness under the secondary seal which could be impractical on the shaft.

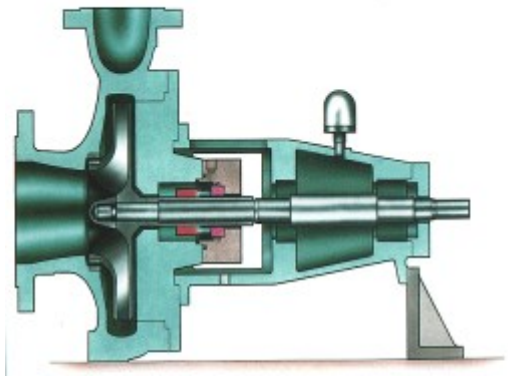


The casing or volute directs the liquid into the eye of the impeller and collects the spinning liquid from the periphery of the impeller, directing it into the discharge port where it is converted from velocity (or kinetic) energy to pressure energy (or head) as it enters the larger volume of this port and slows down.

The casing is usually very robust as it doubles as a pressure vessel and the main pump support by incorporating feet within the casing. The suction and discharge ports must also be of sufficient strength to withstand any loads from the pipe work. Wear and corrosion allowance may also have to be considered.

The back plate is bolted to the pump casing and together with the sealing device prevents leakage from the volute casing to the atmosphere. The main advantage of the back plate is that the pump may be removed for maintenance without disturbing the pump casing. which is fixed to the system pipe work. Note that the hole in the back of the pump casing is larger than the outside diameter of the impeller. This design of pump is known as a back pull-out pump. Flush connections may be included in the back plate or cover plate dependant on the specific application need. e.g., recirculation from the discharge port to the seal chamber, or for seal quenching purposes.





The bearing casing bolts onto the back plate to complete the external shell of the pump, this casing incorporates the second support for the pump. If the bearings are oil lubricated. this housing will contain an oil reservoir. There will be three connections to the reservoir:

1. **An oil filler** at the top of the casing. This incorporates a breather to allow for expansion of the oil and air in the casing.

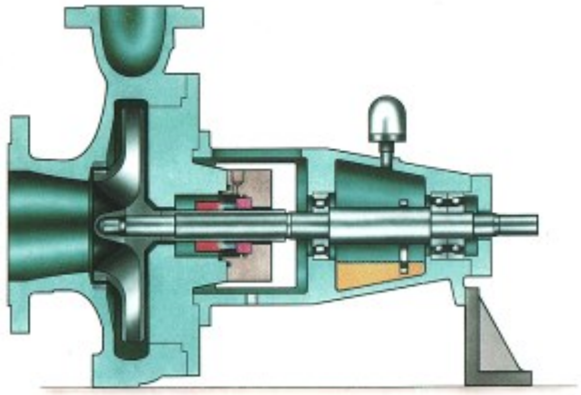
2. **An oil level** plug, which must be removed when the reservoir is being filled or topped up in order to observe when the level is correct. This plug is often replaced by a 'constant level oiler', an automatic device for maintaining the correct oil level in the bearing housing.

3. A **drain plug** situated at the bottom of the casing for the removal of the old oil.

The bearing casing should never come in contact with the pumped liquid and can usually be made from any material. The properties required are rigidity and physical strength, Hence. cast iron or steel are often used except where the environment contains contaminants. such as experienced in Marine environments. which would corrode the chosen material.

The bearings support the shaft within the bearing casing and these are the only support points far the shaft.

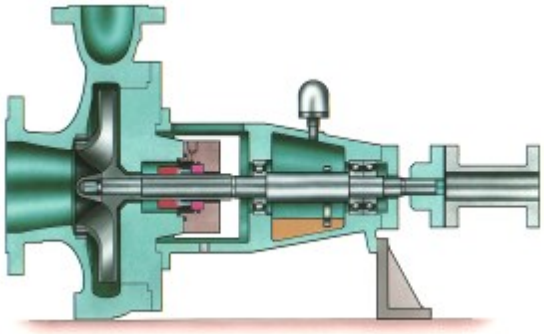
There are two types of bearing required:



1. Thrust bearing to prevent axial and radial shaft movement.
2. Radial bearing to support the shaft but allow axial movement of the shaft due to thermal expansion/ contraction.

The thrust bearing is usually formed by two angular contact ball or roller bearings. In heavy duty applications spherical roller bearings are used.

The radial bearing is usually a single roller or ball bearing. If oil is the lubricant the optimum level is at the centre line of the lowest rolling element in the bearing. In older design pumps. the oil level may be lower and picked up from the reservoir by a thrower ring.
If grease is the lubricant where a cheaper system is desired. bearings will run hotter than if oil lubricated as grease does not dissipate heat as readily as oil

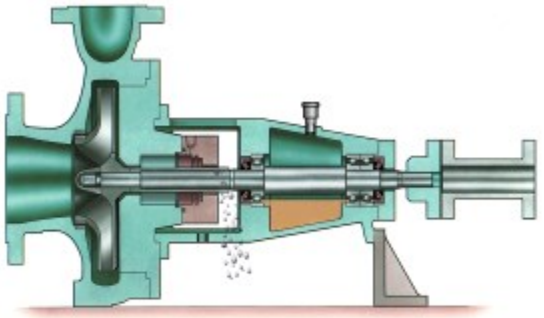


The coupling connects the pump shaft to the driver and has to perform several functions. It must be strong enough to transmit the full driver power; flexible enough to absorb misalignment between the pump and the driver, and may also act as a spacer for easy pump removal.

The main coupling hub is normally keyed to the shaft. A means of flexibly mounting the two halves of the coupling is then used.

There are various points to be noted about the complete pump unit,

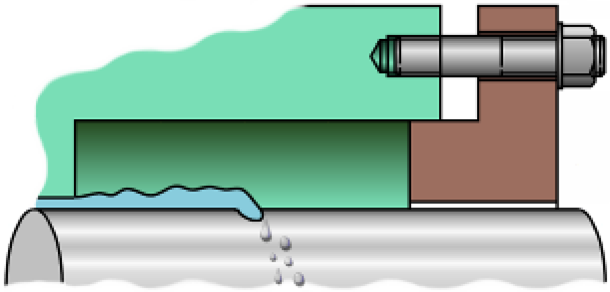
1. Replaceable Wear Rings are often fitted where close running clearances exist between the impeller and the pump casing. All close running clearances reduce liquid transference within the pump unit but damage can occur due to potential contact between the impeller and the casing and also because of the high velocity of the liquid. It is for these reasons that replaceable wear rings are fitted.
2. Balance holes or back vanes are additions to the impeller design that reduces axial thrust on the bearing system.
3. An alternative design in the pump casing of the addition of a double volute outlet reduces the radial thrust generated.
4. Any pumps with seals fitted which are stored for long periods need to be rotated by hand on a weekly basis to prevent shaft deformation and sticking seal faces. Bearings also suffer from deformation if static for long periods (3 to 4 months). The condition is known as 'Brinelling', The rotation should also lubricate the bearings.



In a pump, the shaft that drives the impeller has to pass through a hole in the pump casing. The pumped fluid will leak out, so we must find an effective method of sealing this hole, whilst still allowing the shaft to rotate.

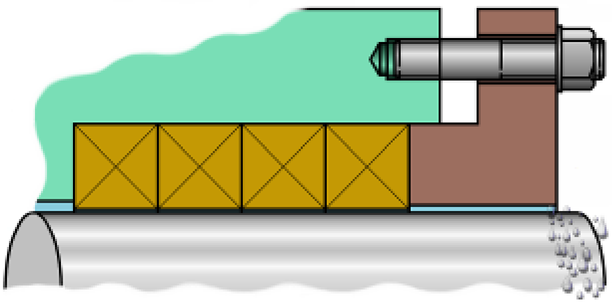
Mechanical Seal Principles

Gland Packing



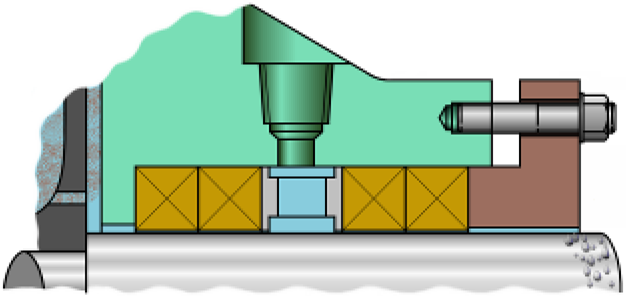
The area of the pump where the sealing device is to be installed is called the Seal Chamber, or Stuffing Box. This latter term was applied when Gland Packing was used. Gland packing was "stuffed" into the "box" to prevent leakage.

There are many excellent gland packing materials available, usually supplied in spools, or as spirals. Just cut off the amount required for each individual ring. Some gland packing materials can be supplied as die-formed rings made to specified dimensions. **Installation Instructions for Gland Packing are included in this manual at Appendix A.**



This stuffing box has four rings of gland packing installed. The packing is compressed by the Gland Follower, which is adjusted by two Gland Studs and Nuts. It is very important to ensure that these nuts are not too tight. The packing must leak a small amount in order to ensure that the packing and shaft/sleeve are cooled and lubricated. To ensure that the packing is lubricated, a small amount of leakage must be visible at all times when the pump is operating.

If the process fluid contains abrasives, these will wear the sleeve rather than provide lubrication. In this case, a Lantern Ring is usually installed in the stuffing box, dividing the packing rings into two groups. Clean fluid is injected at about 1 bar above the stuffing box pressure into the lantern ring. The flow back into the pump will keep the abrasives away from the packing bore. The lubricating leakage to atmosphere will also be the injected fluid, so this method was also used for sealing corrosive or toxic fluids.



Lantern rings are usually made from non-sparking materials such as brass or bronze. In corrosive fluids, PTFE may be used. Note the two small holes to aid removal of the ring from the stuffing box. In metal lantern rings these may be tapped holes. The number of rings either side of the lantern ring will depend on the design of the pump and the pump operating conditions.

There are many disadvantages to using gland packing as the sealing device in rotating equipment, and some perceived advantages, for example:

Disadvantages:

- High leakage
- Shaft/sleeve wear
- Not self-adjusting
- High power consumption
- Open to tampering
- Run-in time required

Advantages:

- Cheap
- Can be fitted without dismantling
- Simple to fit compared to seals

High Leakage. With high leakage, the actors to consider are: health and safety of plant personnel; housekeeping/cleanliness; corrosion of pump and base plate; environmental contamination; cost of lost fluid.

This chart will give some idea of the annual leakage from a packed gland:

One Drop Per Second		
1 Minute Loss	=	0.004 litre
1 Hour Loss	=	0.35 litre
1 Day Loss	=	8.3 litres
1 Week Loss	=	58.0 litres
1 Month Loss	=	255.0 litres
1 Year Loss	=	3,070.0 litres

ø1.6 mm Stream		
1 Minute Loss	=	0.26 litre
1 Hour Loss	=	15.6 litres
1 Day Loss	=	374.0 litres
1 Week Loss	=	2,616.0 litres
1 Month Loss	=	11,337.0 litres
1 Year Loss	=	136,045.0 litres

Two Drops Per Second		
1 Minute Loss	=	0.01 litre
1 Hour Loss	=	0.71 litre
1 Day Loss	=	17.0 litres
1 Week Loss	=	118.0 litres
1 Month Loss	=	512.0 litres
1 Year Loss	=	6,144.0 litres

ø3.2 mm Stream		
1 Minute Loss	=	0.81 litre
1 Hour Loss	=	48.75 litres
1 Day Loss	=	1,170.0 litres
1 Week Loss	=	8,190.0 litres
1 Month Loss	=	35,490.0 litres
1 Year Loss	=	425,880.0 litres

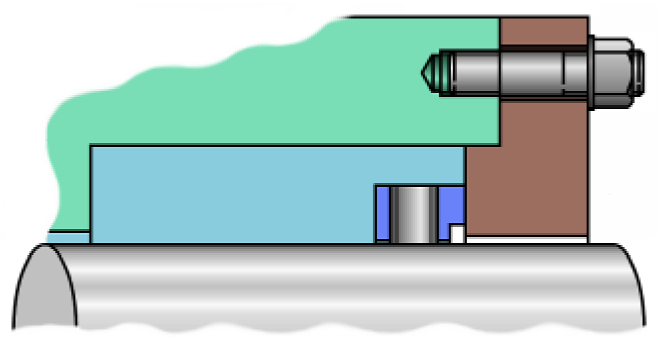
Drops Breaking to Stream		
1 Minute Loss	=	0.08 litre
1 Hour Loss	=	4.75 litres
1 Day Loss	=	114.0 litres
1 Week Loss	=	796.0 litres
1 Month Loss	=	3,450.0 litres
1 Year Loss	=	41,400.0 litres

ø4.8 mm Stream		
1 Minute Loss	=	1.35 litres
1 Hour Loss	=	81.25 litres
1 Day Loss	=	1,950.0 litres
1 Week Loss	=	13,650.0 litres
1 Month Loss	=	59,150.0 litres
1 Year Loss	=	709,800.0 litres

ø6.4 mm Stream		
1 Minute Loss	=	2.93 litres
1 Hour Loss	=	176.0 litres
1 Day Loss	=	4,225.0 litres
1 Week Loss	=	29,575.0 litres
1 Month Loss	=	128,158.0 litres
1 Year Loss	=	1,537,900.0 litres

Shaft/sleeve wear; not self-adjusting; open to tampering. On the worn, shiny portion of this sleeve, the left hand groove was caused by abrasives embedding into the bore of the first ring of packing and acting like a grinding medium. The middle groove was caused by a build-up of abrasives in the bore of the lantern ring as the clean flush had been disconnected. The third and outermost very deep groove was caused by the gland nuts being over-tightened by an inexperienced person, to such an extent that no lubrication or cooling could get to this area, and excessive wear has resulted.

First, we look at the build-up of a very simple mechanical seal to illustrate the function of each of the main components.

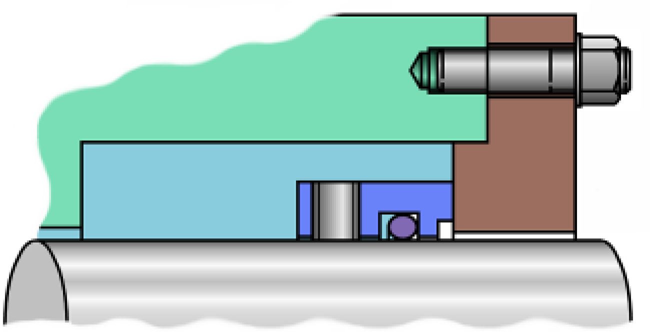


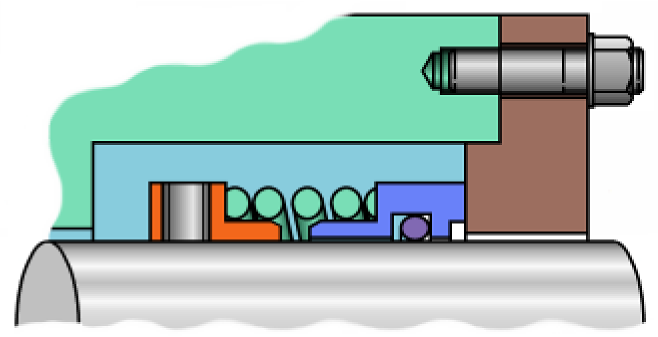
Here, the gland packing has been removed from the stuffing box, the gland follower machined and polished perfectly flat, and a loose metal ring has been slid along the shaft until it is in contact with the gland follower. It has then been locked to the shaft with three grubscrews. Wear now takes place at 90° to the shaft axis between two replaceable parts, and wear on the shaft/sleeve (a common problem with gland packing) has been eliminated.

Even before rotation, it can be seen that this will not form an effective seal, as fluid would leak through the bore of the blue ring.

We can prevent leakage through the bore of the blue ring by inserting a simple sealing device such as an o-ring.

When the shaft is rotated, wear between the two running faces will create a gap, allowing leakage. Also, there is always some clearance in bearings, and axial movement will cause further leakage. Axial movement will also occur due to thermal expansion or contraction of the shaft.

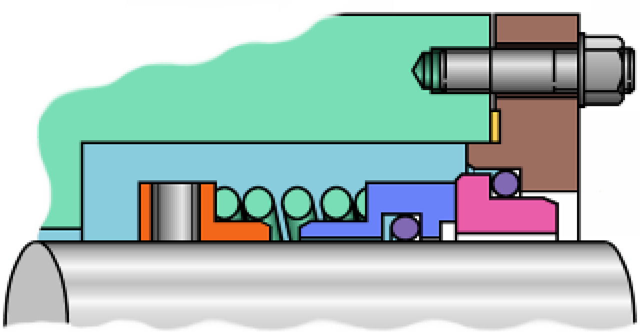




By fitting some form of spring force device, in this case a large single coil spring, any wear or axial movement of the shaft will be taken up by the spring force maintaining contact between the two running faces.

This still leaves us with three problems:

1. Heat generated by the friction between the two running faces must be dissipated.
2. There is a leak path between the stuffing box (seal chamber) face and the gland follower.
3. The gland follower is a large component that must be replaced when worn. It will also be very expensive if made from materials such as tungsten carbide or silicon carbide. It is essential that seal components that have been lapped perfectly flat are not subjected to any bending stresses. In the diagram, the large brown component has been drilled, and has then to be bolted to the face of the seal chamber. This will cause unacceptable distortion of the lapped face, creating a leak path.



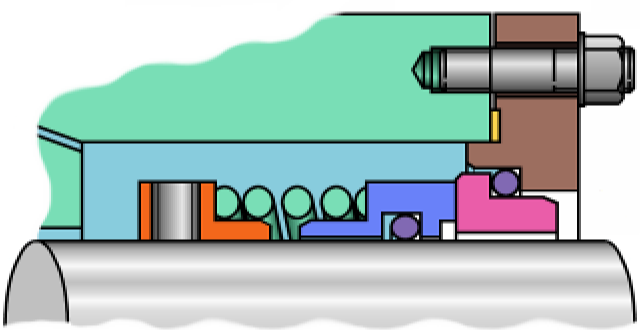
There should always be a gasket or o-ring between the gland plate and seal chamber face to prevent leakage. When fitting cartridge seals, always ensure that there is a sealing device in this position – they can sometimes be left in the box!

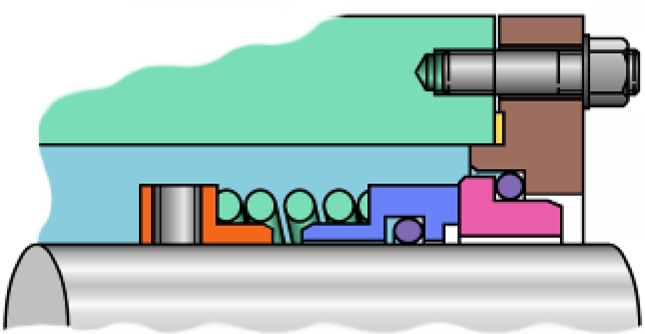
The inserted stationary ring (pink) becomes a less expensive item to replace, and is not subject to any bolting stresses. At lower pressures, this design of stationary ring is ideal as the elastomer o-ring will cushion any slight distortion that may occur in the gland plate.

There is still the problem of how to dissipate the heat generated by the friction at the running faces.

In some very low duty applications cooling may not be required, and a "dead-end box" is used – known as API Piping Plan 2 (as above). However, any air or vapour will be trapped in the seal chamber. When the shaft rotates, the fluid in the seal chamber will also rotate. Because the fluid is heavier than the gas/vapour the fluid will centrifuge to the outside of the seal chamber, and the gas/vapour will collapse to the centre causing the seal to "dry run".

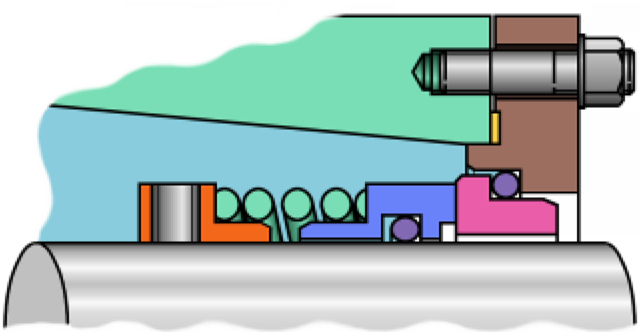
A small drilling at top-dead-centre at the inboard end of the seal chamber will allow any gas/vapour to vent automatically.

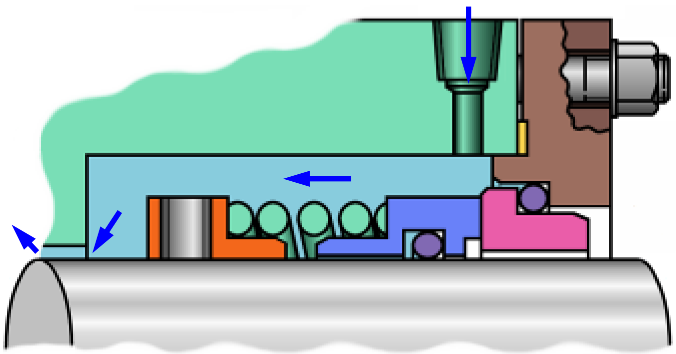




With API Piping Plan 2, it is better to have a straight through seal chamber on dead-end applications to ensure good circulation of fluid to remove any heat.

Better still, an angled or conical seal chamber will increase the flow in and out of the seal chamber, and ensure that no gas/vapour gets trapped in the seal area.





In most horizontal pumps fitted with single seals, heat is dissipated by circulating product fluid from a higher pressure area of the pump into the seal chamber, and back into the pump, maintaining the seal area at about pumping temperature. This is API Piping Plan 11. Later training modules cover how to cool this area where the sealed product fluid temperature is too high to create sufficient cooling effect.

Care must be taken to ensure that this inlet pipe is as close as possible to the running faces, and that the pressure differential is not too high. Often it will be necessary to insert an orifice into the recirculation line. It is important that this orifice is inserted as far away as possible from the port into the seal chamber, otherwise erosion will occur.

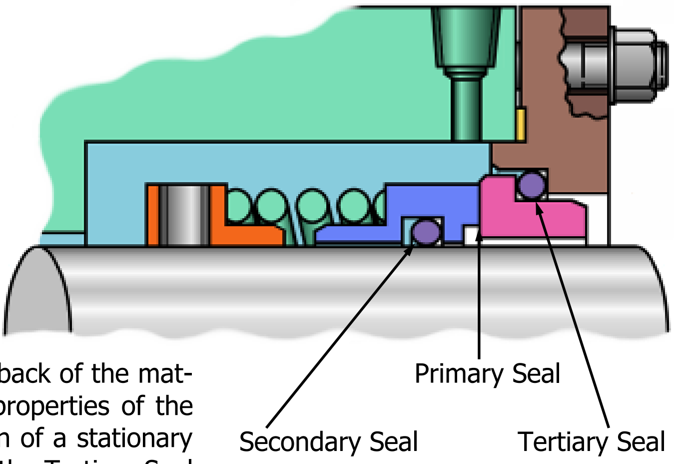
The photograph on the left shows erosion due to a very high pressure flush through the recirculation line from discharge. It is also clear from the grooving in the outside diameter that the recirculation line was not in the correct position over the primary seal interface where there was possibly a dead pocket/high heat area. Note the erosion where the clearance between the primary ring and its retainer is small. There is less erosion where the side holes are machined in the retainer.

The photograph on the right shows similar damage and incorrect positioning of the recirculation line. In this second example, the high pressure recirculation was also laden with solids and abrasives.

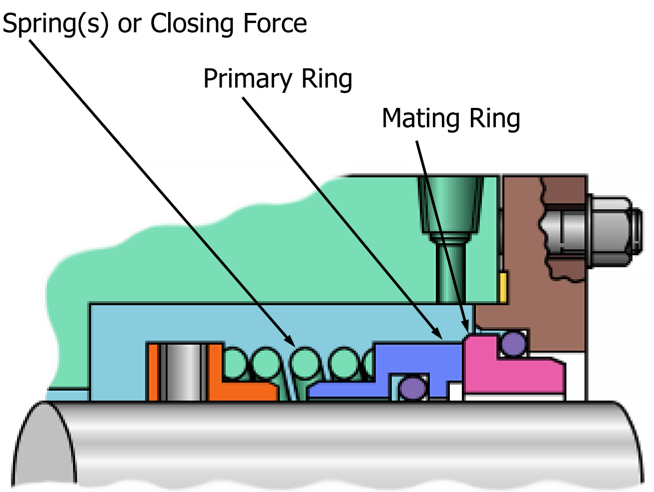
Solution: i) ensure correct positioning of the recirculation line
ii) use orifice plate at pump discharge end of recirculation line
iii) on multistage pumps, take recirculation from first or second stage, not from final stage
iv) use tangential connections, multi-point or distributed flush
v) check for abrasives and take suitable remedial action.

Terminology

In sealing terminology, there are three sealing points in a mechanical seal:



1. **Primary Seal** – the contact area between the Primary Ring and Mating Ring (fluid film).
2. **Secondary Seal** – to prevent leakage passed the Primary Ring, and to allow axial movement.
3. **Tertiary Seal** – where the Mating Ring fits into the housing of the equipment, or the gland plate, and prevents leakage round the back of the mating ring. In some seal designs, the friction properties of the Tertiary Seal are used to prevent any rotation of a stationary Mating Ring. Where the friction properties of the Tertiary Seal are not suitable or sufficient, an anti-rotation pin must be used.



Terminology (Continued)

The main components that go to make up any mechanical seal are:

1. Primary Ring complete with its Secondary Seal.
2. Mating Ring complete with its Tertiary Seal.
3. Some form of spring or closing force:
    - Large single coil spring
    - Small multiple coil springs
    - Wave springs
    - Metal bellows
    - Other spring devices

Primary Ring. This component is usually the softer of the two running faces (most commonly carbon-graphite), and its lapped running surface is narrower in section than the mating ring. This component is the one most likely to wear, and has a wear allowance built in.

Mating Ring. This component is the harder of the two running faces (e.g., silicon carbide), and its lapped running surface is wider than the Primary Ring (in some seal designs it is the same width as the Primary Ring).

Depending on the seal design, either the Primary Ring or the Mating Ring will rotate with the shaft when the machine is operating.

Spring Drive

In some seal designs, the torque is transmitted through the coil spring. These designs are usually uni-rotational, and you must specify a left-handed or right-handed spring, depending on the direction of rotation of the machine to which the seal is to be fitted.

Looking at the driven end of the spring (the end furthest away from the lapped face), the tail of the spring material points in the direction in which the shaft should rotate. If you grasp this spring in a hand so that your index finger is on the tail of the spring and points in the same direction, then the hand in which you are holding the spring is the name of the spring:

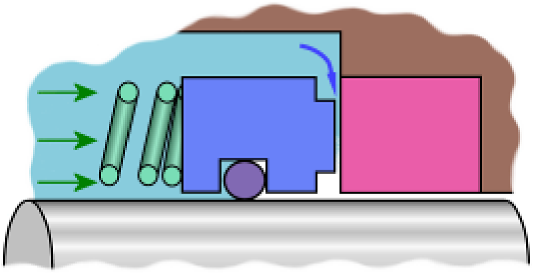
e.g., looking at the driven end of the seal/spring, a left-handed spring will only work on a shaft rotating clockwise, and a right-handed spring will only work on a shaft which is rotating counter-clockwise.

Note: There are two other sealing points associated with mechanical seals:

1. A gasket or o-ring between the gland plate and face of the seal chamber.
2. If the shaft of the machine is fitted with a sleeve, there must be some form of sealing device in the bore of the sleeve to prevent leakage.

Mechanical Seal Principles

The Primary Seal



In most seals, the higher pressure is on the outside of the contact area between the Primary Ring and Mating Ring (Primary Seal). This is to ensure that the hydraulic pressure of the liquid being sealed plus the spring load tends to close the faces against the sealed liquid pressure, which tries to force the faces apart. If the pressure increases, the closing force increases, and vice versa, so the seal is self-adjusting.

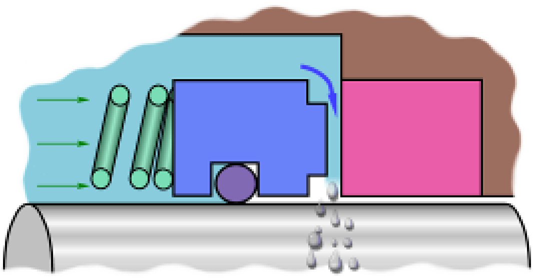
If the higher pressure were on the inside diameter, the spring would be the only force holding the faces together. If the fluid pressure increased beyond the force of the springs, the faces would open causing high leakage. If the fluid pressure dropped well below the force of the springs excessive wear would occur.

If the seal faces do not leak, then the pressure at the inside diameter of the Primary Ring is zero, and there is a pressure gradient across them. This fluid between the faces lubricates them, forms the Primary Seal, and in conventional “wet” seals must be present at all times.

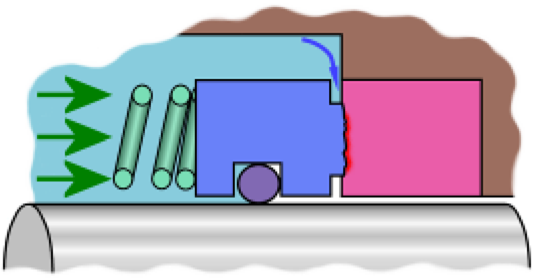
Rule 1

For most seal designs, the maximum pressure should be at the outside diameter of the seal, and the minimum pressure, normally atmospheric pressure, at the inside diameter.

This should be remembered as it will help when looking at multiple seals, cartridge designs and dry gas seals in other Training Modules.



If the closing force on the faces is inadequate, the gap becomes too wide and leakage will occur. Many factors determine the thickness of the fluid film including pressure, temperature, viscosity, specific gravity, speed and nature of the sealed liquid.

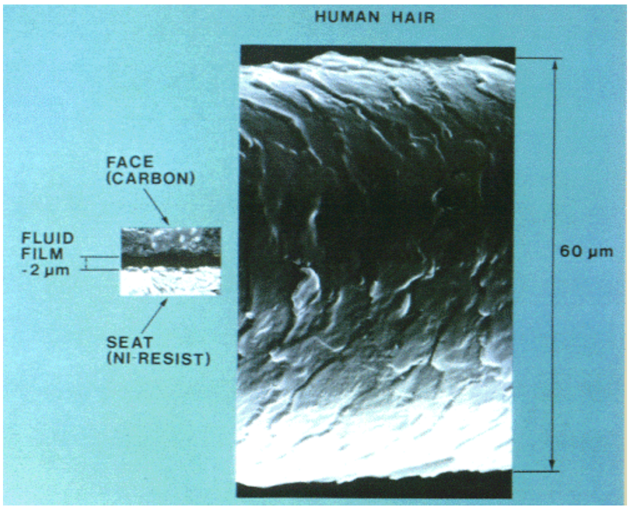


If the closing force is too high, the faces become over-pressurised and the thin fluid film can be squeezed out. The faces then actually touch and excessive wear takes place.

Several elements influence the sealing capability and the amount of heat generated by the faces and there is a strong relationship between the following points:

- Size of the seal.
- Rotational speed (revolutions per minute).
- Temperature of the fluid.
- Nature of the service fluid.
- Surface finish of the seal running faces.
- Materials of construction.
- Pressure acting on the seal faces.
- Surface area of the running track.

Springs assist the hydraulic pressure in keeping faces/seats closed, and are especially important when the machine is switched off or generating a partial vacuum in the seal area.



The lubricating fluid film between the faces is extremely thin. It is physically possible to measure the thickness of the fluid film and it has been found in general to be only about 0.5 to 3 microns thick.

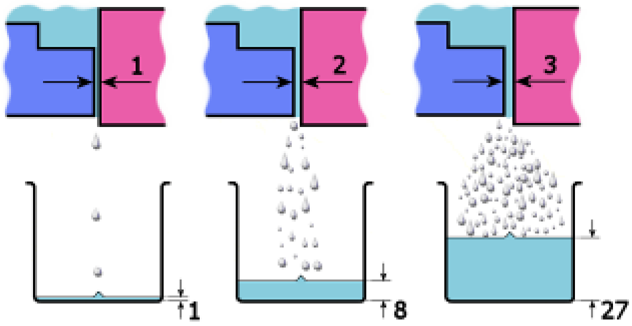
Here we see it compared to a human hair. So, although mechanical seals do leak (if they do not, they are dry running and excessive wear will occur, giving very short seal life), you should not be able to see the leakage as it is so small it becomes vaporised as it leaves the Primary Seal area. By running a test on a closed loop it is possible to measure the actual leakage from seals under varying service conditions.

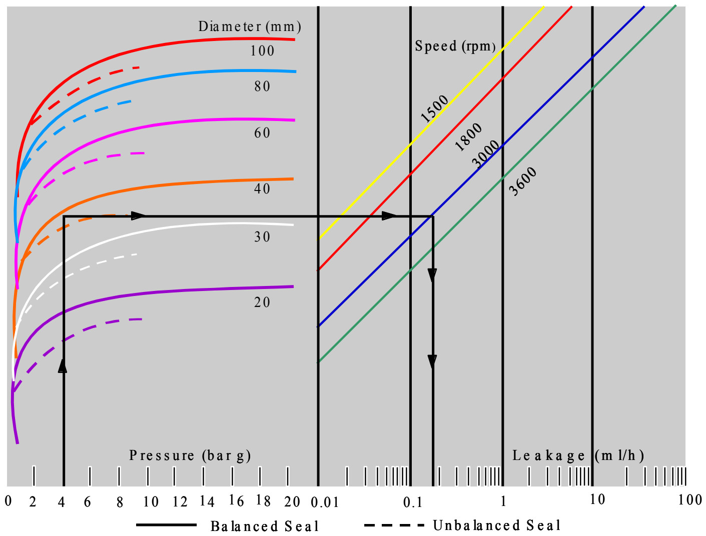
As the fluid film is so thin, it is necessary to ensure that the two faces are as flat as possible to reduce the amount of contact that may occur. The flatness is achieved by the process of Lapping.

For most applications, seal faces are lapped flat to within 1 to 3 light bands.
1 light band = 0.3 micron = 0.0003 mm = 0.00001181 inch.
Human hair = 30 to 130 microns.

Leakage varies approximately as the cube of the fluid film thickness:

1 x 1 x 1 = 1
2 x 2 x 2 = 8
3 x 3 x 3 = 27
4 x 4 x 4 = 64
5 x 5 x 5 = 125
Etc., etc.





This chart shows approximate expected leakage rates from mechanical seals which are operating correctly in a Newtonian fluid.

This is **not** the amount of fluid you will collect in a cup placed under the seal. It is the amount of fluid that you could measure as the loss from a closed loop system. The leakage is not visible, and will be in the form of vapour or gas.

It is possible to engineer the environment round a mechanical seal in order to ensure extended life. A clean, pure, non-abrasive fluid needs to be present in the seal chamber, at moderate pressure, viscosity and temperature. Specialist seals can be designed to cope with dirty or abrasive fluids, high pressures and high temperatures, but if it is possible, it is more economic to engineer the environment rather than engineer the seal. Engineering the environment round mechanical seals is covered in this manual under: Balance; Cooling; Secondary Containment; Multiple Seals; Sealant Systems.

The thickness of the fluid film is very important:

- Too thin - excessive contact causing accelerated wear and premature seal failure.
- Too thick - visible and unacceptable leakage.

The fluid film forming the Primary Seal must be:

- Present at all times.
- Stable - must not vaporise or boil off between the faces.
- Clean - non-abrasive.
- Reasonable viscosity.
- Temperature controlled.
- Acceptable pressure.

Abrasives

This photograph shows typical abrasive damage. The seal is from the sea water circulating pump on a submarine. Abrasives (fine sand and silt particles) have become embedded in the relatively soft carbon-graphite primary ring. This has then ground its way into the mating ring, forming a deep groove. Similar wear can be caused by 'dry running' - see overleaf. Seal failure analysis involves a great deal of investigation of the operating conditions at, and before, the time of failure, and the operating procedures of the equipment.

Another failure mode in the presence of abrasives would be 'gramophoning' of the seal faces where the abrasive particles have been dragged round between the faces causing deep scoring of the previously lapped surfaces. The main clue to abrasive wear is the build-up of solids in and around the seal faces.

The solution to this problem is to prevent the abrasive particles from getting between the seal faces by using:

• Cyclone separator	API Piping Plan 31	
• Filtered recirculation	API Piping Plan 12	
• Clean flush	API Piping Plan 32	*See Appendix B*
• Reverse circulation	API Piping Plan 13	*for details of all*
• Pressurised double seal	API Piping Plan 53 or 54	*API Piping Plans*
• Up-Stream Pumping seal		

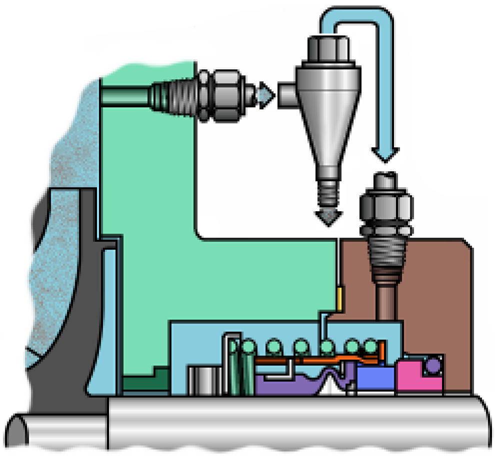
Another way is to cope with the abrasives by using hard face materials, or a mechanical seal designed specifically for abrasive duties. There are seal designs to cope with extremely aggressive abrasive duties such as found in Flue Gas Desulphurisation plants, mineral extraction and processing, and mine de-watering (see overleaf for a typical example).

<u>API Piping Plan 31</u>

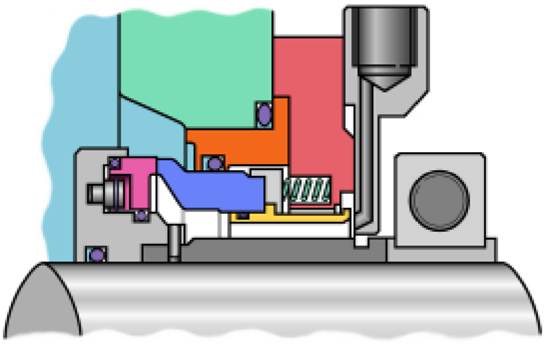
A Cyclone Separator uses centrifugal force to remove abrasive particles from the pumped fluid. They are self-cleaning, so are generally preferred to mesh filters.

Particle density should be at least twice that of the fluid to ensure efficient separation.

The dirty fluid from the bottom of the separator can be returned to the pump suction. It is important that the pressures at both outlets of the separator are the same.



Seal for Aggressive Abrasive Duties



This picture shows a typical aggressive abrasive duty seal.

Design Features:

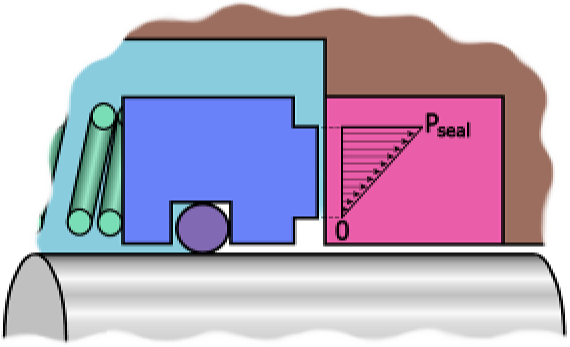
- Wear resistant Silicon Carbide faces.
- Silicon Carbide has controlled porosity for improved lubrication.
- Large open area round faces. No recirculation or clean flush required.
- Product on the outside diameter.
- Springs totally shrouded from, and external to, the product.
- Large open area on inside diameter to prevent clogging from solids that get across the faces.
- Provision for intermittent quenching of the seal bore to remove any solids.
- Cartridge assembly – error free and simplified installation, factory assembled and tested.
- Large split drive clamp. Good torque transmission – no shaft damage.
- Cushioned pin drive to Mating Ring.
- Rotating Mating Ring ensures true running, and prevents solids build-up round faces.
- Inner sleeve/mating ring adaptor easily replaced.
- Wear plate easily replaced (contains secondary o-ring seal).
- Compact design.
- Large axial movement allowance.

Dry Running

This wear pattern could have been caused by the presence of abrasives. A full failure investigation needs to be carried out, including collection of all operating conditions. The seal, with carbon-graphite versus silicon carbide faces, was fitted to a pump operating on a low specific gravity, ultra-filtered hydrocarbon, so abrasive problems were highly unlikely. After a full investigation, it was discovered that the API Plan 11 recirculation pipe had been bent over by someone standing on it. This prevented the product from flowing into the seal chamber to remove the heat generated at the seal faces. The product vaporised round the outside diameter of the running faces, and no fluid could get between the faces to provide lubrication, resulting in dry-running and excessive wear. The carbon-graphite primary ring has worn a deep groove into the silicon carbide mating ring. The theory is that both the carbon-graphite and the silicon carbide start to wear, but the small particles of sharp silicon carbide stick into the softer surface of the carbon. This then creates more wear of the mating ring. Further silicon carbide particles stick into the primary ring face, creating a very effective grinding wheel which forms the deep groove in the mating ring. The excessive heat only occurred at the running faces as there was fluid present in the seal chamber, and the pump suction and discharge telemetry did not indicate any flow problems.

This photograph shows damage on the running surface of a metal mating ring. These radial cracks are only on the running track, and are typical evidence on metallic mating rings (e.g., tungsten carbide, austenitic cast iron) of dry running. The rapid increase in temperature causes the overheated surface to expand rapidly, creating the cracking. These raised cracks then quickly wear away the primary ring if made from carbon-graphite.

Unstable Fluids



As the sealed fluid travels across the Primary Seal area two things happen:

- The pressure drops from seal chamber pressure to atmospheric pressure.
- The temperature of the fluid film rises due to friction and shearing. The actual rise will depend on the various service parameters, but can be around 20°C.

For example: water at 90°C — the fluid film will rise to approximately 110°C as it drops to atmospheric pressure. Under these conditions water turns to steam, expanding rapidly and forcing the seal faces apart. Now there is no friction to generate heat, and the large leakage of water through the faces will cool everything to 90°C. The faces go back together. Heat will be generated, the temperature will rise to 110°C, steam will form, the faces will part, etc., etc. This will continue, creating vibration, spitting and popping of the seal faces, with spurts of steam emanating from the gland plate bore. As the faces will not necessarily part evenly, chipping may be noticed on the outside edge of the Primary Ring running face.

Solution: Fluid in the seal chamber must be at least 20°C below its boiling point at atmospheric pressure. With water, therefore, it should be below 80°C. In many power stations, boiler feed pump seal chambers are cooled to about 65°C in order to achieve a good safety margin.

Viscosity

This chart shows the viscosity limits for various seal types, and the actions to be taken when viscosity is high.

Seal Type	**VISCOSITY (cSt)**								
	Up to 500	**500 to 750**	**750 to 1000**	**1000 to 1500**	**1500 to 2000**	**2000 to 2500**	**2500 to 3000**	**3000 to 3500**	**3500 and Above**
Pusher Seals with Positive Drive	Standard Seal & Mating Ring		Standard Seal & Pinned or Clamped	Refer to John Crane					
Elastomer Bellows Non-Pusher Seals	Standard Seal & Mating Ring		Standard Seal & Pinned or Clamped	Standard Seal with Hard Faces Pinned or Clamped Mating Ring Limited Velocity - See Below					Refer to John Crane
PTFE Bellows Non-Pusher	Standard Seal & Clamped Mating Ring			Refer to John Crane					
Metal Bellows Non-Pusher	Standard Seal & Mating Ring		Standard Seal & Pinned or Clamped	Standard Seal with Hard Faces Pinned or Clamped Mating Ring Limited Velocity - See Below					Refer to John Crane
Recommended Minimum Radial Seal Clearance	Standard	5 mm			10 mm				
Heating at Start-Up	Optional				Recommended				
Maximum Shaft Velocity (m/s)	Seal Design Maximum			10	8	6	4	3	Refer to John Crane

Face pitting.

This can be one of two things:

a) pitting due to the fluid film having too high a viscosity. The shear strength of the fluid can be higher than the shear/tensile strength of carbon-graphite. Carbon particles are plucked out of the surface, and often create scoring on the running face.

Solution: i) heat seal chamber area
ii) use heating in/on mating ring (e.g., hollow mating ring)

b) blistering of the face. Blisters burst and leave pits. High heat generation causes: the impregnant to expand; or product/contaminant that has slowly penetrated the surface to expand quickly.

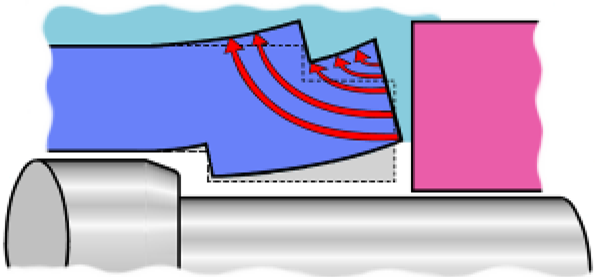
Solution: Ensure no silicone grease is being used during fitting of the seal which could contaminate the lapped surface of the carbon. There are many solutions to this problem, such as: change carbon material to blister resistant grade; reduce face width, reduce spring load, consider increasing balance factor towards 65%; make primary ring and mating ring outside diameters identical to increase cooling flow over the faces; consider multi-point injection or distributed flush; use silicon carbide mating ring to improve heat-soak; if refrigeration compressor application ensure oil is not foaming in seal chamber – ensure good seal chamber venting – ensure no water contamination – check for "coppering" on carbon face.

This seal was operating on a bitumen pump. Bitumen in the seal chamber was heated by a steam jacket to keep it in the fluid state. When the pump was stationary, the interface film cooled and solidified. At start-up, the carbon primary ring sheared off.

Solution: Ensure steam supply or other heating method to pump jacket is maintained when pump is stationary. Use steam heated mating ring also. May also need a steam quench to prevent cold air (such as in the winter) cooling the static fluid film.

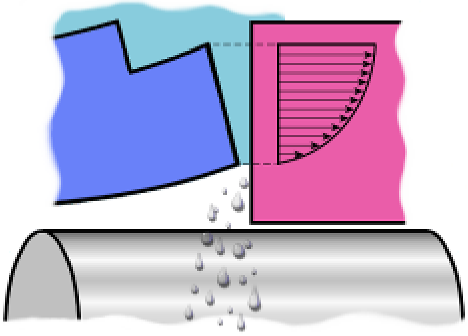
Temperature Control

The fluid film can be affected by the geometry of the Primary Ring of the seal.



Coning Out or Positive Rotation

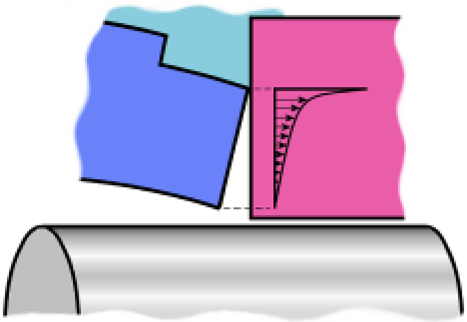
The carbon-graphite material used for the primary ring of most seals is not a very good conductor of heat. Any heat generated by the friction at the seal faces does not soak back through the body of the material. Therefore, thermal expansion only occurs at the front end, causing this "belling" effect. There may be slight, similar distortion on the mating ring, depending on the material used.



More of the fluid pressure can now get between the seal faces, and excessive leakage will be observed when the shaft is rotating. Chipping can form on the inside edge of the primary ring running face. Heavy contact will be noted on the inside diameter of the running track on the mating ring, often fading away to no visible contact on the outside diameter of the contact pattern.

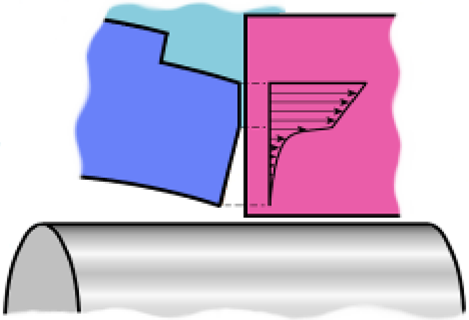
Solution: Improve cooling to the seal; change to materials with better thermal conductivity; check that the seal faces are correctly lapped; check maximum process/ sealant pressure.

Pressure Control

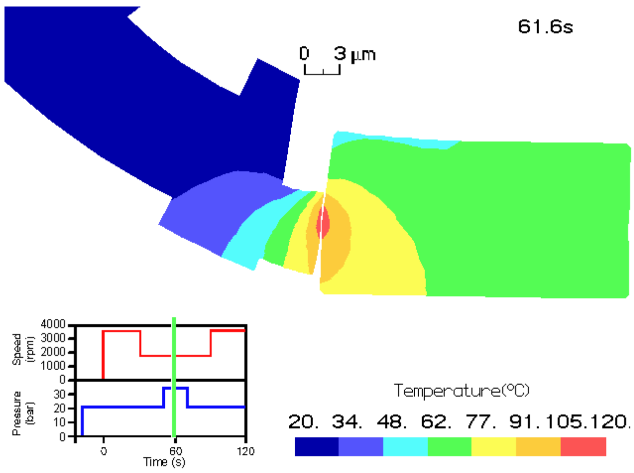


<u>Coning In or Negative Rotation</u>
The Primary Ring may also distort in the opposite direction. It is then termed "Negative Rotation". This is caused by excessive pressure from the process fluid. The Mating Ring may also distort in a similar fashion, so compounding the problem.

Contact occurs between the Primary Ring and Mating Ring, preventing the lubricating fluid film from forming, so excessive wear results, with possible chipping on the outside edge of the running face of the Primary Ring.



Larger scale wear can take place on the Primary Ring if distortion occurs. This has the effect of changing the hydrostatic pressure profile across the faces. Calculating the effect of this can only be accurately done with a computer.



Lack of lubrication due to negative rotation caused by over-pressurisation often means that the frictional heat increases. This in turn creates positive rotation, and thermal cycling can occur. John Crane have a suite of computer programs to evaluate the effects of high pressure/high temperature/speed changes including CStedy/CTrans (as shown to the left). Further details are available from John Crane.

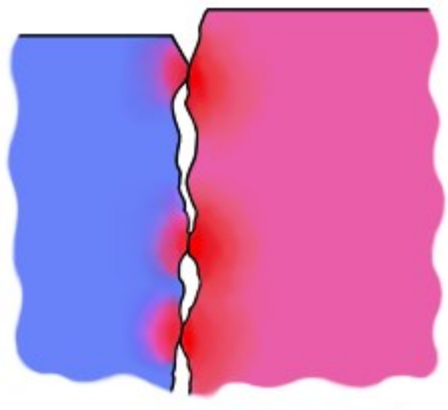
On some high-pressure applications, the fluid film can be increased in thickness to improve cooling and lubrication by the use of Hydropads which are usually machined into the lapped face of the rotating carbon ring. These increase the hydrodynamic forces at the interface film, lifting the two seal running faces slightly further apart, enabling the seal to withstand higher pressures without overheating of the faces.

The disadvantage of this design is that leakage is likely to increase due to the thicker fluid film. The LaserFace™design overcomes this problem. This unique development is discussed later in this manual.

Seal wear can occur when the macroscopic high spots of the Primary Ring and Mating Ring surfaces come into contact with each other.

Under initial start-up conditions, it is normal for a certain amount of "bedding-in" to take place. This wear may result in a better mating surface.

If there is no fluid film, heavy contact will occur creating high levels of heat generation. Excessive wear will take place destroying the surface finishes. This drastically reduces the life of the seal.



Reconditioning of Mechanical Seals

When a mechanical seal has come to the end of its useful life, we recommend that you return it to your local John Crane Customer Support Unit so that it can be assessed to see whether it can be economically refurbished to "as new" condition. Never use seal repairers who do not have the knowledge of the latest seal design updates, materials, manufacturing processes, pressure testing procedures, and who do not have access to genuine manufacturer's spare parts.

Remember: mechanical seals are often sealing dangerous, toxic, inflammable, high temperature, high pressure, environmentally unfriendly products. Premature failure due to incorrect reconditioning practices can be dangerous to staff, plant, production and the environment and be extremely costly.

Handling Mechanical Seals

A mechanical seal is a piece of precision engineering, and should be handled with care.

- Do not unpack the seal until you are ready to install it.
- Avoid touching or handling the lapped faces.
- Use clean tissue paper on the bench to prevent any contamination of the seal.
- Place the seal on the bench with lapped face uppermost.
- Hands must be clean — any minute dirt particles on the lapped surfaces can create wear and leakage.

Seal faces must be installed together clean and dry. Always carefully wipe the lapped surfaces clean with soft tissue and an *approved* solvent just before placing them together in the equipment. Once the faces are perfectly clean, do *not* place any lubricant on them.

Suitable cleaning solvent should:

- Remove grease and dirt.
- Evaporate quickly.
- Not leave any bloom or marking when it has evaporated.
- Not attack any of the seal components.
- Be approved by your Health & Safety Officer.

Test your solvent on a clean piece of glass to check for evaporation time and deposits. If there are any signs of deposits, do not use it on mechanical seals.

Materials for Primary Rings and Mating Rings

Some Common Primary Ring Materials

Carbon-graphite

This is the most common Primary Ring material.

Advantages
- Good self-lubricating properties.
- Promotes a good fluid film.
- Chemically inert – very little will attack this material.
- Wide temperature range.
- Strong in compression.
- Slightly flexible (can also be a disadvantage).
- Easy to machine.
- Relatively low cost.

Disadvantages
- Porous. It must be impregnated with some material to fill the gaps between the carbon and graphite grains. Resin (of various types) is the most common impregnation material in wet seals Antimony is the second most common impregnation material. Antimony impregnated carbon-graphite is the standard material used for dry gas seal Primary Rings.
- The carbon-graphite Primary Ring parameters of temperature and corrosion resistance are determined by the impregnation material. For example: high purity water will attack resin and leach it out, so antimony or some other impregnation material must be used in this application.
- Easily damaged. It must be handled with great care to ensure it is not chipped or scratched, and that no contaminants get onto the lapped surface before/during installation (such as silicone grease).
- Low tolerance to abrasives.
- Low thermal conductivity due to the impregnation process.
- Weak in tension.

Converted Carbon/Silicon Carbide

Where a harder surface is required, carbon-graphite converted to silicon carbide can be used.

- The component is machined from porous carbon-graphite (or moulded).
- It is then placed in an oven containing Silicon Monoxide vapour at 1625-2225°C.
- A chemical reaction takes place between the carbon and silicon monoxide gas, producing silicon carbide.
- The depth of the converted "casing" depends on the time in the oven, and the temperature.

  E.g., 2 hours @ 1925°C gives approximately 0.5 mm depth of silicon carbide conversion.
- Porous carbon-graphite is used, so the resulting material must still be impregnated with resin. It is, therefore, no more corrosion resistant, nor able to seal at higher temperatures, than normal resin impregnated carbon-graphite.

A Converted Carbon Primary Ring. Note the Silicon Carbide "casing".

Advantages
- The hard surface can withstand erosion from abrasives and wear from crystalline build-up.
- For selected applications, it has better wearing properties than impregnated carbon-graphite.

Disadvantages
- The silicon carbide casing is relatively thin, so usually it cannot be re-lapped.

Solid Silicon Carbide.

This is available in two common forms:

Pure sintered material, often referred to as Sintered Alpha grade.

- No free silicon – pure silicon carbide.
- Best chemical resistance.
- High thermal conductivity.
- Lower fracture resistance.
- Superior friction characteristics compared to tungsten carbide.
- Preferred material by John Crane for most standard wet seal applications.

Reaction bonded material.

- Silicon carbide grains bonded together by free silicon – about 10% content.
- Chemical resistance lower than sintered alpha grade.
- Excellent thermal conductivity.
- Better friction characteristics than sintered alpha grade.
- Thermal shock resistance higher than sintered alpha grade.
- Preferred material in API 682 Standard (Edition 1) for Refinery applications.

Silicon Carbide is an extremely hard, but very brittle material, and must be handled with care. Very light in weight - an advantage on larger seals.

Tungsten Carbide (Cemented Carbide).

This is available in two common forms:

Cobalt bonded

- Lower cost.
- Restricted to pH values above 7.
- Attacked by chlorides (e.g., sea water).

Nickel bonded

- Restricted to pH values above 6.
- Better resistance to chloride attack.

Cobalt bonded tungsten carbide mating ring showing results of sea water attack

Tungsten Carbide is a hard but very dense heavy material, which can be a disadvantage on larger seals. Thermal conductivity and chemical resistance are not as good as silicon carbide. Use silicon carbide wherever possible.

Other materials, such as PTFE (and even stainless steel), are used as Primary Ring materials in specialist applications.

Some Common Mating Ring Materials

Austenitic Cast Iron

Cast iron with high nickel content, such as Ni-Resist.

Advantages
- Relatively low cost.
- Easy to machine.
- Good frictional characteristics.

Disadvantages
- Corrodes easily.
- High thermal expansion.
- Shorter wear life.

Ceramic

This is the term usually applied to Aluminium Oxide. The quality of this material has improved over the years, and the content of impurities such as silica have reduced. The preferred grade now has a purity of 99.5%.

Advantages
- Relatively low cost in simple shapes.
- Very hard.
- Very good chemical resistance.
- Excellent wear resistance.
- Good running characteristics with carbon, especially on aqueous fluids.

Disadvantages
- Poor thermal shock resistance (see photograph to the right, showing typical thermal shock damage on a ceramic mating ring).
- Poor dry running characteristics — especially during transient conditions.
- Poor thermal conductivity, poor heat dissipation.
- Brittle.
- Only use on low duty applications.

Silicon Carbide

This is now the most common material used for mating rings due to its hardness and high chemical resistance. The same two forms discussed under “Primary Rings” are used for mating rings. It is used in high duty applications, but also in lower duties where the benefits outweigh the higher initial cost.

The best face material combination is carbon-graphite versus silicon carbide which gives long life in a wide variety of conditions, because of its excellent resistance to thermal shock, transient and boundary conditions, although these conditions should be avoided to ensure long seal life.

In abrasive applications, tungsten carbide is often run against silicon carbide, giving the most effective combination for wear resistance and friction, but tolerance to dry running and boundary lubrication is poor.

Tungsten Carbide

The two grades previously discussed under “Primary Rings” are also available as mating rings. However, silicon carbide should be considered first. Tungsten carbide tends to be favoured in dry gas seals for the rotating mating rings due to its additional toughness compared to silicon carbide.

Other materials, such as Stellite and Chrome Oxide are used as mating ring materials for specialist applications.

Wherever possible, the primary ring and mating ring of a seal should be of dissimilar materials to ensure maximum seal life. Choice of materials will depend on many factors, including service conditions, plant operating procedures and seal design. The seal supplier will recommend the best material combination for each specific application.

Material	JC Code	Vickers Hardness	Thermal Conductivity W/m°C@20°C	Expansion Coefficient X 10^{-6}/°C	Thermal Shock 000's W/m	Density 000's kg/m^3
Silicon Carbide Pure Alpha Sintered	277	2500	125	4.0	24	3.1
Silicon Carbide +10% Si. Reaction Bonded	088	2500 + softer Silicon	150	4.6	35	3.1
Converted Silicon Carbide/Carbon	121	2500 + softer carbon	50	4.0	30	2.0
Tungsten Carbide +6% Cobalt	025	1500-1600	100	5.2	48	14.7
Tungsten Carbide +6% Nickel	005	1300-1500	80	5.6	43	14.7
Aluminium Oxide 99.5% Alumina	059	1500+	26	6.9	6	3.9
Austenitic Cast Iron 13% Ni, 6% Cu	007	200	40	19.3	-	7.3
Carbon-Graphite Resin Impregnated	171	90 (estimated)	12	3.7	10	1.8

Note:

1. Relative density of silicon and tungsten carbides.
2. Relative hardness of silicon and tungsten carbides.
3. Relative thermal conductivity of silicon and tungsten carbides.
4. Thermal shock parameter of alumina ceramic.

Introduction

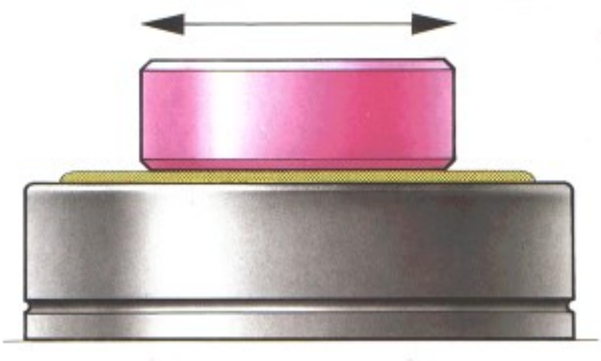
It has been determined by experience that the really flat surfaces of the two rubbing components—the primary ring and the mating ring—of a mechanical seal sustain a fluid film between them that successfully prevents mass escape of fluid from the rotating equipment of various applications in the domestic and industrial spheres of the world.

However, long before that state occurs, it is necessary to produce these flat surfaces. Initially this is achieved by the machining, casting or moulding the components in metal, ceramics, carbons or various carbides. To achieve the flatness and finish required by the mechanical seal it is necessary to lap these components.

The process of lapping is essentially the rubbing of a component against a flat plate with an abrasive material in between. The purpose is to smooth or polish the component by removing the roughness and any surface imperfections.

Mechanical Seal Principles
Lapping and Flatness Measurement

Introduction



Waterproof abrasive paper [wet or dry] is an example of the common use of this process.

The amount of material removed during lapping is very small and mainly depends on two things:

1. The coarseness of the abrasive used
2. The softness of the material being lapped.

Cast Iron components for instance, may take 15 minutes to lap, whilst Tungsten Carbide may take 2

In simple applications there are two types of Lapping Plate used:

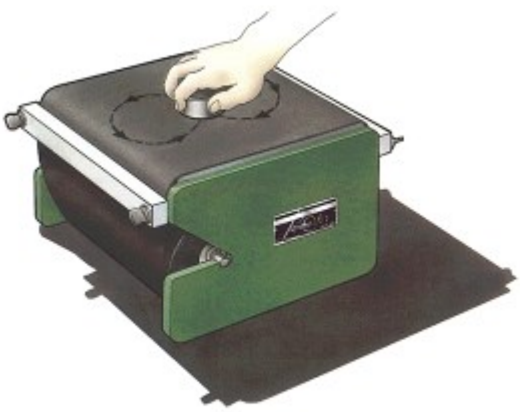
1. Wets Plates
2. Dry Plates

Wet Plates
These plates are normally Cast Iron or Steel, about 250mm square, with shallow grooves machined both ways across the surface at right angles to each other.

Lapping compound is poured onto the surface and the specimen or part to be lapped is pressed onto the plate.

Whilst maintaining pressure on the component it is moved over the surface of the plate using a figure of eight motion. The surplus lapping compound is pushed into the grooves and has to be removed and re -applied frequently.

Specimens rubbed on these plates end up with a matt finish which is sometimes unsuitable for inspection.



Dry Plates

These are used to give a polished finish to components and are similar in some ways to wet plates, as the title suggests, however, no liquid is applied to the surface.

Apart from using flat plates of Cast iron or Ceramic, various fine abrasive papers can be stretched over a plate to provide the correct surface texture.

The paper is normally coated with abrasive powder, such as Silicon Carbide, but when very fine powders are used, the plates can be used for polishing purposes.

The paper itself can be changed when wear takes place or in some applications merely rolled onto a clean area and that portion in used clamped to keep it taut.

Lapping Compounds

Compounds such as:

Aluminium Oxide

Silicon Carbide

Boron Carbide

Are inexpensive compared with Diamond compounds which are used on extremely hard materials including both Silicon and Tungsten Carbides.

The qualities required in a compound are that it retains the cutting edge for a long time and that it is not affected by the lapping liquid which is normally oil based.

General Industrial Use

As the lapping process takes a long time, some kind of motorised plate is essential as hand lapping may take several days to complete.

The component to be lapped is then held in contact with the plate either by hand or in a simple fixture.

Industrial Use

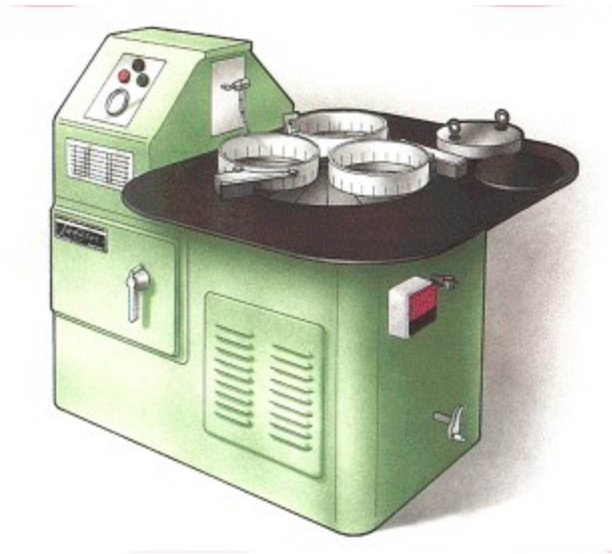
Special machines have been developed for production runs.

These machines, which can be adjusted by the operator keep themselves flat whilst running.

This is achieved by moving the conditioning ring in or away from the centre of the plate.

The use of this machine lends itself to mechanical seal production. The seal faces produced by lapping must not only be smooth in order to operate correctly but the surface achieved must be extremely flat

This is a typical picture of a Lapping Machine. The lapping plate rotates anti-clockwise and on its surface are positioned three or more conditioning rings. These rings, due to friction, rotate in the same direction as the lapping plate. The rings are held in position with adjustable brackets.



The components to be lapped are placed into the conditioning rings and in the presence of an abrasive compound being drip fed onto the plate, material is removed from the surface.

Reasons for Lapping

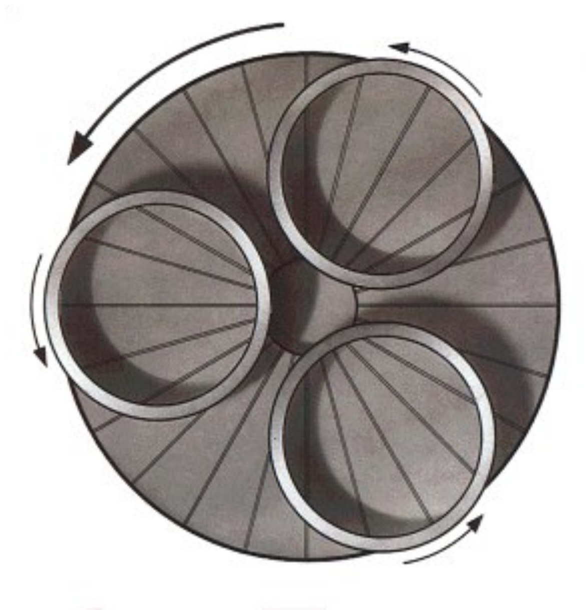
What other benefits do we achieve when we lap?
Are these benefits related to seals or engineering in general?

Here is a list of reasons to lap:

- Flatness accuracy
- Dimensional accuracy
- Finish requirements
- Preparing a clamping surface for distortion free work
- Cost—lapping may prove less expensive
- This parts that are difficult to hold
- Multi-material work pieces that react differently to a cutting tool or grinding wheel
- Accurate control of cutting forces
- Elimination of operations—produces finish and dimensional accuracy simultaneously
- Product improvement—closer tolerances flatness or finishes
- Reduced cost of fixtures— most parts can be lapped without special features

So why do we need to lap?

- Flatness accuracy
- Remove cuts, scratches or other surface blemishes
- To enable two lapped surfaces to support a liquid film to act as a barrier



The conditioning rings serve two purposes:

1. To hold the components
2. To maintain the flatness of the lapping plate.

When components need to be lapped they are place on the lapping plate within the conditioning rings. Sometimes individual fibre discs which are pre-cut for the small components are used to separate the individual parts. As all the parts rotate independently of each other the resultant finish is of a multi-directional lay.

As part of the maintenance programme the actual flatness of the lapping plate needs to be checked.

To achieve this we must polish the dull finish of the lapped test piece to enable light which shines from a Helium filled tube [Monochromatic light] to be reflected through a quartz glass optical flat.

When a component is lapped there are two terms used to describe its appearance

Matt finish [dull and non reflective]

Polished finish [reflective]

Ra—Roughness Average—used in Europe

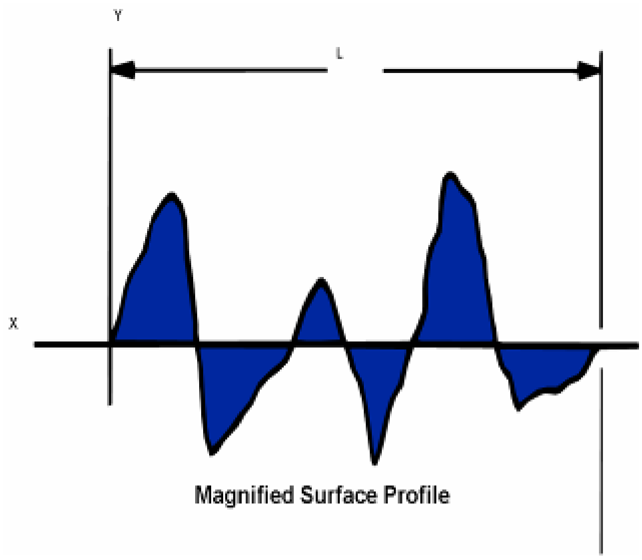
Cla—Centre Line Average—used in USA

RMS—Root Mean Square—Rarely used

As well as the appearance it is also possible to define it dimensionally and the terms opposite are the most commonly used.

The profile of the graph opposite is of an enlarged view of the edge of a surface we wish to check for finish. You can see it is made up of peaks and valleys.

This profile was produced by an instrument called a Profiliometer [Talysurf] , by moving a diamond stylus across the surface to be measured. The instrument interprets the peaks and valleys into a scale that defines the finish by a number. The higher the Ra number the rougher the finish, conversely the closer to zero the more perfect the finish.

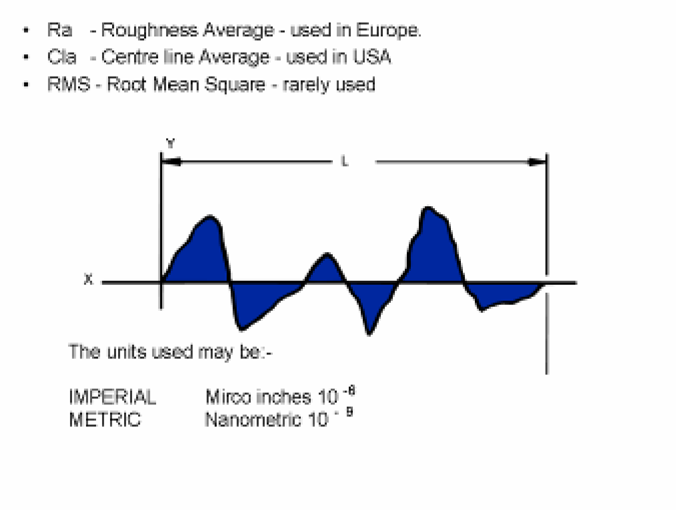


The units used for measurement may be:

Imperial - Micro inches

Metric - Micrometre 10^{-6}

Nanometre 10^{-9}



A wave length of light is 23.2 millionth of an inch. As only one half of the light is used,

1 Light Band	= 11.6 Millionth of an inch
	= 0.0003 mm
	= 295 Nanometres

Flatness

The flatness of a component may be measured using an optical flat and a monochromatic light source and is the reason why it is necessary to polish the matt lapped surface of the component to enable light rays to be reflected through the quartz to give indications of surface variance.

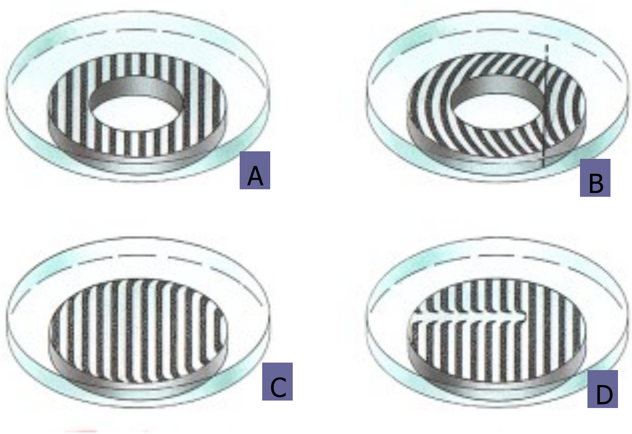
A light band is a unit of measurement to define the flatness of the surface and is half of a single wave length.

The Optical Flat

Cleanliness of the work piece and the optical flat is essential, since the width of a light band is dependant upon the thickness of the air gap between the two surfaces. If there are any dirt particles present the light bands will be very narrow and difficult to read.

NOTE:- The Optical Flat should always be placed in position, or lifted vertically to remove it. Never slide it across the work piece as this action may scratch the surface making it difficult to obtain readings.
Place the work piece under the light source with a lens tissue on top. Now lower the Optical Flat onto the lens tissue and then slide out the tissue.

Apply light finger pressure on the edge of the Optical Flat at several points. Following the procedure above, a series of lines should be reflected. Interpretations of these are given in Figure 11 items A,B,C & D of this module.



Interpretations

Fig. 11A Straight parallel bands. Surface is flat to within 1 light band.

Fig. 11B Tangent intersects two light bands. The component is two light bands out of flat.

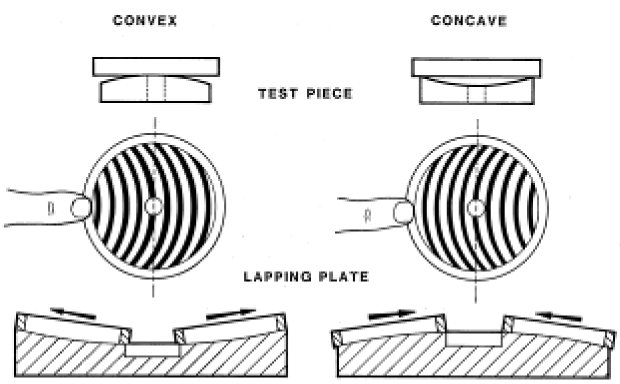
Fig. 11C Component is flat to one light band but falling away at the edges. This indicates that the edges are rounded caused by improper polishing, wrong compound vehicle proportions or excessive pressure.

Fig. 11D Component has been incompletely lapped and low spots are indicated.

Checking Lapping Plate for flatness

The lapping plate should be checked at regular intervals for flatness, and, depending on usage this may be daily, weekly or monthly.

To do this a standard test piece should be made , and for a machine with a 36" diameter lapping plate this should be 250mm diameter and 25mm thick. The test piece should be lapped then checked under an optical flat. If it is found to be not flat, adjustments should be made to the conditioning rings, dependant upon whether the test piece is concave or convex, see figure 12 opposite.



In any event, the conditioning rings should not be adjusted by more than 6mm equally. By this means it is possible to achieve a degree of flatness of the lapping plate, which will guarantee the production parts being lapped to an acceptable flatness.

Compounds	**Alumina Powder grit 8900 [8µm]**
	Boron Carbide Ca 600 grit
[25µm]	
	Diamond Paste [3µm]
Vehicles	**Lapmaster No. 3**
	Gulfcut 36
Polishing Paper	**Carborundum Grade 3/0**

Lapping Compound and Vehicles

The compound and vehicles to be used will depend on the amount of stock [material] to be removed, the degree of finish required and the range of materials to be lapped. Clearly, the compound used must be harder than the work piece.

In general, for low stock removal and good finish Alumina Powder abrasive is recommended for metals but not hard materials.

For hard materials or for fast stock removal Boron carbide is used.

Diamond lapping paste is also frequently used as this gives a finer finish than Boron Carbide.

Remember!
Primary rings and Mating rings are accurately finished. Protect them at all times and keep them clean!

MATERIALS	RECOMMENDED FINISH Primary Ring	 Mating Ring
Carbon	Polish	Matt
Nickel Iron	Matt	Polish
Tungsten Carbide	Matt	Polish
Silicon carbide	Matt	Polish
Ceramic		Polish

Finish — Matt or Polish

Polishing is essential on Carbons used for mechanical seals because the seal must work when in the static as well as dynamic modes and polishing improves the ability to do this.

Other materials also benefit from polishing , but if both the seat and face of a seal are polished they may ‘wring’ together and cannot be separated. This is especially so if both components are made of the same material.

If there is no available means of checking the flatness of a pair of components one method is to try ‘wringing’ both components together. Should the components remain together after ‘wringing’ they are both deemed flat.

The table above indicates the recommended finish that should be applied to primary rings and mating rings and is dependant upon materials of construction.

Summary

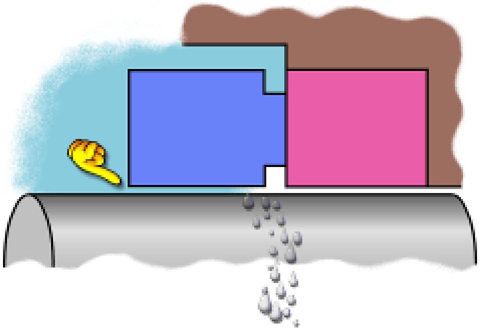
- **Machine lapping is economical and simple**
- **Produces finish and dimensional accuracy simultaneously**
- **Requires no special fixtures**
- **Provides a surface to give the thinnest liquid film**

Mechanical Seal Principles

The Secondary Seal

Introduction

The main purpose of the Secondary Sealing device is to prevent leakage of fluid passed the primary ring, whilst allowing axial movement to take up wear and misalignment. This training module looks primarily at rotating seal units, with rotating Secondary Seals and stationary mating rings, but the theory applies equally to stationary seal units with non-rotating Secondary Seals and rotating mating rings.



There are three Secondary Seal designs in common use:

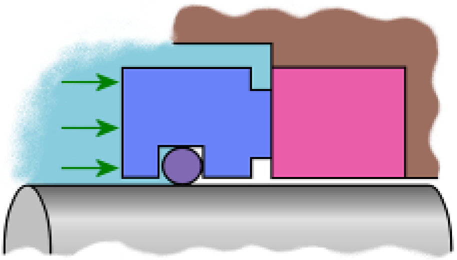
1. O-Rings
2. PTFE Sealing Rings
    - Wedges
    - Chevrons
    - 'C' rings with internal spring energiser
    - PTFE sleeved o-rings
3. Bellows
    - Elastomer
    - Metal
        - Formed
        - Edge Welded
    - PTFE

It is a common practice to describe the seal by its Secondary Sealing device: e.g., Wedge Seals, Bellows Seals, etc.

All secondary sealing devices clearly divide into two main groups, those that slide along the shaft to take up wear and misalignment — Pusher Seals (e.g., dynamic o-rings and wedges), and those that do **not** slide along the shaft — Non-Pusher Seals (bellows secondary seals).

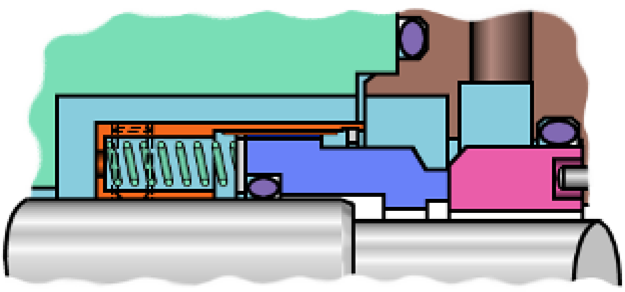
Pusher Seals

As the primary ring moves to take up wear or misalignment, the Secondary Seal (in this example, an o-ring) must slide along the shaft/sleeve, and is pushed by the springs and hydraulic forces – hence the term "Pusher Seal". It is important that the shaft/sleeve has a high quality surface finish to aid this sliding action. (See Chart at the end of this booklet).



Elastomer O-Ring Secondary Seals

O-rings are the most common components used as Secondary Seals in Pusher Seal designs. They are readily available in a wide range of materials to suit most applications.



Here is a typical pusher seal, designed for low emission duties. The primary ring design has been computer optimised, as discussed in a previous training module. Multiple springs ensure an even loading on the primary ring, and result in a compact seal unit. Positive drive is used to prevent any slip, or damage due to torque transmission through weaker components. Cooling flow has been optimised to remove any "dead" areas, and to ensure cooling is applied to the mating ring and primary ring round their full circumference.

Advantages of Pusher Seals

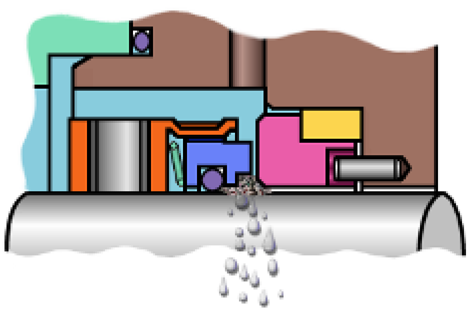
- They tend to start drip leakage giving pre-warning of failure. There is little or nothing in the design that can break and cause sudden massive leakage.
- Primary ring is usually one solid component. No additional stresses are caused by assembling composite items by press-fitting or shrink-fitting. This enables a much higher pressure capability.
- Individual items can be easily replaced, making it simple to change materials, and to replace worn components.

Disadvantages of Pusher Seals

- Sliding secondary seals can “lock” in position, a phenomenon known as “hang-up”.
- Elastomer o-rings can take permanent set, causing hang-up.
- Excellent surface finish required to aid the sliding action of the pusher seal.
- Temperature limited by the available elastomers. Thermal expansion of the elastomers used must also be taken into account. At high temperatures, even though the material of the o-ring is capable of withstanding the temperature, the thermal expansion may be so great as to cause hang-up or even bursting of the carbon ring. At low temperatures, thermal contraction of the elastomer material may cause leak paths. O-ring grooves may have to be specially designed for very low or very high temperatures.

Hang-Up

Where a product can change state as it moves across the interface film, e.g., solidify, crystallise or polymerise, the secondary seal starts to ‘hang up’ on these deposits. As the primary ring wears it is unable to follow-up to keep the faces together. Leakage will gradually increase, deposits build up, and eventually leakage becomes prematurely unacceptable as the faces are held in the open position. In many applications, this problem can be overcome by the use of an external quench using API Piping Plan 62 to immediately remove any small amounts of leakage. (See Appendix B).

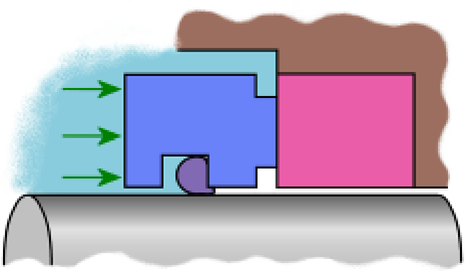


For example:

- Sodium hydroxide solution (crystallisation) – use a cold water quench.
- Hot oil (carbonisation/coking) – use a steam quench.

O-ring seals are very effective, but due to the small clearances required they may only have marginal flexibility. The clearance between the seal and the shaft has to be such that when the shaft/sleeve is on its maximum tolerance and the primary ring bore is on its minimum tolerance there is still adequate clearance to allow for this flexibility.

To increase flexibility, clearances are often increased, but this can cause problems at higher pressures, especially if the equipment maker’s shaft/sleeve has been manufactured to its bottom limit, and the seal maker’s primary ring is on its top limit.

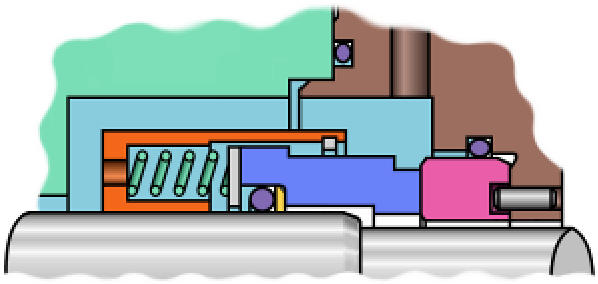


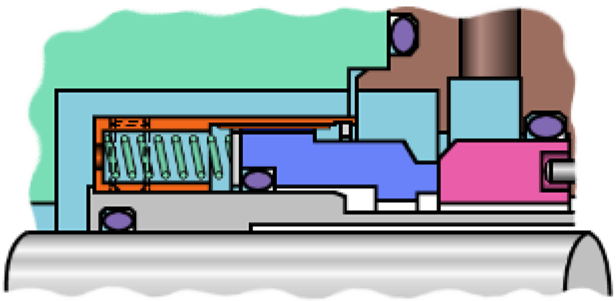
The o-ring is pushed into the corner of the groove by the hydraulic pressure, extruding it into the bore of the primary ring. The o-ring can hang-up on the shaft and prevent the seal from following up as the primary ring wears.

This can be simply prevented by using an anti-extrusion ring on the low pressure side of the o-ring for higher pressure seals. The ring, usually made from PTFE, is sometimes referred to as a “back-up ring”.

Note: It is possible for it to be inserted on the wrong side of the o-ring rendering it ineffective.

The anti-extrusion ring should be installed on the low pressure side of the o-ring. Most PTFE anti-extrusion rings have a small chamfer on the outside diameter. This is to allow for the small radius in the corner of the counterbore in the primary ring. It is important to ensure that the anti-extrusion ring is fitted the right way round.





In cartridge seal designs, where both the primary ring *and* the sleeve going through its bore are supplied by the seal manufacturer, these two components can be made to much tighter tolerances, often enabling the gap to be reduced to a point where the use of an anti-extrusion ring can be eliminated.

The following chart lists some of the more common o-ring materials used in mechanical seals. Maximum and minimum temperatures are given as a guide. These may vary with the many different grades available, and differences between various generic material manufacturers. Many other materials are available for specialist applications. For chemical compatibility information, please consult John Crane.

Material	Typical Trade/ ISO/DIN/ Common names	Minimum temperature in seals	Maximum temperature in seals	Comments
Medium Nitrile	NBR Buna N	-40°C	100°C	General purpose material. Up to 120°C in hydrocarbons
Chloroprene	CR Neoprene	-40°C	100°C	Ideal for refrigeration duties. Some specialist applications.
Ethylene Propylene	EP; EPR; EPDM Nordel™	-40°C	135°C	Ideal for water up to 150°C. Avoid oil/hydrocarbons.
Fluorocarbon*	FKM Viton A™	-30°C	200°C	Maximum 135°C in water. Hardens in high temp steam.
Perfluoroelastomer* (Low temp. grades)	FFKM; Isolast™ Kalrez™Chemraz	-20°C	215°C	Wide range of chemical compatibility.
Perfluoroelastomer* (High temp. grades)	Isolast HT™ Kalrez™Chemraz	-20°C	315°C	Wide range of chemical compatibility.

*<u>*Caution*</u>

These elastomer materials may give off dangerous hydrogen fluoride vapours at elevated temperatures. (Fluorocarbon above 275°C; Perfluoroelastomer above 400°C). These can condense or mix with water to form hydrofluoric acid. If you suspect that an o-ring has been subjected to excessive heat, please take precautions, including wearing suitable neoprene or PVC protective gloves and eye protection. Parts should be neutralised with an alkali solution such as calcium hydroxide.

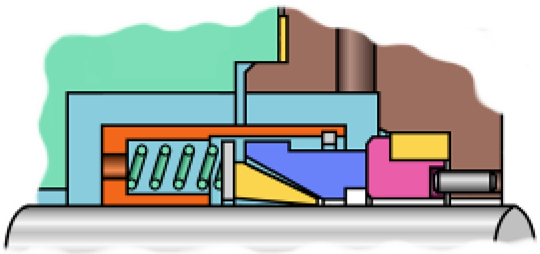
Points to look out for:

- *Has a sleeve been heated to assist removal from a shaft?*
- *Do the stainless steel seal components show any signs of heat discolouration?*
  - *Straw yellow discolouration on stainless steel occurs at 370-425°C, above the danger point of both elastomer materials. All other discolouration occurs at even higher temperatures!*

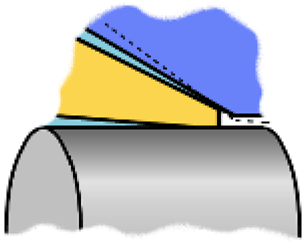
PTFE Secondary Seals — Pusher Seals

Where elastomers are not suitable, PTFE (Polytetrafluoroethylene) can often be used. However, PTFE has an unfortunate property known as "cold-flow". It tends to act like an extremely viscous fluid. In a pusher type mechanical seal, the o-ring groove is designed to give a controlled compression of the section of the o-ring, creating a sealing point on the outside and inside diameters of the o-ring. Too much compression will cause hang-up, too little compression will allow leakage. PTFE is relatively hard, and when in the form of an o-ring can be extremely difficult to fit as one tries to achieve the necessary compression. PTFE also has a high rate of thermal expansion. All this means that it is difficult, if not impossible, to control the compression of the section of the o-ring, so hang-up is more likely. Once a PTFE o-ring has been installed, the compressive forces will cause the o-ring to cold-flow and initial compression will be lost over time. For these reasons, it is unusual to use PTFE dynamic o-rings in mechanical seals. PTFE secondary seals must be designed to be constantly loaded to allow for any cold-flow, and they must be able to compensate for any thermal expansion.

PTFE Wedge Rings



One of the most common forms of PTFE Secondary Seal is the wedge ring. This is actually a cone-shaped component, resembling a wedge in cross-section. The spring pressure constantly pushes the wedge into the cone-shaped bore of the primary ring which forces the nose of the wedge down onto the shaft. The hydraulic pressure has little effect on the wedge, and thermal expansion will only cause the wedge to travel back up the primary ring cone. There is some allowance for flexibility because the angle on the outside of the wedge ring is different to that in the bore of the primary ring to allow for movement caused by misalignment. There is also an angle in the bore of the wedge ring for the same reason. The primary ring can move in relation to the wedge ring and still form an effective seal. The primary ring can actually rock on the nose wedge to compensate for any misalignment. The point loading at the nose of the wedge forms a seal in the primary ring, and an effective sealing band around the shaft.

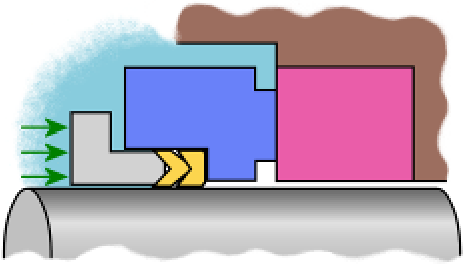


Wedge rings can also be manufactured from exfoliated graphite for high temperature applications.

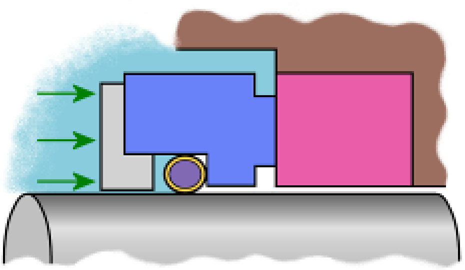
Other PTFE Secondary Seals

There are several other designs of dynamic PTFE Secondary Seals, and we discuss here some of the more common ones.

PTFE Chevrons are constantly energised by the spring force and hydraulic pressure transmitted through the metal follower. PTFE cold-flow is prevented by using a fully-trapped chevron seal. The chevron ring assembly is designed to fit into the same primary ring as elastomer o-ring seals, which can aid inventory reduction.



There tends to be less misalignment allowance with this design, and a change in pressure has a direct effect on the energising and friction forces.

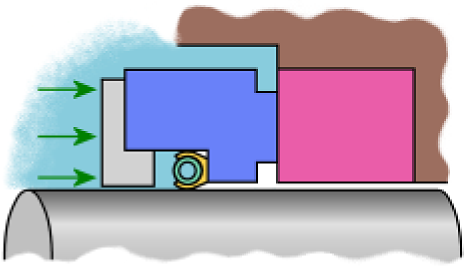


Another way to overcome the incompressibility of PTFE is to use an elastomer o-ring (such as fluorocarbon) that has been sleeved with PTFE. This increases flexibility, reduces the possibility of hang-up and problems from thermal expansion. Again, this component will fit into a standard o-ring counterbore.

The PTFE sleeve is relatively thin and if damaged will allow chemical attack of the elastomer core. Any bending of the seal during installation will cause irreparable damage.

A more flexible Secondary Seal is the fluorocarbon elastomer o-ring that has been coated with FEP (Fluorinated Ethylene Propylene). FEP has similar chemical resistance to PTFE. The coated o-ring is more flexible than a PTFE sleeved o-ring, but is more easily damaged, and has a lower high temperature limit.

The resilience of PTFE can be improved by using a PTFE envelope or 'C' ring. This is initially energised by an internal coil spring, and the hydraulic pressure forces the inner and outer lips outwards to improve the seal. It can be used to replace elastomer o-rings without design changes, allows movement to take up misalignment and is unaffected by thermal expansion.



The chemical resistance is limited by the material of the spring, which is immersed in the sealed fluid. It is easily damaged if slightly bent creating a kink, and thus a leak path, in the sealing lips.

This chart gives the usual temperature limits for dynamic PTFE and FEP Secondary Seals. The stability and strength of pure PTFE at higher temperatures can be improved by the addition of glass filling, usually about 25% glass/75% PTFE

Material	Minimum temperature in seals	Maximum temperature in seals
Pure PTFE	-40°C	230°C
Glass Filled PTFE	-100°C	280°C
Exfoliated Graphite	-40°C	500°C
PTFE Sleeved Fluorocarbon	-20°C	200°C
FEP Coated Fluorocarbon	-20°C	150°C
Spring Energised Pure PTFE Envelope	-20°C	200°C
Spring Energised Glass Filled PTFE Envelope (Higher pressures)	-20°C	200°C

Caution
At elevated temperatures PTFE can give off toxic fumes. Do not incinerate old or new PTFE components. It is dangerous to smoke while handling products made from PTFE.

Non-Pusher Seals

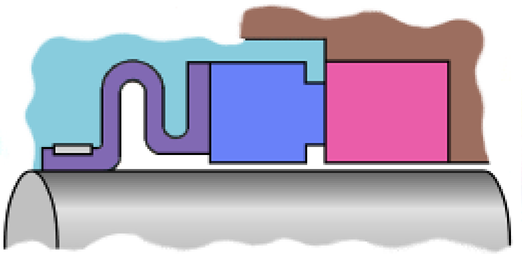
With Non-Pusher seals, the Secondary Seal is formed by a flexible bellows. The actual sealing point becomes remote from the primary ring allowing a great deal more flexibility. The sealing point is fixed on the shaft/sleeve and does not have to slide to accommodate wear of the primary ring, shaft end-play or misalignment. There is no contact between the shaft/sleeve and the primary ring and movement is taken up by the bellows flexing or extending. There is less likelihood, therefore, of hang-up caused by o-rings sticking, or by deposits on the shaft/sleeve. There is also very little hysteresis as friction from a sliding secondary seal component has been removed.

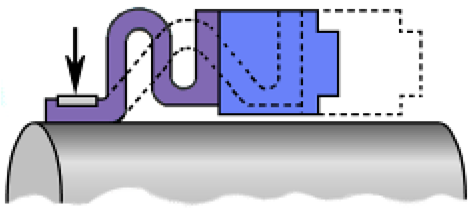
As nothing needs to be pushed along the shaft/sleeve to take up wear, shaft end-play or misalignment this type of seal is described as a "Non-Pusher" seal.

The actual mechanism of sealing the mechanical seal unit to the shaft/sleeve varies dependent upon the duty conditions, seal design and materials of construction.

Elastomer Bellows Seals

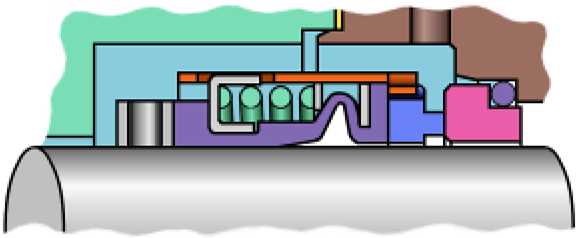
Elastomer bellows Secondary Seals normally grip the shaft with an interference fit created by a metal drive ring. The drive ring creates a squeeze effect on the tail. For this reason, a good shaft surface finish is not desirable, and the use of good lubricant such as silicone grease should be avoided.





It can be seen here how the bellows flexes and expands, allowing the primary ring to move freely, with nothing sliding along the shaft surface.

A typical elastomer bellows seal unit requires a spring to keep the faces in contact. Note the positive drive from the bellows tail, through the metal components, to the primary ring to prevent any torque being transmitted through the elastomer bellows.



The seal type illustrated is a compact general duty elastomer bellows seal, designed to fit into seal chambers to DIN 24960 standard. Note that the spring is mounted within the outer retainer, leading to a reduction in both radial section and axial length.

Elastomer bellows seals are relatively low cost, and ideal for use where space is confined. They are extremely flexible and rugged, and are suitable for most general duties in pulp and paper, petrochemical, waste water treatment and other demanding applications.

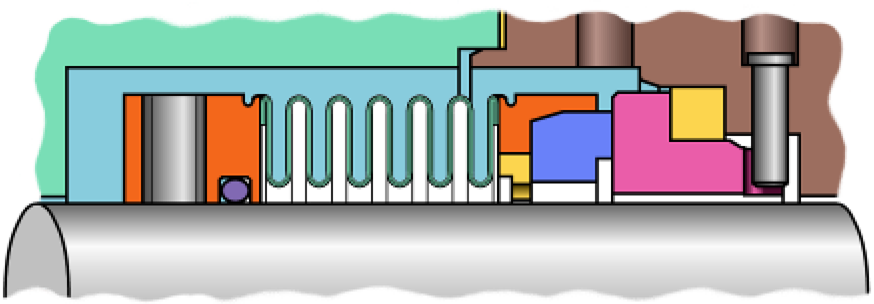
Elastomer bellows are usually available in the following materials:

Nitrile; Chloroprene; Ethylene Propylene and Fluorocarbon.

Formed Metal Bellows Seals

The bellows convolutions are formed either mechanically or hydraulically, or a combination of the two, from a thin welded tube. In a few seal designs, the bellows is welded or press-fitted directly to the shaft/sleeve, but it is more usual to incorporate a locking ring with grubscrews and an o-ring seal in the bore. Also, on some low duty designs, the primary ring is press-fitted directly into the end of the formed bellows. In the design shown below, note that the primary ring carrier or shell is a machined component. This is welded to the bellows core which means that it is possible to manufacture this shell from a different material to the bellows, such as low expansion alloy. This enables higher temperatures to be sealed without the possibility of the primary ring coming loose due to the higher thermal expansion coefficient of the shell.

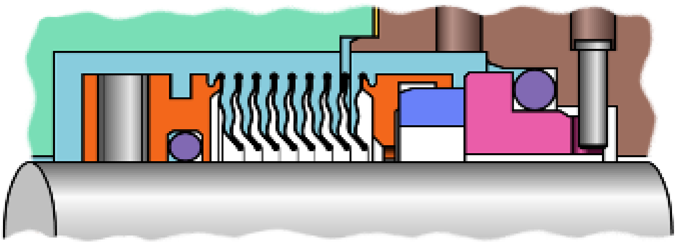
This picture shows a further design advance. The bellows is formed from two concentric metal tubes. This two-ply bellows gives greater Torsional strength, flexibility and lower stress levels in the bellows material.



The profile of this type of seal is very smooth, and can be highly polished for use in hygienic and pharmaceutical applications. Seals are available using formed metal bellows that eliminate all possible “bug-traps” and are designed specifically for use in the food and beverage industries.

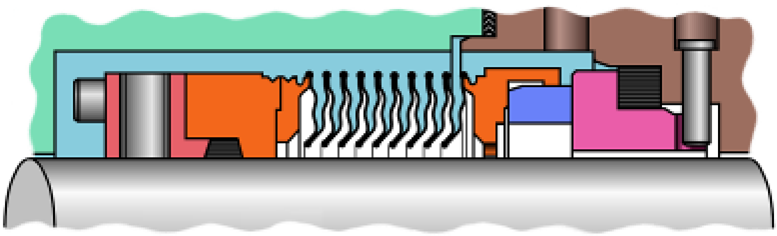
Edge Welded Metal Bellows Seals

Initially developed to solve critical and demanding applications for NASA, welded metal bellows seals are now used in most industries. Bellows plates, which are about 0.15 mm thick (varies with seal type, size and seal manufacturer) are welded together at the inside diameter. A number of welded pairs are then welded together on the outside diameter to form the required number of bellows convolutions. A wide range of metallurgies can be used in the manufacture of these bellows including highly corrosion resistant materials such as Hastelloy™ and Titanium. Edge welded metal bellows can be designed to produce exceptionally reliable, low stress welds. Many metal bellows seal designs are apparently similar, but performance can vary greatly depending on design features, materials and manufacturing quality. Note the 45° tilt at the inside diameter of the plates (removes most of the bending stresses at the weld), the three sweep shape (to prevent “oil-canning”), and the nesting ripple design (computer optimised to give lower spring rate coupled with higher flexibility).



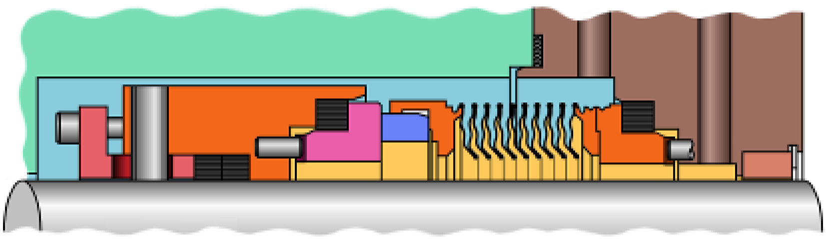
Again, the primary ring carrier shell is a machined component welded to the bellows core so it can be made of a different material such as low expansion alloy which will ensure that the primary ring remains firmly held in position at elevated temperatures.

Where the temperature of the product is above the capability of conventional elastomers, exfoliated graphite Grafoil™packing rings can be used. These are fitted into a counterbore at the back of the seal and small screws force the drive ring/follower into the counterbore to compress the packing, forming a perfect seal on the shaft/sleeve. There is no relative movement between the packing and the shaft/sleeve so there will be no leakage or wear at this point. Graphite packing is also used as the tertiary seal (see Training Module TM-053: Tertiary Seal) to seal the mating ring into its housing. An anti-rotation pin is required due to the low coefficient of friction of graphite. A spiral-wound gasket seals the gland plate to the face of the seal chamber. Seals of this design are able to withstand temperatures of up to 425°C.



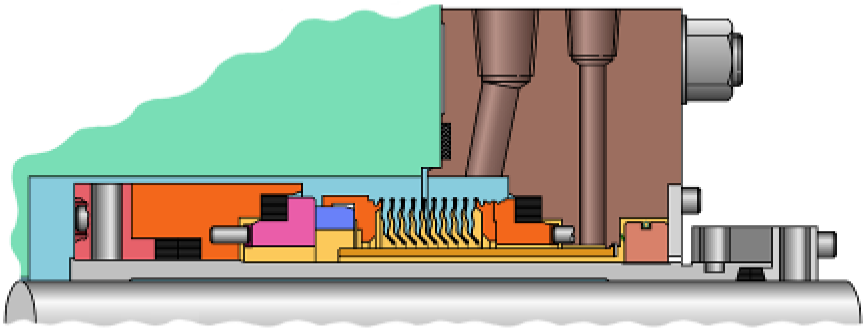
For viscous or thermo-setting applications, welded metal bellows seal designs can incorporate drive lugs under the bellows to provide additional rotational drive, thus reducing Torsional stresses on the bellows.

Stationary seals are preferred in demanding, high temperature applications. Unlike rotating seals which must flex on every revolution to accommodate seal chamber face misalignment relative to the shaft axis, stationary seals adapt to this misalignment by flexing only once during installation. This reduces seal movement, resulting in increased seal life.

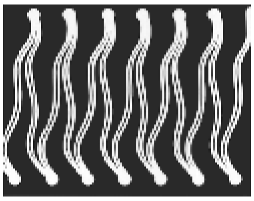


At 2950 rpm, a misaligned stationary mating ring will cause a rotating seal to flex 8,496,000 time per day!

With high temperature hydrocarbons, a steam quench should be injected through the bore of the seal to remove any small amounts of leakage that may occur as carbonised oil and coke deposits can build up on the sleeve and seal components. This picture shows a stationary cartridge seal which is an ideal design for high temperature refinery applications. It is fitted with a bronze anti-coking baffle which directs the steam (shown in yellow) firstly through the bore of the metal bellows, and then past the running faces. Steam and any contaminants are then blown back through the bore of the baffle and out to a drain or safe area. Steam is contained in the seal assembly by a Secondary Containment seal - in this case a floating segmented carbon bush. (See Training Module TM-050: Secondary Containment). If a rotating seal had been used, the steam laden with contaminants would be directed back through the bore of the bellows which would create clogging.



Cartridge seals have many advantages over component seals, including simple, accurate, error-free installation.



Edge welded metal bellows seals are capable of sealing pressures up to 20 bar g, depending on seal size and service conditions. For pressures above this, double-ply bellows are available, increasing the pressure rating to anything up to 65 bar g.

Common materials used for manufacturing the metal bellows used in mechanical seals:

- Inconel: X-750 and 718.
- Alloy C-276 (e.g., Hastelloy C™).
- AM350 Stainless Steel.
- Alloy 20.
- Titanium.

Advantages of Non-Pusher Seals

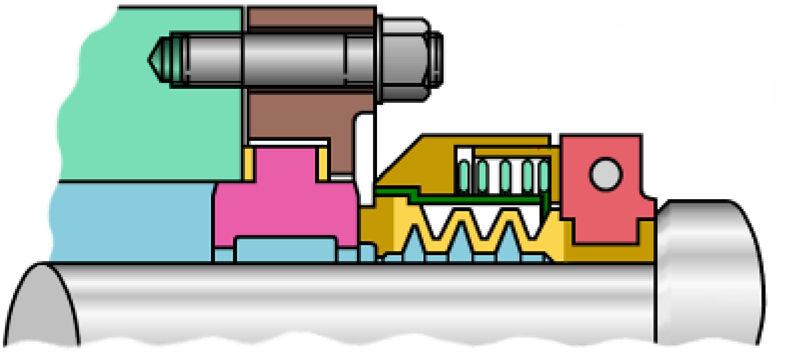
- Excellent for high temperature applications — up to 425°C (or more with special designs).
- No sliding contact — less chance of hang-up occurring
- Non-clogging — a rotating metal bellows seal presents a clean profile to the product, and particles will be centrifuged away from the bellows convolutions.
- Perfect for hygienic applications — formed metal bellows can be polished, with smooth profile.
- Fine machined shaft/sleeve acceptable — the static sealing component does not require a highly polished shaft/sleeve as there is no relative movement at this point.
- Elastomer bellows seals are low cost, whilst being extremely flexible and rugged.

Disadvantages of Non-Pusher Seals

- Possibility of metal bellows fracture — caused by excessive axial movement (due to factors such as product flashing off between the faces or excessive misalignment), although this is now extremely rare due to advances in bellows plate designs and materials.
- Not field repairable — in metal bellows seals, the primary ring is usually a press or shrink fit into its carrier. These must be returned to the manufacturer for replacement. However, this gives the manufacturer the opportunity to thoroughly inspect the seal, including a rigorous leak test of the bellows.
- Lower maximum pressures in the order of 20 bar g — for specialist applications seals can be designed to withstand pressures of up to 35 bar g. This is well within the various seal standards such as DIN 24960 and API 682. Double-ply bellows extend the maximum pressure to 65 bar g.

PTFE Bellows Secondary Seals

For highly corrosive duties including acids, salts and organic compounds, an *externally* mounted PTFE bellows seal may be used. As these seals are externally mounted, clamped mating rings must be fitted. The fluid only comes into contact with seal components made from highly corrosion resistant PTFE and aluminium oxide ceramic.



As plants on which these seals are installed often incorporate ball valves, there is a possibility of pressure surges. It is recommend that a positive abutment be used behind the seal to prevent it from moving along the shaft should excessive pressure be experienced. Note the metal sleeve (coloured green). This is essential to support the PTFE bellows and prevent it from fracturing in the event of a pressure spike.

Most mechanical seals are replaced once they have started to leak unacceptably. As this seal is designed for highly corrosive services, any leakage is unacceptable. For this reason, on the John Crane Type 10T and Type 10R seals there are two setting lines, one green which tells the installer that it is fitted to its correct working length and spring compression, and one red which indicates to the operators that plans should be made to have the seal replaced as soon as possible as it is getting close to the point where it might start to leak. A corrosion resistant clear plastic guard should always be fitted round the seal.

The running face of this type of seal is usually made from glass reinforced PTFE. Versions are available with inserted carbon-graphite or silicon carbide primary rings for abrasive and other duties.

Please refer to manufacturer's literature regarding materials, maximum temperatures and pressures.

In many plants today, for Health & Safety reasons, double or tandem seals are preferred to a single seal for highly corrosive applications.

Shaft / Sleeve Surface Finish.

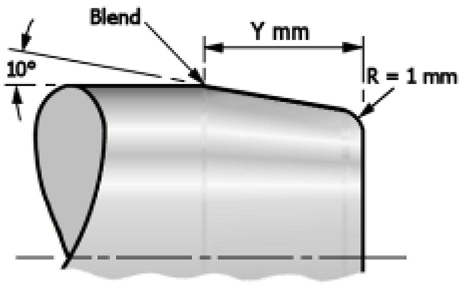
This is critical, and can affect the life of a mechanical seal. This chart gives the recommended surface finish for all types of mechanical seal, but always check with the seal fitting instructions.

Secondary Seal	Shaft/Sleeve Diametric Tolerance	Maximum Ovality	Shaft/Sleeve Surface Finish	Description
PTFE Pusher Seals	±0.05 mm	0.025 mm	0.1-0.25 μm Ra	Ground and polished free from any machining marks
Elastomer O-Ring Pusher Seals	±0.05 mm	0.05 mm	0.3-0.6 μm Ra	Polished free from any machining marks
Elastomer Bellows	±0.05 mm	0.10 mm	0.8-1.2 μm Ra	Fine machined. High quality finish is undesirable
PTFE Bellows	±0.05 mm	0.05 mm	0.8-1.2 μm Ra	Fine machined. High quality finish is undesirable
Metal Bellows	±0.05 mm	0.05 mm	1.2 μm Ra or better	Fine machined or better
Cartridge Seals	±0.05 mm	0.05 mm	1.2 μm Ra or better	Fine machined or better

Shaft / Sleeve Lead-In

It is essential that the leading edge of the shaft/sleeve over which the Secondary Seal has to pass during seal installation is chamfered to a suitable degree to prevent damage to the Secondary Seal, and to ease fitting. Recommended dimensions for this lead-in will be found in the individual seal installation instructions, which should be supplied with each complete mechanical seal.

Typical lead-in examples are given below.



Seal Type (Example Only)	Seal Size	Y mm
Unbalanced Pusher Seals	Up to 65 mm	2.5 mm
Unbalanced Pusher Seals	Above 65 mm	4.0 mm
Elastomer Bellows	Up to 26 mm	5.0 mm
Elastomer Bellows	26 mm — 60 mm	6.5 mm
Elastomer Bellows	Above 60 mm	8.0 mm

Mechanical Seal Principles

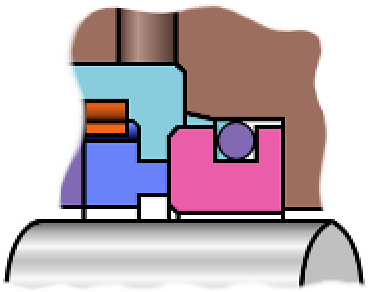
Tertiary Seal

The Tertiary Seal is required to prevent leakage round the stationary seal component, which is usually the mating ring. Where the tertiary seal is made from a material with high coefficient of friction (e.g., Nitrile), it may also be used to prevent rotation of a stationary mating ring. Where the tertiary seal is made from a material with low coefficient of friction (e.g., PTFE), an anti-rotation device is requires, such as a pin.

There are four basic designs of tertiary seal:

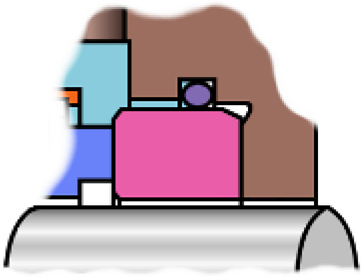
1. O-rings
   Elastomers
2. Rectangular section rings
   PTFE
   Graphite
3. Cup rings
   Elastomers
4. Gaskets or flat joints
   PTFE
   Compressed fibre
   Graphite-metal composite

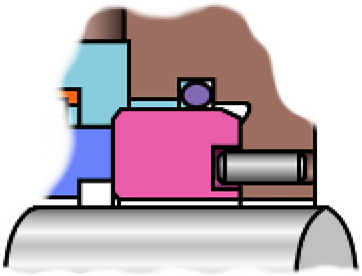
1. O-rings



O-ring tertiary seals are often located in a groove in the mating ring. Good metal-to-metal contact improves heat transfer from the mating ring to the pump casing.

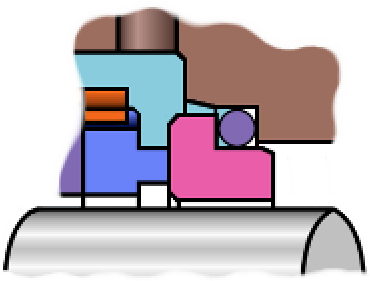
If the mating ring material is very hard (e.g., silicon carbide), an alternative is to machine a groove in the mating ring housing. Great care must be taken to prevent distortion of the surface where the mating ring butts against the bottom of the housing. On high pressure applications, the back surface of the mating ring may be lapped onto the surface of the housing to ensure no distortion will occur due to high hydraulic forces.



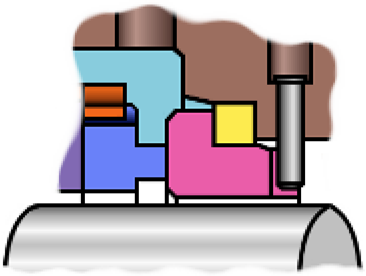


Where the process fluid may be viscous, or where the o-ring material has a low coefficient of friction (e.g., Perfluoroelastomer), the mating ring must be fitted with an anti-rotation pin.

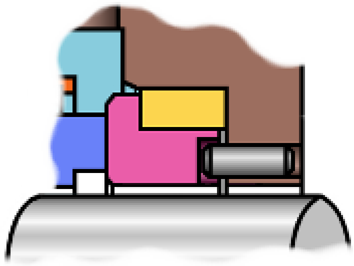
O-rings are versatile, and do not necessarily need a machined groove. Two mating parts can be machined to form an adequate location provided the necessary clearance is maintained to allow movement. The advantage of this design is that the o-ring cushions the mating ring from any small distortions of the housing. The mating ring is now isolated from the housing, so this design would not be used where good heat transfer is required.



2. Rectangular Section Rings



Rectangular section sealing rings in a range of resilient materials can be utilised to great effect. The two most common materials for this type of sealing ring are PTFE (for chemical duties) and exfoliated graphite (for high temperature duties). With these materials, the ring seals not on the outside and inside diameters, but on the front and back faces. This ensures ease of installation into the housing. For example, the coefficient of thermal expansion of PTFE is very high. If the rings were designed to seal on the outside diameter, the thermal expansion created by the heat from the installer's hands would make the ring extremely difficult to fit into the housing. With exfoliated graphite, this material has very little strength. If the ring sealed to the mating on its inside diameter it would break when being placed onto the tail of the mating ring. As both these materials have very low coefficients of friction, an anti-rotation pin is essential.

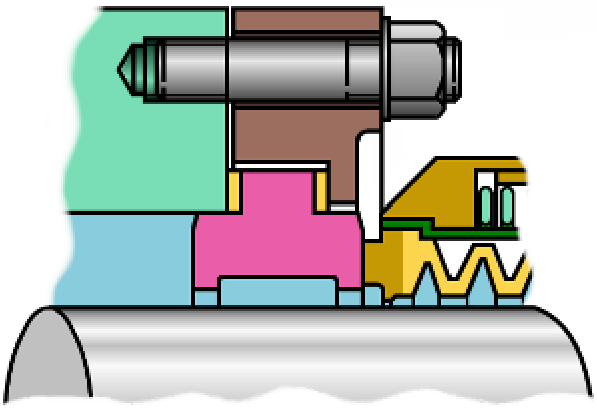


3. Cup Rings

Cup shaped elastomer tertiary seals are very popular in lower cost seal ranges. Hard, but brittle materials such as ceramic and silicon carbide can be used for the mating rings since they are of a simple rectangular section. Fitting tolerances, however, must be precise.

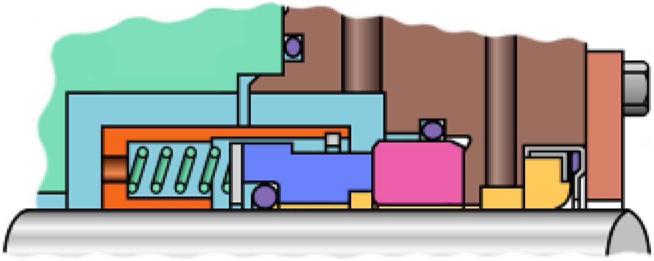
4. Gaskets or Flat Joints

Clamp plates are used to hold flanged mating ring designs against the seal chamber face. Clamped mating rings are used mainly where the mechanical seal is mounted external to the seal housing. Non-clamped mating rings would be pushed out of their housings by the hydraulic pressure. Care must be taken when tightening the clamping nuts to prevent distortion of the mating ring. Note this picture shows the mating ring with a gasket on either side of the flange. The gasket between the mating ring and the clamping plate is necessary to protect brittle materials such as ceramic. It is not required on metal mating rings.



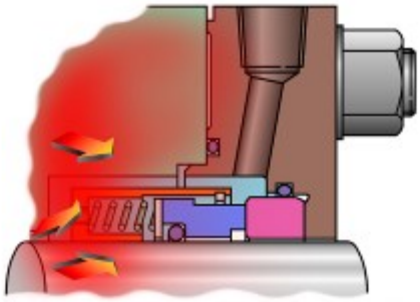
Mechanical Seal Principles

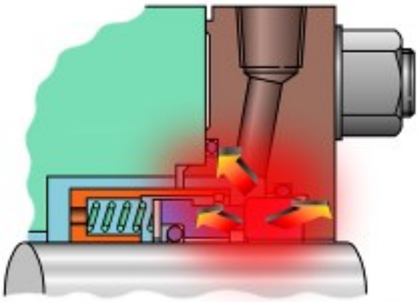
Seal Cooling



When the shaft is stationary no heat is generated by friction between the seal faces.

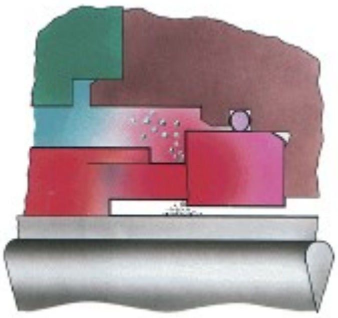
When pumping hot fluids the stationary seal is still affected by heat soak transmitted by conduction along the shaft through the pump body, and through the pumped liquid.





When rotating, the seal generates heat by friction between the primary ring and mating ring. At slow speeds and low pressures the small amount of heat generated can be dissipated into the surrounding area.

Under heavier working conditions the seal will quickly overheat, as a suitable 'heat balance' cannot be achieved, and the sealed liquid can boil.

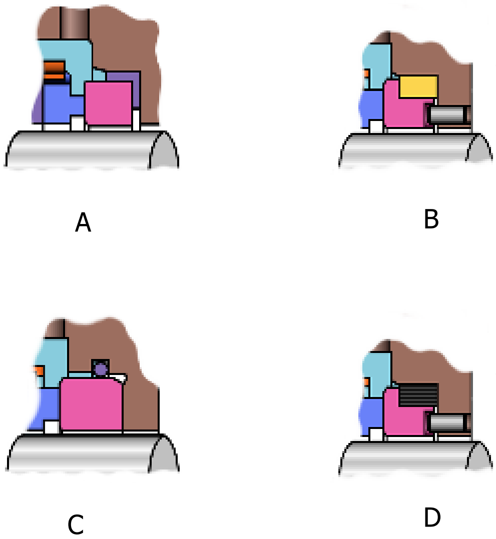


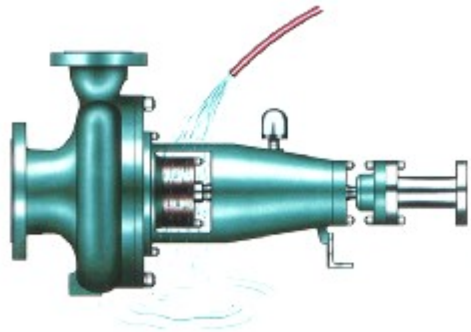
One way in which heat may be dissipated is by conduction i.e. through materials which are good conductors of heat.

This illustration shows various mating ring designs and some aid heat dissipation by conducting heat away into the cover plate.

A&B Seats .., Rubber and PTFE are bad conductors of heat.

C&D Seats ... intimate contact and Crane-foil offer good heat transfer paths.



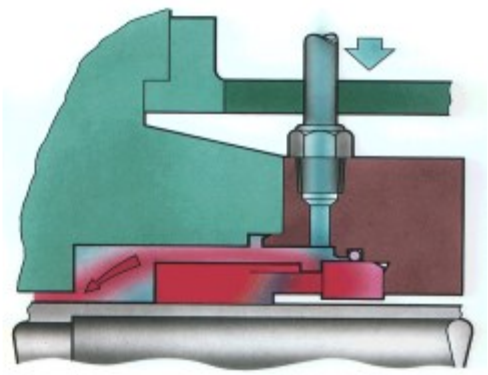


When a suitable heat balance cannot be maintained within the seal chamber. an additional means of cooling must be found,

A simple method of cooling the seal housing is to apply an external source of cool fluid:

A messy and nearly always unacceptable method but a quick solution for a short period.

The problem can be overcome by incorporating a cooling jacket to surround the seal chamber and contain the coolant fluid. This type of jacket can also be used for heating, using a supply of hot water or steam. where the pumped product is very viscous at ambient temperature. When using fluid in the jacket. it should enter at the bottom and exit at the top to ensure all air is vented, If using steam, this should enter at the top and exit from the bottom to ensure any condensate is drained away.



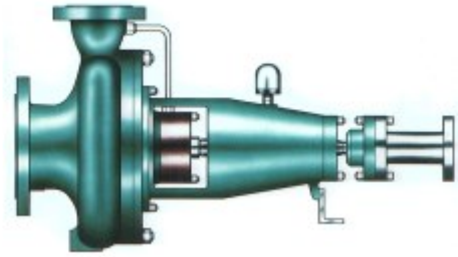
The normal way of dissipating heat generated by the seal faces is to recirculate the fluid which is being pumped. Fluid is taken from the high pressure (discharge) side of the pump and piped directly to the seal faces. This carries heat away from the seal chamber and back to the low pressure (suction) side of the pump. It is important that the entry to the seal chamber is at the primary seal.
Where a pump is mounted vertically, it is normal to use reverse circulation, connecting the pipe from the seal chamber to the suction side of the pump. This ensures that the seal chamber is fully vented and that the seal faces are cooled.

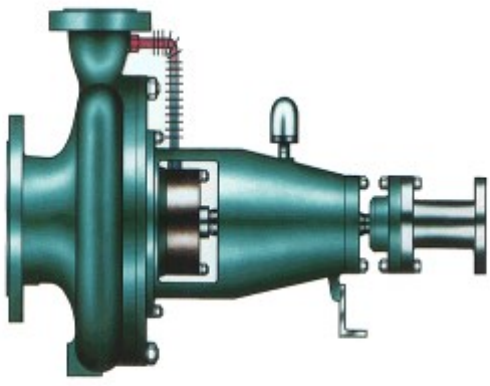
We can assume that a pump is operating correctly when a heat balance is achieved which:

a) guarantees long life to pump and seal components.

b) ensures that the product is stable and well below its boiling point at the seal faces.

c) does not waste energy by removing too much heat from the process.





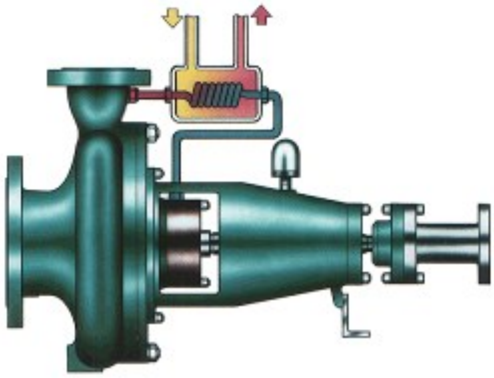
Often we find that insufficient heat is dissipated by simply recirculating the pumped fluid because the fluid itself is too close to its boiling point at suction pressure.

If the extra heat needed to be removed is relatively small. we can improve the heat dissipation characteristics of the recirculation pipe work by increasing its surface area.

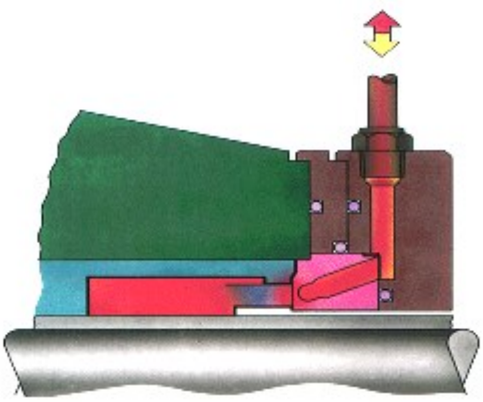
This can be done by either finning, or extending it or both.

If we find that we cannot remove sufficient heat by air cooling the recirculation pipe work then we must consider a heat dissipater,

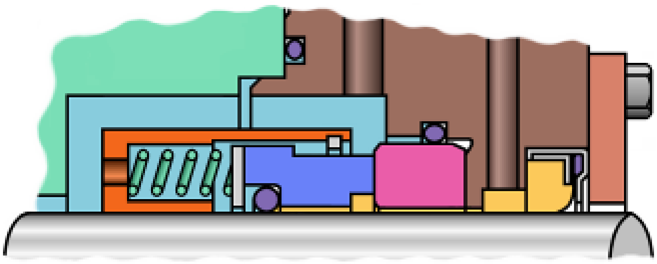
This device works on the same principle but normally uses water to dissipate the heat. Heat dissipaters can be manufactured in various sizes depending on the amount of heat which we need to remove.

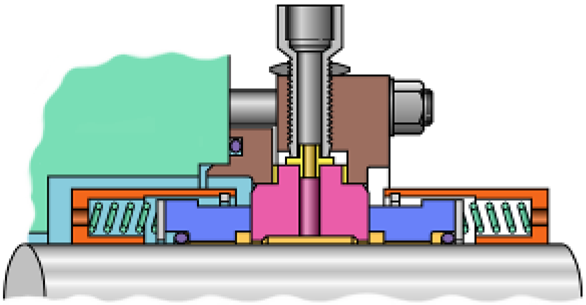


Cooling can be applied to the mating ring itself, This method of cooling is particularly successful when the pumped fluid is stable under pressure within the seal housing. but is brought close to its vapour point when it passes across the faces to atmosphere, (In general. it can be assumed that the fluid film will increase by around 20 deg C due to the friction at this point).
This type of arrangement is also used to heat the area around the faces when the viscosity of the pumped fluid increases as it cools e.g., Bitumen. Bunker Fuel, by injecting steam or hot water.



Another method of cooling the area around the faces is to fit an auxiliary sealing device to the atmospheric side of the seal housing and quench the bore of the seal and mating ring with cool fluid.

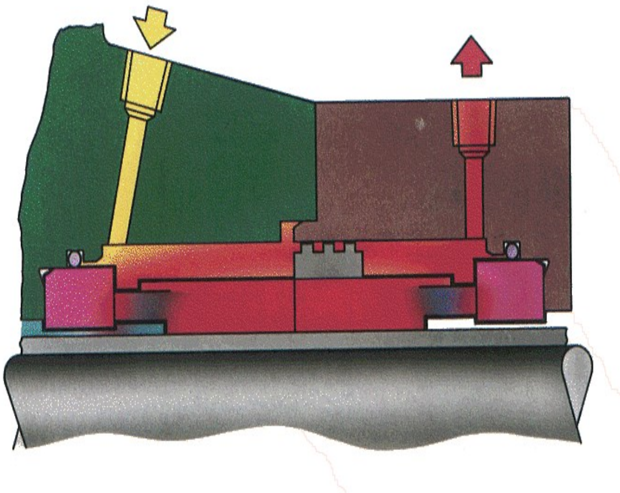




The 'U' type mating ring can be used to provide an atmospheric or low pressure barrier of, coolant between 'face to face' seals, Here, the maximum pressure would normally be 1 bar g to prevent the opening of the outboard seal faces.

Where double seals are required, although the sealant fluid is initially cool. it is normally continuously circulated, and constantly gains heat. Heat is being generated by two seal faces and also transmitted by heat soak from the product,

The seal area becomes very hot and sealant must be circulated to dissipate heat.



Often natural thermosyphon and sometimes the addition of a pumping ring is sufficient to provide an adequate circulation.

Where conditions are not good for thermosyphon a small circulating pump can be used.

This is often the best option as a guaranteed circulation is provided when the process pump is either static, or rotating.

Mechanical Seal Principles

Hydraulic Balance

There are two sources of heat in a seal chamber:

- Heat soak from the product.
- Heat generated by the seal.

In order to reduce the amount of heat generated by the seal friction, we need to recognise the factors involved in the creation of heat at the seal faces:

- Size of the seal.
- Rotational speed (revolutions per minute).
- Temperature of the fluid.
- Nature of the service fluid.
- Surface finish of the seal running faces.
- Materials of construction.
- Pressure acting on the seal faces.
- Surface area of the running track.

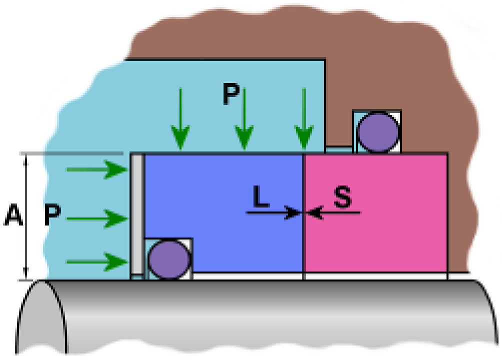
The first two items are normal process equipment requirements which we are unable to change.

We do have a controlling influence on the remaining items at the sealing area. The way we can influence the nature of the service fluid, its temperature and surface finish of the faces are covered elsewhere.

By "balance", seal manufacturers mean balancing the hydraulic forces closing the seal faces and squeezing the fluid film, with an opposite force that will off-load the faces to some degree to maintain a stable lubricating film and lower the face stresses at the running track. This will reduce the amount of heat generated and thus increase the life of the seal.

In this section, we look at the effects of hydraulic pressure on the compressive stress at the fluid film, and see how this can be reduced by changing the shape of the primary ring.

Primary Ring and Mating Ring widths equal



L = Total Load
S = Compressive Stress at Fluid Film
P = Hydraulic Pressure (e.g., 10 kg/cm²)
A = Hydraulic Area (e.g., 2 cm²)
a = Fluid Film Area (e.g., 2 cm²)

$L = P \times A$ $\qquad$ $L = 10\ \text{kg/cm}^2 \times 2\ \text{cm}^2 = 20\ \text{kg}$

$S = \frac{L}{a} = \frac{P \times A}{a}$ $\qquad$ $S = \frac{20\ \text{kg}}{2\ \text{cm}^2} = 10\ \text{kg/cm}^2$

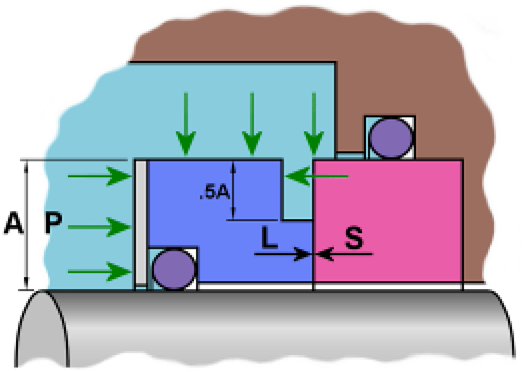
$S = P$ $\qquad$ $10\ \text{kg/cm}^2 = 10\ \text{kg/cm}^2$

When Hydraulic Area and Fluid Film Area are the same, Fluid Film Stress = Hydraulic Pressure

In order to keep calculations simple, in the following diagrams, we have ignored the force provided by the springs. Only the items directly affected by the hydraulic pressure, to change face stresses, are shown. It should be noted that these diagrams are designed to explain the basic theory behind balanced seals, and that figures are not taken from any actual seal designs.

When the hydraulic area, or area over which the hydraulic pressure acts to close the seal faces, is equal to the fluid film area, the fluid film stress is equal to the hydraulic pressure.

Fluid Film Area
reduced by 50%



L = Total Load
S = Compressive Stress at Fluid Film
P = Hydraulic Pressure (e.g., 10 kg/cm²)
A = Hydraulic Area (e.g., 2 cm²)
a = Fluid Film Area

Closing Load = P x A = 20 kg
Opening Load = P x .5A = 10 kg
Total Load L = (P x A)-(P x .5A) = 20 kg - 10 kg
= P x .5A = 10 kg

$$S = \frac{L}{a} = \frac{P \times .5A}{.5A} = \frac{10\ \text{kg}}{1\ \text{cm}^2} = 10\ \text{kg/cm}^2$$

$S = P$ $\qquad$ 10 kg/cm² = 10 kg/cm²

Fluid Film Stress = Hydraulic Pressure

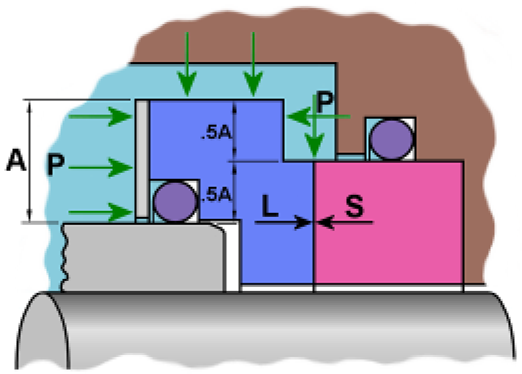
By machining the outer diameter of the primary ring, an off-loading force is introduced, in this case a force of approximately 50% of the closing force. However, the fluid film area is also reduced by 50%, so the fluid film stress is still the same as the hydraulic pressure.

We need to find a way of reducing the closing forces without reducing the fluid film area.

This can be achieved by keeping the faces as they were in the original diagram, but introducing a step into the primary ring, in this case by installing a sleeve on the shaft. We have retained the same closing and off-loading forces as above, giving a net closing force of 50%, but have retained the original face width. The fluid film stress is now theoretically half the hydraulic pressure.

We will see later that the hydraulic pressure *within* the fluid film is approximately 50% of the seal chamber hydraulic pressure. With a fluid film stress also reduced to 50% the faces will become unstable and will open, allowing excessive leakage.

Hydraulically Balanced Seal
(50% Balanced)



L = Total Load
S = Compressive Stress at Fluid Film
P = Hydraulic Pressure (e.g., 10 kg/cm²)
A = Hydraulic Area (e.g., 2 cm²)
a = Fluid Film Area = A (e.g., 2 cm²)

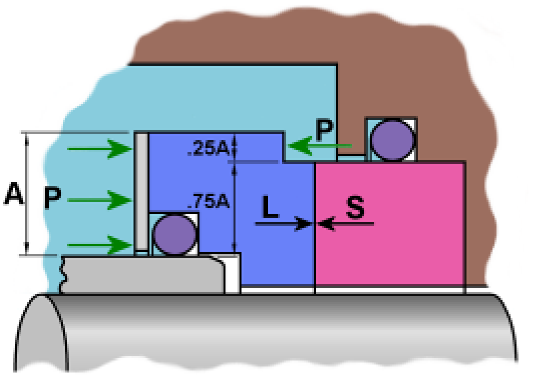
Closing Load = P x A = 20 kg
Opening Load = P x .5A = 10 kg
Total Load L = (P x A)-(P x .5A) = 20 kg - 10 kg
= P x .5A = 10 kg

$$S = \frac{L}{a} = \frac{P \times .5A}{A} = \frac{10\ \text{kg}}{2\ \text{cm}^2} = 5\ \text{kg/cm}^2$$

$$S = \frac{P}{2}$$

Fluid Film Stress = Half the Hydraulic Pressure

Hydraulically Balanced Seal
(75% Balanced)



L = Total Load
S = Compressive Stress at Fluid Film
P = Hydraulic Pressure (e.g., 10 kg/cm²)
A = Hydraulic Area (e.g., 2 cm²)
a = Fluid Film Area = A (e.g., 2 cm²)

Closing Load = P x A = 20 kg
Opening Load = P x .25A = 5 kg
Total Load L = (P x A)-(P x.25A)= 20 kg - 5 kg
= P x .75A = 15 kg

$$S = \frac{L}{a} = \frac{P \times .75A}{A} = \frac{15\ kg}{2\ cm^2} = 7.5\ kg/cm^2$$

S = .75P

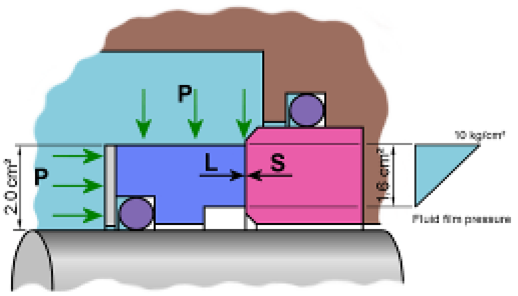
Fluid Film Stress = 75% of Hydraulic Pressure

The normal level of hydraulic balance used by seal manufacturers for a pusher-type seal is approximately 75% of the original stress. A stepped shaft or sleeve is necessary to accommodate a balanced pusher seal. In many seal designs, the step is built into the sleeve supplied with the seal, such as in a cartridge seal design. In that case there would be no need to step the pump shaft or sleeve.

If the high pressure is on the outside diameter of the seal faces, 75% of the area of the seal faces (or fluid film area) will be outside the diameter of the sleeve on which the o-ring (or other dynamic secondary seal) slides. This sliding diameter is known as the "Balance Diameter".

The two diagrams below illustrate pusher seal hydraulic balance using more realistic figures. The hydraulic pressure within the fluid film has also been taken into account. A direct comparison with a typical unbalanced seal primary ring and a balanced seal primary ring is given. It is assumed for these calculations that the pressure-drop across the fluid film is linear. Using the example figures, it can be seen that an "unbalanced" seal has a fluid film stress of less than 100%, and that a 75% balanced seal has a fluid film stress considerably lower than 75% of the unbalanced seal - in this example it is only a third of the stress compared with the unbalanced seal.

Typical Loading in an Operating
Hydraulically Unbalanced Seal



L = Total Load
S = Compressive Stress at Fluid Film
P = Hydraulic Pressure = 10 kg/cm²
A = Hydraulic Area = 2.0 cm²
a = Fluid Film Area = 1.6 cm²

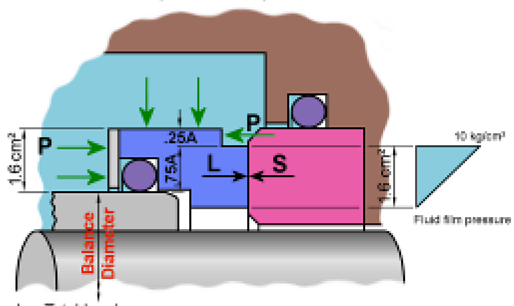
Closing Load (Lc) = P x A = 10 kg/cm² x 2.0 cm² = 20 kg
Opening Load (Lo) = $\frac{P}{2}$ x a = 5 kg/cm² x 1.6 cm² = 8 kg

Total Load (L) = Lc - Lo = 20 kg - 8 kg = 12 kg

$$S = \frac{L}{a} = \frac{12\ kg}{1.6\ cm^2} = 7.5\ kg/cm^2$$

Fluid Film Stress = approximately 75% of Hydraulic Pressure

Typical Loading in an Operating
75% Hydraulically Balanced Seal



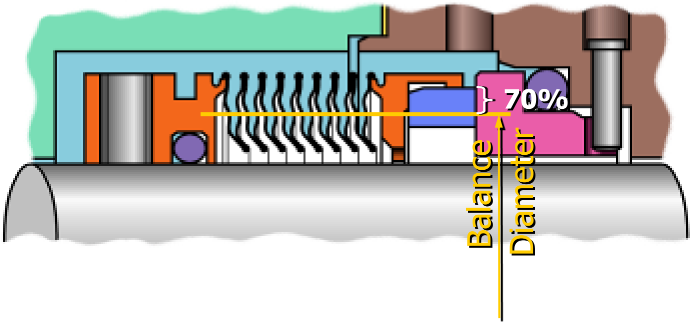
L = Total Load
S = Compressive Stress at Fluid Film
P = Hydraulic Pressure = 10 kg/cm²
A = Hydraulic Area = 1.6 cm²
a = Fluid Film Area = 1.6 cm²

Closing Load (Lc) = P x A = 10 kg/cm² x 1.6 cm² = 16 kg
Opening Load (Lo1) = P x .25A = 10 kg/cm² x 0.4 cm² = 4 kg
Opening Load (Lo2) = $\frac{P}{2}$ x a = 5 kg/cm² x 1.6 cm² = 8 kg

Total Load (L) = Lc - Lo1 - Lo2 = 16 kg - 4 kg - 8 kg = 4 kg

$$S = \frac{L}{a} = 4\ kg = 2.5\ kg/cm^2$$

Fluid Film Stress is approximately 25% of Hydraulic Pressure



Metal bellows seals and, to a lesser extent, elastomer bellows seals, are inherently balanced without the need for a step in the shaft. With metal bellows seals, as pressure acts in all directions, there is an equal hydraulic on-loading and an off-loading effect on the front and back faces respectively of each bellows convolution. The effective "balance diameter" is found to be approximately half-way between the inside and outside diameters of the bellows convolutions. It is generally considered that a more accurate balance diameter is calculated as the root-mean-square of those diameters:

$$D_{Balance} = \sqrt{(0.5(OD^2 + ID^2))}$$

This equation applies with zero differential pressure across the bellows. Differential pressure tends to deform the bellows in such a way as to move the balance diameter towards the low pressure side, thus increasing the balance ratio. The balance ratio for edge-welded mechanical seals is usually approximately 70%, and this value increases as the differential pressure increases.

Benefits of Hydraulically Balanced Seals

Reduced heat generation
- Ideal for unstable and low-Specific Gravity fluids
- Less heat to be dissipated
- Less cooling required

Reduced rate of wear
- Longer life
- Less down-time

~~Reduced power cons~~umption
- Lower running costs

Increased pressure range
- Simple modification required
- Three times or more pressure capability than unbalanced seals

Note: API 682 specifies that balanced seals should be used for ALL applications.

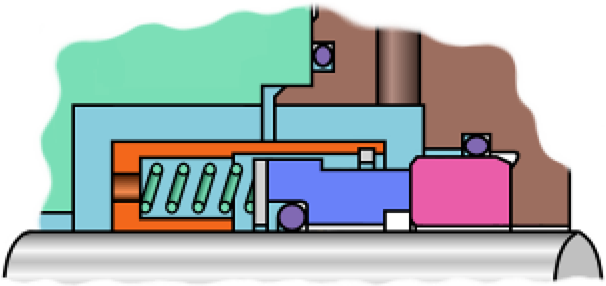
In all calculations in this section, we have ignored the spring force. This is produced by a wide variety of methods, including:

- Single coil-springs
- Multiple coil-springs
- Single wave-springs
- Nested wave-springs
- Continuous coil wave-springs
- "Rat-trap" springs
- Metal diaphragms
- Edge-welded metal bellows
- Formed metal bellows
- Heavy-section elastomer bellows

Spring loads will vary depending on the design of the mechanical seal and the application, but the majority of standard mechanical seals for pump applications will have a spring load calculated to give a fluid film area stress of 1.5 to 2.0 bar. This will ensure that the seal faces will not open should there be a short period of partial vacuum in the seal chamber, for example at start-up. Standard single mechanical seals cannot withstand long periods of partial vacuum in the seal chamber as this causes air to be drawn across the seal faces and dry-running occurs. This would result in very early seal failure due to overheating and excessive wear. See the later section on failure analysis.

Environment of a Mechanical Seal

Multiple Seals

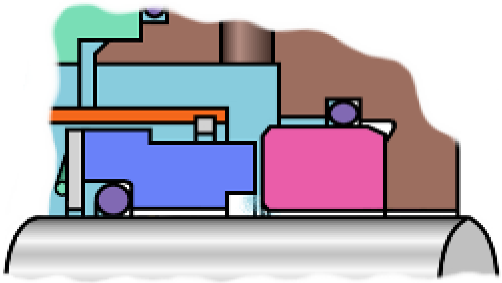


In order to achieve maximum seal life and minimum leakage the following conditions should be targeted:

- Clean, pure, non-abrasive process fluid, present at all times.
- Moderate, non-varying pressure, viscosity and temperature.
- Steady and continuous shaft speed.

A single seal may not be enough

All conventional single mechanical seals utilise the process fluid to provide the lubricating interface film between the running faces. As this fluid film travels across the interface, it drops in pressure, but will travel right across the interface in the form of fluid if the seal is operating correctly. By the time it gets to the atmosphere it will have turned into minute vapour droplets. It can be seen, therefore, that mechanical seals do leak when they are operating correctly, even though only by an extremely small amount, measured in parts-per-million.



There are three basic reasons why a single seal may not be enough.

1. Safety

Although the leakage should not be visible, the small amount of vapour released can build up which is unsafe if the fluid is dangerous to people, the plant or the environment. For safety, we must consider using multiple seals if the:

- Fluid is toxic
- Fluid is flammable
- Fluid is highly corrosive
- Fluid is environmentally unfriendly
- Fluid must be isolated for any other reasons

2. Function

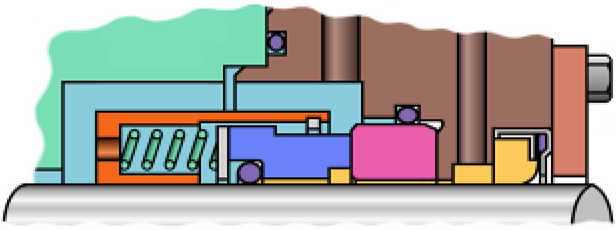
A single seal will not function correctly if the process fluid is not suitable for lubricating the running faces. A separate fluid must be supplied to cool and lubricate the mechanical seals. A single seal will not function correctly if the:

- Fluid changes state (e.g., solidifies, crystallises, polymerises)
- Fluid is not a good lubricant or is a gas
- Fluid is unstable
- Fluid contains dissolved gases
- Pumping temperature is such that ice may form round the seal area
- Operations cannot avoid dry-running due to process conditions and/or operating procedures

3. Cost

A second seal may be required as a back-up to reduce the **cost** of a failure where:

- Process fluid is very expensive
- Machine is critical to the plant operation



A secondary containment device may be used external to the mechanical seal to improve safety or to improve the performance of a single seal arrangement. These devices can contain a quench fluid injected between the process fluid and atmosphere to carry away deposits and dilute leakage, control any leakage when a seal fails, and isolate the product from the atmosphere.

Secondary containment devices are discussed later in this manual.

A seal arrangement incorporating a secondary containment device is commonly used for duties with these properties and conditions:

Crystallising solution	Sodium hydroxide (Caustic)
Polymerising solution	Formaldehyde
Decomposing solution	Hot oil
Cryogenic conditions	Propane
Sterile conditions	Food and Pharmaceuticals

Simple secondary containment devices can provide a solution to a problem. However, non-contacting secondary containment devices may leak unacceptably, and contacting secondary containment devices are more likely to fail before the main mechanical seal. Where a more robust solution is required, multiple seals are usually the answer.

The vast majority of multiple arrangements involve the use of two mechanical seals. This manual will only cover this type of arrangement.

Arrangements

There are two basic multiple seal arrangements:

Unpressurised arrangement
- Low pressure buffer fluid between the two seals
- High integrity secondary containment
- Inboard seal is lubricated by the process fluid

Pressurised double arrangement
- Pressurised barrier fluid between the two seals
- Inboard seal is lubricated by the barrier fluid

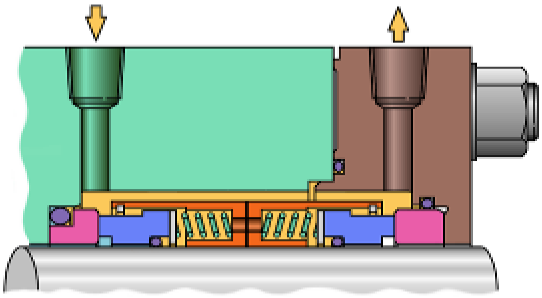
There are many varied designs and applications of mechanical seals to produce a multiple seal arrangement. For example, the mechanical seals can be in four orientations:

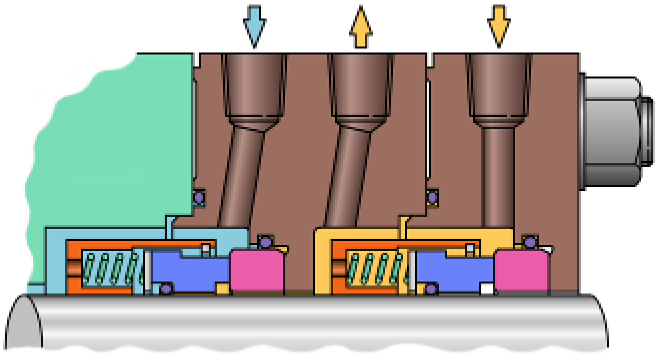
- Face-to-back
- Back-to-back
- Face-to-face
- Concentric

The purpose of this section is to explain when and how multiple seals should be used. We will only examine a few more common arrangements, with fluid (not gas) between the seals. In some of these arrangements, the mechanical seal may be rotating or stationary, depending on the application.

There are various types of multiple seal arrangements. Four common multiple seals are shown here.

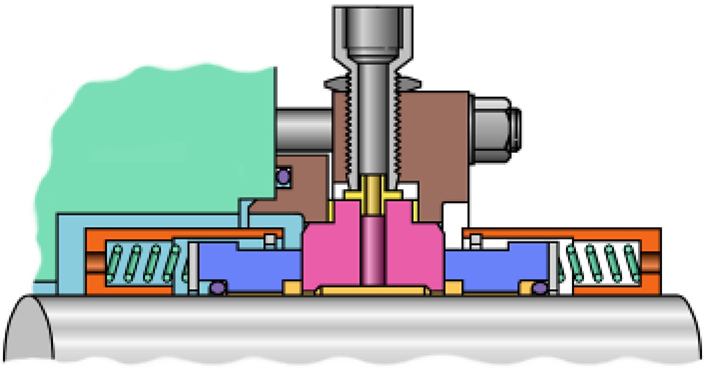
A. BACK-TO-BACK double arrangements normally contain a barrier fluid at least 1 bar or 10% above the process pressure.

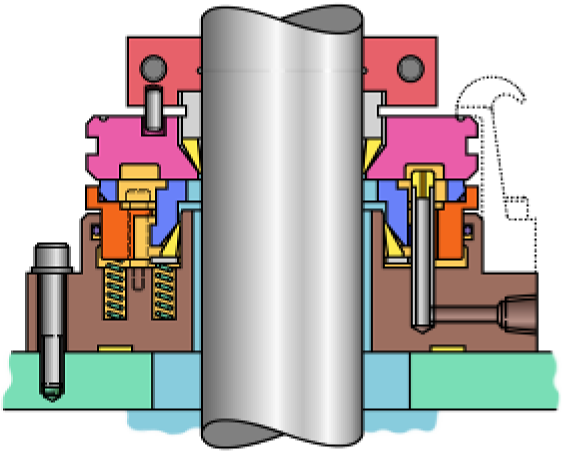




B. TANDEM arrangements are normally unpressurised, but can contain a pressurised barrier fluid when used to reduce differential pressures on high pressure applications.

C. FACE-TO-FACE arrangements, with rotating seals and stationary mating rings are used with low pressure sealant systems often on aseptic duties.





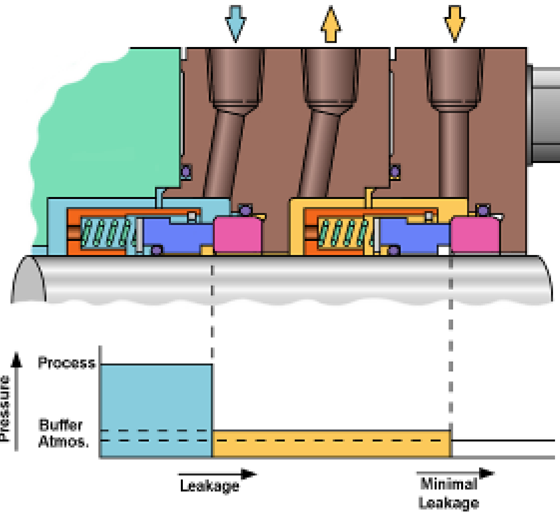
D. CONCENTRIC FACE arrangements are normally pressurised similar to double back-to-back arrangements.

Unpressurised Arrangements

Fluid is circulated between the seals from an external supply at a pressure less than the pressure in the seal chamber. The inboard seal is lubricated by the process fluid (or a fluid injected into the seal chamber and on into the pump from an external source as a flush using API Piping Plan 32), and the outboard seal is lubricated by the buffer fluid, forming a high integrity secondary containment seal. This arrangement uses API Piping Plan 52.

<u>Face-to-Back Seals (Tandem)</u>

A tandem multiple seal is usually arranged with the inboard seal identical to a single seal installation. This seal is lubricated by the process fluid and a recirculation line is usually fitted (API Piping Plan 11). A further seal, similar to the inboard seal, is installed facing the same way, in 'tandem', and they are separated by a suitable buffer fluid which is vented to atmospheric pressure (often to a safe area, vapour recovery system or flare stack).

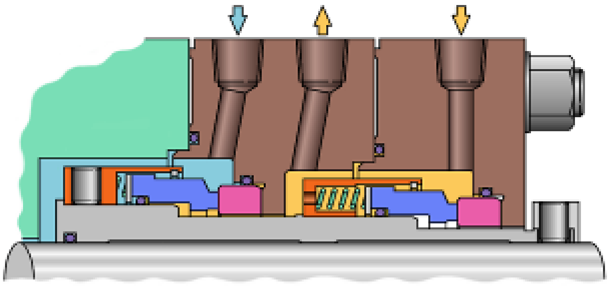


The resulting arrangement is similar to that of a secondary containment device, except that the outboard seal is capable of withstanding the full process conditions if the inboard seal fails. This arrangement is commonly used when sealing process fluids that are suitable to lubricate the faces but any leakage would create a hazard, or may change state or become unstable in contact with air.

The inboard seal is operating under full product pressure, and is lubricated by the process fluid, so if the buffer fluid is circulated from an external reservoir there will be a gradual build-up of contamination in the buffer fluid. This will need to be drained and replaced occasionally. The frequency will be determined by factors such as the leakage rates and hazardous nature of the process fluid. However, if the vapour pressure of the process fluid is higher than the buffer fluid pressure, any process fluid which leaks into the buffer fluid will flash in the sealant supply pot and the vapour can escape into the vent system.

The outboard seal is operating under low pressure conditions, and is lubricated by the cool, clean, low viscosity buffer fluid, which is supplied using a system to API Piping Plan 52. It is acting in the same way as a secondary containment device. The inboard seal, therefore, is likely to fail first and then the outboard seal can operate under full process conditions, if necessary, until it is convenient to shut down the machine. The level in the sealant system will rise, alerting operators to the fact that the inboard seal has failed. The level switch will also operate a valve in the vent line to prevent any product loss.

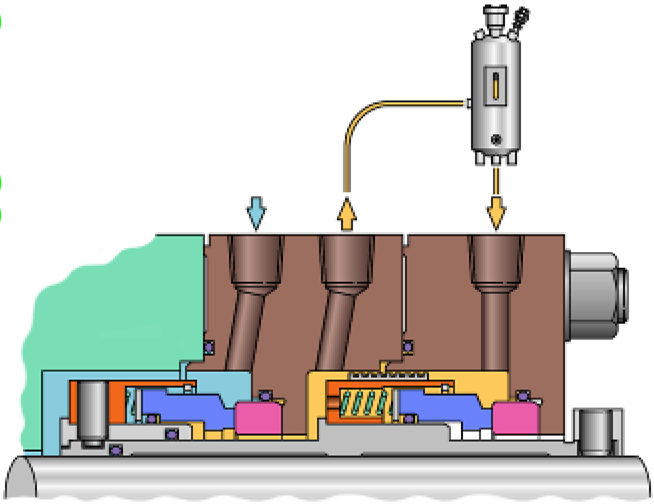
In practise, it is very rare for a mechanical seal to fail suddenly. Usually, leakage gradually increases to an unacceptable level, and this is detected by the level switch in the sealant system.



Here is a typical tandem face-to-back cartridge seal, with two balanced pusher seals of similar design and operating capabilities. Note the API Piping Plans 11 and 52. The buffer fluid is sometimes circulated around the seals by a thermosyphon system. Where this is not acceptable or suitable, the buffer fluid can be forced round the system by a pumping scroll, pumping ring, or a small pump mounted in the line between the sealant system pot and the inlet to the seal chamber.

Note that the balance steps under the seals can cause the seal sleeve to become quite thick in section.

To overcome this problem, we can reduce the overall radial cross-section of the cartridge assembly by using a two-piece sleeve, where this is acceptable, thus enabling the use of the same size seal inboard and outboard. This picture also shows a diagrammatic buffer fluid supply pot, and a pumping scroll to force the circulation of the buffer fluid. The buffer fluid supply pot is fitted with a level switch which can detect failure of either seal:



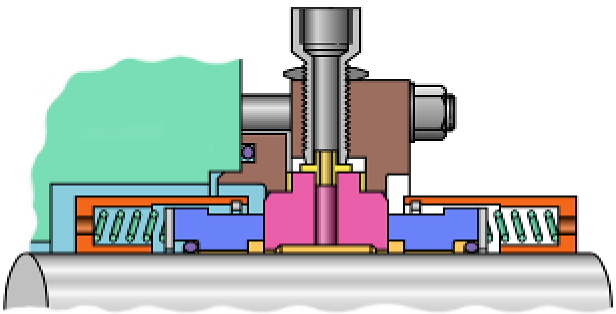
rise in level = inboard seal failure
drop in level = outboard seal failure

With seals arranged in this back-to-face orientation we can be sure that the higher pressure is always assisting the springs to keep the seal faces closed. In normal operation, the process pressure is holding the inboard faces together, and the small amount of pressure in the barrier fluid system is assisting the springs of the outboard seal. If the inboard seal fails, the process pressure is then applied to the outside diameter of the outboard seal, and again assists the springs to keep the seal faces together. (Remember "Rule 1"?).

Where the outboard seal of a face-to-back unpressurised arrangement is being used purely as a secondary containment device (i.e., not also required as a failure back-up) the outboard seal may be of a lower cost, simpler design. For example, where the process fluid is likely to crystallise on contact with the atmosphere, a low-pressure low-flow water quench can be contained by a simple elastomer bellows seal.

For very high pressure applications, a Tandem seal arrangement may be used to distribute the process pressure between inboard and outboard seals. The buffer fluid is introduced between the seals usually at half the pressure of the process fluid at the inboard seal, thus the full process pressure is shared between the two seals. This arrangement is commonly found on high pressure pipeline duties.

Face-to-Face Seals (Rotating Seals)



The main advantage of this arrangement is that it is relatively short, so will often fit where a face-to-back seal would not. Because the outboard seal is internally pressurised the arrangement is only suitable for low pressure buffer fluid, unless a seal with reverse pressure capability is used. If the outboard seal was a conventional unbalanced pusher seal the buffer fluid pressure would have to be limited to a maximum of about 1 bar g, otherwise the seal faces would open, causing excessive leakage to atmosphere. Because of this, the arrangement is essentially a single seal plus a secondary containment mechanical seal. The outboard seal would not be able to seal against full process pressure in the event of an inboard seal failure. This installation is sometimes used to protect an aseptic processes from contact with the atmosphere.

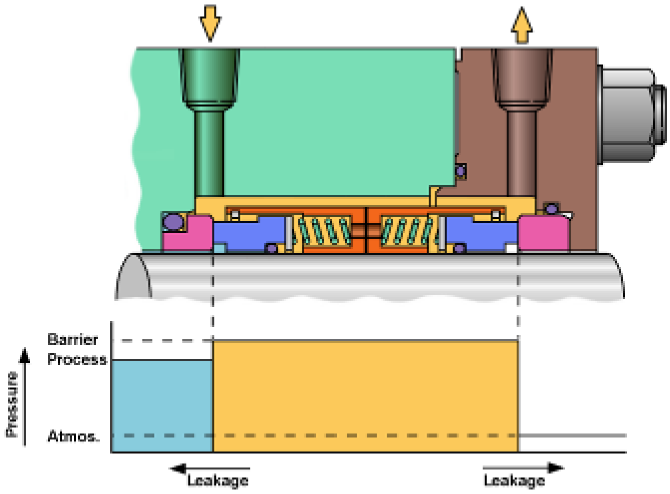
Pressurised Arrangements

Fluid is circulated between the seals from an external supply at a pressure higher than the pressure in the seal chamber. Both seals are lubricated by the barrier fluid. This arrangement uses API Piping Plan 53 (A, B, C or D) or Plan 54.

Back-to-Back Seals (Double)

The most common arrangement for a pressurised double seal is two seals installed back-to-back.

The barrier fluid, contained in a seal pot, is circulated round the system either by thermosyphon, by a pumping device installed in the buffer fluid space between the seals, or by an external pump.



The pressure between the seals should be maintained at a minimum of 1 bar or 10% (whichever is higher) above the maximum process fluid pressure at the inboard seal. This seal arrangement is not dependent on the process fluid to lubricate the inboard seal faces because the positive barrier fluid pressure ensures that the faces are lubricated by the barrier fluid.

This means that this double arrangement may be used to seal gases, unstable fluids, highly toxic, abrasive, corrosive, and viscous fluids.

There are three basic API Piping Plans for this seal arrangement **(See Appendix B for all API Plans)**:

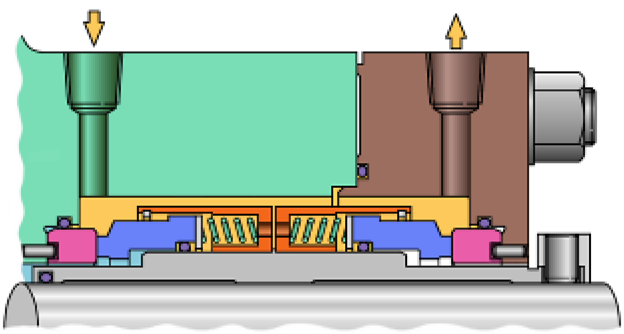
API Plan 53A
This is the basic pressurised external barrier fluid reservoir, supplying clean fluid between the seals. The pressure is maintained by an external supply of nitrogen gas.

API Plan 53B
This plan differs from Plan 53A in that pressure is maintained in the seal circuit through the use of a bladder-type accumulator. This system isolates the nitrogen supply from the barrier fluid. This is essential at higher pressures as the nitrogen could go into solution in the barrier fluid. The nitrogen would come out of solution where there is a drop of pressure in the system, namely at the seal fluid film interface. (Like opening a carbonated drinks bottle!). This would cause severe dry running and flashing at the seal faces.

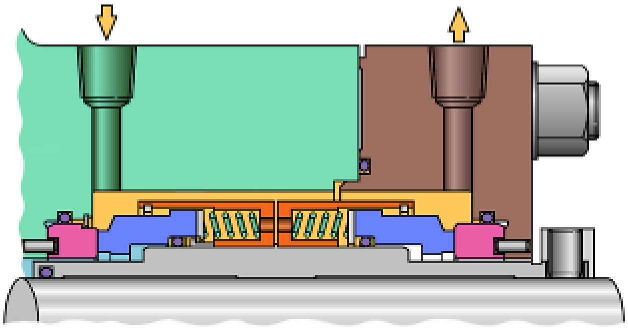
API Plan 53C
This plan uses as piston-type accumulator to maintain pressure of the barrier fluid above the process fluid pressure. This system is ideal where there are large fluctuations in the process pressure as the barrier pressure is kept at a set differential pressure above the process pressure.

A pressurised double seal arrangement using Plan 53 is dependent on having the barrier fluid pressure maintained at the proper level. If the barrier fluid pressure drops, the system will begin to operate like a Plan 52 tandem arrangement due to the pressure reversal. Both seals are being lubricated by the barrier fluid, but the outboard seal is likely to fail first as the pressure differential is highest at this seal. Seal failure will result in a rapid loss of barrier fluid and pressure, so the machine must be shut down immediately there is a seal failure, or loss of barrier fluid pressure for any other reason. There are ways of preventing pressure reversal causing seal faces to open, but we must remember why it was necessary to select a pressurised double seal in the first place. It is always better to eliminate the cause of pressure reversal if possible.

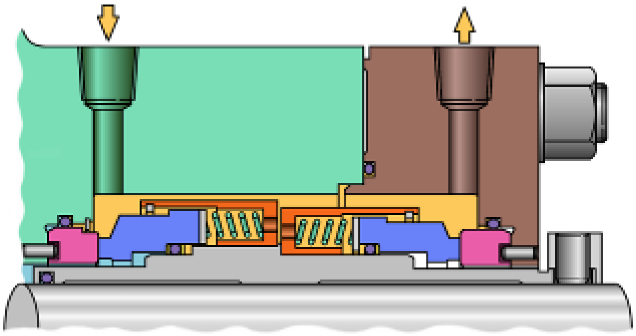
Reverse Pressure



If the barrier fluid pressure was lost from this seal arrangement, the process pressure would push the inboard balance diameter o-ring back against the thrust ring, compress the springs, and the o-ring could become dislodged and the faces open. The higher pressure would get up behind the mating ring and blow it out of its housing. Even if a short period of pressure reversal is experienced when the shaft is rotating, the mating ring could become dislocated from its ant-rotation pin, and it would rotate. If the barrier fluid pressure was subsequently restored, it is unlikely that the mating ring would correctly locate back onto this pin, creating massive misalignment.

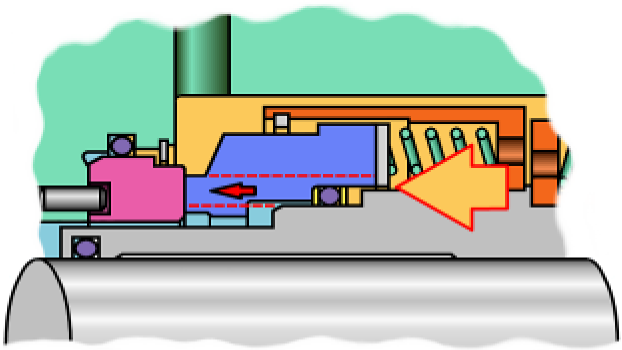


By isolating the springs from the o-ring by a simple spacer sleeve fitted between the o-ring and the seal retainer, and retaining the mating ring in its housing, the problem is overcome. The balance diameter shifts when the o-ring moves back, from the shaft/ sleeve diameter to the outside diameter of the o-ring/counterbore of the primary ring and the positive pressure in the bore of the primary ring acts to keep the faces closed. The retaining washer in front of the mating ring prevents this component from moving out of its housing and becoming dislocated. Note the anti-extrusion rings installed on both sides of the inboard secondary seal o-ring. When designing mechanical seals to take pressure reversals, consideration must be given to the bursting strength of the primary ring.

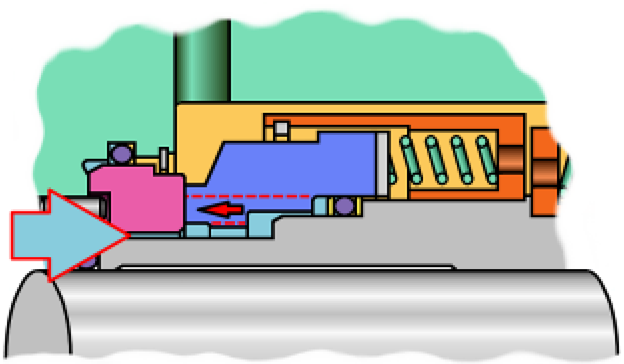


In many modern seal designs, the small sleeve mentioned above is replaced by a step on the seal sleeve to produce the "double-balance" feature. In this picture, the cross-section of the primary ring has also been increased to withstand the higher internal pressure which occurs with pressure reversals.

Double-Balance Feature

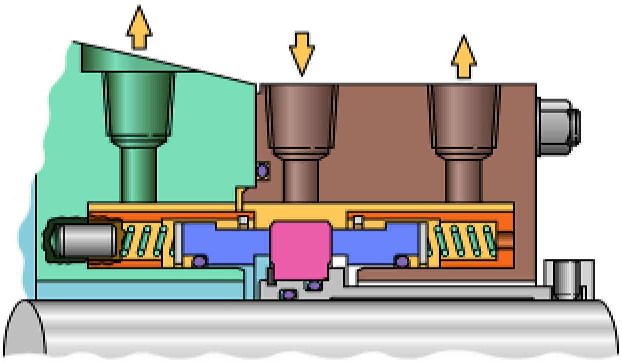


When a balanced pusher seal is operating correctly as the inboard seal of a pressurised back-to-back arrangement with the higher barrier fluid pressure at the outside diameter of the seal faces, the "balance diameter" is at the sliding contact surface of the inner diameter of the secondary seal o-ring. The hydraulic pressure effectively acts over the area between this diameter and the outside diameter of the running face of the primary ring to hold the faces together. This area is usually about 75% of the area of the running face in a pusher seal.



If the barrier fluid is lost, or the process pressure becomes higher than the barrier fluid for any reason, higher pressure is then on the inside diameter of the running faces. The process pressure will push the secondary seal o-ring back against the step on the sleeve. The sliding or balance diameter is now on the outside diameter of the o-ring. As pressure acts in all directions equally, about 75% of the running face area is still under hydraulic load, but now from the process fluid, and the seal faces remain closed. This simple design feature enables the secondary seal o-ring to move with pressure changes such that the balance diameter also moves to suit, and positive hydraulic pressure is always acting to keep the seal faces closed.

Face-to- Face Seals (Rotating Mating Ring)

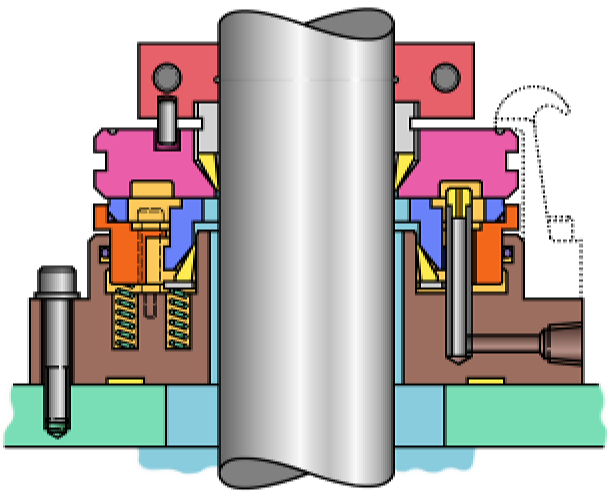


This is similar in appearance to a face-to-face rotating seals unpressurised arrangement, but actually operates as a pressurised double seal. The seals are stationary with a rotating mating ring, and are externally pressurised.

As there is only one mating ring (or one assembly containing two mating rings), the whole seal assembly can be made more compact.

This configuration is ideal for high speed applications as only the simple mating ring assembly rotates, and the seal unit components are stationary. Also, it is easier to ensure that the mating ring rotates squarely to the axis of the shaft as it is mounted on a long sleeve. At high speeds, it is difficult, or impossible, for the seal to flex to follow any misalignment, creating leakage and short seal life. For these reasons, this arrangement will be found in applications such as high speed water injection pumps.

Concentric Face Arrangement



Another pressurised double seal, this arrangement is designed for sealing top entry agitator vessels. It is not practical to use a back-to-back double seal arrangement because of large radial shaft deflections, and the length of such an arrangement.

A rotating mating ring runs against the stationary inboard and outboard primary rings which are lubricated by normally positively circulated pressurised barrier fluid. The outboard seal is mounted concentric to the inboard seal, considerably shortening the arrangement. This allows the agitator manufacturer to mount his bearings closer to the top of the vessel. The concentric seal can accommodate larger shaft run-out and deflections due to the increased clearance in the bore of the stationary seal housing.

Versions are available for glassed or enamelled vessels and shafts, where no metallic components of the seal come into contact with the process fluid.

Summary - Multiple Seals

Advantages of Pressurised Double and Unpressurised Tandem Arrangements

Pressurised Double Arrangement	Unpressurised Tandem Arrangement
Will seal abrasives, viscous, highly toxic and unstable media, gas, or vacuum Any leakage will be barrier fluid into the process, so no contamination of barrier fluid Outboard seal is likely to start to leak first, giving early warning of seal failure. No process leakage Will not be upset by dry-running conditions in the pump (e.g., starting with closed suction valve)	Simple sealant system - no pressurisation equipment required As product recirculation is commonly used at the inboard seal, less heat is accumulated As the buffer fluid is not pressurised, there is less heat generated, therefore less to remove Inboard seal is likely to fail first. Outboard seal can be designed to seal full process conditions, giving time to complete the process run before shutting down the pump

Disadvantages of Pressurised Double and Unpressurised Tandem Arrangements

Pressurised Double Arrangement	Unpressurised Tandem Arrangement
Complex sealant system requiring pressure supply, control and monitoring Need to remove additional heat generated by pressurised barrier fluid Higher power consumption Seal failure results in a rapid loss of barrier fluid. The system must usually be shut down on failure	Sealant system, although simple, must normally be vented to atmosphere, or safe area Buffer fluid will become contaminated by the process fluid Can only be used if the process fluid is suitable for lubricating the inboard seal faces

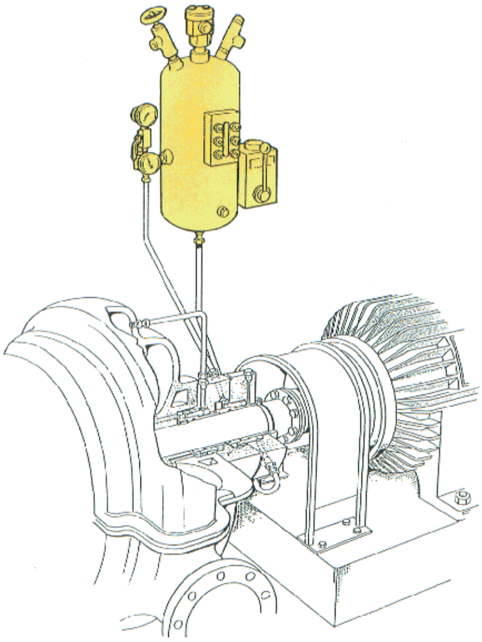
Recommended reading: American Petroleum Institute Standard 682: "Pumps - Shaft Sealing Systems for Centrifugal and Rotary Pumps".

Environment of a Mechanical Seal

Sealant Systems

Sealant systems are required to:

- Supply low pressure buffer fluid to unpressurised multiple seals
- Supply high pressure barrier fluid to pressurised double seals
- Supply low pressure quench to secondary containment seals
- Cool the buffer/barrier fluid where required
- Monitor seal condition/leakage



Typical Sealant System

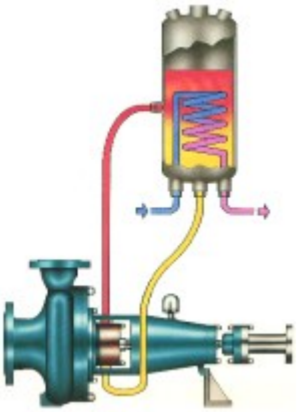
There are two basic types of sealant supply systems:

Vessel-type reservoir. Fluid is moved round the system by thermosyphon, pumping device in seal chamber, or external pump

Self-contained pumped system (from atmospheric reservoir)

Vessel Based Systems

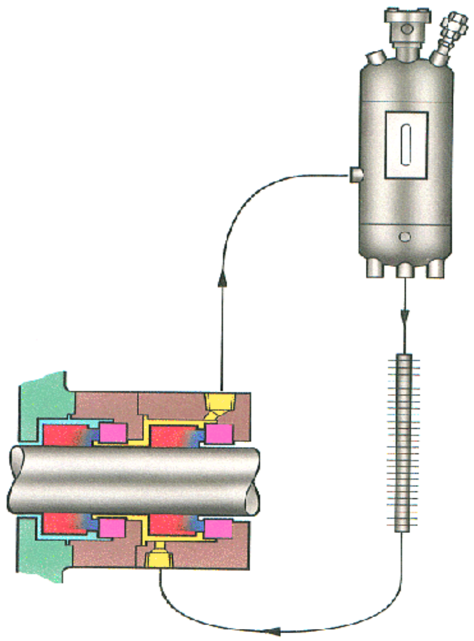
Systems are normally manufactured in stainless steel for maximum barrier fluid compatibility, with an average working capacity of around 6 litres and pressure ratings of 30 bar g, They are welded in accordance with ASME VIII, and have 1/2" screwed connections,



They should be mounted at least 0,6 meters above the seal chamber to increase the thermosyphoning effect and the pipe work should have the largest bore that is practical (normally 1/2") and have rising runs,

Where conditions are not good for a thermosyphoning, a pumping ring or scroll can be added in the seal chamber, or a small circulating pump can be added in the pipe work,

Vessel systems can usually only supply sealant to one seal chamber.



Typical Tandem Sealant System

The vessel normally contains a non-pressurised barrier fluid, Circulation of this fluid is usually by thermosyphoning,

If the inboard seal fails, the barrier fluid becomes pressurised against an orifice on the vessel. This pressure can be indicated using a pressure gauge or pressure switch,

A level switch can signal an alarm either with falling sealant level (outboard seal failure) or rising sealant level (inboard seal failure),

Tank Based Systems

These system normally incorporate an electric motor and remove the need for internal pumping device. A positive displacement pump can be used to pressurise and circulate the barrier fluid. The pressure is easily controlled by a needle valve in the lid-mounted manifold block.

The barrier fluid is filtered using the suction strainer and prevented from over-pressurisation by a relief valve . One system can supply more than one seal chamber.

Heat exchanger can be mounted in a between the pump suction and seal return chambers.

Vessel System Advantages:

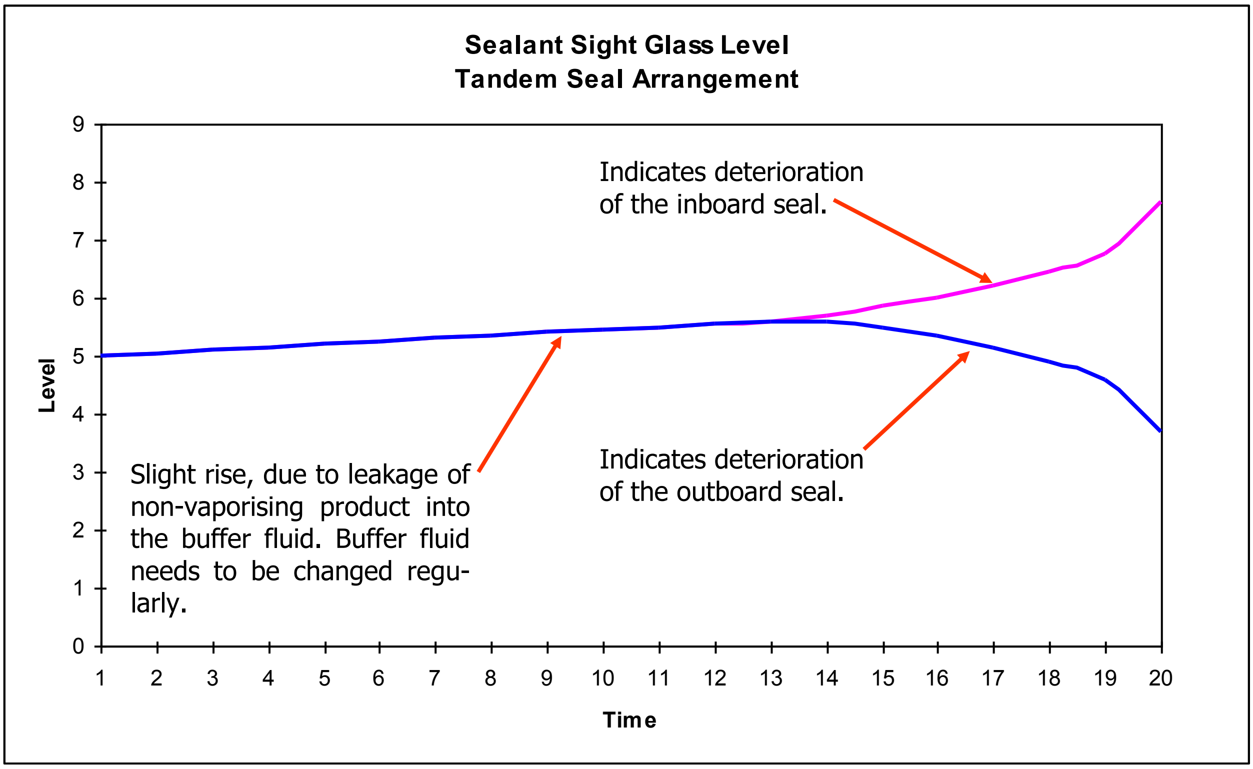
Simple maintenance	- System simply drained and flushed clean.
Long Life	- No moving parts to wear out.
Material traceability	- Simple construction using readily documented material.
Stainless Steel	- Variety of acceptable barrier fluids,
Barrier fluid isolation	- Barrier fluids may be isolated from atmosphere,

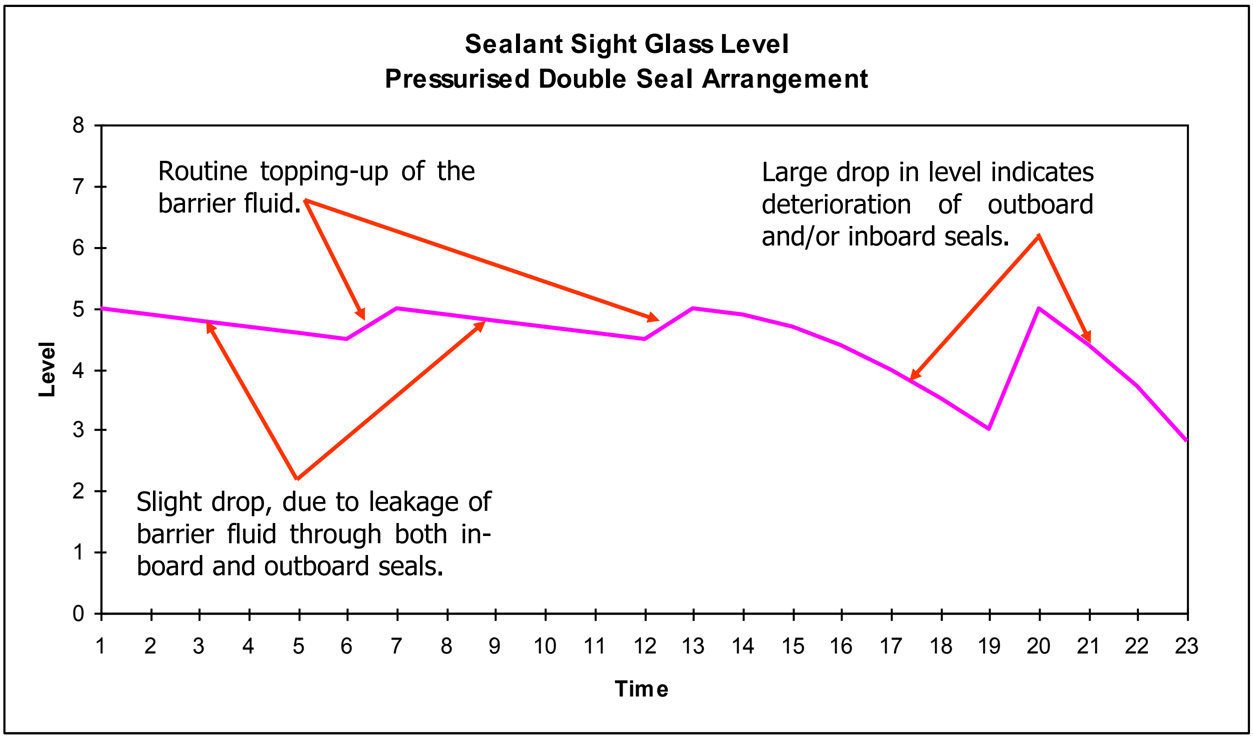
Tank System Advantages:

Pressure versatility	- High pressure capability.
Positive displacement	- Barrier fluid 'pushed' through seal chamber.
Multi-server	- One system may supply more than one seal chamber.
Heat removal	- Good heat removal with a heat exchanger due to increased flowrate

Monitoring

By careful observation of the buffer/barrier fluid levels in the supply system, the condition of the mechanical seals can be monitored. By this method, early warning of seal failure can be obtained. The mechanical seals can then be replaced at a planned shutdown, rather than having to replace them in an emergency after failure has occurred.





Typical Systems

Skid-mounted Vessel System, complete with external pumps, cooler, level switched and controls

API 610 System with flanged connections

API 682 System with threaded connections

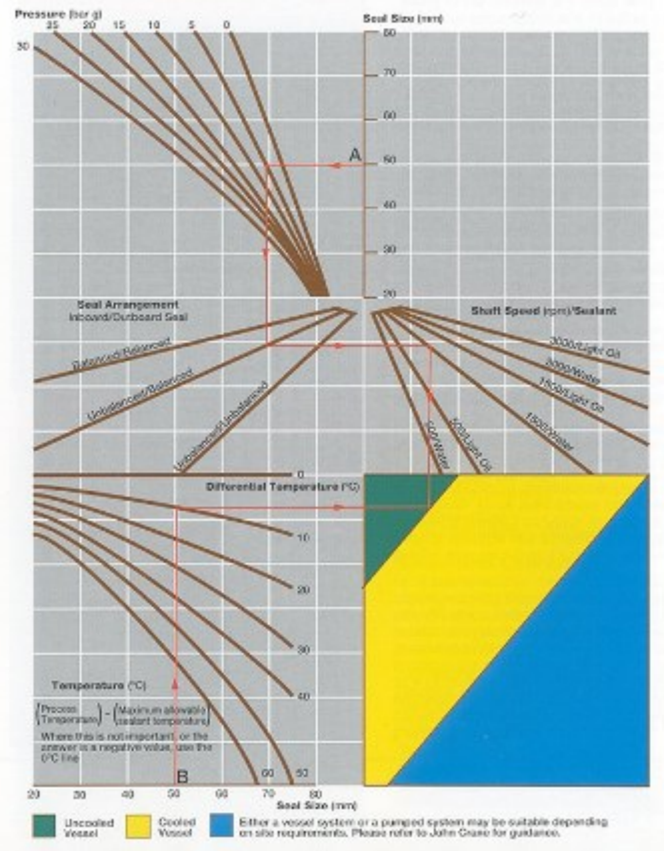
Pumped Systems

High pressure system designed to lubricate and cool seals in oil field export pumps in a tropical climate

Simple pumped system

See Appendix B for details of API Piping Plans

Chart 1. Vessel System Application Guide



The graph can be used to select a suitable sealant system for pressurised double seal installations.

The following information is required:

- Seal arrangement
- Seal size
- Sealant pressure
- Shaft speed
- Sealant fluid [water or light oil]
- Any cooling required [ΔT-temperature differential]
- Whether seals are balanced or unbalanced

Note: Barrier fluid is at least 1 Bar or 10% higher than process pressure, whichever is greater.

See Appendix H for A4 Chart

Other Sealant Systems
In addition to the standard tank and vessel systems, the following are available:

Fail Safe Sealant System: A tank based system which continues to maintain barrier fluid pressure in the event of a sealant system power failure.

Gas Panel: Used with Type 2800 series seals and provides a means of controlling and monitoring a buffer gas to and from the seal arrangement.

Hand pump / Accumulator Set: A simple form of pressurising a seal chamber with barrier fluid. This set has a limited reservoir and minimal instrumentation.

Multi-Server: Skid mounted custom built sets to supply sealant to more than one seal.

- Sealant Systems provide an independent barrier fluid source for multiple seals
- Standard Sealant Systems are Tank or Vessel based
- Vessel System 200 - Uncooled
  100 - Low duty / low cost system
  300 - Cooled system
- Tank System 600 - Positive circulation
  6SA - Cooled tank system
- Vessel Systems are manufactured and welded in accordance with ASME V111
- Sealant pressure is normally at least 1 Bar or 10% above process pressure, whichever is greater.
- All systems are inspected and tested prior to despatch

Environment of a Mechanical Seal

Secondary Containment

Uses for Secondary Containment

1. Control of hazardous emissions
   When using a single mechanical seal, the permitted emission levels for a given hazardous product may be exceeded, posing a threat to personnel and the environment. Legislation governing the control of such products is becoming increasingly demanding.

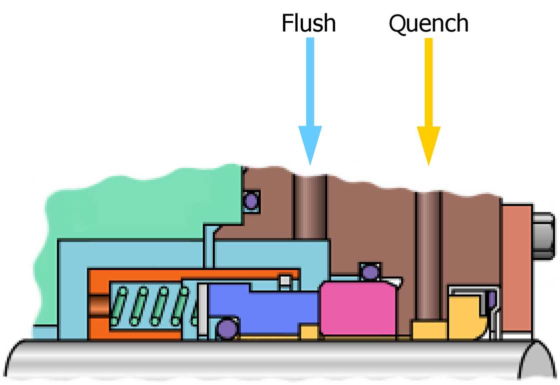
2. Control of leakage due to seal failure
   Although mechanical seals leak in normal operation, the negligible levels may mean that a single seal is acceptable. Leakage caused by seal failure is usually unacceptable, but can be prevented, reduced or contained by fitting a suitable secondary containment device. This ensures a safe working environment.

3. Isolating a sensitive product from the atmosphere
   Seal life can be extended by quenching with a clean fluid or steam. This is done to prevent a sensitive product degrading, oxidising or becoming volatile as it comes into contact with the atmosphere. It may also be required to prevent air contamination of the product.

There is sometimes confusion between the terms “Quench” and “Flush”.

A **quench** is used, where necessary, with secondary containment devices and is:

- Low pressure (<0.5 bar g)
- Low flow (0.5 litres/min)
- External to main seal
- No contamination of process
- No dilution of process



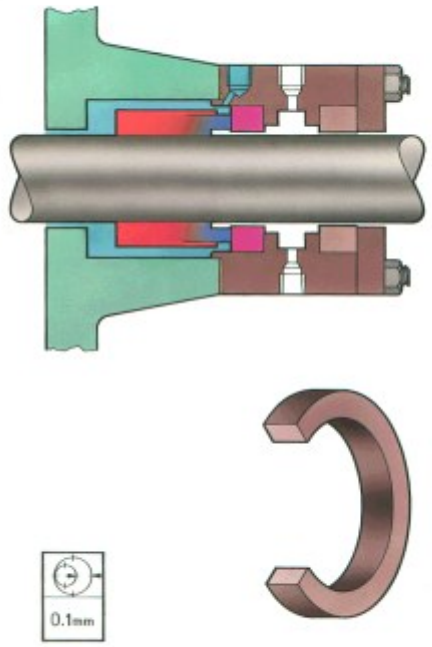
A quenching fluid may be introduced into the area between the mechanical seal and secondary containment device. A lubricating quench is essential when using contacting lip seals or compressed gland packing as secondary containment. Quenching aids heat dissipation when product re-circulation alone is insufficient. The quench fluid carries excess heat and product contamination away from the seal to a safe area or vent/flare stack. An intermittent quench (usually steam) may be used to clean the quench area at regular intervals and/or when a seal fails. This method of quenching requires the use of a non-contacting secondary containment device.

Note: A liquid quench should be introduced into the bottom of the gland plate and removed from the top to ensure all air is vented. Any air trapped in this area will collect around the shaft when the machine is rotating, preventing the quench fluid from doing its job.
A steam quench should be introduced into the top of the gland plate and drained from the bottom to ensure removal of any condensate.

If a sleeve is fitted to the pump shaft, this should extend right through the secondary containment device so that any leakage through the bore of the sleeve can be identified as such.

A **flush** may be a separate external supply of fluid, or process fluid and is:

- High pressure (>1.0 bar above process pressure)
- Injected into seal chamber
- Dilutes pumped product
- Cools pumped product

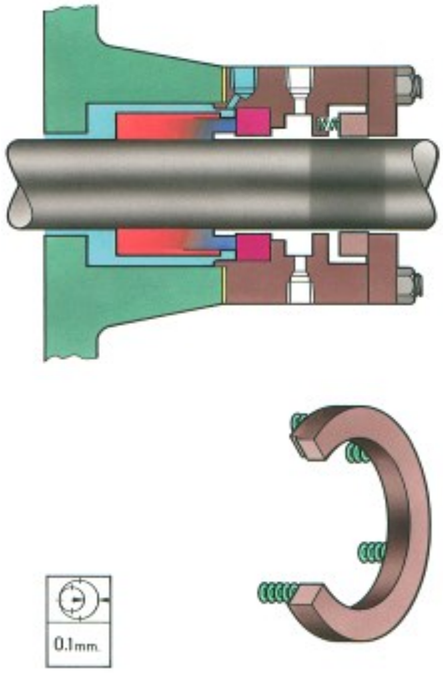


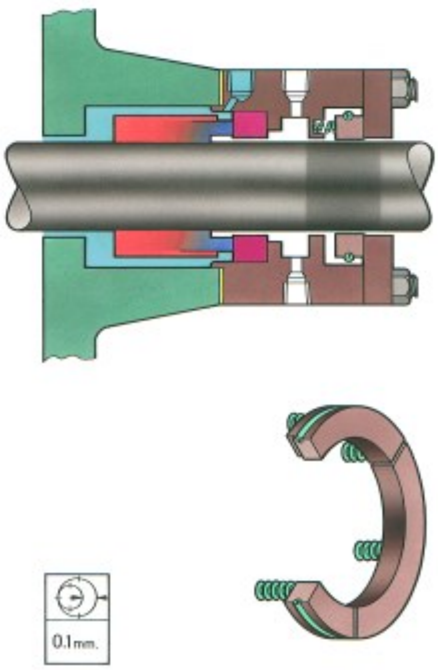
Fixed Bush

- Non-contacting
- Fixed bush (throttle bush, disaster bush)
- Non-sparking material (bronze, carbon, PTFE, etc)
- Minimum requirement for API 610 & 682
- Leakage reduction
- Radial shaft movement will create wear
- Hardened shaft/sleeve recommended
- Use API Plan 61

Floating Bush

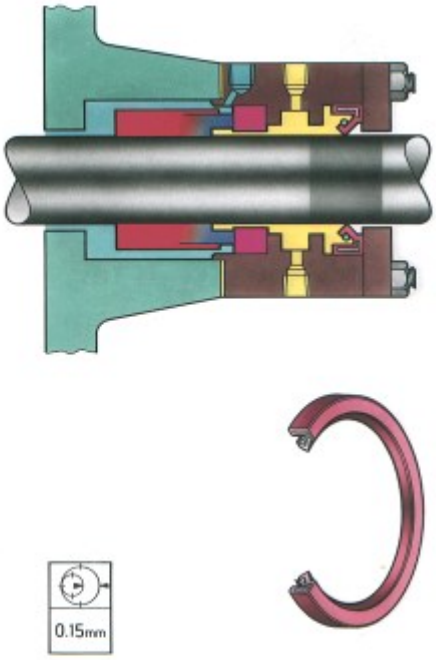
- Non-contacting
- Floating bush
- Carbon material
- Minimum requirement for API 682 ‘FL’ specification
- Leakage reduction
- Less wear if shaft moves radially as bush is floating
- Longer life
- Can be used for steam quench
- Hardened shaft/sleeve recommended
- Use API Plan 61 or 62





Segmented Bush

- Non-contacting when run-in
- Floating bush
- Carbon material
- Segmented design allows shaft to wear its own close clearance
- Leakage reduction
- Good with steam quench
- Less leakage
- Longer life
- Easy to replace without dismantling pump
- Hardened shaft/sleeve is essential
- Use API Plan 61 or 62

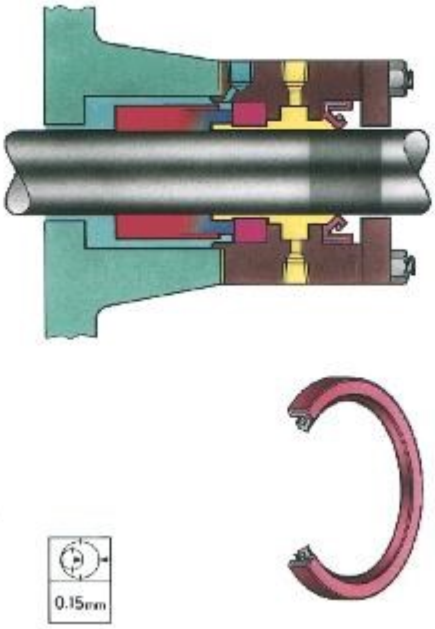


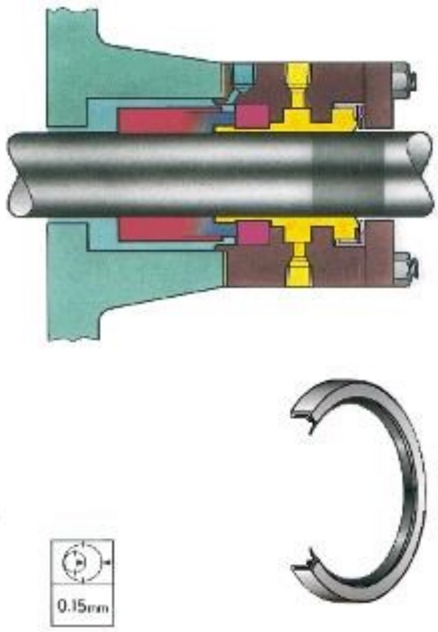
Lip Seal

- Contacting
- Various lip seal types
- Isolates product from atmosphere
- Dilutes leakage
- Contains low pressure leakage
- Must supply lubricating quench at all times
- Low pressures only
- Hardened shaft/sleeve recommended
- Use API Plan 62 or 51

Lip Seal - Reversed

- Contacting
- Various lip seal types
- Isolates product from atmosphere
- Dilutes leakage
- Contains low pressure leakage
- Can be used with grease quench, via grease gun
- Reversed lip seal acts as pressure relief valve when pumping in new grease
- Low pressures only
- Ideal for vertical pumps where partial vacuum may be drawn at start-up
- Hardened shaft sleeve recommended
- Use API Plan 62 or 51





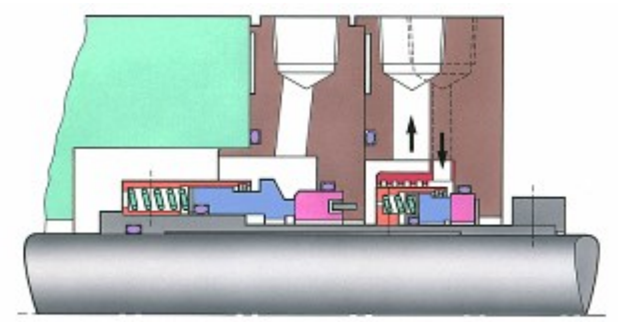
Metal Cased Lip Seal

- Contacting
- Various plastics (including reinforced PTFE) encased in 316 stainless steel casing
- Isolates product from atmosphere
- Dilutes leakage
- Contains low pressure leakage (can be designed to take up to 10 bar g)
- Must supply lubricating quench at all times
- Versions available which will operate with water
- Hardened shaft/sleeve is essential
- Use API Plan 62 or 51

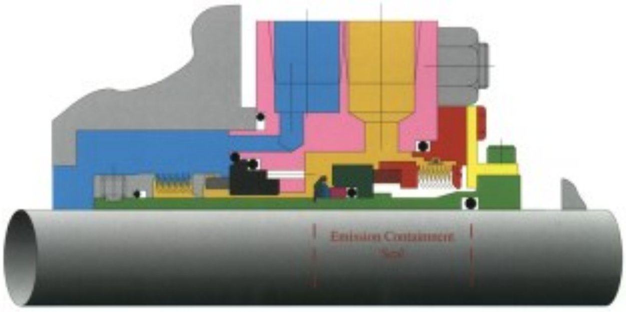
Other Secondary Containment Devices

Unpressurised seal arrangements give high integrity secondary containment, as previously discussed, but they require sealant supply systems and buffer fluid levels must be monitored.

This simple arrangement has a very short, non-contacting, gas lubricated spiral groove faced outboard seal. Low pressure, low flow, nitrogen is injected into the space between the two seals. This lubricates the seal faces and provides their lift and separation, and blows any vapour and gas emissions from the main mechanical seal up the vent to a safe area or vapour recovery system.

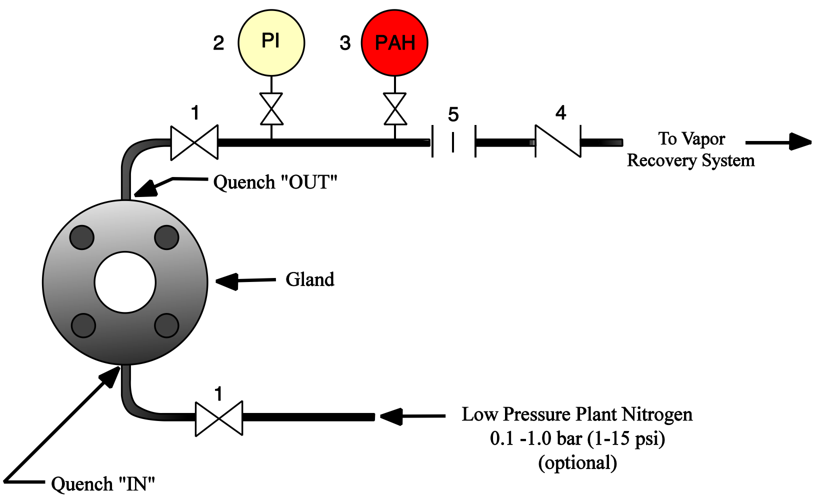


There are available a wide range of very lightly loaded, dry-running seals which act as secondary containment seals.



The outboard seal here is a very lightly loaded metal bellows seal which runs without any lubrication. It is also capable of running as a conventional “wet” seal when the main inboard seal eventually fails. As above, optional low pressure, low flow, nitrogen can be injected to blow any emissions to a safe area. A typical piping diagram is shown below.

1. Block Valve
2. Pressure Indicating Gauge
3. High Pressure Alarm (optional)
4. Check Valve. Opens at .1 bar (1-2 psi) ∅P
5. 3mm (1/8") Orifice



Summary

Secondary containment devices are used:

- To contain hazardous emissions
- To control leakage due to seal failure
- To blanket a sensitive product from atmosphere

There are two main types

- **Non-contacting** gives a reasonable measure of fluid control is seal failure occurs
- **Contacting** enables equipment to be run for a short period after seal failure

Benefits of containment seals:

• **Emission Reduction**
Escape of Volatile Organic Compounds (VOC) to the atmosphere can contribute to local and global air pollution. Legislation and Regulation in some countries restricts the level of these emissions and many users have a Corporate policy to assist in reducing VOC leakage levels.

• **Hazard Reduction**
Excessive leakage of toxic, flammable and/or explosive gas from a primary seal failure is a major potential plant hazard. Containment is an important seal feature, both to minimise the hazard and the impact on plant operation.

• **Capital Cost Reduction**
Unpressurised Dual Seal arrangements with Plan 52 buffer liquid systems provide excellent gaseous leakage containment but need costly support systems. Containment seals minimise the complication and specification of the auxiliary equipment and offer lower overall cost solutions compared with other dual arrangements

Mechanical Seals

Handling and Installation

<u>Handling</u>

Mechanical seals contain precision finished components, and must be handled with great care:

- Do not unpack the seal until ready to fit. This is necessary to prevent dust, dirt and grease from getting onto the lapped faces, and to avoid unnecessary handling especially from passers-by!
- Avoid touching or handling the lapped surfaces
- Place the seal on a bench with the lapped face uppermost, on tissue paper. Do not place lapped surface onto the bench, nor place the seal on its side - it will roll away.
- Keep hands clean when fitting a mechanical seal, or wear clean latex surgeon's gloves - the type without any talcum powder on the surface.
- Always wipe the lapped surfaces perfectly clean with a tissue and solvent* just before placing them together. Ensure all solvent has evaporated before the faces come into contact with each other.
- Do not wipe any lubricant onto the faces

*Use a suitable cleaning solvent. It should:

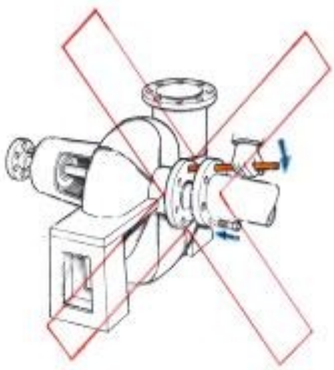
- Remove all traces of grease, dirt and marks
- Evaporate fairly quickly (contact cleaner evaporates too quickly)
- Not leave any bloom or marking on the surface when it has evaporated
- Not attack any of the seal component materials, such as elastomer o-rings
- Be approved by your Health and Safety Officer

Test your selected solvent on a clean piece of glass to check for evaporation time and whether any deposits are left behind. If there are any signs of deposits, do not use it on your mechanical seals.

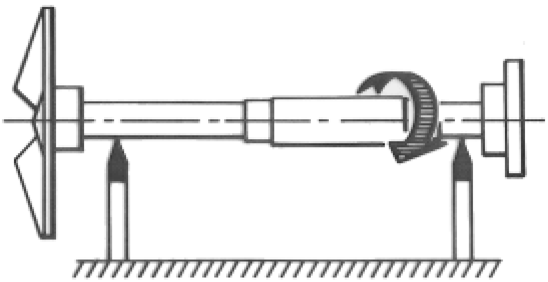
<u>Pump Assembly Checks</u>

Always read the installation instructions supplied with the mechanical seal. It is important that the pump or other machinery into which the seal is to be installed should be in excellent condition in order to achieve maximum seal life. See the charts pages 31 and 32 for shaft/sleeve tolerances and surface finishes.

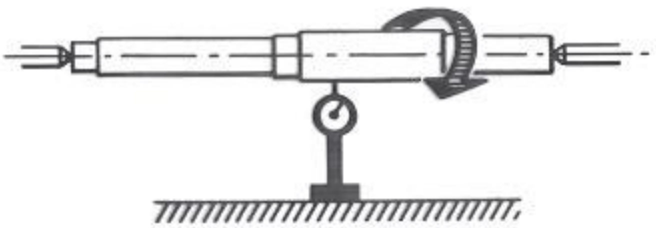
Accurate alignment at both ends of the machine will ensure long bearing and seal life.



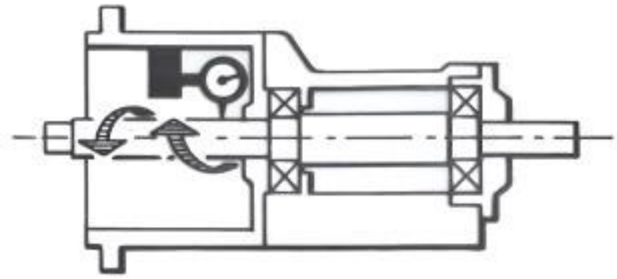
The shaft or sleeve surface must be in good condition with no pits, scratches or set-screw marks.



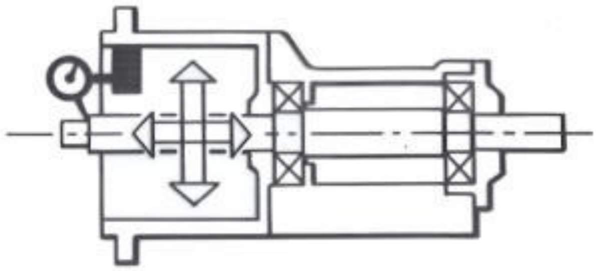
Rotating assembly should be correctly balanced
VDI 2060 and ISO 1940



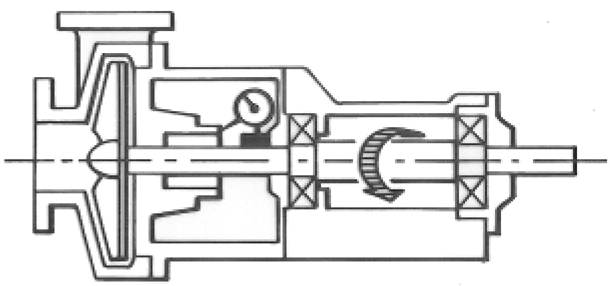
Shaft should be straight
Maximum TIR 0.05 mm



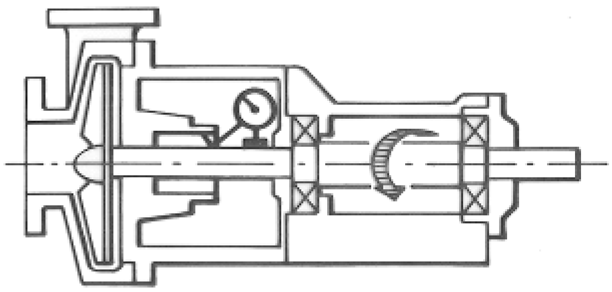
Shaft Run-out
1450 rpm 0.08 mm TIR; 2900 rpm 0.05 mm TIR



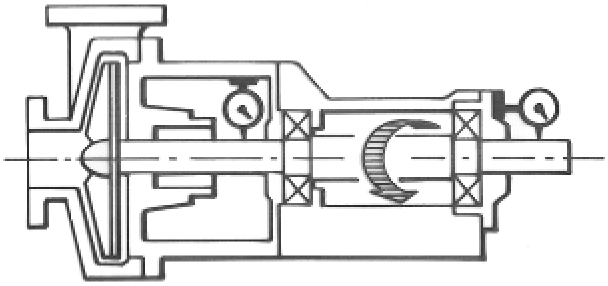
Axial and Radial Bearing Clearances
1450 rpm 0.08 mm TIR; 2900 rpm 0.05 mm TIR



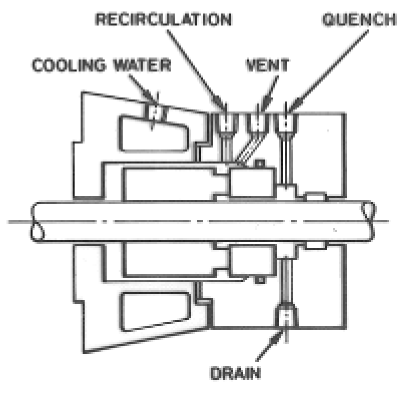
Seal Chamber Face
1450 rpm 0.08 mm TIR; 2900 rpm 0.05 mm TIR



Concentricity of Seal Chamber Bore to Shaft
0.15 mm TIR



Final Check - Seal Area and Coupling End
1450 rpm 0.08 mm TIR; 2900 rpm 0.05 mm TIR



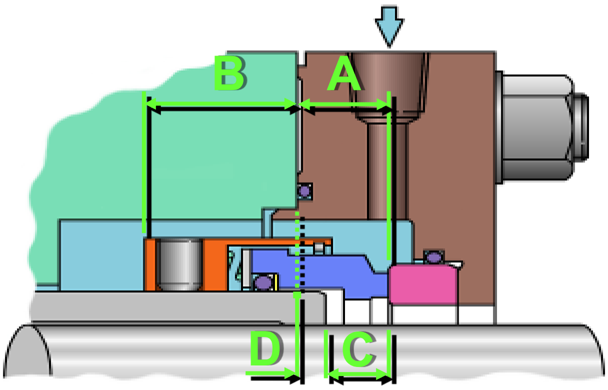
Ensure piping is attached to the
correct connections

Installation

There are many points in the procedure for installing non-cartridge seals where errors or damage to the seal can occur. There are many measurements to be taken, and the seal must be installed before the pump is assembled, which means that it is impossible to check whether the seal has been installed correctly.

Typical installation procedure for a **non-cartridge seal**:

- Mark position of face of seal chamber on sleeve
- Dismantle pump
- Lubricate tertiary seal
- Fit mating ring in gland plate - ensure fully home and square to axis
- Check mating ring is correctly located on anti-rotation pin
- Measure distance from front of gland plate to mating ring lapped surface, taking care not to damage the lapped face (A)
- Mark position of mating ring on the shaft
- Look up step dimension (C)
- Reinstall sleeve and check dimension (C)
- Look up seal working length in fitting instructions
- Subtract dimension (A) from seal working length to get (B)
- Measure (B) from mark on sleeve towards impeller
- Mark shaft in this position
- Measure from this mark to end of the shaft, or nearest step towards impeller and note dimension (Z)
- Carefully wipe lapped face of mating ring perfectly clean
- Place gland plate on shaft, taking care not to damage or dislodge the mating ring
- Lightly lubricate shaft and secondary seal
- Slide seal unit onto shaft, ensuring it is the right way round
- Wipe lapped face of the primary ring perfectly clean, taking care not to damage the surface
- Fit seal (Z) from the end of the shaft or shaft step, ensuring it is perfectly square to the axis of the shaft
- Evenly tighten set screws, ensuring that the seal unit is concentric to the shaft/sleeve
- Assemble the pump, taking care not to damage the rotating seal unit
- Offer gland plate to face of seal chamber. Check gap, before compressing seal, with dimension given in fitting instructions
- If this gap is incorrect, fully dismantle pump, and start again!
- If the gap is correct, tighten the nuts on the gland studs
- Cross your fingers!



Common questions asked after fitting a **non-cartridge seal**:

- Was it clean?
- Did I look up the correct seal?
- Is the mating ring square?
- Did I measure accurately, and correctly?
- Will it work?
- For how long?

Fitting instructions in Appendix C
Type 2, Type 109, Type 8B1, Type 515E, Type 680

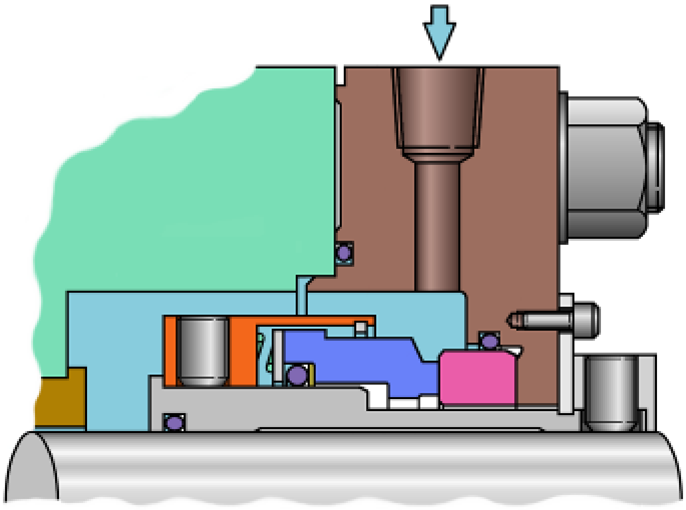
Typical installation procedure for a **cartridge seal**:

- Dismantle the pump
- Lubricate the seal sleeve o-ring
- Slide cartridge onto shaft/sleeve
- Assemble the pump
- Tighten nuts on gland studs
- Tighten set screws
- Remove setting devices

Common knowledge after fitting a **cartridge seal**:

- The faces are perfectly clean
- The seal is set to the correct working length
- The seal is square on the shaft
- The mating ring is correctly fitted and located on its anti-rotation pin
- The seal has been fully pressure-tested in the "as fitted" configuration
- You know it will work
- No premature failures due to installation errors or problems

All this also means that a cartridge seal can be installed much faster than a non-cartridge seal. API Standard 682 specifies that all mechanical seals should be cartridge assemblies.



Cartridge seal attributes:

- One-piece assembly "out of the box"
- Fully pressure tested in "as fitted" configuration
- Lapped faces pre-assembled and in contact - cannot be damaged
- Seal unit installed on long sleeve so will be square to axis of shaft
- Driving screws accessible from outside the seal chamber
- Seal is accurately pre-set to its working length
- Simple, quick, error free installation

Typical Installation Instructions for a cartridge seal are given in Appendix D

Spiral Groove Technology

Fluid Sealing with Gas Barrier

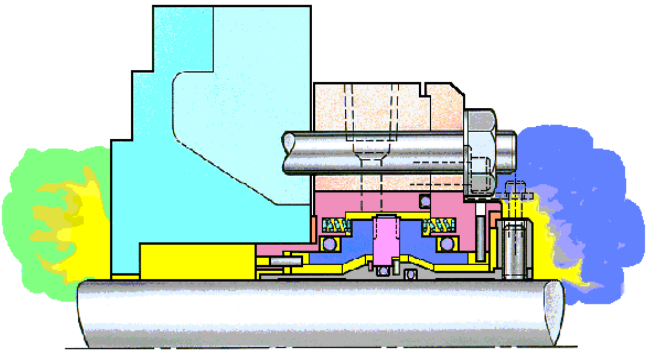
The new Spiral Groove technology ensures that the stationary and rotating seal faces do not contact each other during operation. When the technology is applied to mechanical seals for pumps, the seal is:

- Double pressurised arrangement
- Cartridge seal
- Gas lubricated
- Non-contacting faces
- Zero emissions
- Simple gas barrier system

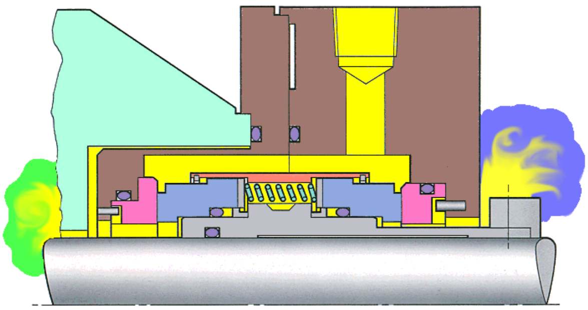
Giving the following benefits:

- Full compliance with emissions legislation - 100% containment
- Double-balance feature - reverse pressure capability
- No wear
- Vastly extended mean time between planned maintenance
- Lower repair costs
- Lower energy costs - less than 10% of an equivalent “wet” dual seal
- No heat generated, so no cooling required
- Ideal for heat-sensitive and high purity process fluids
- No expensive wet barrier system
- Barrier system can be remotely located
- Pump cavitation and dry-running do not affect seal performance
- Easy to install
- Low total life cost

The spiral grooves are the heart of the non-contacting design. These are machined to a precision depth into the lapped face of the mating rings. A suitable pressurised inert gas is injected between the seals. As the shaft rotates, gas flows into the tip of the spiral groove and is compressed as it travels to the sealing dam, the ungrooved portion of the mating ring. At the sealing dam, the gas expands as the pressure drops. The resultant film pressure gives an opening force greater than the closing force, even if the barrier gas pressure is momentarily reduced, maintaining the gas film between the gas and the process. At normal pump speeds, the grooves can be machined on either the rotating or stationary faces, and still have the same lift effect. Therefore, the cartridge seals may be either back-to-back rotating seals, or face-to-face stationary seals.



Face-to-face stationary seals, suitable for small-section seal chambers.



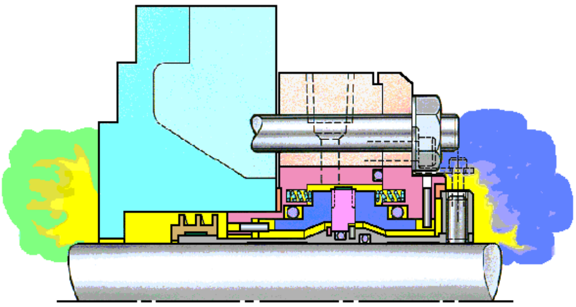
Back-to-back rotating seals, suitable for large-section seal chambers.

Face-to-face stationary seals, suitable for small-section seal chambers.

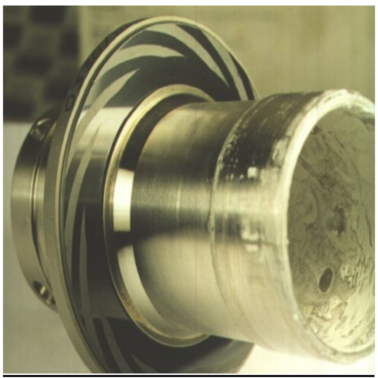
Back-to-back rotating seals, suitable for large-section seal chambers.

The simple gas barrier supply system can be seen in this photograph. This supplies clean, dry gas to the non-contacting seals.

Spiral groove pressurised seals are capable of sealing fluids containing up to 2% solids by volume without any modifications. With small modifications, they can be adapted to handle up to 20% solids by volume.

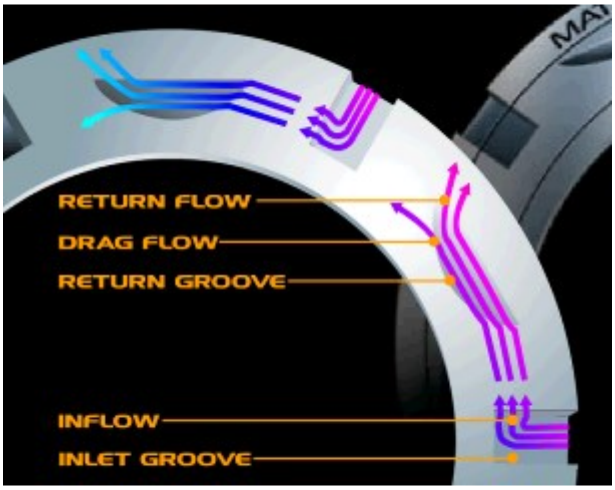


Here, a face-to-face seal has an expeller fitted to pump away any solids that may find there way into the seal chamber. This has proved to be a very effective modification, as can be seen from the series of photographs below. The photograph on the left shows the sleeve and mating ring from a seal without the expeller fitted after running in a fluid containing 18% solids by volume. The photograph on the right shows a sleeve and mating ring that was fitted with an expeller, after running on test in the same slurry.



LaserFace™

On pages 21/22 of this manual, we discussed the problem of high pressure, and other poor lubricating environments, where it is necessary to induce a thicker fluid film by increasing the hydrodynamic forces between the seal running faces. Hydropads can be used, but they create increased leakage which is often unacceptable.



John Crane have developed their LaserFace Technology to overcome this high leakage problem.

A series of inlet grooves and return crescents are machined into one of the running lapped faces, as shown in the left picture above. These grooves are only microns deep. The inlet grooves operate in much the same way as the Hydropads, but most of the fluid in the thick film produced between the running faces is returned to the seal chamber by the action of the crescent shaped return grooves.

Benefits of this exciting new technology include:

- Allows single seal to be used close to SVP
- Increased seal reliability
- Vastly reduced friction levels
- Low wear rates - enhanced MTBPM
- Low leakage levels - comparable to plain face seals
- Adaptable to existing seal designs
- Bi-directional operation
- Excellent in low vapour pressure applications

As this is a proprietary design, further details can be obtained from John Crane International.

Mechanical Seals

API 682

The American Petroleum Institute Standard 682 covers shaft sealing systems for centrifugal and rotary pumps. Edition 1 was published in 1994 and was primarily for the refinery industry. Edition 2 was published in July 2002 with a much broader scope, covering not only the refinery industry but also the natural gas and chemical industries.

The general parameters are:

- Shaft diameters 20 mm to 110 mm
- Temperature -40°C to +400°C
- Pressures up to 42 bar

The aims of the standard are:

- Maximum reliability and availability of equipment
- Meet emissions legislation
- Reduce costs through standardisation and reliability
- Improve safety
- Consistent seal application based on accumulation of knowledge of users and suppliers
- Seal interchangeability

Edition 2 states the following objectives:

1. All seals should operate continuously for 25,000 hours without replacement
2. Containment seals should run for at least 25,000 hours (wet or dry seals) at any containment seal chamber pressure equal to or less than the seal leakage pressure switch setting (not to exceed a gauge pressure of 0.7 bar and for at least 8 hours at the full seal chamber conditions
3. All seals should operate for 25,000 hours while complying with local emissions regulations, or 1,000 parts-per-million as measured by the EPA Method 21, whichever is more stringent

There are three seal types:

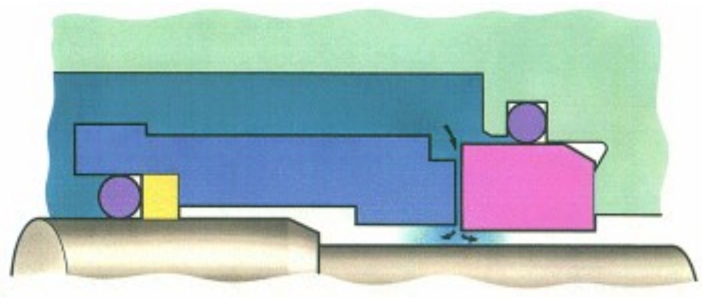
- Type A - balanced, inside mounted, cartridge design, pusher seal with multiple springs, rotating seal unit. Secondary seals to be elastomer o-rings
- Type B - balanced, inside mounted, cartridge design, non-pusher seal with metal bellows, rotating seal unit. Secondary seals to be elastomer o-rings
- Type C - balanced, inside mounted, cartridge design, non-pusher seal with metal bellows, stationary seal unit. Secondary seals to be flexible graphite

There are three seal arrangements:

- Arrangement 1: single cartridge seal with secondary containment
- Arrangement 2: dual unpressurised seals with forced circulation
- Arrangement 3: dual pressurised seals with forced circulation

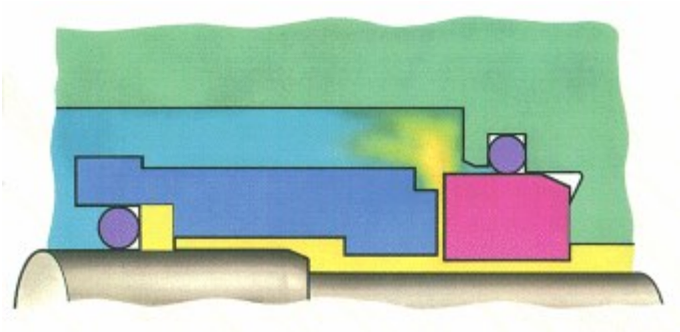
See Appendix B for Piping Plans

Mechanical Seal Principles
Upstream Pumping



A conventional seal has lapped faces which are flat to within 2 helium light bands. It operates by utilising the process fluid as the interface fluid providing a gap of 0.5 to 3 microns.

Up-Stream Pumping is so called because this principle is reversed and a chosen fluid is 'pumped' from the atmospheric side of the seal primary ring 'Upstream' to the higher pressure process fluid.

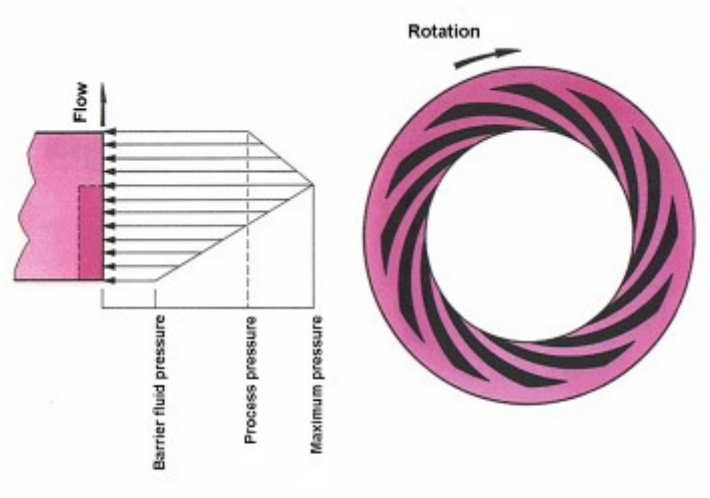


This can be achieved by using a series of spiral grooves extending approximately two thirds across the mating ring track. The un-grooved part of the mating ring forms a sealing dam when the seal is stationary.

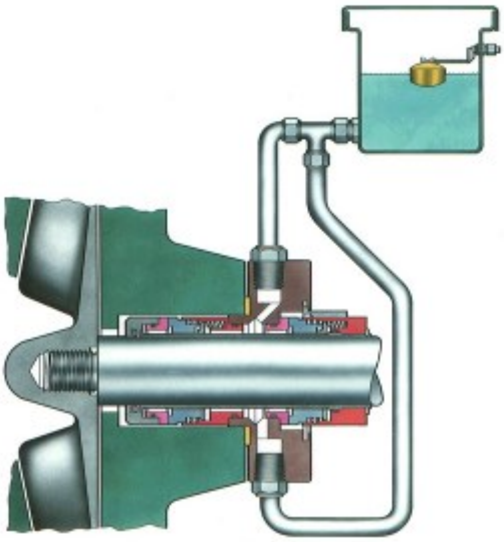
When dynamic [the seal is uni-directional only] , these spiral grooves cause a barrier fluid to be induced towards the outside mating ring diameter, where it meets the resistance of the sealing dam.

Pressure is increased, causing the flexibly mounted primary ring to lift off, setting the sealing gap.

Maximum pressure occurs at the outside diameter of the Up-Stream Pumping grooves and is always higher than the process pressure when the seal is operated within the design parameters.



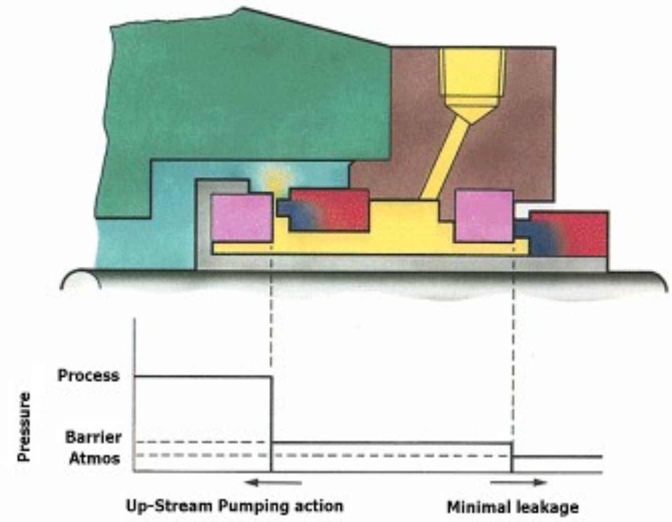
This principle solves the problems and dangers involved in sealing toxic, hazardous and abrasive fluids, because nay 'leakage' is inboard to the fluid being pumped.



To contain the barrier fluid, a second seal is used and both are mounted in a cartridge form. This technology can be used in many standard seal cartridge arrangements.

The inboard Up-Stream Pumping seal is supplied with barrier fluid from a simple tank system at atmospheric pressure.

If the barrier fluid is lost, the seal runs in the same way as a conventional seal.



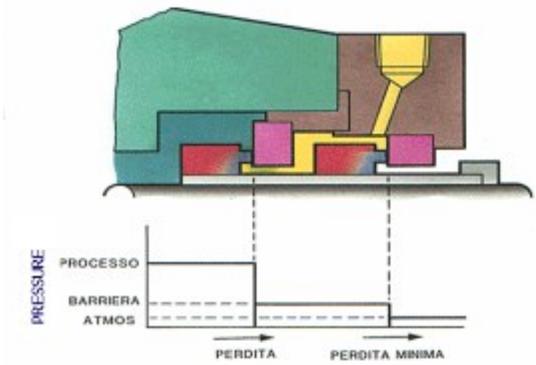
Little or no leakage occurs at the outboard seal because it is only sealing against a head of barrier fluid from the tank.

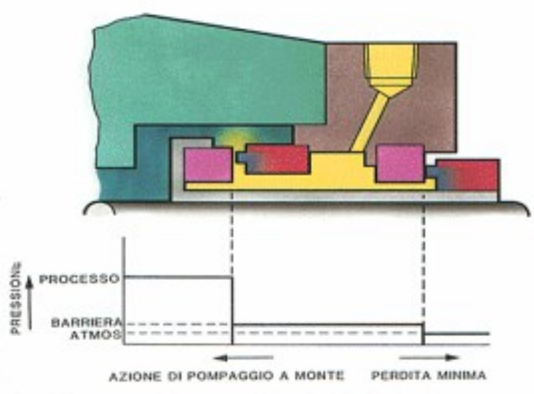
Barrier fluid is drawn from the area between the two seal arrangements into the process by the pumping action of the spiral grooves located on the inboard mating ring.

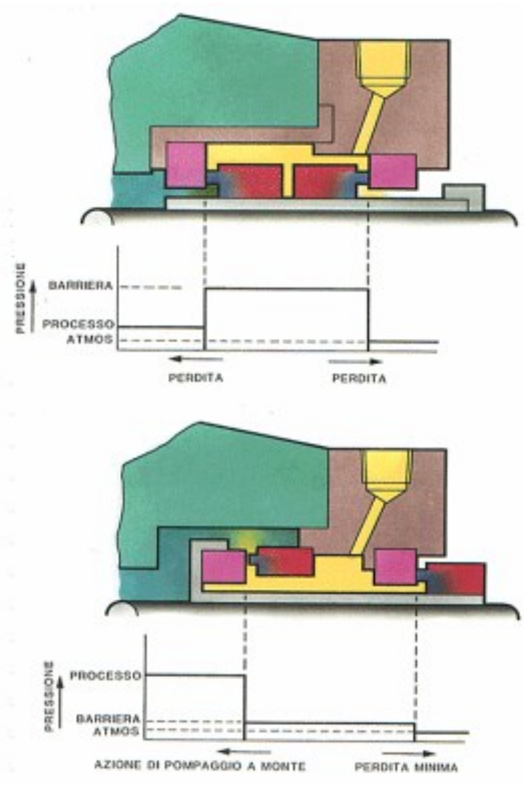
Up-Stream Pumping is a multiple seal arrangement which combines the benefits of both back to back and tandem sealing.

In a tandem seal installation the barrier fluid is lower than that of the process fluid and may become contaminated, eventually leaking to atmosphere.
A tandem seal installation also requires venting, which, can be expensive and will contribute to total plant emission.

In the case of Up-Stream Pumping, although the barrier fluid is contained at atmospheric pressure, considerable pressure is generated between the sealing faces pumping barrier fluid towards and into the process fluid.







When, as with double bask to back seals, the barrier fluid is maintained at a greater pressure than the process fluid, it should not become contaminated.

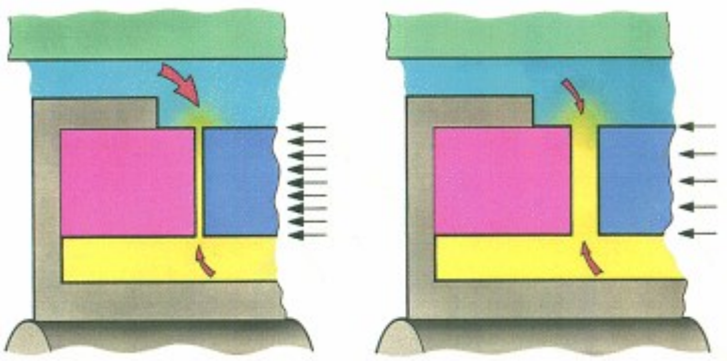
The spiral grooves within the Up-Stream Pumping arrangement ensure that the fluid between the inboard seal daces is not process fluid.

Unlike Up-Stream Pumping where the pressure is generated across the inboard seal faces, the pressure is maintained between two seals. High heat load may result especially at the outboard seal. As this seal deteriorates, loss of barrier fluid pressure may cause the inboard seal to open.

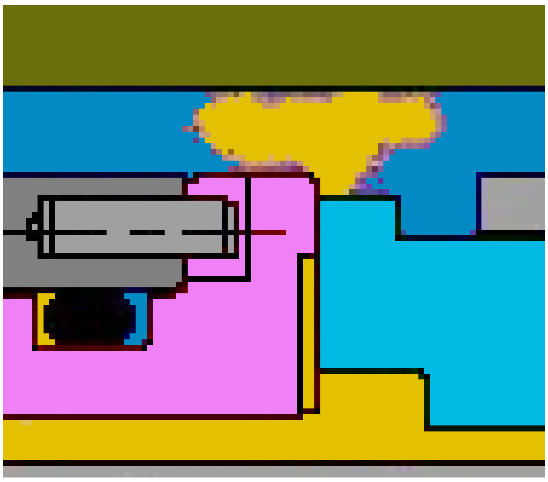
The Up-Stream Pumping seal generates low frictional heat due to the non-contacting mode of operation [the operating gap is larger than a conventional wet running seal] .

The fluid being pumped across the faces will remain stable within the performance parameters of the seal.

The unit is designed to produce a controlled rate of flow which maintains optimum face separation at all times in order to ensure full lubrication.

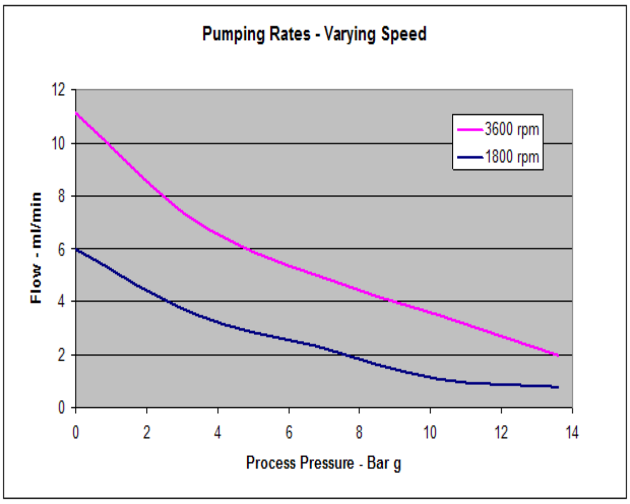


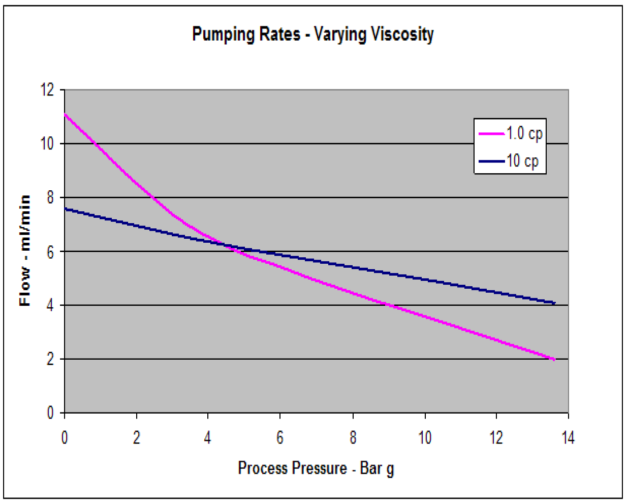
If pressure increases in the pump, the seal faces are momentarily squeezed closer together, the pumping efficiency of the groove increases, fluid film pressure increases and optimum face separation is achieved. If the pump pressure reduces, the seal faces open slightly, the pumping efficiency of the grooves reduces and again optimum face separation is achieved.



The Up-Stream Pumping seal design is such that a small amount on barrier fluid will be forced into the process.

This has the effect of flushing the inboard faces, eliminating the need for a continuous flush system. The barrier fluid should however be compatible with the process fluid.





Process temperature	40°C	Process temperature	40°C
Barrier Pressure	Atmospheric	Barrier Pressure	Atmospheric
Barrier temperature	40°C	Barrier temperature	40°C
Viscosity	1 Cp	Speed	3600 rpm

As can be seen from the charts above the pumping rate of barrier fluid into the process varies, but higher speeds usually result in high pumping capacity.
Higher viscosity flattens the pumping head curve.

Advantages over conventional multiple seal arrangements:

Tandem

Requires no product recirculation - Seal is self flushing by pumping action of the spiral grooves

Eliminates contamination of barrier Fluid by process fluids, minimising Potential emissions. - Positive pumping action of spiral grooves

Back-to-back-double

Eliminates the requirement for a Pressurised barrier fluid. - Spiral grooves generate a pressurised barrier fluid

Reduces the requirement for cooling - Low face stress and heat generation by non-contacting inboard Faces and un-pressurised outboard seat.

Advantages over Glandless Pumps

Allows easy retrofitting of existing pumps – Cartridge unit

Greater pump efficiency - Low starting and running torque / direct drive impeller

Ability to handle abrasives and high viscosity. - Barrier fluid provides the interface fluid

Longer bearing life - Bearings run outside the process in a conventional bearing housing

The design benefits of Up-Stream Pumping

Up-Stream Pumping has design benefits which greatly improve seal reliability and safety particularly in the areas of application stated overleaf.

Up-Stream Pumping Applications

Application	Advantage
• Food and Pharmaceuticals	No contamination from wear
• Fine abrasive slurries	Non contacting faces lubricated by clean barrier fluid
• Multiphase / Phase change	Stable barrier film maintains lubrication
• Low lubricity / high vapour pressure	Non contacting faces lubricated by clean barrier fluid
• Maintenance	Zero wear of USP seal plus reduced wear outboard seal
• Variable conditions	The USP seal is independent
• High speed	Pressure/Velocity heat generation and flush rate reduced
• Toxic or hazardous process	No product leakage to barrier fluid or atmosphere

The Up-Stream Pumping principle can be applied to other seal designs.

Summary

The main benefits of Up-Stream Pumping are long seal life and excellent reliability from:

Low heat generation

Self flushing design

Low running and staring toque

This gives significant advantages over existing technology with:

Energy savings

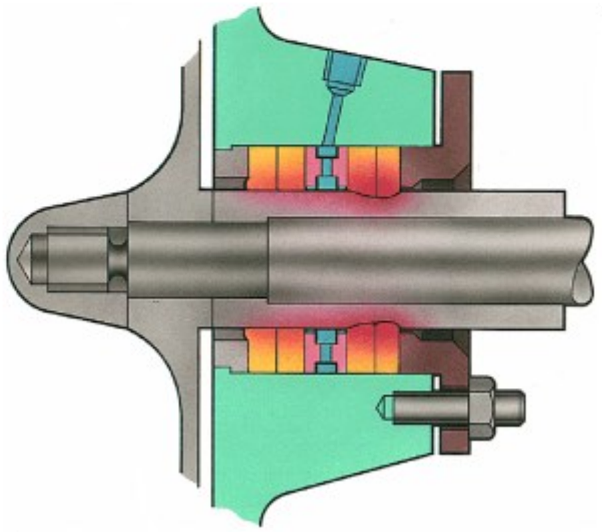
Low cost, simple hardware

Minimal maintenance

Safety

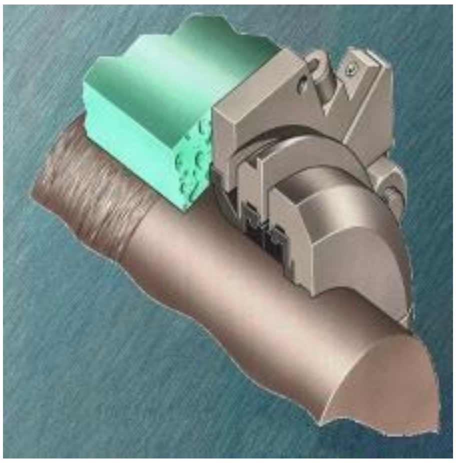
Spilt Seals
Type 37FS Features & Benefits

The Type 37FS [shown opposite] and FSB are fully split mechanical sealing arrangements and offer many benefits over traditional Gland Packing and other split and un-split mechanical seals.



The use of traditional gland packing is rapidly declining, mainly due to the following factors:

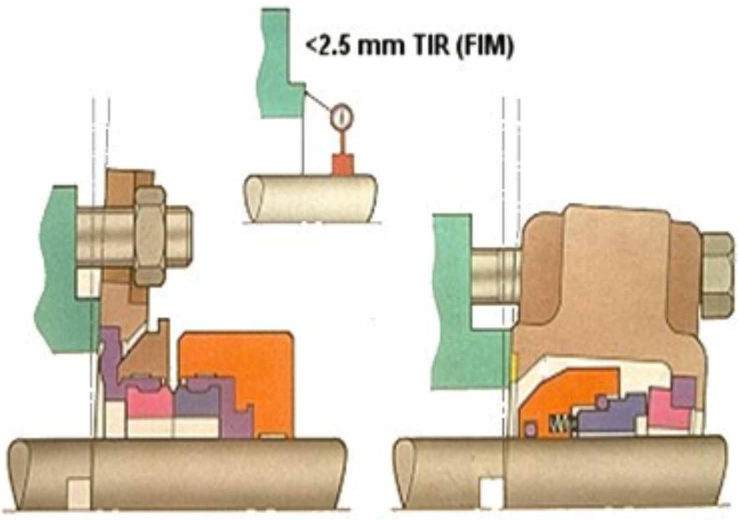
- Gland packing requires regular adjustment and replacement.
- Gland packing must leak to operate correctly
- Gland packing rapidly wears the shaft or sleeve.
- Gland packing consumes approximately 5 times the power of a typical mechanical seal.
- Shaft run-out will increase leakage and severely reduce the life of gland packing.



Because the Type 37FS family is fitted outside the stuffing box or seal chamber, the seal is unaffected by any shaft damage caused by previously fitted gland packing.

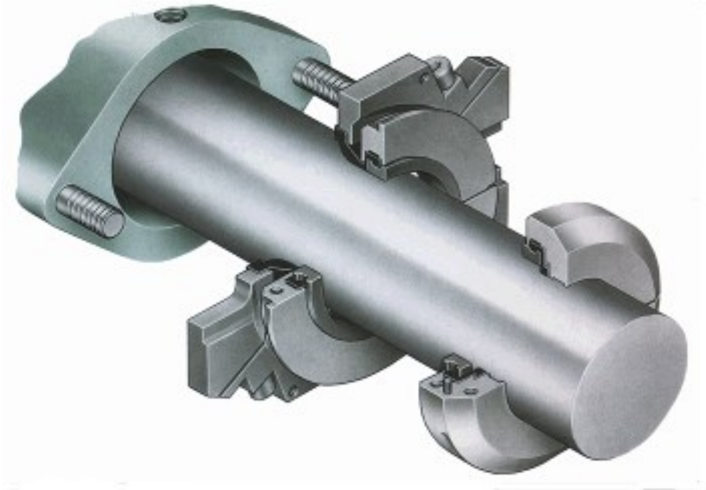
John Crane have manufactured 'O' Ring designs of split seals for many years but this design has been found to have severe limitations, particularly on equipment with large shafts and associated operating conditions. The illustration opposite shows how the bellows provides greater flexibility and the ability to handle:

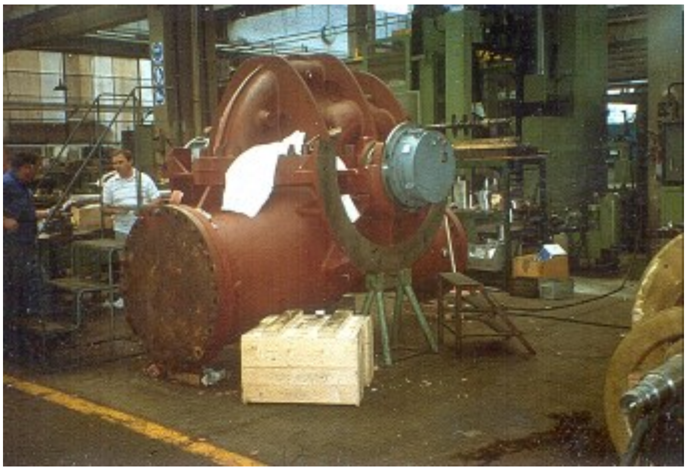
- Shaft run-out
- Misalignment
- Out of squareness of stuffing box face



The unique Type 37FS bellows provides the following benefits which give much enhanced reliability:

- Eliminates the need for springs which can easily clog.
- Fewer parts for ease of installation
- No gasket is necessary—the flange of the elastomer bellows seals against the stuffing box face.
- No special shaft finish required



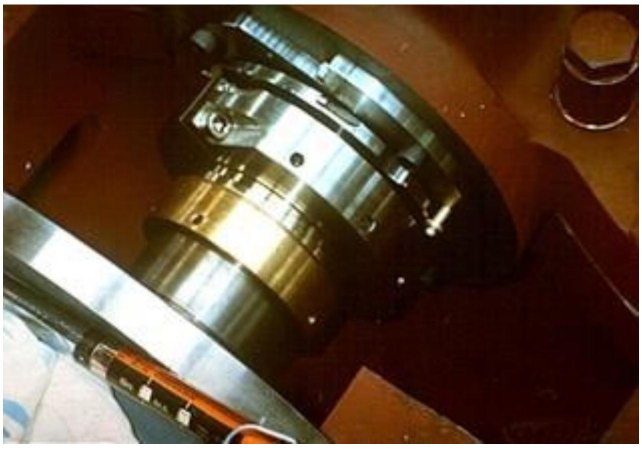


[Example of horizontally split case pump]

The following examples illustrate the adaptability of the 37FS family for retro-fitting gland packing. The size and condition of the equipment are typical.

We recommend use of the Type 37 FS for all rotary equipment which is either inaccessible or wherever it is difficult or time consuming to dismantle the equipment.
The Type 37FS is successfully installed on a wide range of rotating equipment such a mixers, pulpers, agitators and large pumps.

Typical shaft finish after removal of gland packing



A Type 37 FS fitted

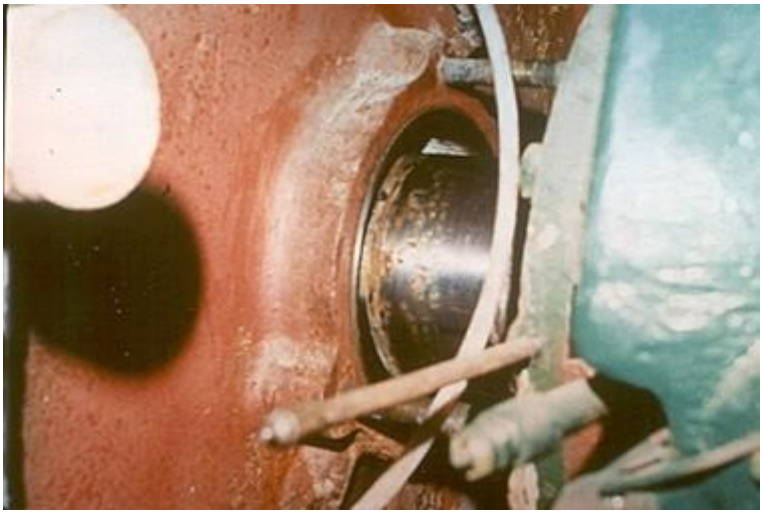
[Large centrifugal waste water pump]

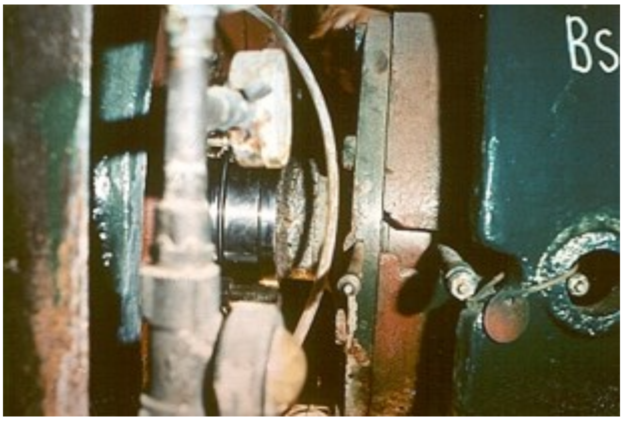
The illustration shows a large waste water centrifugal pump which was previously packed and is now fitted with 6 1/2” Type 37FS split seal.

To disassemble this pump and reassemble with a non -split seal or to replace a gland packing sleeve would normally require 6 working days to complete.

The Type 37FS was installed in 2 hours.

Typical shaft condition





Type 37FS fitted

The Type 37FS & FSB are available for inch or metric shaft sizes between 35mm to over 350mm. Maximum temperature is 82°C, maximum pressures are 37FS—5.5 Bar G, 37FSB—14 Bar G and maximum shaft speed 1800 rpm.

Summary and benefits of the Type 37FS & FSB

- Maintenance free and increased reliability
- Will not damage the shaft / sleeve.
- Low power consumption
- Minimal preparation of the equipment is required
- Leak free operation

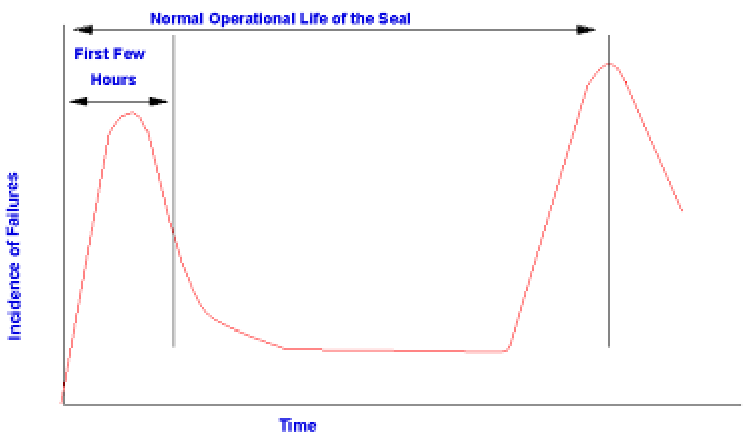
Mechanical Seals

Failure Analysis

Firstly, it is necessary to define what we mean by a “failed” seal. Obviously, a seal that is leaking is not working correctly, but the amount of “acceptable” leakage will vary depending on the application and the industry. The chart on page 17 gives an indication of the expected leakage from a seal that is operating correctly on Newtonian fluids.

This list, in ascending order of leakage rates, gives an idea of some of the considerations:

- Environmental considerations and legislation
- High hazard or pollution risk
- Process contamination
- Minimum levels of cleanliness
- Economic considerations
- Exceeding removal capacity
- Exceeding supply capacity



The “bath-tub” curve is often used when discussing failures. Some mechanical seals fail in the first few hours after start-up. These are most likely due to errors in installation. These can be cured by using cartridge seals. Some will be due to incorrect start-up procedures. Others may be during plant commissioning, where often foreign objects and rubbish in pipe-work damage the seals. By far the greatest majority of seals fail between this first few hours and the expected time of failure (25,000 hours according to API 682, but anything up to 8 years has been reported as the meantime between failures (MTBF) on some plants. These failures are most likely to be caused by operating problems and upsets. This can only be improved with good training, especially of operation staff.

There have been several surveys into the cause of premature seal failures, and we give below the result of two of them. Whilst the figures are different, they show similar trends.

Seal Failure Analysis		Institute of Mechanical Engineers	Global Chemical Company
Design	Seal selection and design ~ Pump selection, design & suction circuitry	28%	15%
Maintenance	Installation, alignment, bearing & pump condition ~ Seal fitting & installation tolerances	24%	20%
Operation	Process upsets, off-duty operating, priming, mal-operation ~ Seal flushing, cooling, quenching	48%	65%

It is important to collect ALL the evidence when a failure occurs, including visual evidence. Do not clean the seal (unless there is a possibility of hazardous contamination). Debris and other detritus in a seal can often help in pinpointing the cause of failure. Photographic evidence is also helpful if a seal must be cleaned, or re-installed quickly.

Here are some items to consider:

- Modus Operandi of equipment
- Start / stop sequences
- Maintenance arrangements
- Vibration signature
- Prior to start-up failure
- Cartridge pressure test
- Dynamic failure < 1 hour
- Operating hours
- Sudden failure or progressive leakage increase
- Failure following process change
- Failure related to operational change
- Start / stop history and operation
- Drive end / non-drive end bias
- MTBF - consistent or irregular
- Comparable duty with different history
- Anything unusual
- Actual operating conditions at failure
- Installation dimensional check
- Note the good points

Seals are selected on the basis of information supplied at either the enquiry or order stage. In the majority of cases the selection has to be made during the plant design phase. As a result of the timing of this selection the data supplied may be theoretical and is often incomplete, not taking into account all phases of the plant operation. A proportion of seal problems are caused by variations to the operating conditions from those originally specified. The changes may be to temperature, pressure, speed, product or the presence of abrasives, not previously mentioned. Due to this possibility, if problems are experienced, it is wise always to check the actual operating conditions against those originally specified, before assuming seal failures are the fault of the seal supplier. Changes in operating conditions may give rise to vaporisation, rapid face wear, or incompatibility of the components, which in turn leads to premature seal failures

A percentage of failures result from incorrect installation of the seals and/or components i.e. damaged carbon primary rings, trapped or cut secondary seals (o-rings) etc. However such faults usually result in heavy leakage when the equipment is pressurised. On stripping the seal for inspection the cause of the failure is usually quite visible and easily rectified.

The condition of the equipment to which the seals are fitted also has a bearing on failures. Bad coupling alignment may set up vibrations affecting bearing life, which can ultimately lead to seal failure. Vibrations can deflect shafts at critical speeds, causing the same effect. Smaller pumps and machines may not be routinely checked for vibration, and blocked or broken impellers may cause the shaft to gyrate and the resulting shaft displacement may result in seal failure.

On between bearing pumps, frequent removal of the bearing housings may result in misplaced shafts if the locating dowel holes have not been re-reamed and the dowels replaced. If seal failures persist on this type of application, it is essential to check that the shaft is concentric with the bore of the seal chamber and that the face of the seal chamber is square to the shaft centre line. The cause of seal failure from this condition may be difficult to detect, so pump assembly checks are very necessary.

There is a number of seal failures where the cause will be clearly evident on inspection and these are relatively easy to deal with. However failures with no apparent cause or visible defect are difficult to analyse. In these cases the type of leakage and the running characteristics of the machinery may indicate the cause of the problem.

On pressurising the equipment the seal should hold static pressure. If very bad leakage is observed this may indicate no face contact or a badly damaged component. Leakage in the form of a steady drip may be coming past one of the secondary seals or across the seal faces. On most products, static leakage is not tolerable and should be corrected before satisfactory seal performance can be obtained.

Note: Very heavy leakage is normally from faces and not from secondary seals.

If the seal holds static pressure but leaks after start-up, several matters should be considered:

a) Does it leak immediately or only after one or more hours of running?

b) Does the leakage stop when the pump is shut down? (This could indicate a swashing movement of the rotary parts of the seal caused by out of squareness, which would have to be located by alignment checks, and then corrected.)

c) How long has the pump been in service? If the leakage occurs during commissioning and vaporisation at the seal faces is indicated, it could be caused by 'off spec' start-up conditions. This problem should not re-occur once the plant has settled down. It could be caused by an incorrect seal selection. If seal failures occur after periods of successful operation and more than one attempt to solve the problem has failed, it is reasonable to assume that there has been a change in operating conditions.

d) If vaporisation at the seal faces is suspected, the product characteristics should be checked.

e) If the failure proves difficult to identify, the equipment should be checked for such things as out of balance, coupling alignment, pipe-strain, etc.

f) Could it be a double-ended pump with the failure occurring at one end only?

g) Could it be that A, B and C pumps are all operating on the same nominal duty but the failures are occurring on only one pump?

Precise identification of the seal failure mode can only be achieved by consideration of the leakage behaviour, which can give many valuable clues to the causes of the problem. Subsequent visual inspection of components will confirm the initial diagnosis. In this way the cause can be established and the cure implemented.

From our many years of service experience we have put together the following notes that accompany photographs of the more common type of seal failures. The notes highlight the failure symptoms and suggest remedies to cure or minimise the problems.

VAPORISATION

Vaporisation occurs when the heat generated at the seal faces is not removed effectively and local boiling of the interface film takes place. On hydrocarbon duties, this may be indicated by a popping or puffing noise, followed by heavy vapours. On other applications e.g. water, or where vaporisation is only slight, the symptom would not be visible during operation, but evidence will be found on components.

Symptoms

Light pitting of the carbon ring leading into comet trails in the direction of rotation. The vaporisation causes a chip to dislodge from the surface and this chip is ground away by the rotating faces and creates the 'comet trail'.

This shows (right) a Stellite metal ring that has cracked during vaporisation at the faces. The cracks are induced by large temperature changes that occur during the vaporisation process. Carbon dust deposits around the atmosphere side of seal caused by the pitting and eruption of the carbon face, which is then blown out with the vapour.

Where the seal is working under only slight vaporising conditions the seal life could extend to weeks. The failure symptom would be indicated by the carbon ring, which would have worn back considerably, but still be in a smooth condition.

Rectification

- Check product conditions against the original pump specification or order, for seal selection and materials.
- Narrow faced carbon should be used.
- Check circulation lines for blockages.
- Where coolers are used these should be checked for blockages, both through the cooling coil and water jacket around the coil.
- There are occasions where slight vaporisation could be prevented by increasing the circulation to the seal.
- Check for correct venting of the seal chamber. Air can collect around the seal faces, reducing the cooling.
- On some installations it is possible to alter the seal chamber pressure.

According to the operating conditions and subject to Engineering agreement, it is possible to prevent vaporisation by increasing or sometimes decreasing the pressure.

DRY RUNNING

Dry running occurs when no, or insufficient, liquid exists between the seal faces.

Symptoms

Severe wear and grooving of the carbon seal ring. The metal seal ring face shows polished grooving and, sometimes, radial cracking and discolouring. Other overheating symptoms: hardening and cracking of o-rings.

This shows a Stellite seal ring that has actually 'blued' with the heat of dry running.

Rectification

- Check suction flow and filters.
- Check circulation line is not blocked. If no circulation line is fitted check pumping conditions and fit circulation line according to requirements.
- Increase circulation flow.
- Ensure seal chamber is correctly vented.

Note: This condition is often confused with abrasive wear. It is not necessary to have abrasives present for these symptoms to occur and there will be no evidence of abrasives building up around any part of the rotary seal ring particularly in the area of the o-rings

COKING

Coking occurs when minute quantities of the fluid film leakage tend to carbonise under the atmosphere side of the seal, causing the sliding member (the rotary seal ring) to hang up or jam on the shaft or sleeve, preventing face contact.

Although the collective coke particles are present with the seal running under normal conditions, face contact is maintained because the product temperature and friction helps keep the associated waxes and gums reasonably soft. Solidification of the particles takes place when the equipment is shut down or on standby. On re-starting, seal leakage can occur. It has been known for a seal under the above conditions to 'make up' after a short running period because the rise in temperature softens the coked product, and, combined with slight axial movement, allows the seal faces to regain face contact.

Symptoms

Build up of product inside the sliding member of the seal prevents movement. Rotary seal rings that have coke build-up may be difficult to remove.

Rectification

Use a permanent low-pressure steam quench (1 – 2 bar g) to the atmosphere side of the seal, entering through the quench connection in the seal plate. This prevents the coke and wax particles from solidifying. A high temperature lip seal in the back of the seal plate aids the efficiency of the quenching and considerably reduces the leakage of steam, preventing it from entering into the pump bearing housing. The drain hole in the seal plate should always be clear to prevent a steam pressure build up. The steam quench should be applied well before start-up as it helps remove the existing particles and prevents a build up. Steam quenches should, ideally, be left on permanently, during the life of the seal, whether or not the pump is operating.

SLUDGING

Sludging is associated with the sealing of high viscosity liquids. The problem can be acute on pumps sealing hydrocarbon liquids at temperatures above ambient temperature. When the seal is shut down, the viscosity of the liquid in the interface film increases as the temperature drops. Problems then arise on restarting the equipment, when the shear stress between the seal faces can exceed the rupture strength of the carbon, and particles are pulled form the face and ground in between both mating seal faces. Sludging can also be caused when the interface film partially carbonises due to overheating.

Symptoms

- Cavities in the carbon face from which particles have been pulled.
- Polished wear track, or scoring, on hard face.
- Distortion of driving device.

Rectification

- Check that the viscosity of the product is within the capabilities of the seal, in accordance with John Crane Seal Selection limits.
- Check that the circulation to the seal is adequate and that the line is prevented from solidifying during shut down.
- To overcome start-up problems, it is necessary to pre-heat the seal area. This can be done by steam tracing or electrical heating circulation lines, steam or hot oil through the pump jacket, and a permanent low pressure steam quench under the seal via the quench connection in the seal plate.
- A seal plate incorporating a heating chamber can be used, through which steam or hot oil can be passed.

BONDING

Bonding is a similar type of phenomena to that of sludging, a bond is formed between the faces usually after the pump has been standing for a period and on starting particles are pulled from the carbon face. In vary bad cases, the carbon face can be torn off. The symptom can also be acute on cold Freon gas compressors.

BLISTERING

Very similar symptoms can occur to the carbon face under the action of "blistering". It is not easy to separate the mechanisms for sludging, bonding and blistering but blistering is normally associated with stop-start applications.

Symptoms

Particularly with high viscosity products and with high pressures it is possible that, on start-up, high local heating occurs in the initial few seconds. This can cause a rapid expansion of liquid that may have been partially absorbed into the seal face surface. This rapid expansion causes high stress in the carbon and under extreme conditions can cause failure. This failure may be observed initially by a shiny bruised effect or blister in the surface, and at a later stage will manifest itself as a crater where the blister has broken away from the surface and passed through the seal faces.

Rectification

Difficult to solve but attempts to keep the viscosity of the product low by heating is effective and careful choice of seal face materials is often necessary to build in the maximum resistance to blistering. In severe cases the mechanical seal manufacturers should be consulted.

In general terms the better seal face materials i.e., the ones with the higher thermal conductivity, produce less blistering against carbon counter-faces

ABRASIVES IN PRODUCT

If it is known that abrasive particles are to be present in the pumped product the initial seal selected for the duty can be made to accommodate them.

Particles of a certain diameter, and the ever-present scale and dirt, can be kept away from the seal faces by clean circulation to the seal. However failures related to abrasives are usually caused by the minute particles that enter the faces via the interface film, heavily scoring the faces and creating high leakage. It should be noted that virtually no wear takes place off the total area of the carbon face.

Symptoms

- Carbon face can show deep small scratches of an uneven pattern, but usually in the direction of shaft rotation.
- Metal face could be worn and polished.
- Deposits on product side of secondary seal.

Rectification

- Introduce a clean external injection from another source, e.g., API Plan 32.
- Use a cyclone separator, e.g., API Plan 31 (if particle size and density fall within separator specification).
- Use wear resisting face materials i.e. solid tungsten carbide or silicon carbide.
- Use dual pressurised seal arrangement, e.g., API Plan 53.
- Use Up-Stream Pumping seal design.
- Use reverse circulation, e.g., API Plan 13.

FACE DISTORTION

If seal leaks statically, or immediately on start-up and there are no visible faults on the components when stripped down for inspection, the seal faces should be checked for distortion. A flatness test will indicate that the face is not flat, but the readings are very difficult to interpret. Lightly rubbing or machine blueing a face on a flat surface should indicate high spots or uneven wear tracks.

Distortion can be caused by incorrect assembly. The holes in the back of the face shown in the photograph contacted the three anti-rotation pins, which protruded too far. This was the result of the use of an incorrectly sized joint ring. In this case the uneven wear track was easily visible.

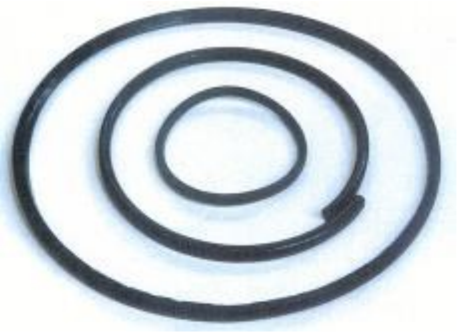
Rectification

- Re-lap the components on a lapping machine wherever possible, or in situ by using a flat disc, or fit a new seal face making sure the seal is fully cleaned before re-assembly.
- Ensure faces are assembled correctly, and not under stress from another component.

HARDENED OR CRACKED O-RINGS

Most o-ring materials become hard, brittle and crack when over-heated. Cracks in o-rings can also be found on low temperature duties where there is incompatibility with the sealed fluid.

Symptoms



- Rubber o-rings hardened and cracked
- PTFE o-rings discoloured blue/black.

Rectification

- Check circulation to seal, ~~including cooler where fitted, for blockages.~~
- Check suction flow and filter to pump.
- Check for possibility of dry running.
- Check compatibility.

SLEEVE DAMAGE OR MARKING AS A SYMPTOM OF SEAL FAILURE

There are many ways of finding clues to causes of mechanical seal failure. One such clue can be given by marking on the shaft sleeve. If a shaft is particularly eccentric or misaligned, it can cause contact between the o-ring landings on the inside of a rotary seal ring and the shaft sleeve. Often this contact is only on part of the circumference, as with a gyrating shaft, and it may be an indication of a condition that causes seal face leakage, due to imposed misalignment between the seal faces.

Marking which occupies only part of the circumference of the shaft sleeve is usually an indication of an eccentric or gyrating shaft.

A misaligned stationary seal ring will cause the rotary seal ring to oscillate relative to the shaft sleeve when running. This usually results in contact marking around the complete circumference and is often accompanied by wear on the inside diameter of the secondary seal.

A shaft that is bent, often gives rise to two marks, one on the front landing of the rotary ring and one on the rear landing, diametrically opposite.

An eccentrically mounted shaft can cause an extremely hydrodynamic seal if the eccentricity is excessive.

All of these effects normally cause leakage only when running, and often this leakage disappears when the machine is static.

Sleeves affected in other ways can cause seal leakage. These vary from chemical damage, to incompatibility, to rough unpolished surfaces on incorrectly manufactured sleeves. These often occur when a user manufactures his own sleeves, and is not fully aware of the importance of a smooth highly polished sliding area under a pusher seal (a good surface finish is not necessary under non-pusher metal bellows seals, and must not occur under elastomer bellows seals which must grip the sleeve tightly).

This photograph (left) shows a sleeve which has been attacked chemically, and a large part of the sliding surface has become detached.

This photograph (right) shows a sleeve, and its reflection in a mirror. It can be seen that the marking is principally on one side of the sleeve, the reflection indicating no marking on the other side.

Symptoms

- Severe marking on shaft or sleeve, or inside of rotary seal ring, corresponding with o-ring landings as described above.
- Corrosion can be found generally on the product side of the sleeve if chemical incompatibility is a factor. Unless the seal is leaking badly, the atmosphere side is often undamaged.

Rectification

- Check pump or equipment for drive alignment. Check for the possibility of vibration from failed, or failing bearings, which could cause either increased vibration or misalignment.
- Check for a bent or misaligned pump shaft, and also the assembly of the components into the machine.
- Consider alternative materials of construction. If the marking on the sleeve is caused by a slight movement in combination with abrasives consider using a hard deposit such as chrome oxide or tungsten carbide on the sliding area.

ELASTOMER INCOMPATIBILITY

Elastomer attacked, softened, or swollen.

This photograph shows o-rings which have been attacked on the product side of their section.

This photograph shows a new ethylene propylene bellows on the left, and a similar bellows, originally of the same size, which has been attacked by lubricating oil.

Symptoms

A reduction in the cross section of the elastomer from the product side, or a general swelling and softening of the complete elastomer section.

Rectification

- Check the actual product conditions against seal selection; ensure materials have been used.
- Ensure the correct lubricant is being used to assist fitting the seal, e.g., do not use hydrocarbons to assist fitting ethylene propylene o-rings.
- Ensure that the correct cleaning solvent is being used during the assembly/installation of the seal, e.g., acetone should not be used on fluorocarbon o-rings.

SAFETY BUSH

An API 682 specification throttle bush fitted in the back end of the seal gland plate is designed to contain and minimise sudden leakage.

Symptom

Bush badly worn to the extent of the material reaching melting point.

Rectification

- Renew bush and re-machining bore out after pressing into seal plate to 0.025 inch diametrical clearance.
- Use non-sparking metal, or carbon bush. PTFE is less desirable due to its thermal expansion properties and extrusion under load.
- Check for bent shaft or bearing failure, and correct.

SPRING DISTORTION AND BREAKAGE

Some seals incorporate the use of a single spring drive. In operation, the unidirectional spring should always tighten its coils and grip the components. If, for any reason, the wrong-handed spring is fitted, or the pump is rotated in the wrong direction, e.g., turbines backwards, the spring will tend to uncoil, slip, distort, and may even crack across the coil section. This type of failure occurs more frequently when springs are fitted incorrectly on high viscosity duties where excessive torque is present at the seal faces due to sludging or bonding.

Symptoms

- Radial cracking on the inside section of a spring coil.
- Broken spring coil.
- Wear marks on the ends of spring coils, also on the neck of the spring sleeve and rotary seal ring, caused by the spring slipping.

Rectification

- Check for correct handed spring.
- Check that the pump rotation is correct. If backwards turbining is experienced or suspected, check non-return valve in the discharge line between A and B pumps.

CARBON RING EROSION

Carbon ring erosion occurs when the pressure of the circulation to the seal is excessive, or it contains abrasive particles.

Symptoms

- Eroded groove all the way round the rotating carbon face.
- On stationary faces, a groove is sometimes seen adjacent to the circulation inlet in the seal gland plate.

Rectification

- Fit an orifice plate or flow controller in the circulation line, at the discharge end of the line.
- If a multi-stage pump, take the circulation line from first or second stage.
- Use a tangential connection or distributed flush for the circulation line entry into the seal plate.
- Fit a cyclone separator if abrasive content in the product is excessive.

RUPTURED METAL BELLOWS

These show in two forms. Firstly, a complete rupture of the bellows, usually caused by bonding of the high viscosity fluid film between the seal faces where, on start-up, the bond strength is greater than the design torque of the bellows. The second type of failure is when the bellows fails because the welded seams split; usually initiated by chemical attack, or vibration.

Rectification

- On high viscosity products ensure there is maximum auxiliary heating to the seal area i.e., seal chamber jacket and seal gland plate.
- Ensure this heating is on at all times, including during standby periods.
- Ensure the equipment is subjected to the maximum pre-heating or product circulation period.
- Use a seal with positive drive, removing torque from the bellows, on high viscosity applications (See picture bellow).
- Use a permanent low pressure steam quench.
- Check for sources of vibration.
- Is "boundary lubrication" possible, causing stick-slip, or torsional vibration?
- Is circulation line pressure too high causing radial vibration on a rotating seal?

INCOMPATIBILITY OF SEAL BODY MATERIAL

The photograph shows heavy corrosion of the seal body. Note that the corrosion is not present on the atmosphere side of the secondary seal o-ring. The mechanical seals functioned correctly until the advanced stage of corrosion shown was reached.

Rectification

Re-select the materials to be compatible with the actual service conditions in use now.

CARBON INCOMPATIBILITY

Area of the carbon ring in contact with the product is badly pitted and porous, caused by chemical attack of the impregnation material.

Rectification

Select correct seal materials for product compatibility against actual operating conditions, and all chemicals that are pumped, including any cleaning chemicals.

O-RING EXTRUSION

Extrusion occurs when part of the ‘0’ ring is forced through a close clearance gap between components, caused by excessive pressure (sometimes aggravated by overheating or softening due to incompatibility). On extremely high pressures, extrusion shows up as a deep cut adjacent to the moulding flash in the I/D of the ring with the result that some of the material of the ring is forced through the clearance. Extrusion can be caused where seal parts have been reconditioned to sizes beyond their design tolerance, creating a larger clearance between the components.

<u>Symptoms</u>

O-ring cut, and in some cases peeled off like an outer cover. This photograph shows an extreme case of ‘0’ ring extrusion where one section of the o-ring has unpeeled itself into and through the clearance to a low pressure. Sometimes these o-rings can be rolled out into flat sheet showing the scrolling effect during extrusion. Not all cases of extrusion are this severe

One, or two lips formed on cross-section. This photograph shows examples of extrusion both on rubber and PTFE o-rings. It is interesting to note that the extrusion can take place not only towards the atmospheric side of the seal. but also backwards towards the seal product itself. This occurs because there is a magnification of pressure due to the hydraulic forces within the design of the seal, and hence the pressure within the o-ring can exceed the pressure of the sealed fluid. On the PTFE example, the inside diameter has become blackened and charred because of overheating. This was, undoubtedly, one of the principle reasons for extrusion on this o-ring.

<u>Rectification</u>

- Check components on final assembly.
- Ensure parts are to original makers design tolerances.
- Check drawing or fitting instructions for possibility of omitted anti-extrusion ring.
- Try and establish if failure occurred during a malfunction.
- Check the product conditions for changes from original specification.

COLLAPSED SEAL

Whenever, and wherever installing mechanical seals, they should be left in the box until you are ready to fit the seal, and the equipment on which the seal is to be fitted has been fully prepared.

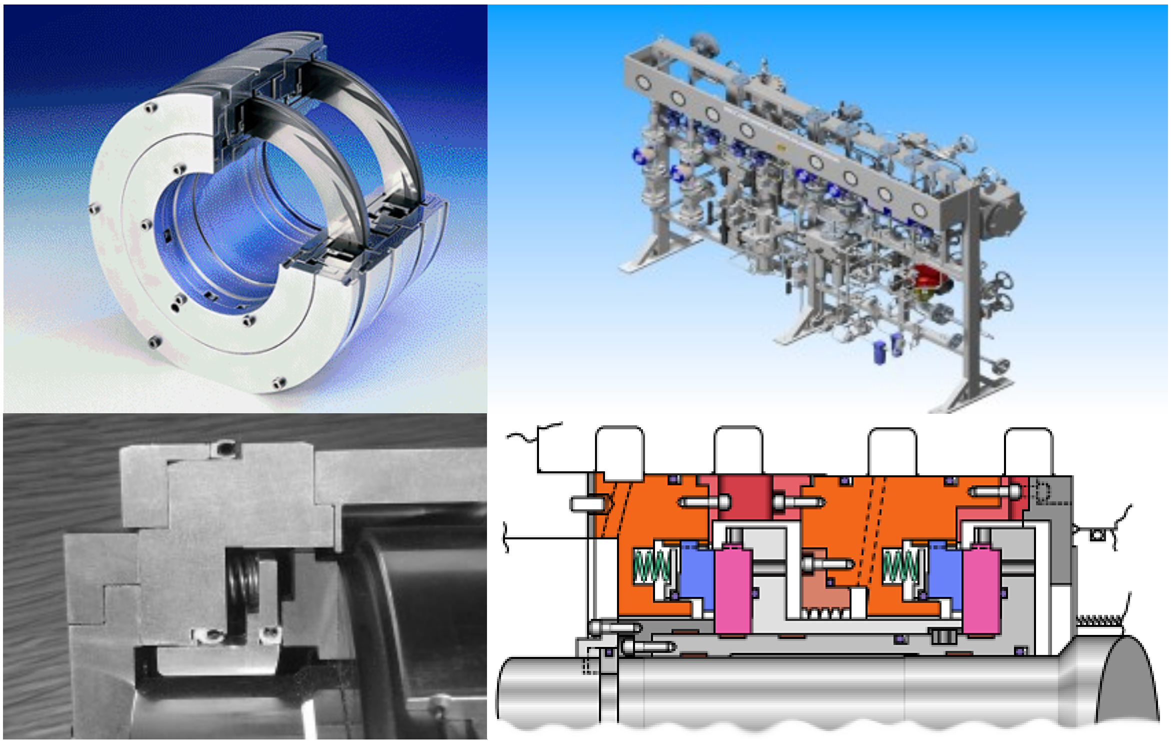
When the seal is removed from the box, it should be placed onto tissue paper, with the lapped face uppermost. Do not place it lapped face down on the tissue, and never place the seal on its side.

In this case the fitter, working out on site, removed the seal from the box, placed it on its side, and turned to undo the impeller nut. The seal rolled into the roadway and a forklift truck ran over it!

<u>Rectification</u>

- Buy a new seal.
- Send the fitter on a John Crane Training Course – soon!

Dry Gas Seals



Course Manual

Dry Gas Seals

Contents

An Introduction to Dry Gas Seal Operation
Introduction

Nearly thirty years ago the sealing of centrifugal compressors was revolutionised by the introduction of type 28 Dry Running Gas Seals. During the mid 70's a leading pipeline operator in Canada carried out a survey into all the compressor failures that had occurred during the previous few years. The findings showed that approximately 80% of compressor failures were due to seal oil system faults.

The installation of Type 28 dry gas seals to a compressor in 1976 proved the capability of this non-contacting seal which has completely solved the problems that this operator was experiencing with the compressors

Dry gas seals are now accepted world-wide as a mature product handling gases on a very wide variation of plants and compressors.

The advantages of dry gas seals over wet seal systems can be defined as:

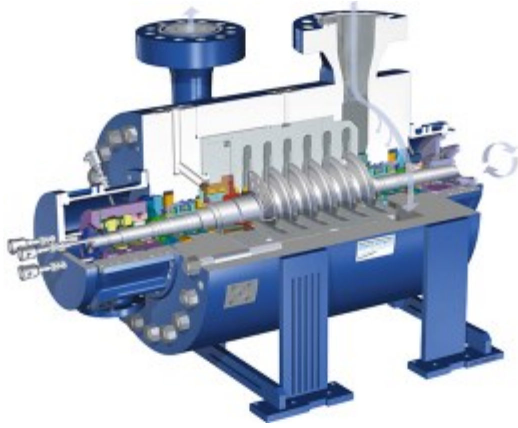


Figure 1: Typical Compressor (sectioned)

Cartridge Seal Design

Being a cartridge design the seal is relatively simple to fit. But care should be taken to ensure the seal is not damaged using poor techniques.

No Wear

Because a thin film of gas separates the seal surfaces while the shaft is rotating, seal wear is avoided. This

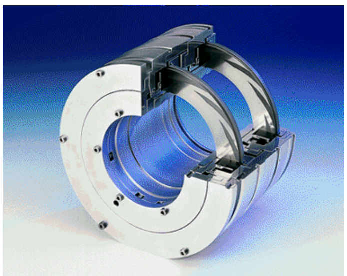


Figure 2: Typical Gas Seal (sectioned)

No Oil System

In dry running seals, the complex, heavy and expensive oil feed systems are replaced by clean compact control and monitoring systems. As well as space and weight savings, capital costs can be reduced. The dry lubricated system cuts maintenance and removes the need for lubricating oil. As a result contamination of the process gas is effectively eliminated, improving product quality and efficiency.

Low Power Consumption

Wet lubricated seals generally absorb significantly more energy in the form of viscous shear. Dry lubricated seals offer virtually no resistance, reducing frictional losses by up to 98%, leading to significant power savings. Typically less than 5Kw.

Improved Rotor System Stability

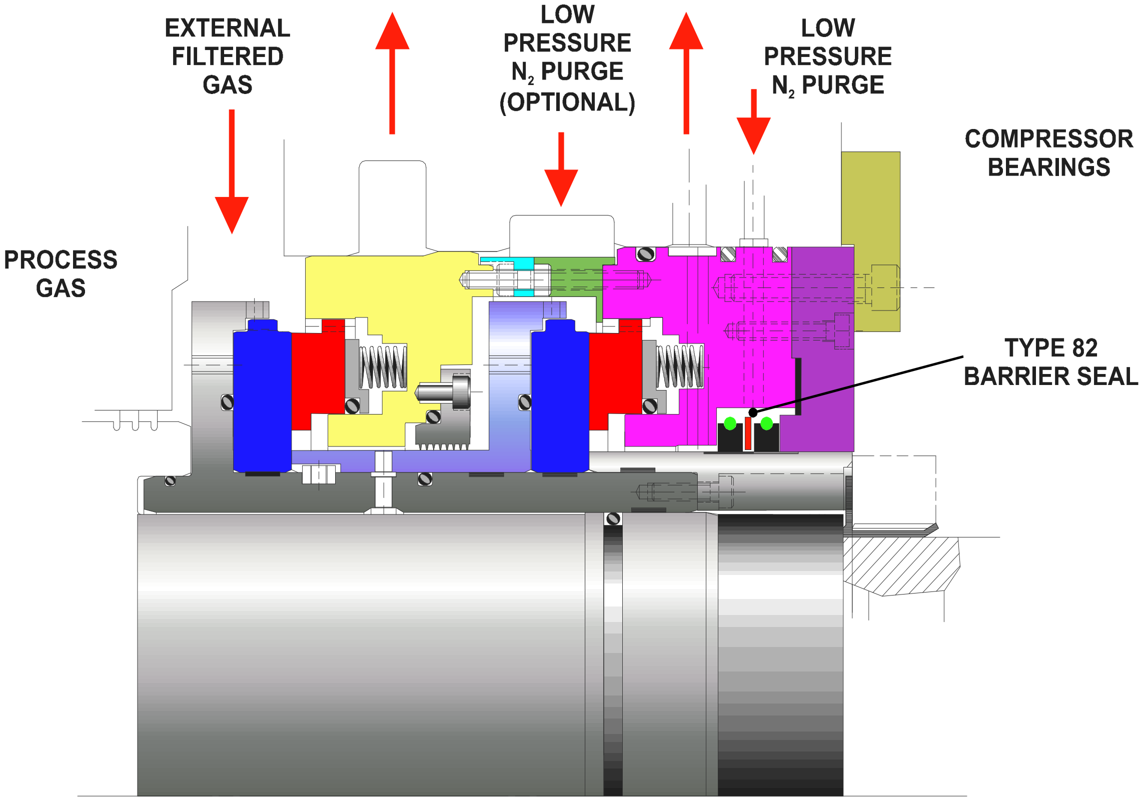
Traditional oil ring seals can be unpredictable leading to excitement of the shaft and rotor instability. Dry Gas Seals are very predictable and will not affect rotor stability.

Improved Operational Safety

Elimination of the wet seal system improves operational safety by eradicating any dangerous build up of hydrocarbon gas in the seal oil.

The majority of compressor sealing applications use a tandem seal configuration as shown in Figure 3. The Type 28AT tandem seal is a cartridge design where all components are held within a metal retainer. The stationary section of the seal comprises a spring-loaded carbon primary ring with an o–ring sealing between the back face of the carbon primary ring and the metal sleeve.

Figure 3



TANDEM TYPE 28AT SEAL CARTRIDGE WITH TYPE 82 BARRIER SEAL

The rotating section of the seal comprises a mating ring, normally manufactured from Tungsten Carbide or Silicon Carbide, with grooves machined in to the running face. This component is contained within a metal shroud and driven through drive lugs or flats machined on the outer diameter of the mating ring.

An Introduction to Dry Gas Seal Operation
Principles of Operation

The primary ring floats on a minutely thin film of gas generated by the logarithmic spiral grooves in the surface of the mating ring. As the mating ring rotates, this creates a hydrodynamic effect that draws gas towards the root of the grooves and forces the two faces apart to form the dynamic seal.

The John Crane groove design is a logarithmic spiral. Computer animation shows that there is an optimum groove angle that, during operation, will generate maximum lift. This optimised groove profile is patented by John Crane.

Figure 4: Spiral Groove

By contrast a radial slot will generate lift, but this may be insufficient to ensure that the sealing gap is maintained at all times, thereby increasing the risk of contact, particularly during transient conditions. It can also be seen that pressure generation will reduce when the groove angle is more acute.

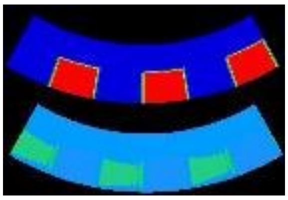


Figure 5: Radial Groove

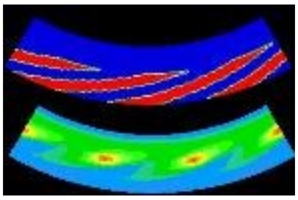


Figure 6: Optimised Spiral

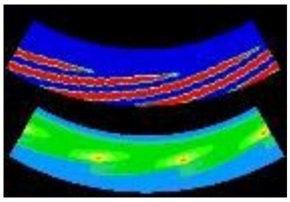


Figure 7: Acute Spiral

A Bi-directional groove design has been developed for applications where there is a real risk of a compressor running in reverse

The groove profile uses the key features of the optimised spiral to maximise lift and separation of the seal faces. The optimised groove angle is also a critical design feature of the seal.

The Type 28AT Dry Gas Seal utilises a short primary ring for maximum operational flexibility.

Figure 8

High pressure applications are sealed with the Type 28XP Dry Gas Seal. The key difference with the Type 28AT seal is the L-shaped carrier design behind the primary ring. This allows for polymer rings to be incorporated, eliminating the potential for O-ring extrusion.

Polymer rings can sustain unlimited decompression rates and have an extremely high resistance to chemical attack, allowing most corrosive gases to be sealed.

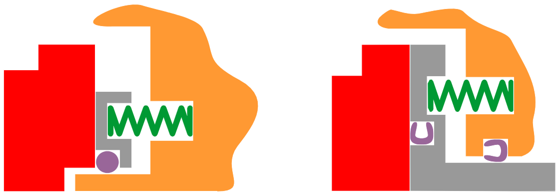


Figure 9: Type 28AT

Under design conditions the forces acting upon the seal in operation can be graphically represented by those shown in figure 11 producing an operating sealing gap of approximately 0.003mm

Normal Gap

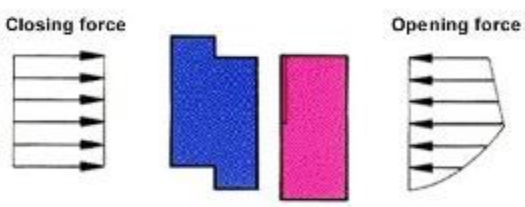


Figure 11

The closing force is a result of the system pressure acting behind the primary ring plus a small force from the springs. The opening force is a combination of the system pressure, plus the force generated by the spiral grooves as the mating ring rotates with the compressor shaft. Equilibrium in operation, with the designed sealing gap, is achieved when the

Reduced Gap

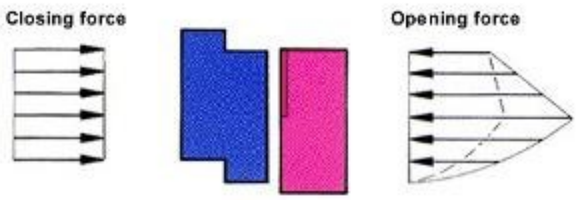


Figure 12

Whilst conditions remain steady and the faces remain in the same parallel relationship, the seal will continue to operate in the mode indicated. Should, however, there be some disturbance which results in a decrease in the sealing gap, the pressure generated by the spiral grooves consid-

Increased Gap

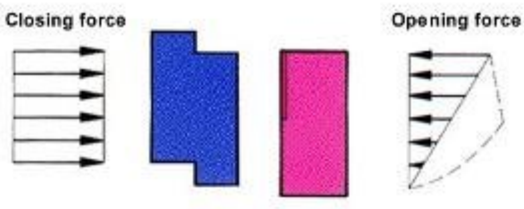


Figure 13

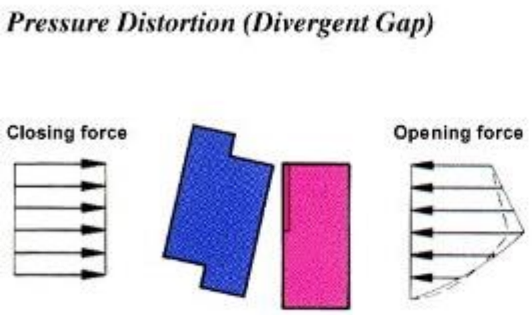
Similarly should the upset cause the gap to increase as shown (figure 13), a reduction in the pressure generated by the grooves will occur. In each case the closing force remains constant and so whichever situation is apparent, equilibrium is quickly established and the design sealing gap restored. This restoring mechanism is known as film

The significant increase in film stiffness with small gap changes ensures that the seal is insensitive to pressure or mechanical disturbances, and there is no direct contact between the face and seat, regardless of system and mechanical upsets.

An Introduction to Dry Gas Seal Operation
Self-Aligning Mechanism For Radial Film Stability

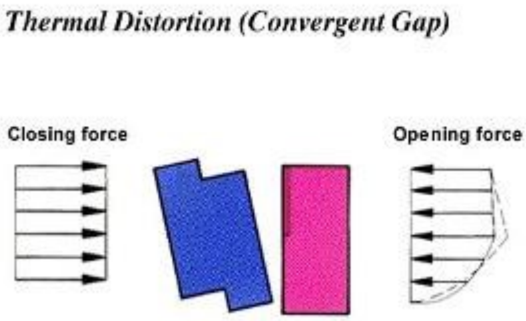
Under ideal conditions the hard rotating ring is perfectly flat and normal to the axis of rotation, however, in practice this is impossible to achieve. There will always be some angular misalignment, whether from manufacturing tolerances or movements of the shaft in operation. The mechanisms within the seal that produce such high levels of film stiffness compensate for these conditions and quickly re-establish equilibrium and film stability.

Ideally there should always be a parallel presentation between the face and seat but in practice angular variations occur. Generally pressure deformations tend to close the faces at the outside diameter producing a divergent gap.



The angular variation brings the outer half of the primary ring closer to the mating ring and the aerodynamic pressure generation rises. As the gap widens in the inner half the pressure profile reduces. The changes in pressure distribution between a parallel and divergent film results in a returning moment, restoring parallel presentation to the operating gap

Figure 14: Pressure Rotatio



Thermal deformations tend to close the faces at the inside diameter producing a convergent gap. The resultant changes in pressure profile again combine to restore equilibrium.

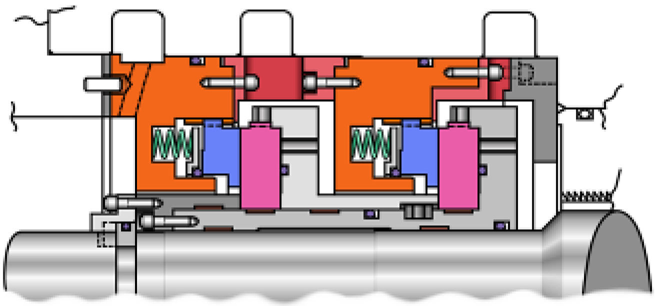
The net result of a highly stiff fluid film is that the optimised spiral groove seal can maintain a minimum running clearance without risk of face contact whilst compensating for a wider range of shaft displacements.

John Crane

An Introduction to Dry Gas Seal Operation
Tandem Seal Configuration

Tandem seal arrangements are a much-favoured solution to most gas sealing problems and ideally suited for flammable, hazardous and low toxicity gases. A tandem seal arrangement may have two or more seal modules oriented in the same direction behind each other.

Figure 16: Tandem Seal



The tandem arrangement may be used to share the sealing load or more commonly one seal handles the full system pressure while the outer seal runs as a standby or back-up seal while functioning as an additional barrier between the process gas and the atmosphere. Very high pressures may require Type 28XP high pressure seals or triple tandems for ultimate safety where the two inner seals share the sealing load, while the out seal is a back-up and barrier seal.

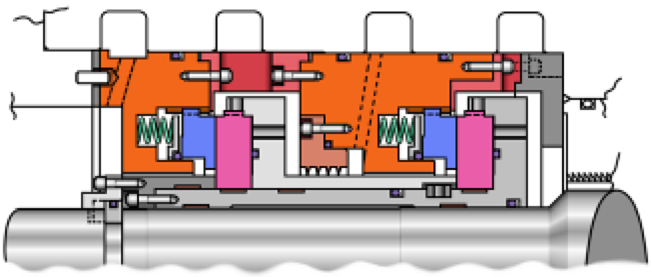


Figure 17: Tandem Seal with Intermediate Labyrinth

A variation of the tandem seal includes a labyrinth in the interspace between the seals. Utilised on more toxic applications, the intermediate labyrinth is purged with inert gas (Nitrogen) directing all the process gas leakage to the primary vent, ultimately to be flared.

There is now an extensive range of gas seal products with variants for pumps, blowers, fans and steam turbines, as well as centrifugal compressors and turbo-expenders.

The type 28AT uni-directional seal is suitable for operating pressures up to 123 bar g, temperatures up to 250°C dependent on elastomer material and shaft sizes to 305 mm. The Bi-directional seal has a similar operating range.

Type 28XP seals are used extensively on high pressure applications and will accommodate pressures of up to 200 bar across one sealing stage.

An Introduction to Dry Gas Seal Operation
Control and Monitoring

The Dry Gas Seal does not require complicated ancillary support equipment. In most cases, all that is required is a simple control and monitoring system comprising fitters, flow metering devices and pressure instrumentation

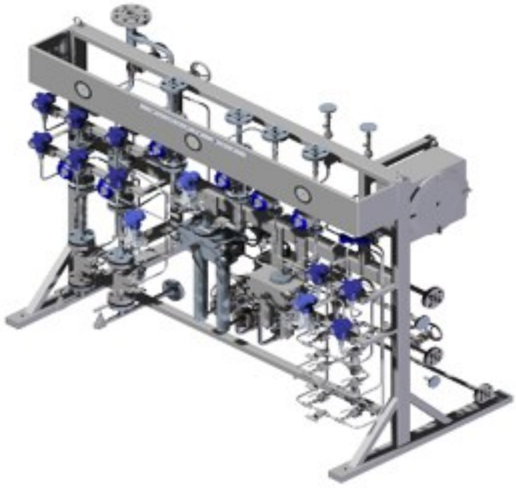


Figure 18: Gas Seal Control and Monitoring Panel

The role of the control and monitoring system is to control the environment of the gas seal, monitor performance and initiate alarms or shutdown should a mal-operation occur.

A single control system will usually supervise several seals in operation. Figure (19) shows a typical flow diagram for a tandem seal with intermediate labyrinth. Continuous monitoring of seal leakage will immediately detect a malfunction in either the inboard or the outboard seals and initiate alarms while the process gas is still safety contained

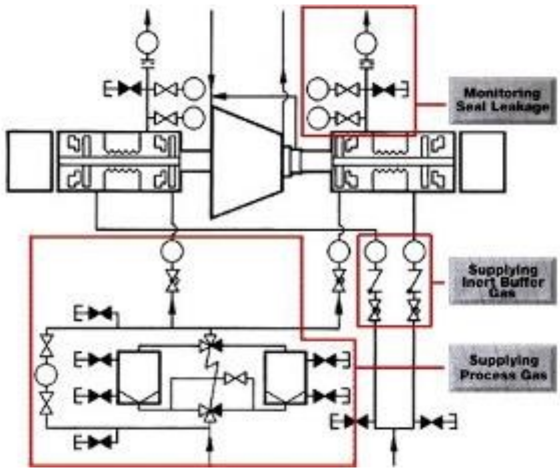


Figure 19

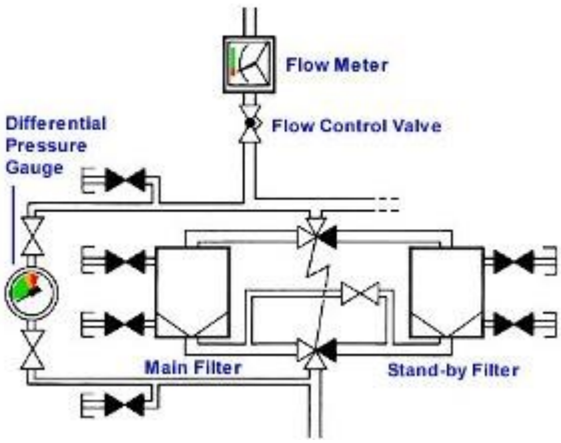


Figure 20: Process Gas Filtration

It is essential that the Dry Gas Seal operates in a clean dry environment. This is normally achieved by circulating the process gas from compressor discharge via one of two 5-micron filters and injected inboard of the seal cartridge at a rate of flow greater than the normal leakage rate of the inboard seal. Flow of the filtered gas may be controlled either with a simple restricting orifice or flow control valve.

During static pressurisation of the compressor, the filtration system is flooded with gas. As soon as the compressor has developed a head of pressure at its discharge, the flow of process gas through the selected filter will commence.

A 'differential pressure gauge' monitoring upstream and downstream pressures across the selected filter will determine filter condition. A 'differential pressure high' signal will alert the operator of the need to change the filter.

Gas seal integrity is confirmed by the systems leakage monitoring instrumentation. While statically pressurised, gas seal leakage is usually slight.

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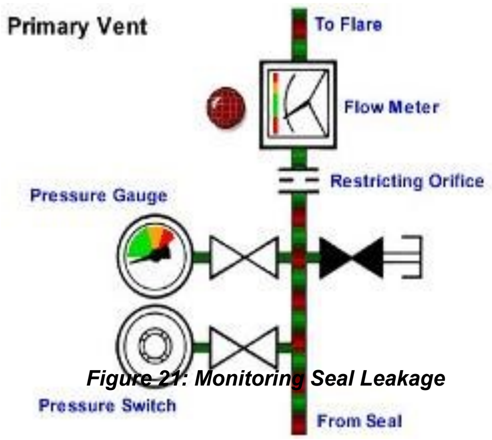


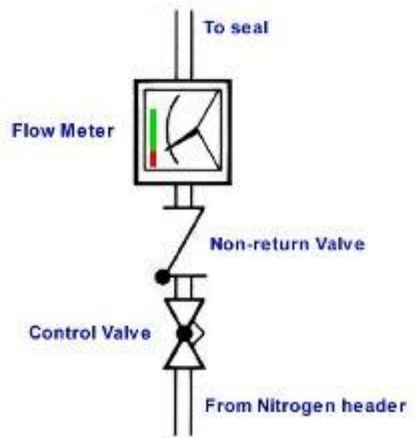
Figure 21: Monitoring Seal Leakage

Under conditions of normal dynamic leakage, a flow will be registered in the primary vent. A reduced primary leakage rate is indicative of an outboard seal malfunction. An inboard seal malfunction will cause an increase in primary leakage. A flow meter with high and low leakage alarm signal will give the operator warning of the malfunction

Should a serious breach of the inboard seal occur, a restricting orifice will restrict leakage through the primary vent. A trip signal is generated by the pressure increase upstream of the orifice.

Tandem seals with intermediate labyrinth are often purged with an inert gas (Nitrogen). This ensures that process gas leakage from the inboard gas seal is directed to the primary vent.

The outboard seal operates on the inert gas and process gas is prevented from entering the bearing area of the compressor. Inert gas flow is controlled by a simple control valve and monitored by a flow meter. A low flow alarm provides warning of inert gas head failures.



An Introduction to Dry Gas Seal Operation
Key Operational Considerations

Experience has shown that the majority of seal malfunctions are caused by contamination of the seals by solids and liquids. It is particularly important for reliable operation to keep the Dry Gas Seal clean and dry. This can normally be achieved by circulating the warm gas from compressor discharge via the filter to the seal chamber.

Dirty Gas

In some instances the process gas may be considered particularly dirty and during periods of standstill there may be a risk of the dirty gas entering the seal cavity. In such cases it may be appropriate to utilise a clean buffer gas. This must be compatible with the process gas and be available at a pressure higher than system pressure. The buffer gas is circulated to the seal chamber via the filter gas supply system.

Gas Condensate Liquids

Condensing of the compressed gas occurs mainly when there are significant quantities of heavy hydrocarbons present. These will condense out when the temperature is below the dew point of the gas.

Gas condensate liquids can present themselves as an oily, sticky substance. When present in the seal area this can coat all seal components. The liquid will congeal and clog, preventing free movement of the seal components.

Condensing of the compressor gas is most likely to occur:-

A. When the filtered gas stream pressure reduces through a throttling device such as a restriction orifice or pressure regulator. As the gas expands the 'Joule Thompson' effect causes it to cool and the heavy hydrocarbons condense out as liquid.

B. As a result of cooling when the compressed gas is circulated through pipework from compressor discharge via filters to the seal area.

C. During static settle-out conditions when the compressor casing is pressurised and the temperature drops below the dew point of the gas.

Small quantities of condensate can be tolerated by the Gas Seal due to the heating effect of the small gap during operation. The seal generates a small temperature rise (typically 20°C) that is normally sufficient to "boil away" condensate.

For applications involving substantial amounts of heavy hydrocarbons various solutions to prevent the formation of gas condensate liquids including the following:

I. Maintain the temperature at a level above the dew point of the compressor gas. Heat trace filter gas pipework if necessary or source clean gas supply from a warmer gas stream of the compressor.

II. Keep length of filter gas pipework between compressor discharge and seal chamber to a minimum. Lag pipework to reduce heat loss.

III Minimise pressure differential across throttling device in filter gas line, which will limit the cooling effect of the gas. Consider receiving the compressed gas from an intermediate stage of the compressor.

IV. Install coalescing filters in the filter gas line whereby the free liquid is removed from the gas. It should be noted that this will only remove free liquid and not heavy hydrocarbons entrained in the gas.

V. Control the seal environment by circulating a filtered dry external buffer gas to the inboard Dry Gas Seal.

VI. Avoid or minimise duration seals are subjected to high pressure settle-out condition.

VII. Install a double seal design with a pressurised Nitrogen barrier gas between the seals.

Avoidance of gas condensate liquids may require any one or a combination of the above solutions dependent on risk or severity. A full analysis of the process gas should be made to address the potential for liquids to condensate out.

Sour Gas

The application of Dry Gas Seals to Sour Gases has been extensive. Many of the Natural Gas applications in offshore environments and Hydrocarbon rich applications on Refineries include quantities of hydrogen sulphide or sulphur.

Critical considerations are material selection, leakage hazards and environmental limitations.

Materials are selected in compliance with NACE specification MR0175, which specifies acceptable materials, the requisite heat treatments and the maximum permitted hardness, for avoidance of sulphide stress cracking.

Generally, for Natural Gas and Hydrocarbon recycle applications, tandem seals are preferred.

To prevent sour gas leakage towards the compressor bearings, an intermediate labyrinth may be installed between the seals and purged with Nitrogen to ensure that under normal operating conditions sour gas leakage through the inboard seal is directed to primary vent to be flared (Figure 17 shows a typical arrangement).

In extreme Sour Gas applications it is normally required that the compressor gas is fully contained with no venting to atmosphere. In such cases, either an external buffer gas is circulated to the inboard seal or a double seal design is selected with a pressurised Nitrogen barrier gas between the seals.

Hydrate and Ice Formation

Under certain conditions hydrates may form in hydrocarbon gases. This occurs when molecules of water attach to elements of the hydrocarbon gas to form crystals. The conditions promoting hydrate formation are shown as follows:

1) Gas is at or below its water dew point with “free” water present

2) Low temperature

3) High pressure

Secondary conditions include:

- High gas velocities
- Pressure pulsations
- Any type of agitation
- Introduction of hydrate crystals

Cold ambient shutdown conditions are those more likely to cause hydrate formation in the gas seal cartridge, when the inboard seal is pressurised to compressor casing settle-out pressure.

There is also potential for icing if there is a release of water from the gas during high pressure cold ambient shutdown conditions. As the gas containing the 'free' water or vapour expands across the inboard seal faces, the Joule Thomson expansion cooling can potentially form ice.

Applications where there is a risk of ice or hydrate formation during prolonged periods of pressurised shutdown should incorporate a purge of the gas seal interspace with 'warm' buffer gas flow prior to compressor start-up or some method of gas seal cartridge warm through.

Wet Chlorides

Wet chloride is an aggressive contaminant. Material distress takes the form of pitting and stress corrosion cracking. Tungsten carbide and stainless steels can degrade in the presence of wet chloride contamination so alternative materials such as duplex stainless steels, hastelloy and silicone carbide are normally selected.

It can be seen that where gas conditions exceed the design criteria of the seal (as in very dirty gases) the seal environment is adjusted to assure long and trouble free life.

An Introduction to Dry Gas Seal Operation
Analytical Tools

As the leader in the field of gas lubricated seals, John Crane has developed a unique suite of computer analysis tools. Known as CSTEDY™ and CTRANS™ these computer tools make it possible to accurately predict all aspects of seal performance under a wide range of conditions.

Earlier programmes solved for two-dimensional axisymetric systems for liquid seals, looking at heat transfer, temperature distribution and complete finite element analysis of seal face deflections. The currency programme has been extended to cover the Gas Seals that have both compressible sealing mediums, and special features such as the spiral grooves on the mating ring. These comprehensive computer tools have been developed to predict seal performance under both steady stage and transient conditions.

The analysis tools have a wide range of uses due to the ability to accurately predict seal performance for specific operating conditions. They are applied extensively for new product optimisation, analysis of existing seals for a new application and analysis of seals for application to off duty conditions. Should the need arise it can be used to identify quick solutions when troubleshooting.

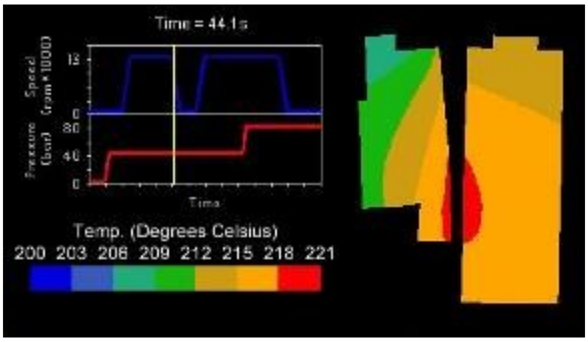
A fully integrated solution is achieved involving interface hydraulic/contact pressures, temperature distribution, film thickness pressure and thermal distortion.

The sophisticated mathematical model takes account of a wide range of system parameters and projects the seal operating performance during both normal and abnormal operating conditions. Input criteria include operating pressures, temperatures, shaft speeds, gas properties , seal geometry and materials. The programme will provide predictions of seal leakage, friction / power consumption, film thickness and operational stability.

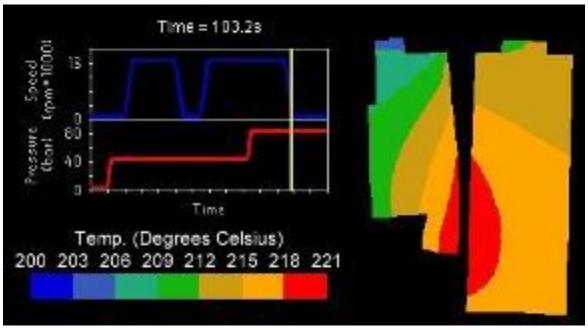
By utilising accurate mathematical modelling tools to design and stimulate seal designs, John Crane is able to meet customer requirements for a reliable product achieving greater productivitym, reduce equipment downtime, and lower maintenance costs whilst complying with environmental and safety requirements.

The sequence of frames in Figures 23 to 26 illustrate the capability of CTRANS™. The output from the computer model shows a typical operating cycle for both inboard and outboard seals. The computer can run these frames on a continual basis showing actual movements and deflections experienced by the sealing faces. For the purposes of this paper, however, four still frames are shown

The sealing gap is shown with time and pressure and temperature scales shown left. Figure 23 shows operating conditions of 40 bar pressure and 13,000 rpm on the inboard seal of the Tandem cartridge



Figures 23



Figures 24

The pressure is suddenly increased to 80 bar, as shown in Figure 24, which results in changed face deflections and a rapid change in temperature gradient due to the higher thermal conductivity of the gas at higher pressures

Figures 25 and 26 show the outboard or low pressure seal of a Tandem arrangement. Figure 25 shows normal operation for this seal at low, or flare line, pressure and 13,000 rpm. The outboard seal is different to the inboard seal in that it is designed to operate continuously at low or ambient pressure conditions. Under these conditions the high heat rotation is now countered by pressure rotation, and the spiral grooves are designed to increase lift to counter these effects.

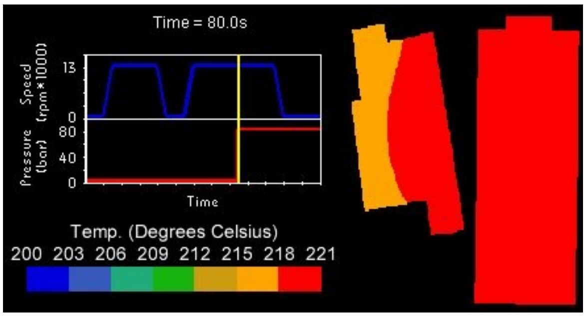


Figure 25

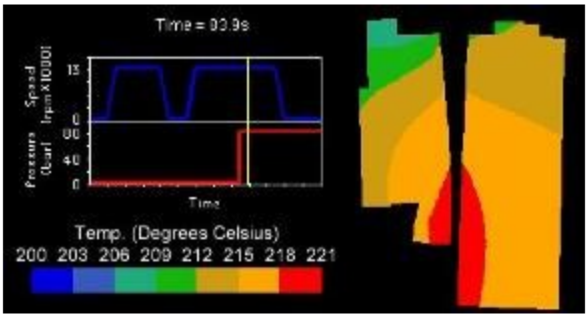


Figure 26

A high thermal rotation is shown in Figure 25 that rapidly changes when an inboard seal failure is simulated. The sudden increase in pressure shown I figure 26 has the effect of lowering the seal temperature, due to the higher leakage and heat conductivity of the high pressure gas, and increasing pressure deflection, which counters the remaining thermal rotation. This has the effect of reducing face running angles and producing a more parallel running gap.

The computer model illustrates very effectively the seal's response to varying operating conditions and is probably the most powerful tool available to the seal designer today. Excellent correlation exists between predicted and actual seal performance and the model has been extensively used to improve the performance of existing seal designs.

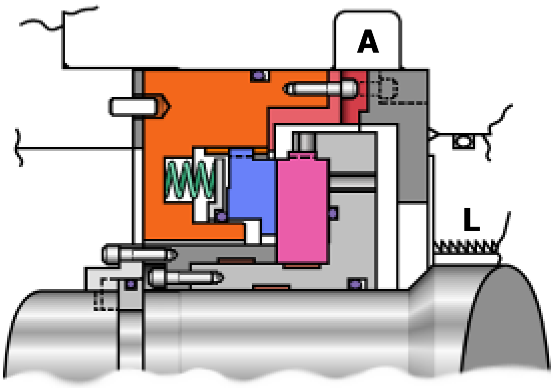
Field and test problems can be swiftly analysed, significantly reducing time penalties, and higher performance seals designed to achieve successful first time testin

Spiral Groove Technology

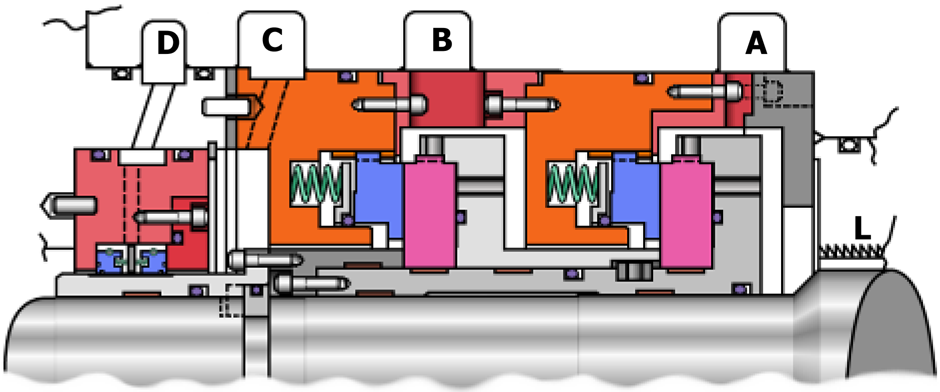
Seals for Gas Compressors

This subject has already been covered in an earlier part of this course, so we give here just the basics of dry gas, non-contacting mechanical seals for turbo-machinery. The theory is similar to that discussed in the previous section, except that now the process gas is used as the "lifting" gas, after it has passed through a control and monitoring system, which also cleans the gas. Single seals can be used for safe gases such as in nitrogen compressors. However, the most common arrangement is the face-to-back tandem, with two seals. (Other arrangements are available, including face-to-face pressurised double seals). For safety reasons, many applications also include a nitrogen injection, in order to prevent process gas leaking out to the atmosphere.

The basic seal has a rotating mating ring, usually manufactured from solid tungsten or silicon carbide. This has the spiral grooves machined on the running face. Against this runs a stationary, hydraulically balanced carbon primary ring. This component is very short, and has a wide radial cross-section making it a very stiff, distortion-free, ring. All rotating components are dynamically balanced to ensure no vibration occurs, and they are centralised with each other and the compressor rotor by "tolerance strips".

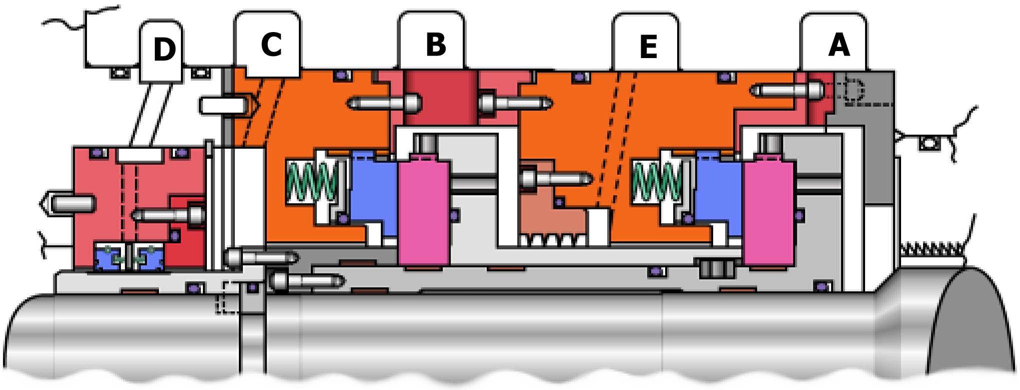


Clean, dry process gas is injected into Port A. Most of this gas returns to the compressor, via the machine's labyrinth (L), in order to prevent dirty process gas from flowing into the seal chamber. A small amount of gas is taken across the seal faces by the spiral grooves, which further compress the gas. When the gas reaches the sealing dam on the mating ring face (the end of the spiral grooves) it expands, lifting the seal faces apart, and drops to virtually atmospheric pressure. Due to the Joule-Thompson effect, the gas also drops in temperature. In a single seal, this small amount of gas is vented to the atmosphere.

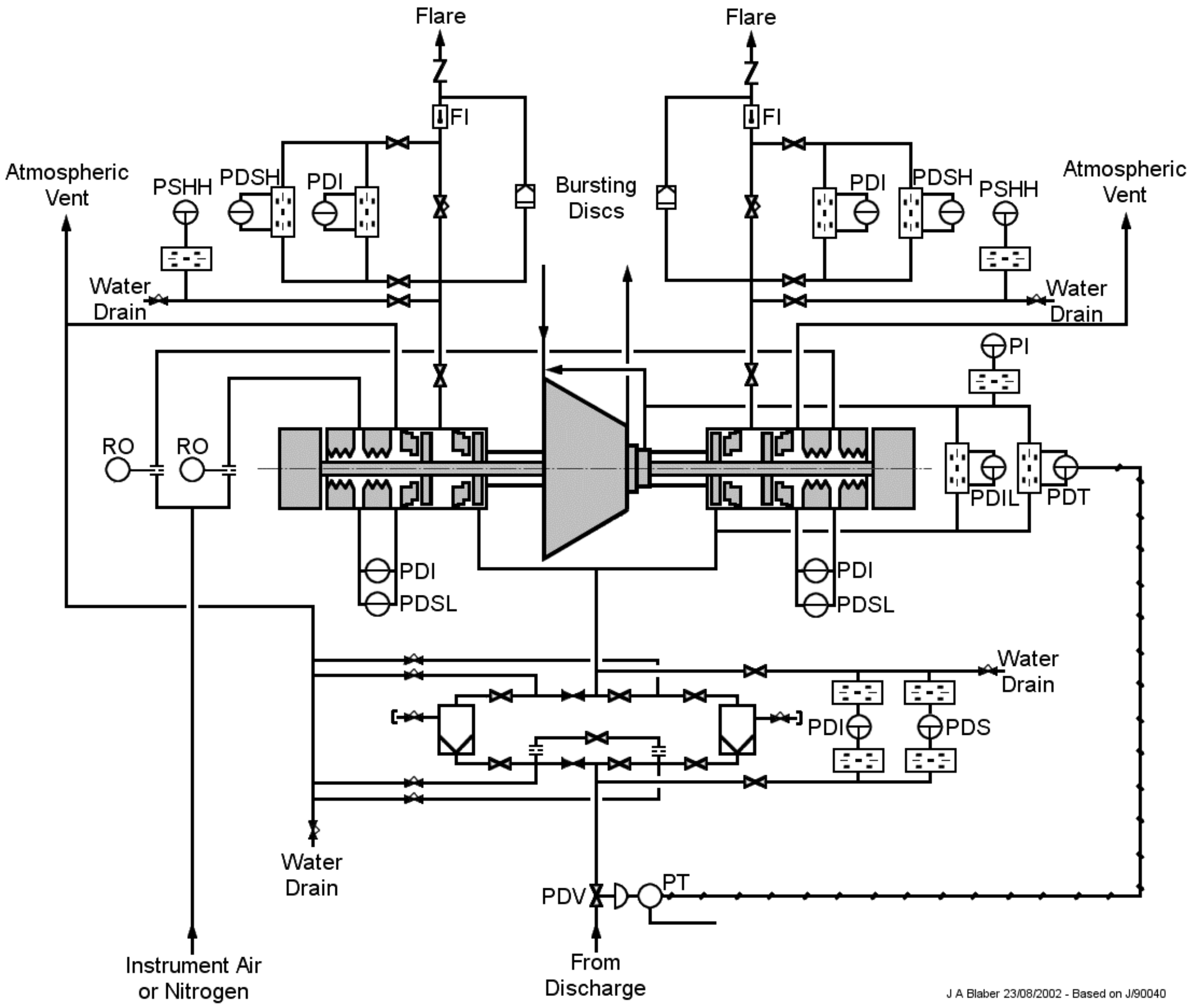


Where the process gas is toxic or flammable the leakage must be contained. If the compressor is a critical machine, a back-up seal must be provided. An unpressurised tandem seal solves both these problems. As before, the clean process gas is injected into Port A, with most gas passing back into the compressor via the labyrinth (L). The remainder of the gas passes across the inboard seal faces, dropping to just above atmospheric pressure. Most of this gas exits through Port B to the flare stack. A small amount crosses the faces of the outboard seal to lift these faces apart, and then exits out of Port C, or towards the compressor bearings. In most modern applications, there is a double labyrinth or double carbon containment seal fitted between the outboard seal and the bearings. Nitrogen or some other suitable low pressure gas (<0.5 bar g) is injected into Port D. The flow of this gas splits, with some of it going towards the bearings to prevent any bearing oil getting into the outboard seal and causing premature failure, and the rest of the gas flows towards the mechanical seal, forcing any process gas coming across the outboard seal faces out through Port C, either to atmosphere or a safe area.

Should the inboard seal fail, the outboard seal will act as a back-up, enabling a controlled shutdown to be carried out safely.

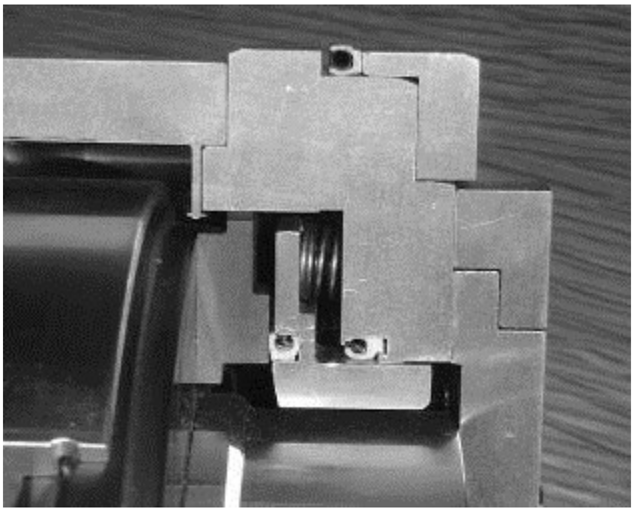


This seal arrangement has the addition of a labyrinth between the two seals, and an additional Port E. This is an extremely safe arrangement, as no process gas will travel passed this labyrinth. Process gas enters Port A as before, but now all the leakage from the inboard seal is vented to the flare stack. Low pressure nitrogen is injected into Part B at such a pressure/flow as to ensure that nitrogen flows through the labyrinth at a minimum speed of 3 m/s, thus ensuring that all process gas is forced up the flare stack vent. A small amount of nitrogen passes across the faces of the outboard seal to operate the lift, joins the nitrogen coming from Port D, and exits through Port C.

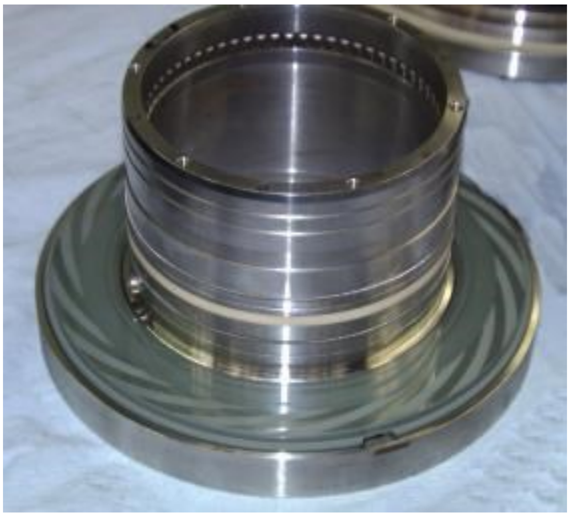


This is a typical P&ID showing gas flows and instrumentation for the simpler tandem seal arrangement shown on the previous page.

High pressure designs now incorporate PTFE/polymer seals in place of elastomer o-rings which were subject to explosive decompression. This picture shows a cross section of a high pressure dry gas non-contacting compressor seal.

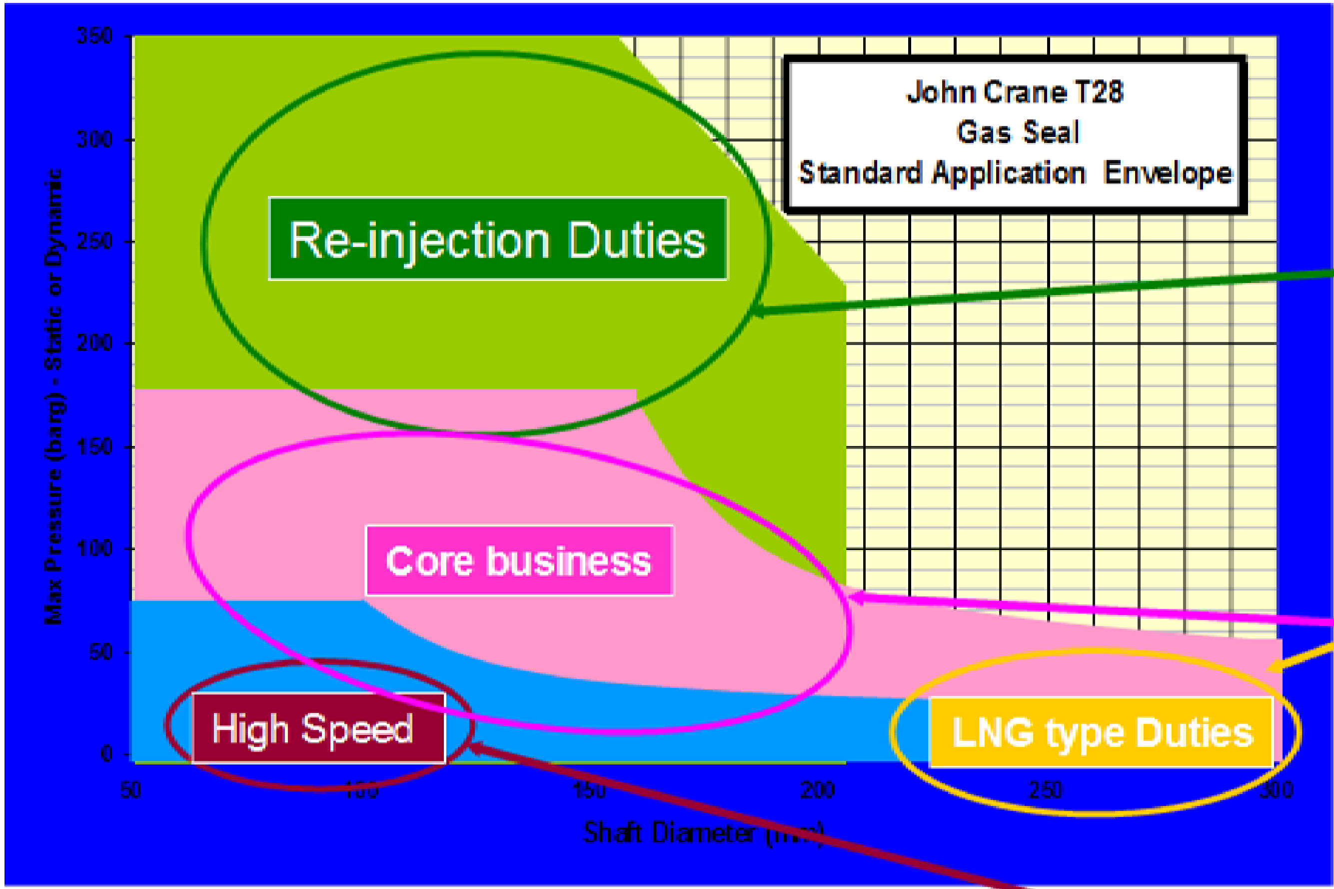


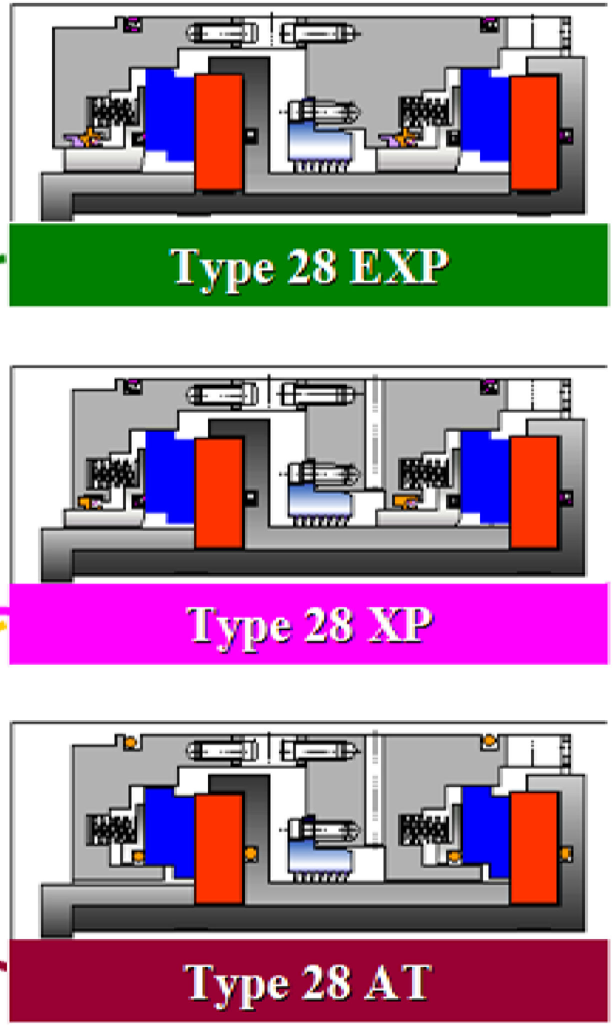
Groove designs are also available to enable bi-directional operation.
The photograph on the left shows uni-directional spiral grooves on the silicon carbide mating ring of a high pressure seal. The picture on the right shows a seal rotor assembly with optimised bi-directional grooves on tungsten carbide mating rings.



The photograph on the left shows a sectioned model of a Type 28XP seal used for Liquified Natural Gas (LNG) applications. It is the largest sized Gas Seal ever manufactured with a Shaft diameter of 350mm.

John Crane Range of Standard gas seals





John Crane dominate High Technology Market sectors

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Seal Operating Parameters

Seal Type	Max. Press. Bar g ❶	~Max. Speed M/s	Max. Temp. °C	❺ Max. Size shaft ø mm	❷ Min. Speed M/s <10 bar g	Max. Static Reverse Pressure bar g	Max. Decomp Rate bar/min >40°C
28AT	82	100	200	240	2.3	0.5	❸ 4 to 20
28BD	82	100	200	240	3.5	0.5	❸ 4 to 20
28XP	170	140	200	280	2.3	10.0	100
28XPBD	170	140	200	280	3.5	10.0	100
28EXP	350	140	200	200	❹ 1000 rpm	10.0	100
28EXPBD	Not available						

❶ Max. pressure varies with size and temperature
❷ Speeds measured at the Balance Diameter
❸ Depends on material of o-ring, pressure and temperature
❹ Barring or Ratcheting not permitted
❺ Larger sizes can be engineered

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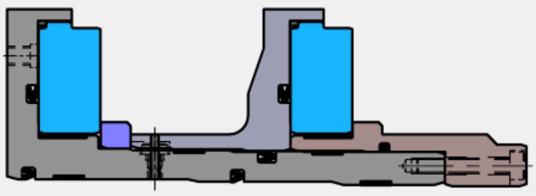
AURA – The Next Generation of John Crane Gas Seals

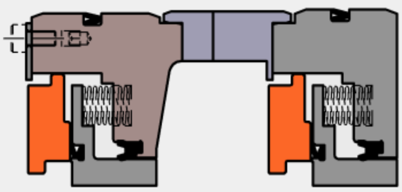
- Expanded product range
- Simplified and common global design
- Increased reliability
- Improved performance envelope

AURA – Innovation

- Optimised rotor design paired with application-appropriate stator
- Common interface geometry



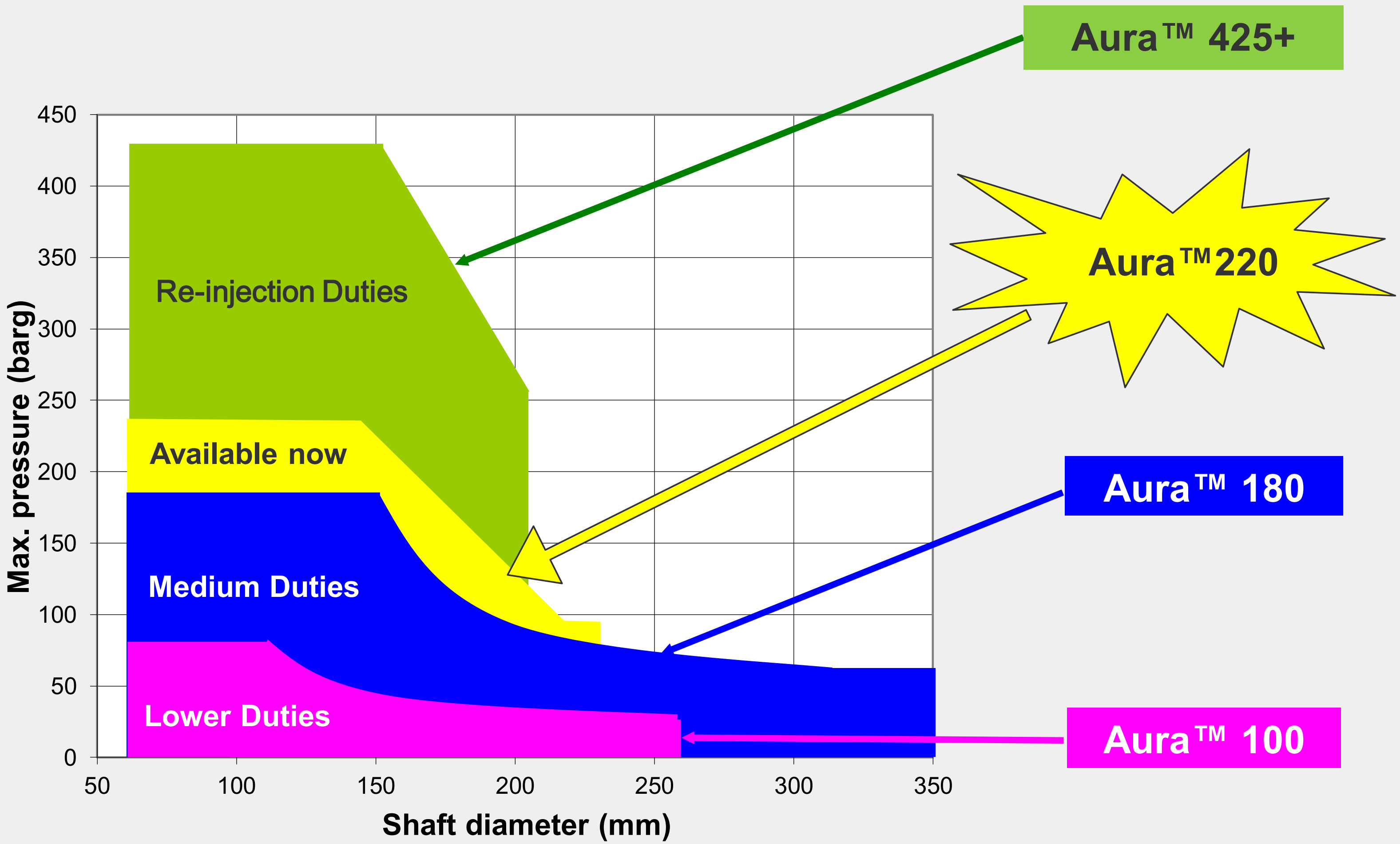
Optimised rotor: common design for all AURA™ seals



Stator changes based on application and performance

New global standard incorporates experience from unrivalled installed base

AURA – Technology Roadmap



smiths

John Crane Proprietary

The following section concerns the Installation and Removal of Gas Seal Cartridges into Compressors. It features GE (Nuovo Pignone) and Dresser Rand installation and Removal tooling and techniques. Although there are many other systems used they all are based on those featured.
The Installation and Removal tooling is generally supplied by the Compressor manufacturer not the Gas Seal manufacturer. The basic procedure is however the same whichever manufacturer is used.

Note:
One of the most critical operations in the Installations of Gas Seals is that of lubrication. Failure to you the correct lubrication can result in damage to both the Gas Seal and the Compressor.
John Crane recommend the following Anti-seize compounds for use on the shaft and in the bore of the Seal sleeve:
Copper Based, Nickel Based or Molybdenum Based Anti-seize Compounds.
Silicon Grease must **not** be used on either the shaft or in the bore of the Seal Sleeve as this can result in serious damage.
All external O-rings should be lubricated using a thin coating of Silicon Grease.
All external Polymer Seals should not be removed or lubricated.
If in doubt please contact John Crane.

Copper Anti-seize Compound (left) and
Silicon Grease (above)

Prior to any Installation or Removal of a Gas Seal Cartridge it is essential to have the Gas Seal Installation Drawing available for the end being worked on. These drawings are found in the Compressor Manuals and provide important information relating to the Gas Seal Cartridge.

Gas Seal Installation using GE Nuovo Pignone tooling.

Prior to any installation being carried out at either end of the Compressor it is first necessary to establish the correct axial running position of the shaft relative to the housing. Once this position is established it is necessary to axially lock the shaft in this position to prevent any movement either axially or rotationally. When installing the Non Drive End Seal it will be necessary to lock the shaft as indicated above. Typically when installing the Drive End Seal the shaft is locked by the Thrust Bearing. In order to check for any shaft movement it may be useful to use a dial gauge set at zero when the shaft in its correct running position.

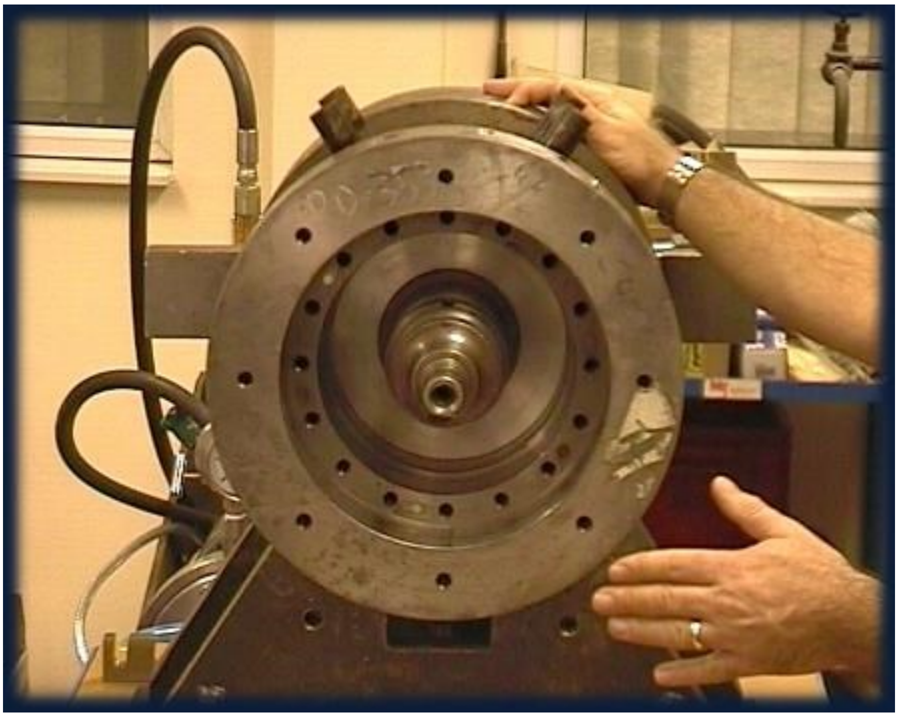
Ensure that the sealing region of the Compressor is thoroughly clean. All cleaning of pipework should be done away from the seal to prevent dirt entering the sealing region. At this time it is recommended to visually inspect the condition of the Seal gas Filters. If they appear to be clogged they must be replaced with new elements.
There should be no sharp edges or burrs on the shaft as this will damage the Gas Seal Sleeve bore and o-ring / polymer seal.

To establish the correct working position of the Gas Seal it will be necessary to measure the working length of the seal with its Setting plates installed. This working length has a tolerance of +/- 0.5mm. If the length is outside this tolerance then it will be necessary to contact John Crane.

Further measurements are carried out on the Compressor. These will be taken from a machined datum surface as indicated above. For better accuracy it is advised to taken three readings and to calculate the average value.

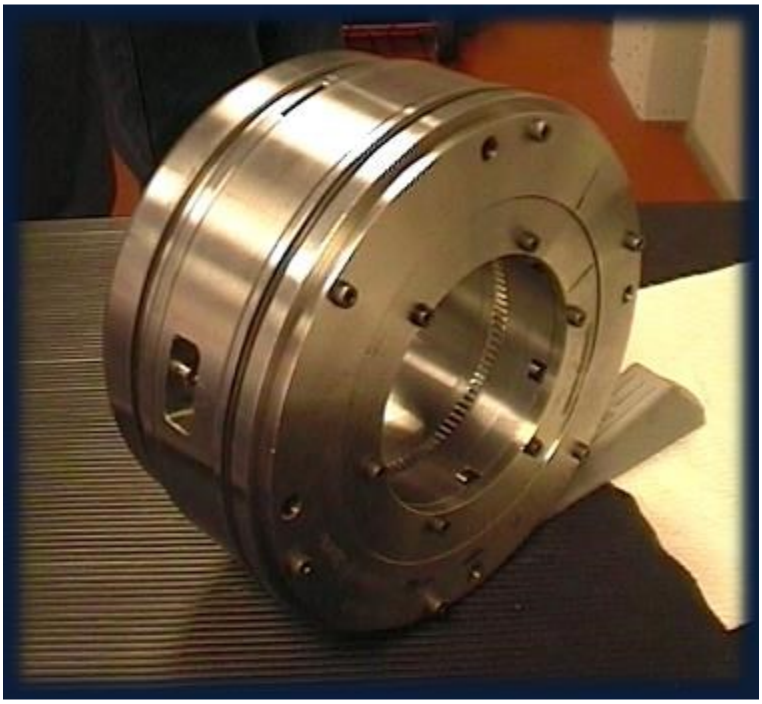
Once the measurements have been taken it will be possible to calculate the thickness of the shaft shim. The nominal dimension is given on the Installation drawing but this needs to be checked each time a seal is to be installed. If the shim needs to be machined it must be ground flat and parallel prior to installing onto the shaft.

The shim has a small clearance with the shaft and needs to be carefully installed up to the shaft abutment face. This surface will be the first point of contact of the Seal Cartridge during installation.



After the cleaning process has been successfully completed it will be necessary to turn the shaft so that the keyway slot is at top dead centre. This will make it easier to locate the key in the seal rotor.

It now will be necessary to lubricate both the shaft and the housing. The shaft should have a slight discolouration from the anti-seize compound only to lubricate the seal sleeve bore o-ring. Excessive lubrication will lead to a “wiper effect” which results in a build up of lubricant at the inboard end. The housing also needs to be lubricated using a slight discolouration of anti -seize compound to prevent any galling from occurring. Ensure that no anti -seize compound get into any of the inlet / outlet ports.



The Gas Seal Cartridge should be wrapped in plastic and stored in a blue flight case. It should be labelled to indicate the shaft direction as well as its individual Cartridge number. After careful removal of the wrapping inspect the seal to ensure that there is nothing missing and that the tolerance strips are firmly attached in the sleeve bore.

It will be then time to exercise the seal to check its free axial and rotational movement. First unscrew both sets of setting plate screws by 2 whole turns and then apply even hand pressure to the seal stator to compress the seal, then slowly release pressure, the seal should expand as the springs extend. Second rotate seal stator relative to seal rotor in the correct shaft rotation. There should be no resistance between the rotor and stator.
Note: on some seals there is a small locking screw to lock the two plates.

After repeating procedure 4 times, remove setting plates by removing both sets of setting plate screws.

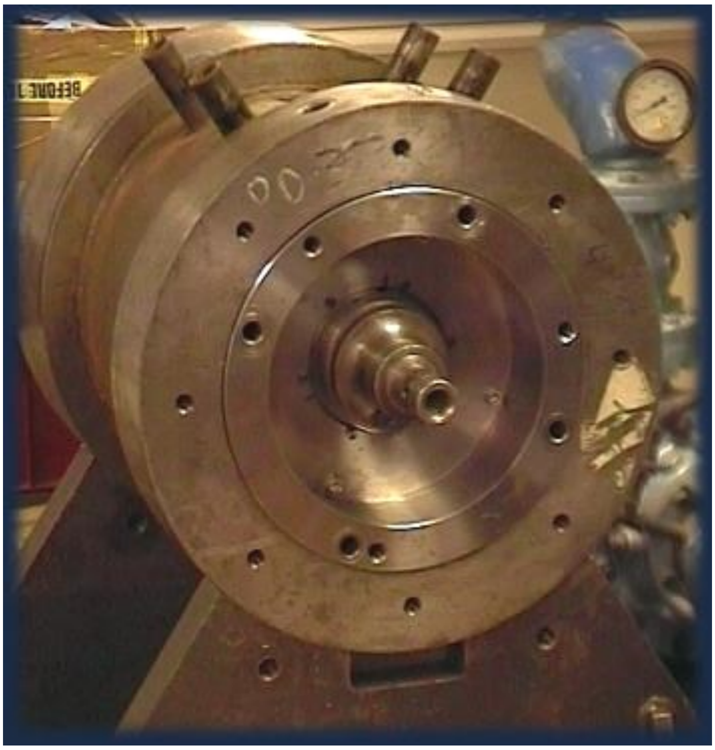
Install fitting tool onto seal cartridge. Note: some fitting tool designs replace the setting plates and are fixed directly onto the seal. Other designs have the fitting tool fixed onto the seal with the setting plates in place.
When tightening both sets of screws, loosen them by 1/8 turn to allow for the seal rotor and stator to self centre during installation operation.

Lubricate the bore of the seal by lightly applying anti-seize compound on the peaks of the tolerance strips. Wipe off excess compound. There is no need to lubricate any other bore surfaces.

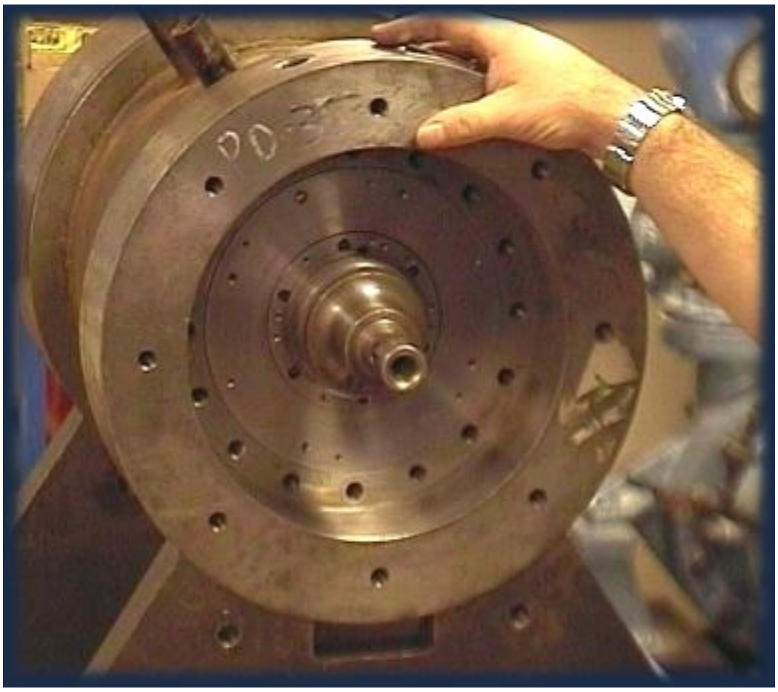
Remove all external o-rings and lubricate using suitable silicon grease. Remove any excess grease after refitting in their respective grooves.
If the seal is a type 28XP or 28EXP then do **not** remove any polymer seals and do **not** grease them.

Install a shaft fitting sleeve if provided at this stage. Lift seal cartridge on to shaft either by hand or by suitable lifting equipment. Ensure that 4 guide rods are in position as indicated above. Fit first few mm by hand pressure only. This will engage the shaft bore o-ring / polymer seal on the shaft.

Then use correct fitting tools to jack seal evenly into position. Ensure that only one person jacks the seal into position. Also check the reference dimension to determine if that shaft has moved axially in the process. Correct shaft to normal running position if necessary. Ensure that key has engaged in its slot in the shaft. If it is not located correctly it will be necessary to remove the seal.



When the seal is fully installed, remove all installation tooling and unscrew all fitting tool screws. Again ensure that reference dimension is correct.



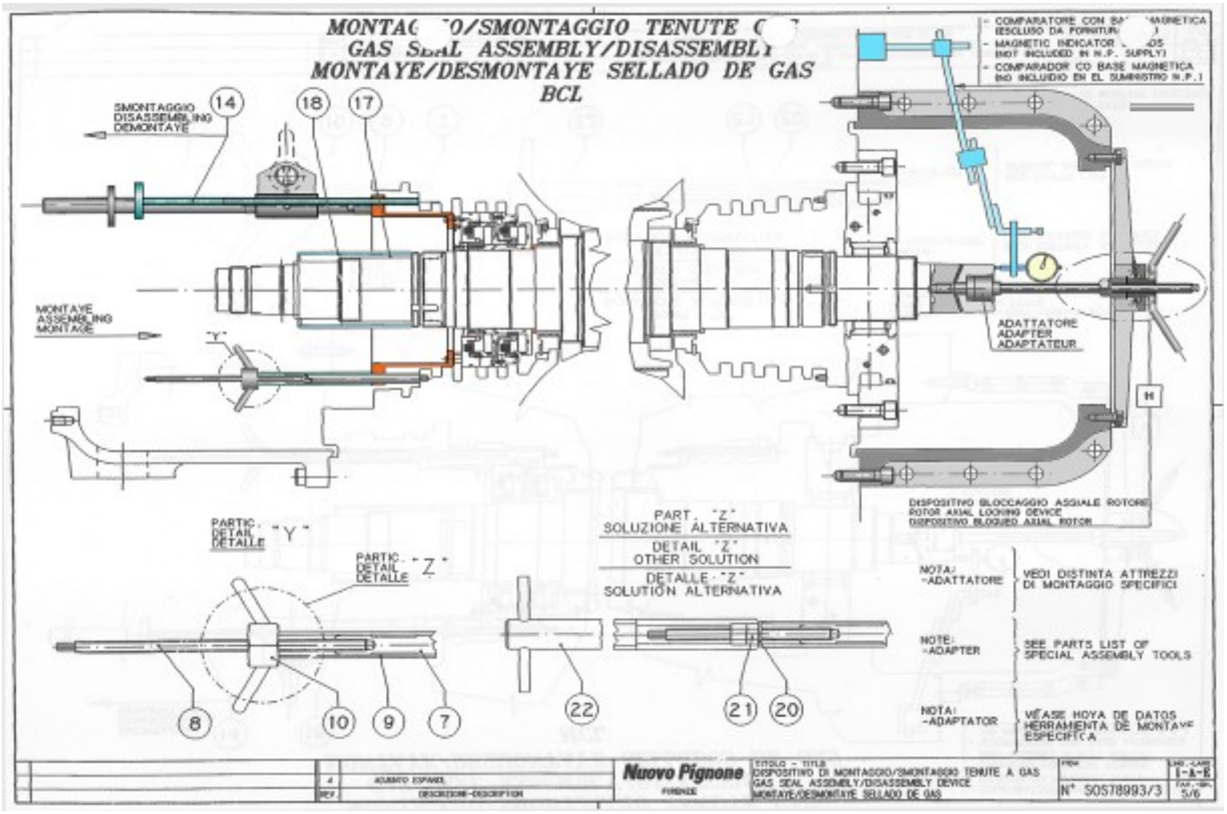
Once the setting plates have been removed check that seal stator and rotor are in the correct axial positions. If this is incorrect will be necessary to re-move the seal to establish the reason for the inaccuracy. If everything is correct then proceed by installing the separation sealing arrangement and the bearings.

Gas Seal Removal
using GE Nuovo Pignone tooling.

Install fitting tool onto seal cartridge. Note: some fitting tool designs replace the setting plates and are fixed directly onto the seal. Other designs have the fitting tool fixed onto the seal with the setting plates in place.
When tightening both sets of screws, loosen them by 1/8 turn to allow for the seal rotor and stator to self centre during removal operation. Install a shaft fitting sleeve if provided at this stage.

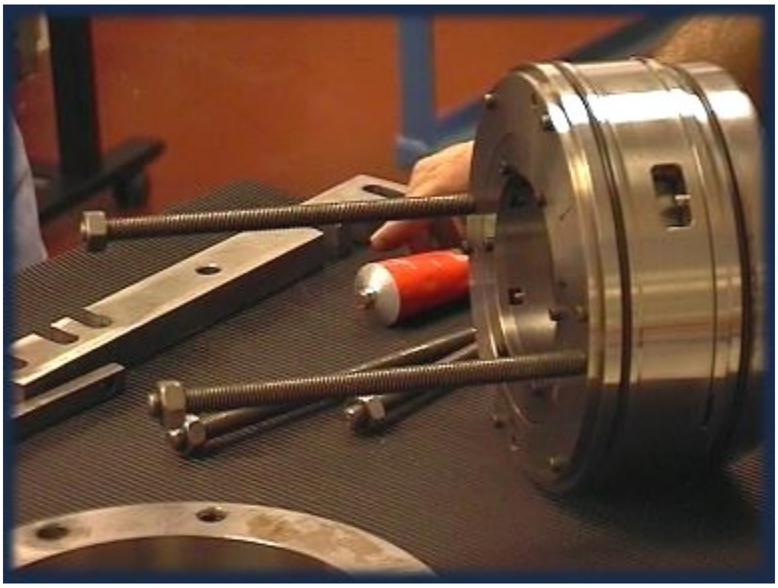
Then use correct fitting tools to remove seal evenly. Ensure that only one person removes the seal. Also check the reference dimension to determine if that shaft has moved axially in the process. Correct shaft to normal running position if necessary.

Once seal cartridge is free on the shaft either lift it off by hand or use suitable lifting equipment.



Gas Seal Installation using Dresser Rand tooling.

Ensure shaft is in the correct axial position. After the seal chamber has been thoroughly cleaned and all sharp edges removed, screw in the shaft stud into the tapped hole in the shaft. Check all shim calculations and machine shims as necessary.

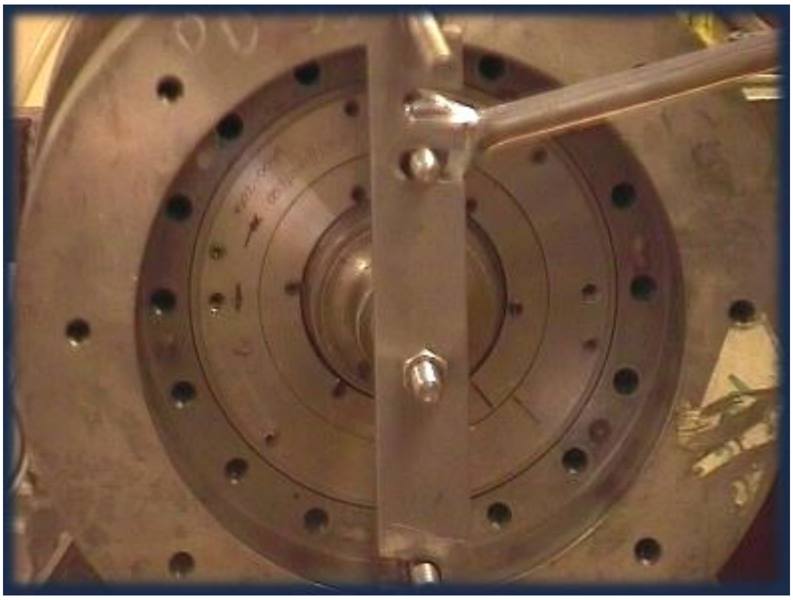


Carry out all seal preparations as stated on pages 60 to 63. After exercising and lubricating the seal as stated on pages 63 to 65 replace setting plates onto outboard end of seal. Screw in two stud bars into the setting plates as shown. Fit two nuts and then washers onto both studs.

Fit a protective sleeve (if available) onto the shaft. Carefully lift seal cartridge onto the shaft using both stud bars. Ensure that seal is square to both shaft and housing. In addition ensure that all pins and keys are correctly aligned. Use measuring equipment to check that the seal is square by measuring in four positions.

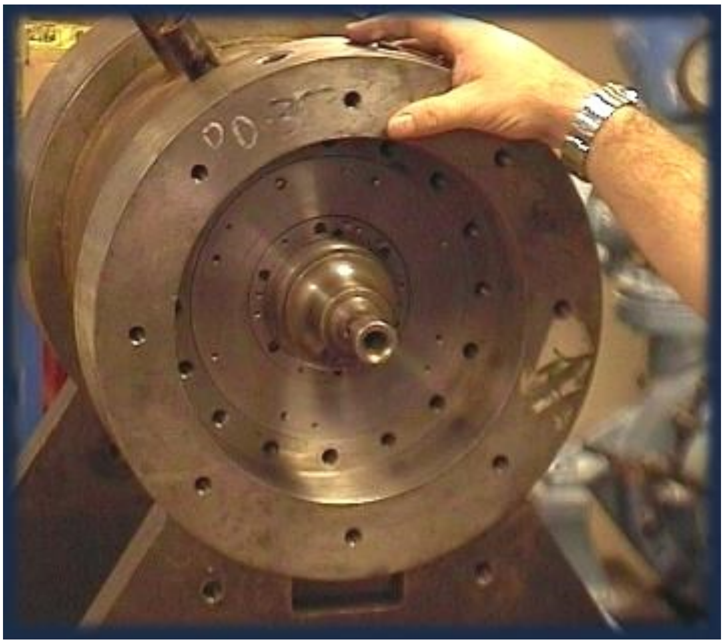
Attach Installation bar onto shaft stud and tighten. Fit two stud bars into housing and lock axial position of shaft using two nuts either side of tool. Ensure that nuts and washers on inner set of stud bars are on the seal side of the jacking tool. Again check that seal cartridge is square to housing in four positions.

Using a spanner unwind evenly the two nuts on the two inner stud bars in order to install the seal into the casing. One person should do this operation. Ensure that seal does not rotate whist jacking and that studs do not unwind from the seal itself. At regular intervals check for square-ness of seal to housing.

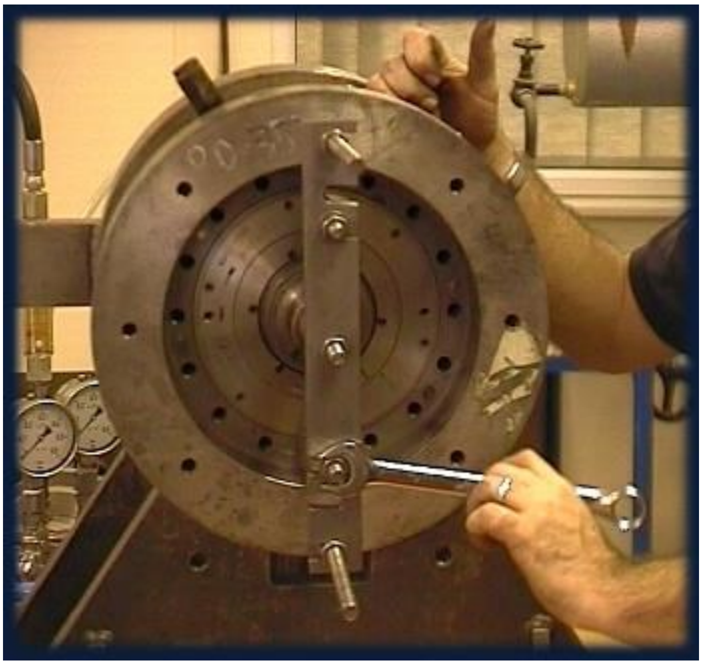


When the seal is fully home the nuts should both go tight. Remove the jacking equipment and setting plates. Once removed proceed with the installation of the Separation sealing device and the remainder of the sealed end.

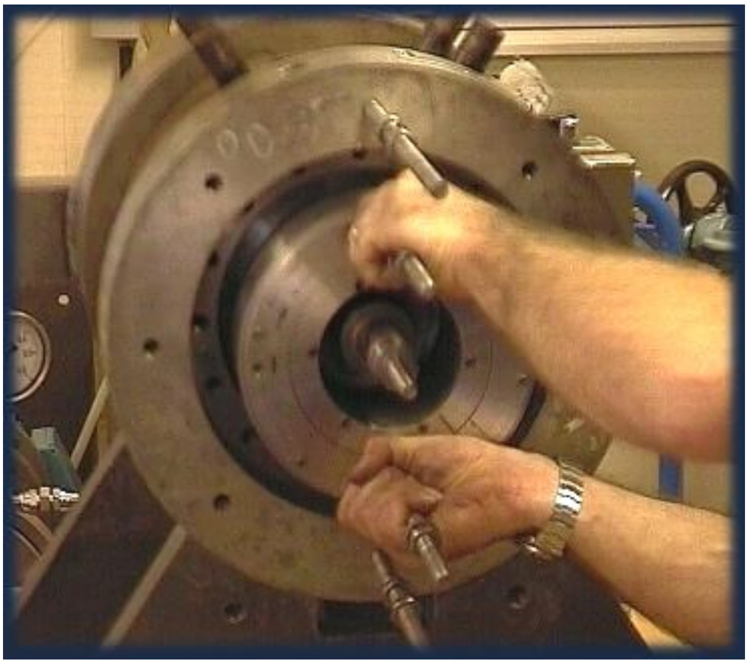
Gas Seal Removal using Dresser Rand tooling.



First ensure that the shaft is in the correct axial position and then fit the seal setting plates. Fit the shaft stud and attach the jacking tool to lock the shaft. Screw in two stud bars through the tool and into the seal setting plates. Then fit two washers and then nuts onto these stud bars. Fit a protective sleeve (if available) onto the shaft.



Using a spanner wind evenly the two nuts on the two inner stud bars in order to remove the seal into the casing. One person should do this operation. At regular intervals check for square-ness of seal to housing.



When seal is free remove jacking bar from shaft stud and outside stud bars.
Carefully lift seal cartridge off shaft by pulling on bars as shown.

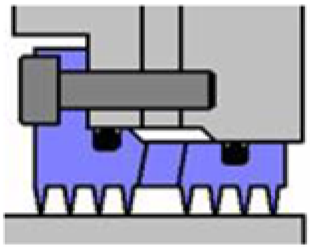
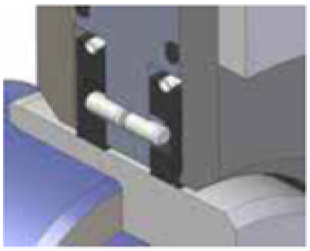
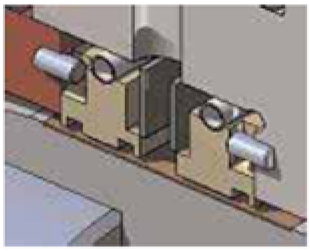
Separation Seals

The seals which prevent ingress of oil from a compressor bearing into the outboard gas seal, and process gas from the outboard stage of the gas seal from entering the bearing cavity, are referred to by several names including:

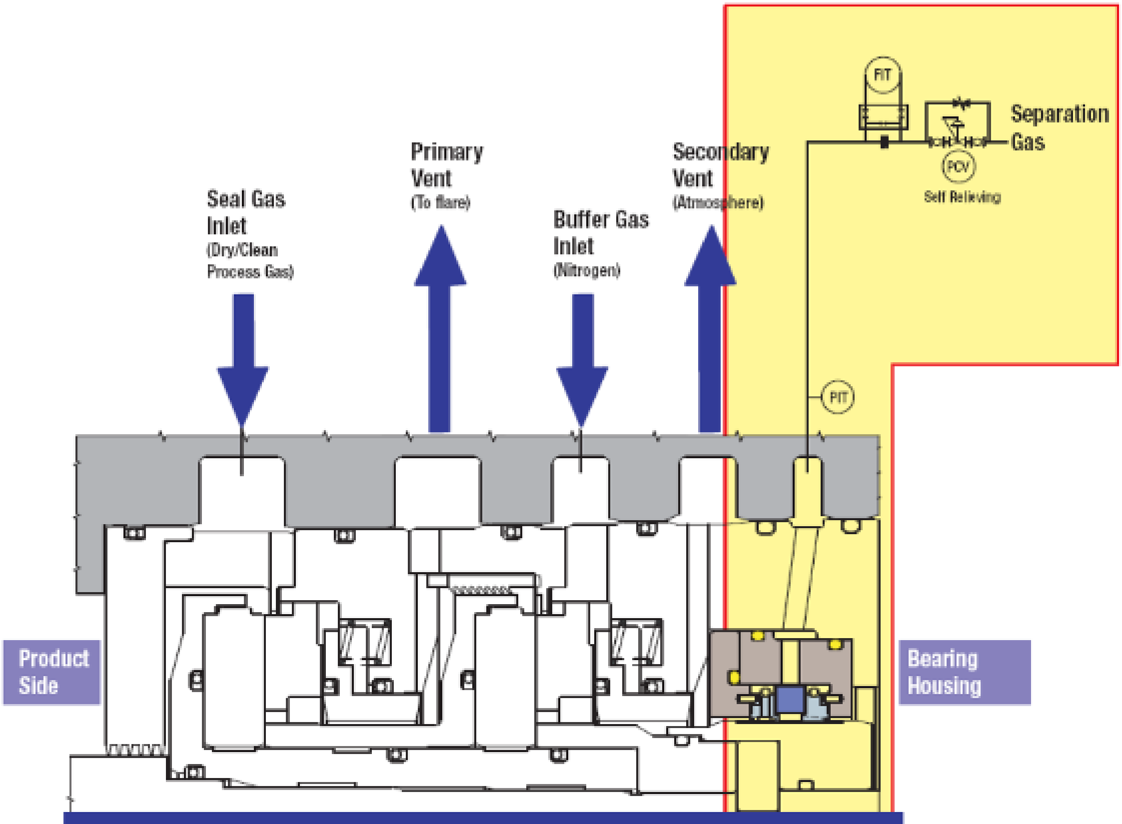
- Tertiary Seal
- Barrier Seals
- Clearance Seals

The API have reviewed this area and our current advice is that they will consolidate on the Term:

"Separation Seals"

Labyrinth	**Clearance Bush (Type 93FR & Type 93HL)**	**Contacting bush (Types 82 & Type 83)**
Advantages: Simple design Simple installation Disadvantages: Very high flow rates No restriction to Oil when not energised	Advantages: Cartridge design Lower dynamic (hot) flow rates than labyrinths Disadvantages: High cold/static flow rates Limited restriction to Oil when not energised	Advantages: Cartridge Arrangement Lowest Flow Rates Positive restriction to oil when not energised Disadvantages: Wearing parts



Typical arrangement with a Type 28XP Tandem Seal

All Separation Seals need to be continuously supplied by a gas such as air or nitrogen at a set level of pressure. This acts to prevent oil migration from the bearings and outboard seal leakage in the bearing housing.

The supply pressure can be controlled by manually setting a needle valve or by the use of a Pressure Control Valve (PCV) as shown above.

Appendix A

Installation, Operation and Maintenance Instructions

John Crane
Engineered Sealing Systems

Installation, Operation and Maintenance

Instructions for Type 28 Gas Seals Used in Turbo-Machinery Applications

This applies to the following John Crane seals:
Type 28 AT
Type 28 XP

John Crane EAA
361-366 Buckingham Avenue,
Slough, SL1 4LU,England, UK

Printed copies are uncontrolled documents

This seal may only be installed, commissioned and maintained by an authorized plant machinery specialist, paying close attention to these instructions and all other relevant regulations. Failure to do this relieves the manufacturer from any liability or warranties.

JC Ref N°: IOM-0250-ver-1.3

CONTENTS

OTHER NECESSARY SUPPORTING DOCUMENTATION:

This IOM must be read in association with the appropriate John Crane seal installation drawing to the latest issue. Consult the equipment supplier's manual for other information relating to operation of the seal and its associated systems. Appropriate alarm and trip levels for leakage need to be available for assessment of the seal's performance.

For ATEX compliant applications the operational specification, as laid down in the contract of supply (and summarised in the ATEX Compliance Technical Data Sheet) must be made available to the operator.

1. INTRODUCTION TO THE JOHN CRANE TYPE 28 GAS SEAL

These instructions describe the Gas Seal's preparation for fitting, operation in, and removal from the compressor. Seal cartridge disassembly, is not covered, as the complete cartridge should be returned to John Crane if refurbishment is required.

The seals are robust in operation, however incorrect handling or assembly fitting can easily damage them. It is therefore recommended that the seals are fitted by a John Crane trained and approved Technician.

If there are any Gas Seal problems that need an urgent response from John Crane EAA, there is a Hotline Number 44 (0)1753 224400 (answer phone out of hours). Emergency line 24 hours + 44 (0)7889 653846.

2. DECLARATION OF INCORPORATON

E.C. MACHINERY DIRECTIVE (98/37/EC AND ITS AMENDMENTS)

DECLARATION OF INCORPORATION

SECTION 1.0 – MACHINERY DESCRIPTIONS

Type 28 Series Dry Running Gas Seal

SECTION 2.0 – APPLICABLE STANDARDS

BS EN 292 Parts 1 and 2

SECTION 3.0 – INSTALLATION, OPERATION AND MAINTENANCE MANUAL

The document of which this declaration is a part.

SECTION 4.0 – DECLARATION

We John Crane UK Ltd, declare that the product described in Section 1.0 above, is intended to be incorporated into machinery or assembled with machinery to constitute relevant machinery as covered by the Machinery Directive and in compliance to other directives where applicable, e.g. the ATEX Directive (94/9/EC).

The component assembly covered by this declaration must be incorporated in accordance with the application guidance supplied by the vendor and the specified conditions in the contract of supply. It must be installed in accordance with its appropriate Installation, Operation and Maintenance Manual described in *Section 3.0* above. It must not be put into service until the relevant machinery into which it is to be incorporated has also been declared in conformity with the provisions of all applicable Directives.

Ian Goldswain
For and on behalf of John Crane

3. SAFETY

The safety notes refer to the arrangement supplied. They can never be exclusive, and must be used in conjunction with the relevant safety regulations for the machine, auxiliary equipment, plant and sealed fluid.

This manual relates to matters effecting the operators of machinery incorporating dry gas seals. It covers areas directly associated with the Installation, Operation and Maintenance of these components. For other related subjects, such as the correct operation of associated systems or guidance when conducting hazard analysis (as may be required under the European ATEX Directives) consult the equipment supplier or their manual.

3.1 Warning Symbols

The following symbols are used in this instruction manual to highlight information of particular importance:

Danger

Mandatory instructions designed to prevent personal injury or extensive damage.

ATTENTION — Special instructions or information to avoid damage to the seal or its surroundings.

Note: Information for easy installation and efficient operation.

3.2 Safety Instructions

ATTENTION

Any working practice that compromises safety is to be avoided.

At all stages in work relating to the seal reference must be made to the installation drawing, which should accompany this document.

In the event of an operating problem the machinery must be switched off immediately and made safe! Problems must be solved promptly.

A small controlled leakage will occur during normal seal operation. In case of a worn or defective seal the leakage will increase. The leakage may be hazardous or toxic, and a safe collection system is required.

Hot surfaces have to be protected against accidental contact.

In order to avoid unforeseen hazards do not make unauthorised changes to the gas to be sealed, the specified duty, or the seal parts.

Follow the local relevant guidelines for the safe and environmentally friendly disposal of assembly lubricants, supplied fluids and scrapped components.

Compounds containing PTFE, fluorocarbons and perfluoroelastomers should never be burned as the fumes and residues are highly toxic. If this or gross over-heating occurs protective gloves must be worn as hydrofluoric acid may be present. As a general safeguard, protective gloves should be worn if handling failed seal parts.

4. SEAL PREPARATION

If seal cartridge installation or compressor commissioning (first start-up) takes place later than 24 months after shipment from John Crane, the seals must be inspected by a John Crane trained and approved Technician.

Seal fitting and removal tools are not supplied by John Crane.

For installation the entire outer surface of the seal cartridge including the bore should be clean and dry. The cartridge should not be cleaned with any fluids or solvents that may enter cartridge or attack vunerable components such as 'O' rings.

If the cartridge is dirty this may indicate the seal has been contaminated and DO NOT PROCEED with installation. Consult John Crane EAA.

All secondary seals ('O' rings or polymer seal) on the outside and bore of the cartridge should be carefully examined for damage such as cuts or crushing. If damaged, they should be replaced with a correct part from the spares pack, in-accordance with the installation drawing.

Apply silicon grease sparingly to outer diameter 'O' rings.

ATTENTION

Silicon grease must not be used on sleeve shaft 'O' rings.

4.1 Exercising the Seal

Prior to seal installation it is recommended that the seal cartridge is exercised to check for free axial and rotational movement.

<u>TO CHECK FOR FREE ROTATIONAL MOVEMENT</u>

a) Position the seal cartridge on a firmly supported surface with the setting rings uppermost

b) Loosen off all the setting ring caphead screws by two complete turns.

c) Remove the anti rotation screw from setting rings.

d) Rotate the seal stator relative to the seal rotor by one complete turn. If this is not possible, there is a problem within the seal cartridge and John Crane should be contacted.

<u>TO CHECK FOR FREE AXIAL MOVEMENT</u>

a) Support the main sleeve as shown below.

b) Remove the setting rings by removing all securing screws.

c) Apply even hand pressure to the seal stator (as indicated in Figure 1) such that the seal is slowly compressed into itself.

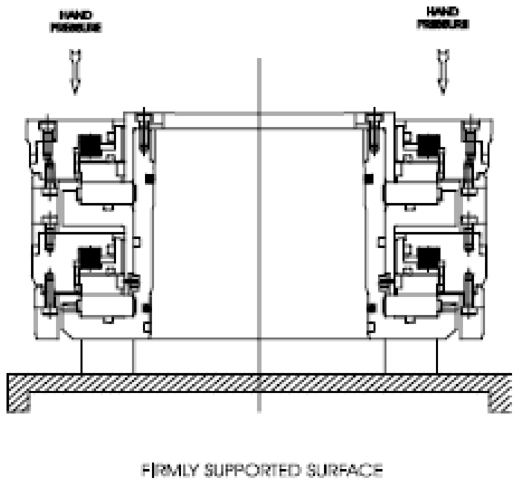


Figure 1

d) When fully compressed slowly release the pressure.

ATTENTION

Use minimum pressure necessary to avoid the risk of impact damage when maximum movement limit is reached.

e) Repeat steps (c) and (d) three times. If it is not possible to compress the seal or it fails to return to it's original position when hand pressure is released, there is a potential problem within the seal cartridge and John Crane EAA should be contacted.

f) Ensure that the rotor and stator are in correct orientation according to the installation drawing and refit the setting rings and all securing screws.

Note:

With large or heavy seals the weight of the static parts may overcome the spring force, which may prevent the static parts from returning to their original position.

4.2 Compressor Preparation

ATTENTION

The compressor casing must be adequately earthed and electrical earth continuity must be maintained. See equipment manufacturer's manual for details.

The following text assumes that the compressor is ready to accept the Dry Gas Seal, and that the required fitting/extraction tools are available.

John Crane recommend Rocol anti-seize compound for shaft/seal sleeve assembly lubrication. (Alternatives such as Molycote MAY be used). This compound should be applied sparingly, and only at the seal sleeve/compressor shaft interface. Silicone grease is used for lubricating stator interface 'O' rings. This too must be applied sparingly.

UNDER NO CIRCUMSTANCE MUST SILICONE COMPOUND BE USED FOR SHAFT/SEAL SLEEVE ASSEMBLY LUBRICATION OTHERWISE SHAFT SLEEVE GALLING CAN RESULT.

Excess grease and compound must be removed, and care taken to prevent ingress of these lubricants into the sealing area during cartridge installation. The suitability of alternative lubricants should be confirmed with John Crane.

a) Clean the entire area into which the seal fits. Ensure that it is free from defects and any rough or sharp edges that could damage the seal on installation.

b) If the dry gas seals have not been previously fitted to the compressor or if there is a new rotor, check that the seal operating envelope(s) are in accordance with installation drawings

c) Apply a film of Rocol anti-seize compound to the shaft.

d) Axial positioning shims or spacers may be provided. If supplied these need to be adjusted by appropriate machining to achieve the correct rotor to stator working position. Once adjusted, fit in the compressor or to the cartridge as shown on the installation drawing.

4.3 Fitting the Seals

Ensure that the direction of rotation shown on the seal corresponds with actual shaft rotation for the end being fitted.

a) Using appropriate lifting equipment carefully position the seal on the shaft.

b) Align any keys, pins, slots or holes on the stator and rotor as appropriate.

c) Using the fitting tools provided by the compressor manufacture (or, if unavailable, studding and jacking bar) carefully and evenly jack the seal assembly into position.

Take care to ensure that the seal is kept square to the shaft to prevent it "binding".

Note:

It may be necessary to axially lock the shaft to prevent it from moving whilst jacking the seal into position.

d) When the seal is fully engaged home, remove all fitting tools.

e) Remove the seal setting rings and their cap head screws as per installation drawing.

f) Lock the seal stator to the housing and the rotor to the shaft as shown on the installation drawing.

Note:

Any barrier seal or separation seal parts should be fitted in accordance with the suppliers IOM.

Note:

It is essential that both the seal rotor and stator are located in correct axial position as identified on the installation drawing. The setting plates can generally be used to verify this.

Any axial miss-alignment of the seal rotor to the stator will potentially result in seal failure.

Note:

It is essential that the rotor is effectivly locked onto the shaft and any locking devices are fully tightened.

Note:

Any retaining screws should be tightened to the torque's shown in table 1 below

Table 1: - Tightening Torque Values

Torque Size	Steel Screw N.m JC Code ****/301/050	Stainless Steel Duplex Screw N.m JC Code ****/310/015
M4	4.6	1,9
M5	9.4	3,9
M6	15.9	6,6
M8	38	16
M10	77	32
M12	135	55
M14	215	90
M16	340	140
M20	663	275

4.4 Removal of Seals

It is assumed that the compressor is in a state where the dry gas seal can be removed.

a) Carefully study the installation drawing. All devices that secure the seal stator or rotor to the compressor casing or shaft must be removed.

b) Fit the seal setting plates as identified on the installation drawing.

c) Fit the seal removal tools provided by the compressor manufacturer (or, if unavailable, suitable studding and jacking bar) to facilitate seal removal.

d) Using these tools extract the seal cartridge from the compressor housing.

Note:

It is imperative that while extracting the seal cartridge, it is kept square to the shaft axially. Other wise it may bind onto the shaft.

Note:

It may be necessary to lock the shaft to prevent it moving axially during this operation.

If the seal is removed without the appropriate setting rings severe damage may occur to the seal cartridge.

e) Using lifting equipment, remove the seal cartridge from the shaft.

4.5 Replacement of Tolerance Rings

The majority of Type 28 seals are fitted with tolerance rings located in the bore of the seal sleeve. The tolerance rings fit into grooves machined into the bores of the sleeve and are glued in place. If damaged, the tolerance strip can be changed though this may adversly affect the seal balance.

It is essential the strip is fully located in the appropriate groove. If excessive adhesive is used shaft damage can result.

The method of installing a tolerance ring is as follows:-

a) Remove the old tolerance ring and all traces of adhesive from the groove in the sleeve bore.

Exercise extreme care when handling tolerance strip. The edges may be sharp.

b) Carefully cut a new tolerance ring (between "waves") to slightly longer than required.

c) Curl to provide the best possible fit in groove, and trim the length such that a 5mm gap is established between the ends of the tolerance ring. As shown in *Figure 2*

Figure 2

d) Chamfer both cut ends, as shown in *Figure 2*

e) Clean the tolerance ring and the sleeve bore grooves with a suitable solvent.

f) Ensure gap aligns with T-balance mark. See figure 3. The tolerance strip is held in place by a two part quick acting adhesive such as Loctite Multi Bond.

g) Apply adhesive sparingly to 5 waves only of one end of the replacement tolerance ring.

h) Apply hardener to groove.

i) Locate and hold tolerance strip in place until set (approximately 60 seconds).

j) Repeat steps (g) to (i) for the other end of the replacement tolerance ring ensuring that it is pushed back well into its groove.

k) When the adhesive is set, remove any excess from the bore.

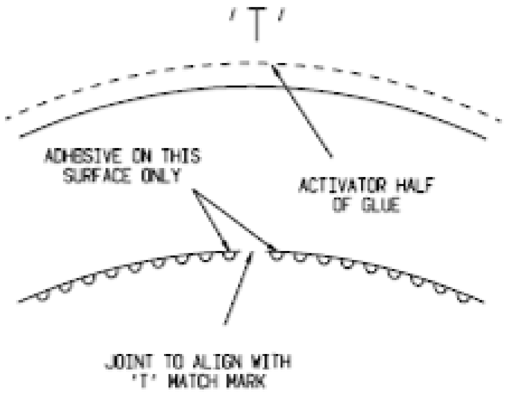


Figure 3.

5. COMMISSIONING PROCEDURE

These tasks should be undertaken following seal cartridge installation and prior to starting the compressor.

5.1 Static Test

a) With the drive coupling disconnected, rotate the compressor shaft to ensure freedom of movement.

b) If a barrier or separation seal is installed, commission the separation gas system and ensure that it is functioning correctly.

c) Pressurise the casing incrementally up to line pressure. Record the inboard seal leakage against each pressure increment.

d) If excessive leakage is observed, i.e. close to or greater than alarm settings, the compressor must be depressurised (*see table 2 for AT seals*) and the cause of the high inboard leakage rectified.

5.2 Dynamic Test

a) Commence normal start procedure, Periodically record inboard seal leakage rate during the first 4 hours of operation.

b) If excessive leakage is observed, i.e. close to or greater than alarm settings, the compressor must be depressurised (*see table 2 for AT seals*) and the cause of the high inboard leakage rectified.

Table 2
Maximum permissible decompression rates to avoid damage to 'O' rings

	Temperature			
Pressure (Bar)	<40ºC bar/min	<95ºC bar/min	<150ºC bar/min	<205ºC bar/min
<82	No Limit	20	20	20
82 - 103.5	20	20	20	8
103.5 - 124	20	20	8	8
>124	8	8	8	4

Note 1: These rates are based on hydrocarbon gases used on 90 IRHD (Crane Code 387) elastomers. For 70 IRHD Viton (Crane Code 134) elastomers the decompression rate is <4 bar/min regardless of the conditions.

6 COMPRESSOR OPERATION AND MAINTENANCE WITH DRY GAS SEALS

The seal must not be subjected to conditions or substances outside the scope of those specified in the contract of supply.

The seals are designed to cover the widest range of operating parameters and require virtually no maintenance. It is recommended that the seal leakage is continuously monitored and recorded using the compressor's logging system. A trend of increasing leakage may give forewarning of a seal problem.

ATTENTION

SEAL LEAKAGE. The seal should not be operated if the leakage is greater than factory set shutdown trip levels.

Note:

CHECK MONTHLY for oil in the atmospheric vent lines between the outboard labyrinth barrier seal and the dry gas seal. Drain any oil in these lines, and rectify the cause.

Note:

FOR PERIODS OF SHUTDOWN or if the compressor is stored for a prolonged time the seal cartridges should be isolated by blanking off all connecting ports.

ATTENTION

The direction of rotation is determined by the grove profile, which is shown on the installation drawing.

Reverse rotation is permissible on bi-directional Dry Gas Seals.

With uni-directional Dry Gas Seals **reverse rotation must be avoided.** (Particular situations may be acceptable. Check with John Crane Technical Department)

ATTENTION

Under static conditions, to avoid condensation of the process gas and possible freezing at the seal faces, it may be necessary to reduce the applied sealing pressure. This will depend on the gas composition and the temperature at the seal. John Crane can advise on this if required.

ATTENTION

When decompressing the compressor, either statically or dynamically, take care not to exceed the maximum permissible decompression rates specified in table 2, where 'O' rings are in contact with the pressurised gas.

ATTENTION

REVERSE PRESSURE under static conditions can result in increased static leakage and may damage T28-AT seals. Under dynamic conditions, reverse pressure will result in seal failure and severe damage to any T28 seal.

Whilst **SMALL QUANTITIES OF BEARING OIL** and/or hydrocarbon condensate on the sealing surface are generally not detrimental to Gas Seal operation, the general ingress of these substances into the seal should be avoided to permit good seal performance and long life. If it is suspected that significant quantities of oil/hydrocarbon condensate or any other debris has contaminated the seal. Do not use the seal it should be returned to John Crane UK for inspection, cleaning and refurbishment.

ATTENTION

It is the practice of some compressor operators to "wash" or clean their machines internally whilst in service. John Crane cannot recommend that such practices are applied to dry gas seals because of the danger of loosened contamination lodging in key areas of the seal. Where compressor washing is practiced the seal should be buffered by a suitable clean gas supply fed inboard between the seal and the machine's labyrinth. Corrosive or reactive chemical must not be used. Only chemicals which have been specified in the contract of supply should be allowed to come into contact with the seals.

7 STORAGE AND TRANSPORTATION OF SEALS

7.1 Storage

a) John Crane Type 28 Gas Seals should always be stored as complete cartridge units with the setting rings or plates properly fitted as shown on the installation drawing.

b) Prior to dispatch every seal is packaged in a purpose built case with a foam lining. If the seal is delivered as a complete cartridge then the crate is suitable for subsequent shipment. Seals and/or spare elastomers should be stored in their original case indoors in a clean dry environment at a temperature between 15 and 25°C.

c) If the seals are to be stored inside the compressor for long periods (more than 8 weeks), it is necessary to ensure the seal faces are not filmed with oil. If this is suspected (i.e. oil is found in the line between the seals and the bearing labyrinth or barrier seal). Attempts should be made to turn the compressor shaft manually. If the shaft fails to rotate freely the seals should be removed for inspection and cleaning by a John Crane EAA trained Technician.

d) If the compressor is shipped with the seals in situ the shaft should be restrained to prevent movement and potential seal damage. All compressor connections should be sealed off, after ensuring that the atmosphere within the machine is dry. Preserving oil should not be allowed to contact the seal.

e) Gas seal cartridges need routine refurbishment at regular intervals. After the agreed period of operation, they should be returned to John Crane for 'O' ring replacement and general inspection.

ATTENTION

Maximum storage and installed service time combined is 10 years.
Maximum installed service time is 5 years.
At the end of these periods the cartridge should be returned to John Crane for inspection and refurbishment.
Note that the length of the successful installed service period will vary depending on the application. With intermittent and wet or dirty processes the serviceable life is significantly reduced. Seal condition should be assessed in operation by monitoring leakage levels and trends.

7.2 Shipping of the Seal Cartridge

a) It is essential that the seals are suitably packaged and always transported in the purpose built case supplied by John Crane.

b) The seals should be assembled as a cartridge unit with the setting plates fitted (preventing movement between the seal rotor and stator).

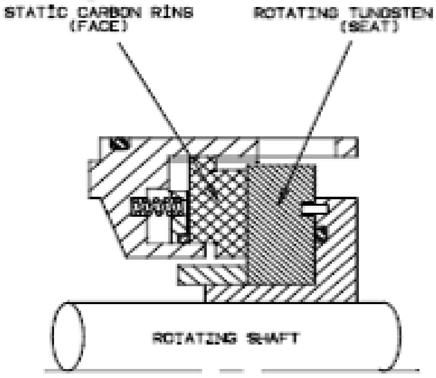
e) Any "loose" items placed inside the transportation case must be securely wrapped to prevent transit damage.

f) For other shipping procedures see Appendix II

APPENDIX I. PRINCIPLE OF OPERATION

Simply explained, the seal typically comprises of an 'O' ring sealed carbon FACE, located in a stainless steel retainer, spring loaded against a rotating carbide SEAT, fixed to the shaft, as seen in *figure I* below.

Figure I



Sealing of the fluid is achieved at the radial interface of the rotating and stationary rings by a unique and ingenious method. The sealing surfaces are lapped to a high degree of flatness, but the rotating carbide ring has a series of logarithmic spiral pattern grooves on its running face.

With rotation, gas is drawn inwards towards the root of the groove, called the sealing dam. The sealing dam provides resistance to flow, increasing the pressure. The generated pressure lifts the carbon ring surface out of contact with the tungsten carbide ring by a precise amount, typically 3 microns. The gap between the radial faces is set when the closing forces, hydrostatic pressure and spring load, equate to the opening forces generated within the fluid film.

If a disturbance occurs which results in a reduced sealing gap, the pressure generated by the spiral grooves considerably increases, as illustrated in *figure II.*

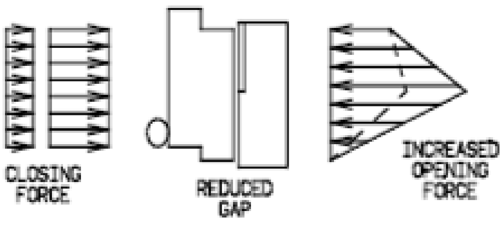
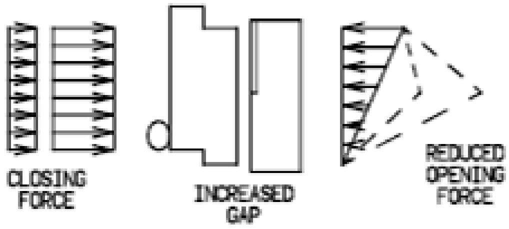


Figure II

Similarly, if an upset causes the gap to increase there is a reduction in the pressure generated and the seal regains its equilibrium very quickly as shown in *figure III*

Figure III



The result of this mechanism is a highly stable yet very thin fluid interface between the static face and the rotating seat. This results in the two surfaces being kept apart and not touching under normal dynamic operating conditions. In turn this leads to a long life, reliable seal with no wear at the interface.

To achieve this John Crane EAA have invested heavily in advanced technology and have built up a considerable and unsurpassed fund of knowledge and experience in rotary shaft gas sealing applications.

There are many principles governing the seal's performance only a few of which are explained in the previous paragraphs.

Further information on this topic is available from John Crane EAA Ltd.

APPENDIX II. RETURNING SEALS TO JOHN CRANE

In the event that seals have to be returned to John Crane EAA.
The following information must be made available on the Shipping Documentation.

1) Installation Drawing Numbers.
2) Cartridge Numbers.
3) Value (for insurance only).
4) Commodity Code 84842000000.

a) All seals returned from outside the European Community, must be assigned to our agents:-

COMPASS INTERNATIONAL
UNIT 1 FALCON WAY
OFF CENTRAL WAY
FELTHAM
TW14 OUQ

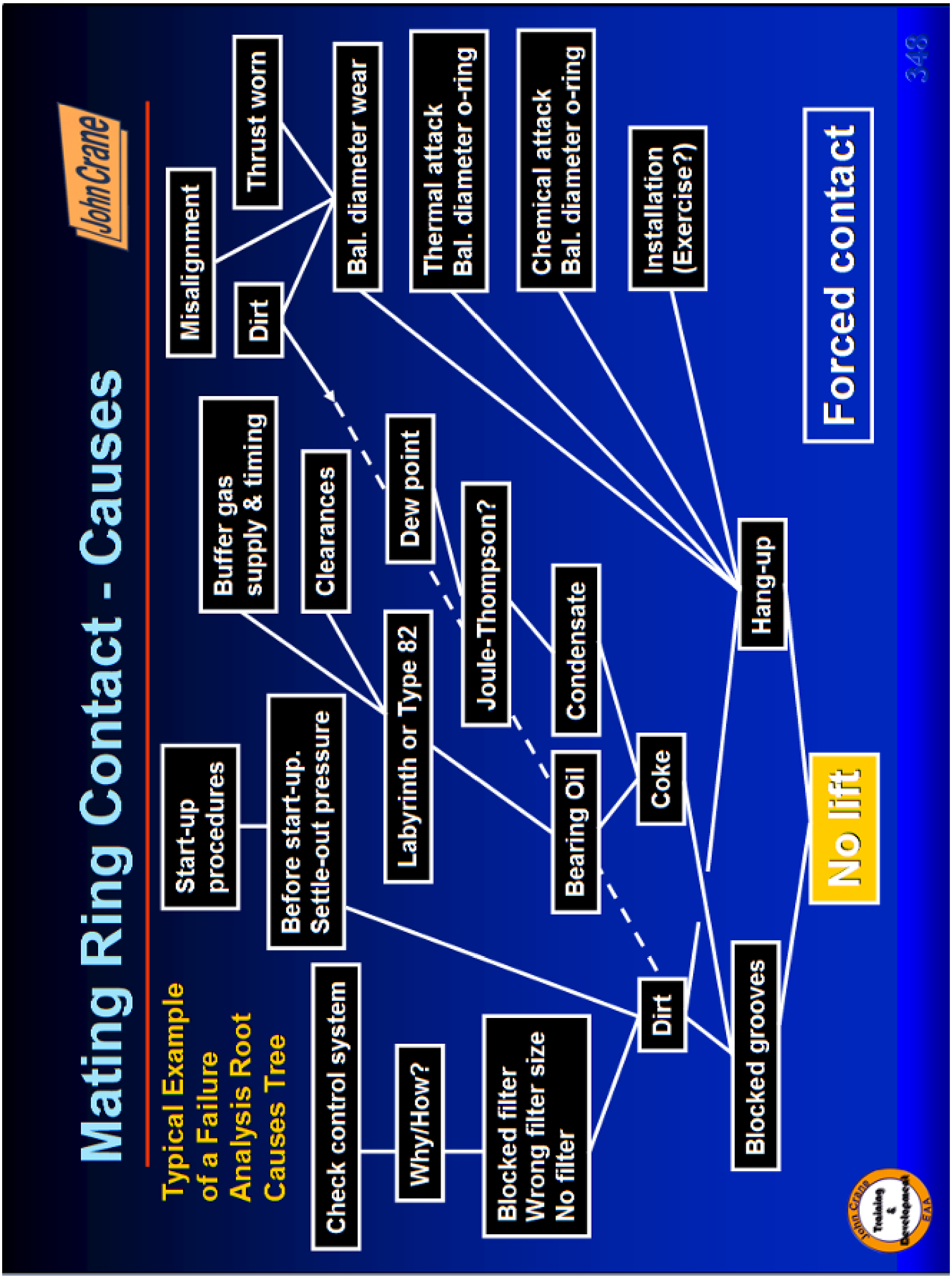
Telephone Number: +44 (0)208 8441991.

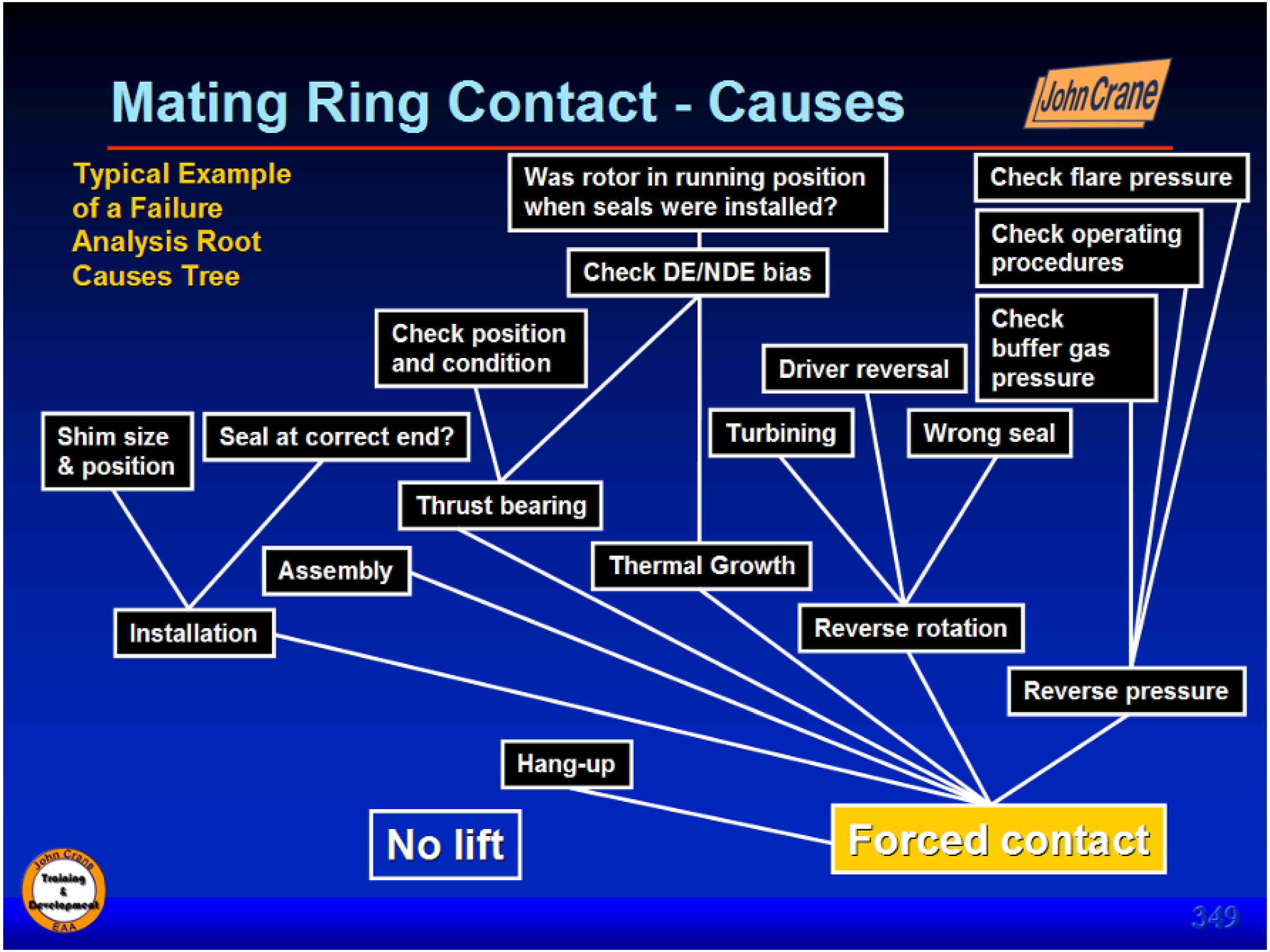
b) And advise John Crane Slough of the shipment details.

c) If the seals are returned using a courier company (such as Fedex, DHL, TNT etc) they should be consigned to John Crane Slough.

Appendix B

Failure Analysis Tree—Examples


Mating Ring Contact - Causes
Typical Example of a Failure Analysis Root Causes Tree
John Crane
Check control system
Why/How?
Blocked filter
Wrong filter size
No filter
Start-up procedures
Before start-up. Settle-out pressure
Misalignment
Thrust worn
Dirt
Buffer gas supply & timing
Clearances
Labyrinth or Type 82
Dew point
Joule-Thompson?
Condensate
Bearing Oil
Coke
Dirt
Blocked grooves
Bal. diameter wear
Thermal attack Bal. diameter o-ring
Chemical attack Bal. diameter o-ring
Installation (Exercise?)
Hang-up
No lift
Forced contact
348


Mating Ring Contact - Causes
John Crane
Typical Example of a Failure Analysis Root Causes Tree
Was rotor in running position when seals were installed?
Check DE/NDE bias
Check position and condition
Shim size & position
Seal at correct end?
Thrust bearing
Assembly
Installation
Thermal Growth
Turbining
Driver reversal
Wrong seal
Reverse rotation
Check flare pressure
Check operating procedures
Check buffer gas pressure
Reverse pressure
Hang-up
No lift
Forced contact
John Crane Training & Development EAA
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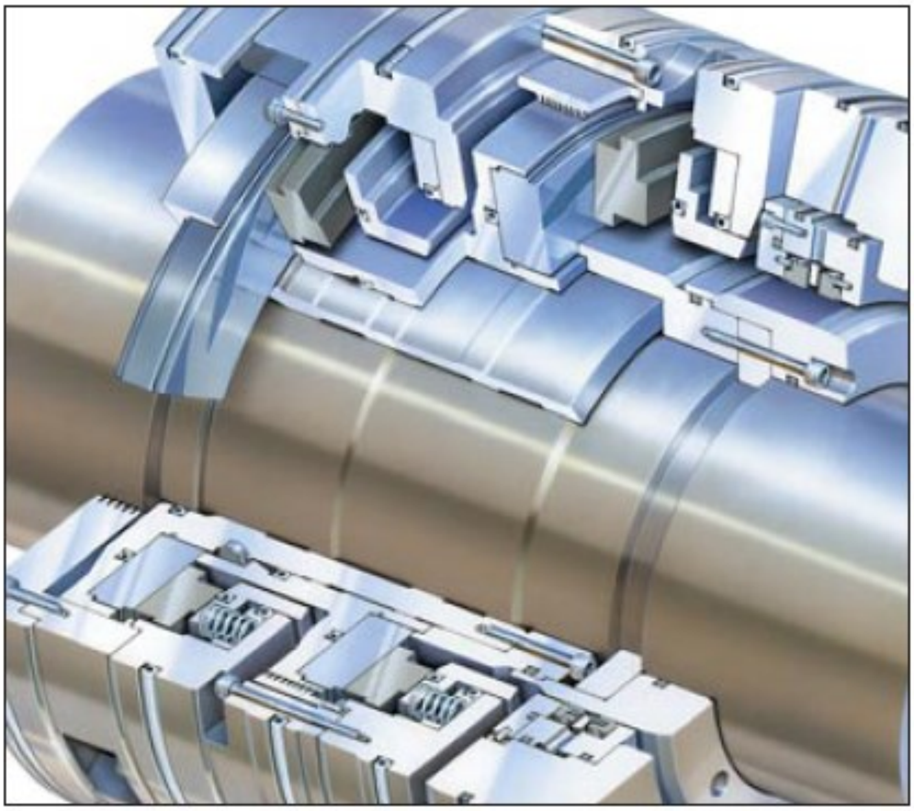
Appendix C

Dry Gas Seal Data Sheets

TYPE 28 Compressor Seals

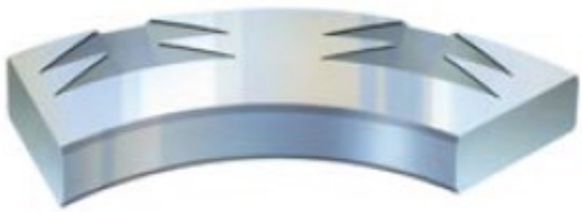
Dry-Running, Non-Contacting Gas Seals

28AT/28XP/28EXP



Standard Unidirectional Groove Design

Optional Bidirectional Groove Design



Product Description

Type 28 compressor dry-running gas seals have been the industry standard since early 1970 for gas-handling turbomachinery. Utilizing John Crane's patented spiral groove pattern, these seals are non-contacting in operation.

- During dynamic operation, the mating ring/seat and primary ring/face maintain a sealing gap of approximately 0.0002 in./5 microns, thereby eliminating wear.
- These seals eliminate seal oil contamination and reduce maintenance costs and downtime.
- Single, double opposed, and tandem cartridge seals are capable of handling a wide variety of gas sealing applications in the gas collection/transmission, refining, chemical and petrochemical processing industries.

Design Features

- Shrouded mating ring prevents secondary damage in the event of a mating ring fracture.
- Low-level leakage can be vented to a safe area, used as fuel to drive equipment, or returned to process via a low-pressure ejector.

Performance Capabilities

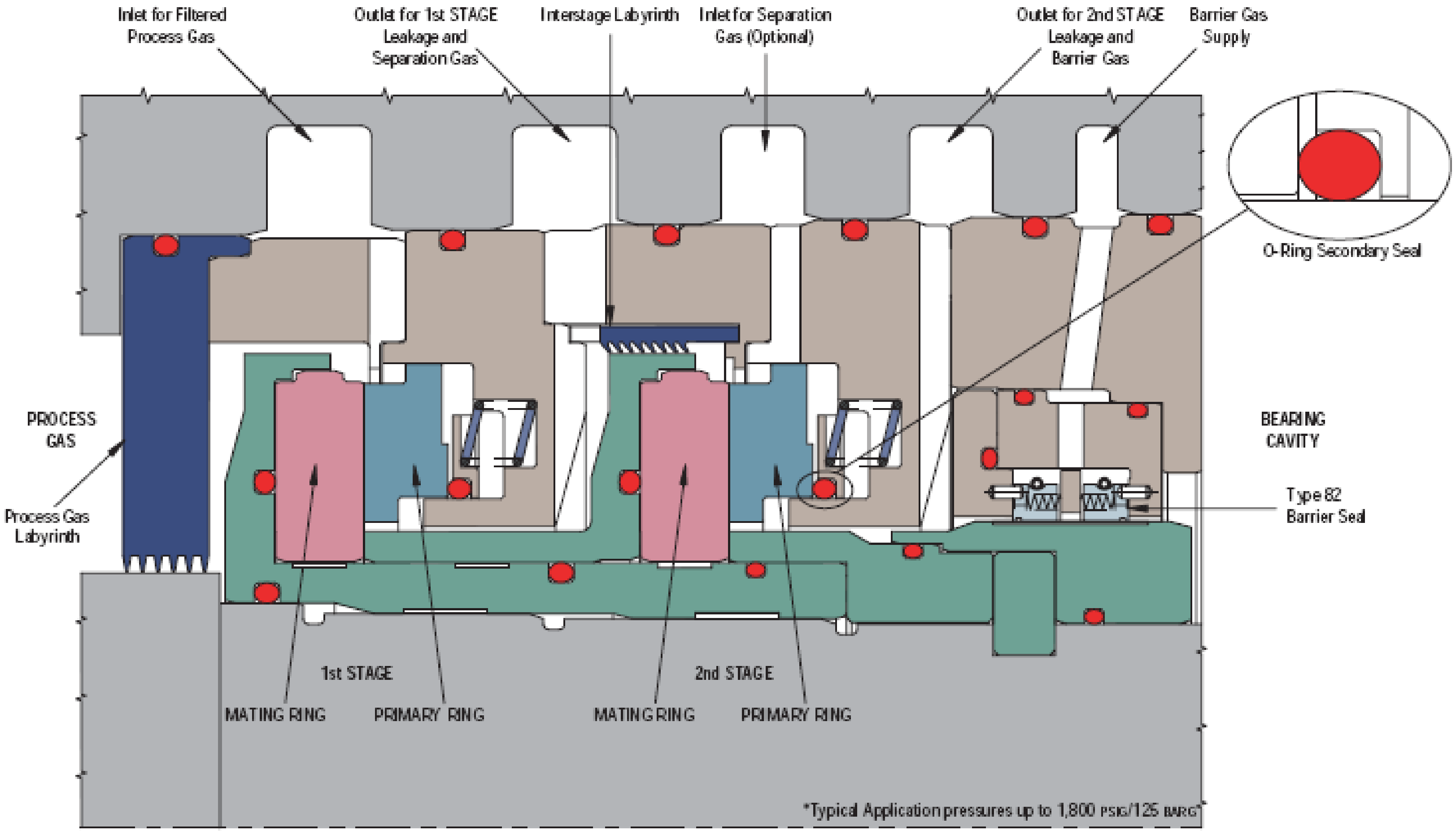
- Temperature: -140°C to 315°C/-220°F to 600°F
- Pressure: Up to 450 BARG/6,500 PSIG across single stage
- Speed: Up to 200 m/s/ 660 fps
- Shaft: Up to 330 mm/13 in.

*Contact DryGasSeals@johncrane.com for more information about exact application requirements.

TYPE 28AT Compressor Seals

Dry-Running, Non-Contacting Gas Seals

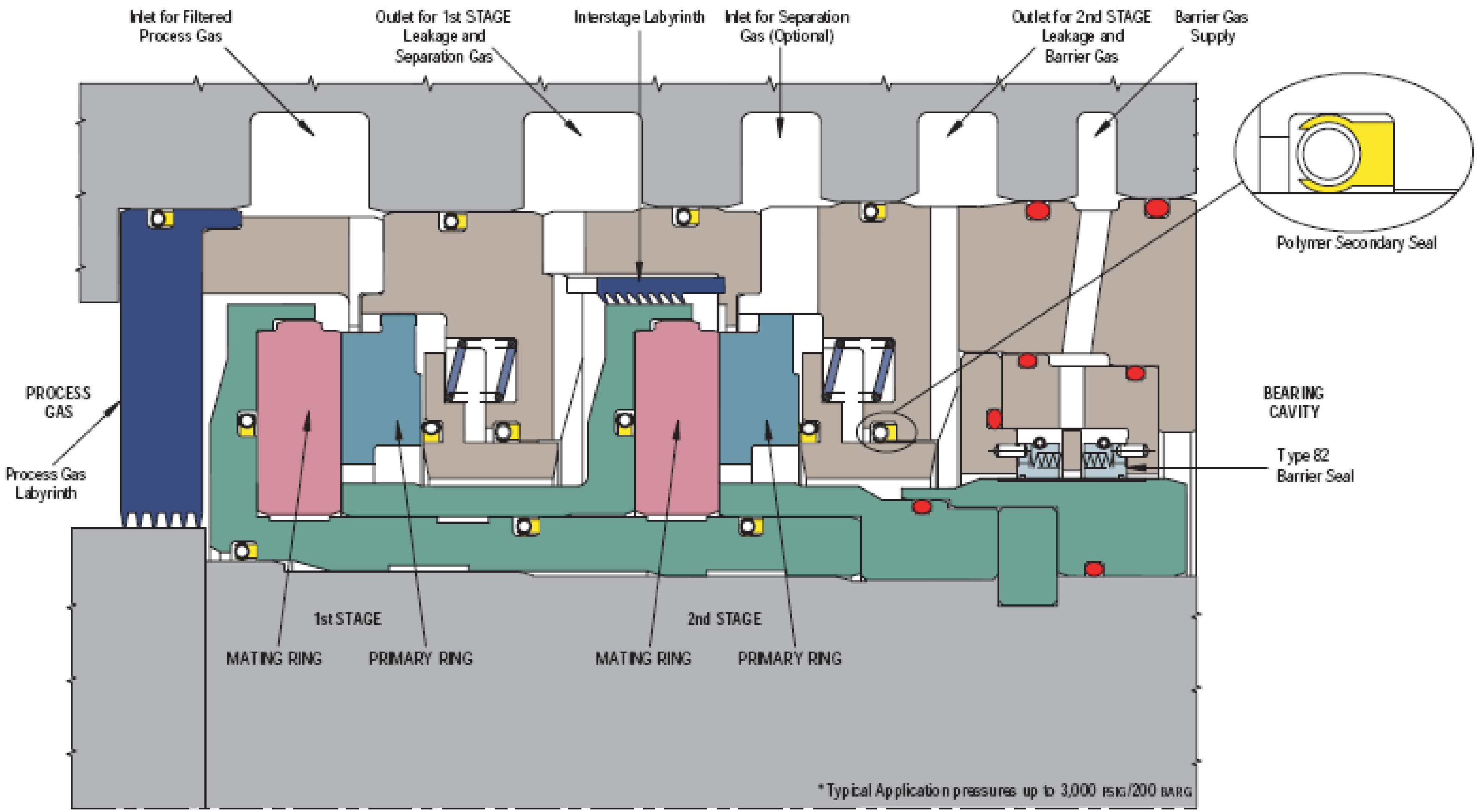
Type 28AT Tandem Arrangement



TYPE 28XP Compressor Seals

Dry-Running, Non-Contacting Gas Seals

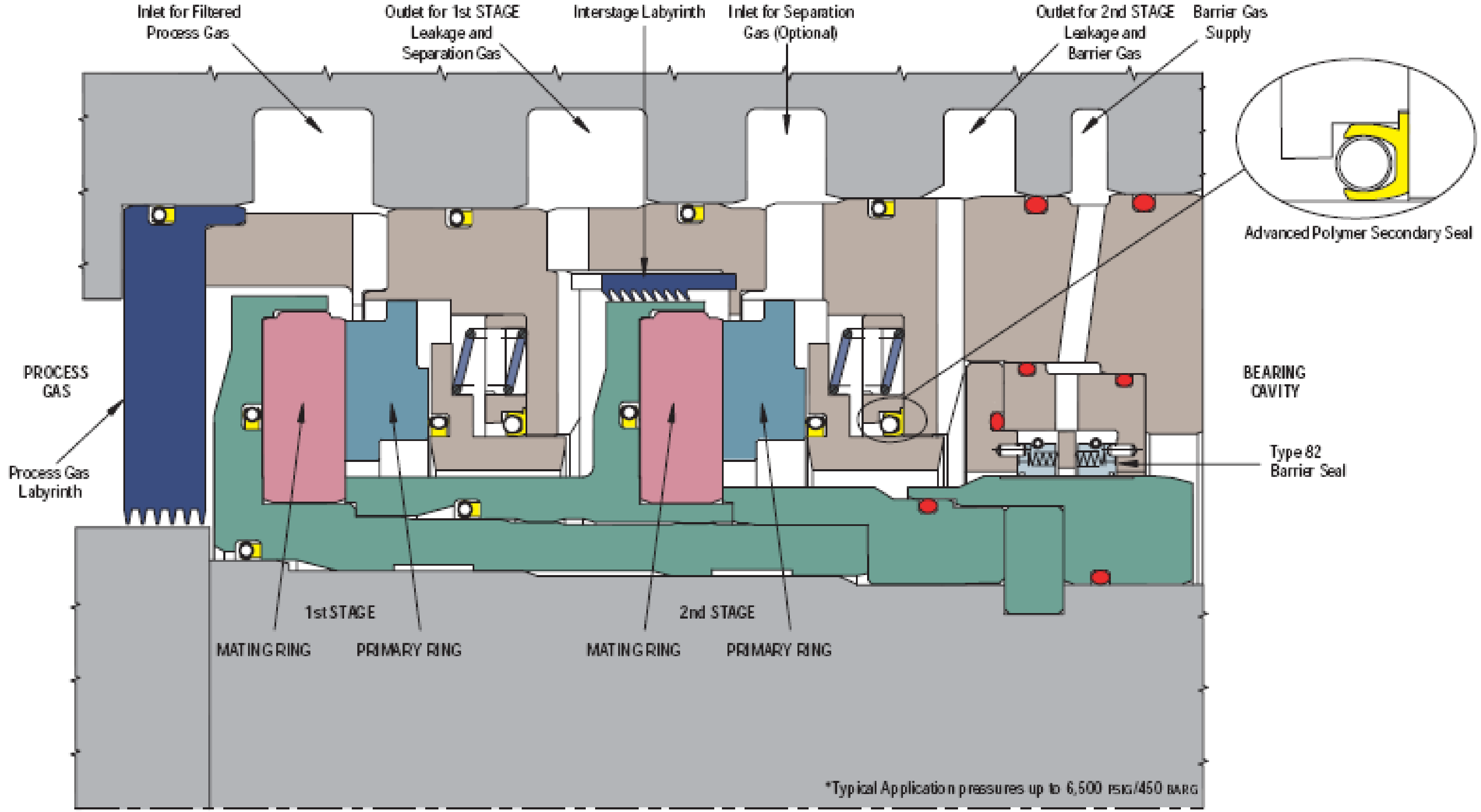
Type 28XP Tandem Arrangement



TYPE 28EXP Compressor Seals

Dry-Running, Non-Contacting Gas Seals

Type 28EXP Tandem Arrangement



AURA™ Compressor Seals

Dry-Running, Non-Contacting Gas Seals

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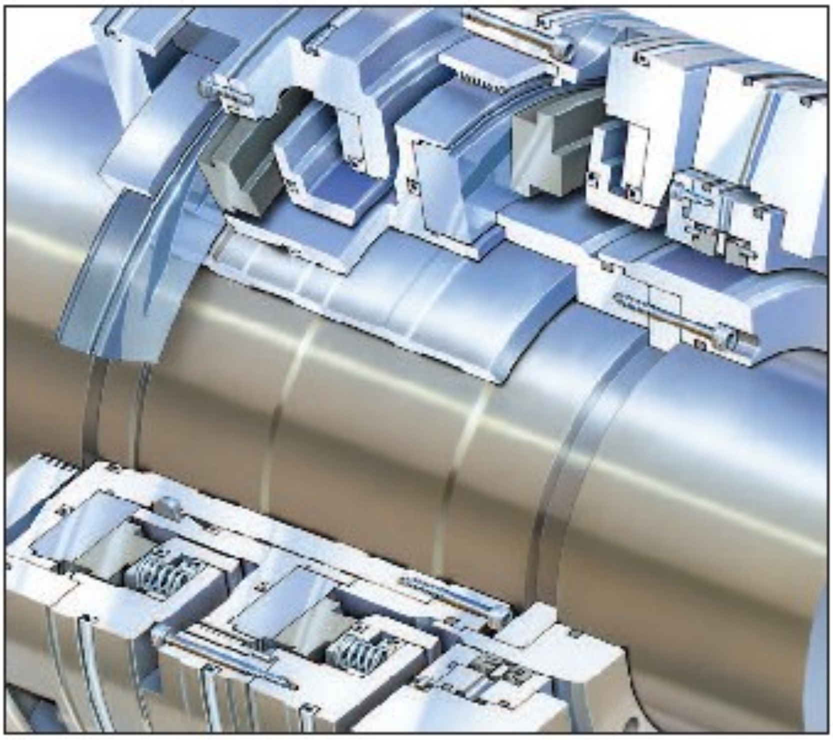
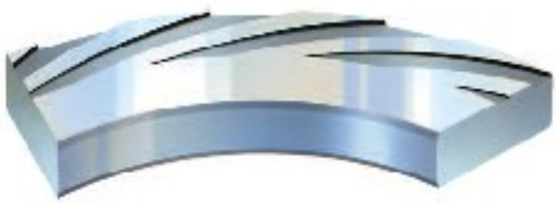
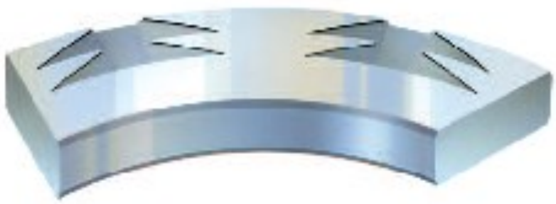




Unidirectional Groove Design



Bidirectional Groove Design

Product Description

AURA™ 220 is the foundation of the next generation of John Crane gas seals that reduces seal operating and transaction costs, extends maintenance intervals and reduces spares inventory. Utilizing John Crane's patented spiral groove patterns, these seals are non-contacting in operation.

- Simplified and common global design
- Increased reliability
- Improved performance envelope

Design Features

- Balance diameter sealing - patented design includes component to seal retainer housing and carrier
- Polymers behind face and seat - for reduced emissions and wider performance envelope

Performance Capabilities

- Temperature: -50°C (-58°F)/220°C (356°F)
- Static Pressure: Up to 220 bar (3190 psi)
- Speed: Up to 459 fts/140 m/s
- Seal Size: 100-276mm (3.937"-10.875")

*Contact **DryGasSeals@johncrane.com** for more information about exact application requirements.

Published by: John Crane Global Training and Development Group

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