

API 682 mechanical seals are versatile and provide emissions sealing

AW Chesterton Co is introducing a new line of API 682 mechanical seals and systems for centrifugal pumps used in the oil and gas industry.

According to the firm, these seals aim to set a new benchmark for versatility, standardisation and performance.

Many of the critical pump-sealing applications in the oil and gas industry – for example, in refineries and on off-shore platforms – require seals that meet high-performance standards. With fluid sealing requirements ranging from light hydrocarbons to crude oil, sealing versatility and standardisation are also critical. More end-users are looking for alternatives to single-use, engineered sealing systems to promote standardisation and streamline logistical supply.

The company says that it has assembled proven and innovative features into one seal assembly to meet API 682 requirements. Designed to deliver high reliability and emission control performance, the new API seal line combines proven Chesterton technology with extensive research and development qualification testing.

The technology offers a versatile approach to refinery sealing. The A182 single seal is designed to meet tough emissions requirements

through an advanced seal face design and low heat generation. Its stationary spring arrangement offers superior performance compared with conventional API 682 seals, claims the firm. Critical seal ring micro-polishing enhances long-term sealing while standard multi-port injection enhances seal cooling.

The A2382 is a high-pressure tandem seal that is capable of operating in both Arrangement 2 and Arrangement 3 pressure modes, unlike older conventional units which require two different seal designs.

This proven technology increases reliability and standardisation, and it reduces risk in plant operation. It also simplifies the seal selection process throughout the plant and platform. Its low heat generation combined with enhanced thermal management results in cooler operating conditions in tough high-pressure applications.

The A182 Single and A2382 Dual can be coupled with environmental control packages to create superior sealing systems, says the company. Numerous pre-engineered packages are available to promote ease-of-use and standardisation.

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AW Chesterton Co's new line of API 682 mechanical seals aim to set a new benchmark for versatility, standardisation and performance.

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Chesterton A2382 dual seal raises the performance bar while lowering costs

AW Chesterton Co has developed a new API 682 (ISO 21049) dual seal one-seal design that is versatile enough to be used for multiple arrangements because of its compliant, dual balance mechanism.

Dual seals are seal configurations that use two seal ring sets per assembly. These seals are designed for use within two different types of arrangements.

In common industry practice, each arrangement requires a different dual seal design. However, the innovative dual seal design of the Chesterton A2382 is versatile enough to offer seal standardisation within both arrangements.

Geometric double balance technology enables one seal to perform optimally in both pressure modes, eliminating the necessity for multiple wet-lubricated dual seals. As a result, plant inventory within this area can be reduced by almost half.

Versatile design reduces costs and maximises safety

Beyond cutting sealing costs, the A2382 also simplifies installation procedures and maximises safety.

Eliminating the need for multiple seals allows for more focused installation training. Consequently, the margin of error during installations is reduced. Testing reveals that the design actually increases seal reliability during pressure transients, further improving overall safety.

In addition to the aforementioned benefits, the A2382 surpasses the API 682 (ISO 21049) specification – the standards used to help plants minimise hazardous emissions.

The API 682 and ISO 21049 standards were created to standardise approaches to sealing rotary pumps with mechanical seals.

Seal applications on pumps and mixers within refineries are complex and hazardous. Applications vary widely with pressure, temperature and fluid types.

Pressure transients can cause mechanical seal upsets if the seal designs are not optimised for pressure reversals. In an effort to address these concerns and meet standards, various seal balance methods are implemented in mechanical seal designs to accommodate pressure variations and enhance dual-seal reliability.

This mechanical seal specification is used primarily in the petroleum, natural gas and chemical industries. Three primary mechanical seal arrangements are defined in the standard: Arrangements 1, 2 and 3 (**Table 1**).

Arrangement 1, which identifies a single mechanical seal, provides a fluid-tight seal between a rotating shaft and stationary pump housing. Single mechanical seals consist of a rotary and a mating stationary seal ring.

Arrangement 2 dual seals pressurise the space between the inboard and outboard seals at a pressure that is lower than seal chamber pressure – this is achieved through the use of a buffer fluid. This arrangement is typically used to prevent the slight leakage that would result if an Arrangement 1 single seal was used in the application.

Arrangement 3 dual seals pressurise the space between the inboard and outboard seals at a pressure that is higher than the seal chamber pressure. These seals use an externally supplied barrier fluid to lubricate seal faces. A pressurised gas – usually nitrogen – creates the external fluid pressure on the barrier fluid. The inboard seal is lubricated by the barrier fluid.

Seals must withstand mechanical and hydraulic forces

When an assembly is operating, numerous mechanical, hydraulic and temperature effects are in motion around the seal. Torque, vibration, seal-face frictional heat and process temperatures are among the many variables that must be taken into consideration.

Proper preventative measures must be taken in order to minimise mechanical distortions and the associated effects that result from these forces. Seal designers address these concerns while still ensuring precise contact between seal rings and minimising leakage.

Because high sealing pressures are common, hydraulic forces acting on a mechanical seal are of particular interest within the petroleum industry. Circumferential forces around the seal can cause deformation of seal parts, which is a big concern as the impact on face flatness is critical to seal reliability. The API 682 standard specifies reliable sealing performance at pressures up to 42 bar (615 psi).

The existence of axial loads acting on the seal is another hydraulic force that is important to evaluate. In the case of a single seal, it is essential to assess the hydraulic forces acting to close the seal as well as the forces acting to open it. The axial pressures acting on the surfaces can be resolved to determine the net axial force acting to close the seal.

It is possible to estimate the opening force, or the function of the pressure drop across the seal faces. Advanced iterative FEA analyses assist mechanical seal engineers in optimising seal face geometry and seal balance for enhanced performance.

By reducing the area exposed to hydraulic forces, closing and opening forces can be bal-

Arrangement	Seals	Pressure between the seals	Fluid between the seal faces
1	Single	NA	NA
2	Dual	Lower than seal chamber	Buffer
3	Dual	Higher than seal chamber	Barrier

Table 1. Three primary mechanical seal arrangements are defined in API 682 (ISO 21049): Arrangements 1, 2 and 3.

anced for the application. **Figure 1** identifies a single seal design that is unbalanced while the design shown in **Figure 2** reduces the hydraulic closing axial force by attaching a shoulder to the rotating shaft.

The balance ratio for single seals is defined by the following formula:

$$B = \frac{(D_o^2 - D_i^2)}{(D_b^2 - D_i^2)}$$

where D_o is the seal face outside diameter; D_i is the seal face inside diameter; and D_b is the seal face balance diameter.

The balance diameter of pusher seals equipped with O-rings is generally defined by the sliding surface interface of the O-ring and the sealing surface.

The API 682 and ISO 20149 standards ensure that seal performance is maintained by requiring extensive testing on multiple fluids. The tests consist of dynamic, static and cyclic phases to simulate normal running, upset, standby and start-up/shutdown pump operations. Seals that are not balance-optimised can have lower performance levels associated with starting torque and seal frictional heat, especially in Arrangement 3.

Dual seal balance is crucial to performance

Because of the additional fluid pressure acting between the inboard and outboard sets of seal faces, dual seals have a more complex hydraulic pressure arrangement than single seals.

The inboard set of seal rings is subject to pressure from the process as well as an external barrier/buffer fluid. Reliable leak-tight performance similar to single seals is required, but pressure fluctuations associated with changes in process pressure and barrier/buffer pressure also must be accommodated.

If dual seals are not hydraulically balanced to accommodate upset conditions in addition to normal operation, the risk of a breach and/or sealing contact loss is heightened. When primary seal rings are no longer in contact and are forced open, leakage will occur.

Early dual seal designs were capable of performing under only one pressure condition. These seals were identified as single-balanced dual seals. Under process upsets, during which the seal chamber pressure changes or when auxiliary buffer/barrier systems cannot maintain set pressures, the inboard seal can be hydraulically compromised and lose sealing contact, resulting in failure. It is evident that

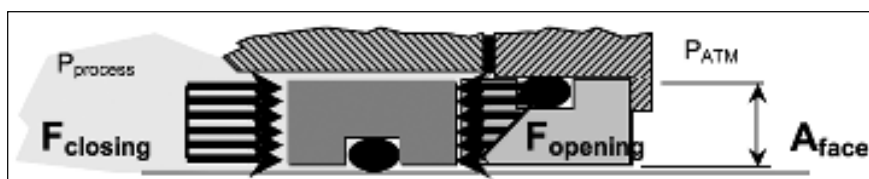


Figure 1. Single unbalanced seal arrangement.

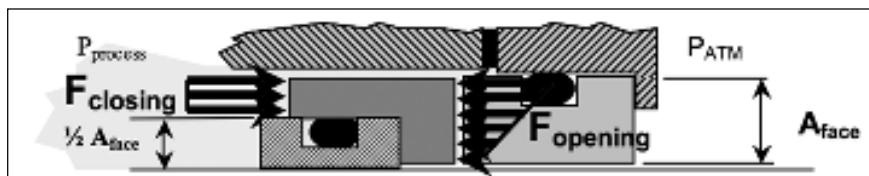


Figure 2. Single balanced seal arrangement.

a dual seal capable of accommodating pressure reversals as well as sealing hazardous and high-pressure fluids is a necessity.

The A2382 is able to handle a wide range of pressure conditions by operating reliably in pressure Arrangements 2 and 3. Moreover, it has the unique ability to operate under optimised balance conditions in both pressure arrangements, which eliminates the need for multiple seals and reduces the potential for failing emissions levels.

Available dual balance options

Dual seal designs that ensure optimum operation in only one pressure mode are commonly used in industry.

The seals in these arrangements do not open in reverse pressure, which reduces the need to focus on the second pressure mode. The design enables the arrangement to operate at a balance ratio comparable to that of a single seal. In these cases, API 682 qualification testing can be completed with an optimised balance ratio for one arrangement. The advantage of this design is that the balance ratio and face geometry are independent of one other and give the seal designer the freedom to optimise performance for the arrangement selected.

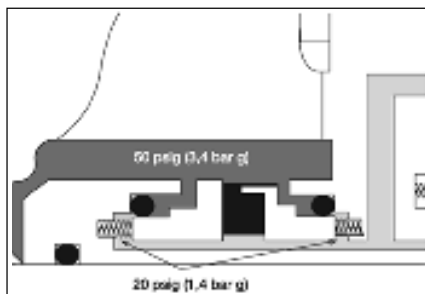


Figure 3. Arrangement 2 O-ring shift.

While the balance is less optimal in the second arrangement, it will prevent pressure reversals from opening the inboard seal.

The clear downside to this dual balance method is that the optimal balance and seal geometry have not been established within the second pressure arrangement. As a result, the seal is unlikely to achieve sufficient results in performance testing when operated in reverse pressure mode. Separate seal designs are required to maintain adequate operation within both pressure modes.

O-ring designs

Using a shifting O-ring is one way to eliminate the necessity for multiple seals in Arrangements 2 and 3. This creates a balance shift within the seal, allowing for exceptional seal balance in both arrangements.

Currently, the O-ring design is used within numerous seals in industry and yields excellent performance results. When positioned in a wide groove, it can shift with the pressures in Arrangements 2 and 3 (**Figure 3 & Figure 4**).

The O-ring diameter is consistent with the shift in balance and the seal does not open during pressure reversals. In fact, the seal balance can be engineered to produce optimised hydraulic balance and reliable operation in both arrangements.

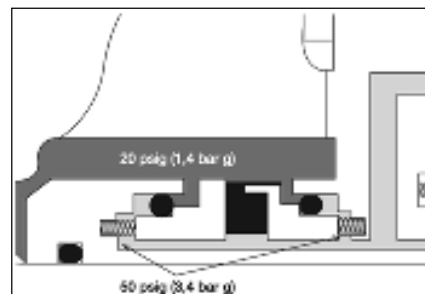


Figure 4. Arrangement 3 O-ring shift.

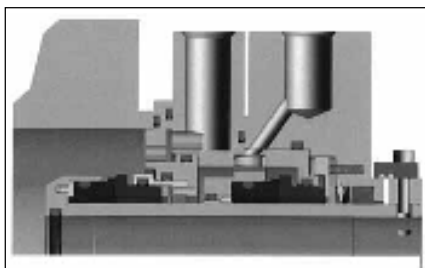


Figure 5. API 682 arrangements 2 and 3 dual seal.

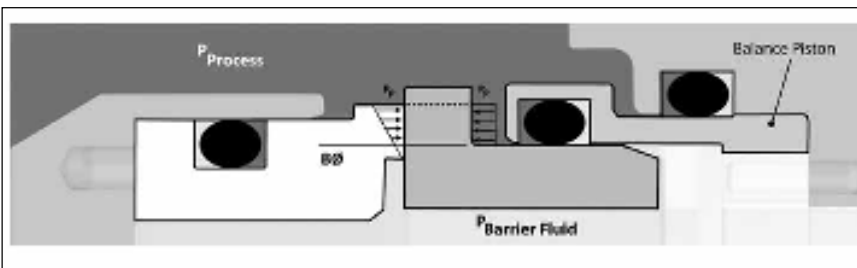


Figure 6. Arrangement 2 balance diameter.

The main drawback to the O-ring dual balance option is its reliance on precise geometric design and placement.

The seal face must be wide enough to support the required balance areas above and below the O-ring as well as inside and outside of its diameters. Therefore, the face width must be engineered to accommodate the O-ring cross-section.

Since the face width is dependent on a hydraulic balance arrangement, there is little room for design flexibility. These factors often result in wide seal face designs that cause higher frictional heat generation. This balance option is appropriate for lighter duty applications. It is not suitable for light lubricating and vaporising fluids at high speed and pressure, and is unlikely to meet API 682 and ISO 20149 seal regulations under such conditions.

Currently, metal bellows dual seals do not have the ability to optimise seal balance for Arrangements 2 and 3 with one design. In order to preserve sealing integrity in reverse pressure conditions, pressure reversals typically require the seal design to maintain lower than optimal balance ratios.

The A2382 seal, which uses a new dual balance mechanism, has been developed to

comply with API 682 (ISO 21049) standards and performance testing that requires only one seal design for both Arrangements 2 and 3 (Figure 5). The new double balance mechanism minimises the performance tradeoffs and maximises standardisation.

The seal balance is optimised for both Arrangements 2 and 3. Only one seal is required to perform optimally in both pressure modes using geometric double balance principles. The seal uses a pressure balance piston to set balance for both arrangements. Under operation, this piston applies optimised balance in both pressure modes. The piston responds by defining one of two hydraulic piston areas.

Arrangement 2 functionality

During operation with a buffer fluid, the fluid pressure in Arrangement 2 is lower than the process pressure.

Typically, there is a relatively high pressure differential across the sealing interface in this arrangement, which makes the optimisation of seal rings essential.

The piston area and associated balance diameter is defined by the inside diameter of the O-ring located on the inside diameter of the piston (Figure 6).

This particular sealing arrangement and its associated balance diameter are identical to that of a single seal. Consequentially, the seal rings designed and tested in accordance with API 682 single seal protocols can be used (Figure 7).

Pressure transients can be easily controlled when the balance system based on ID or OD pressurisation takes over.

The seal design easily accommodates pressure reversals of up to 0.275 MPa (2.75 bar) (40 psi), which is safely within the range specified by the API 682 standard Arrangement 2 for maximised safety.

Unlike the limitations associated with other dual balance options, the piston can optimise balance easily. For example, shifting O-rings have fixed diameters and do not allow seal rings to be optimised for both Arrangements 2 and 3. Geometric dual balance enables the thickness of the piston wall to be varied so both balance diameters can be optimised.

Arrangement 3 functionality

During operation with a barrier fluid, the fluid pressure in Arrangement 3 is greater than the process fluid pressure.

In this arrangement, the O-ring sealing on the outside diameter of the piston defines the second piston area. And, just as in the scenario illustrated earlier, the balance system takes over based on ID or OD pressurisation (Figure 8).

When equipment is operating dry with high barrier fluid pressures, large pressure differentials can occur across the inboard seal faces. However, these conditions are easily managed in this arrangement.

Similarly, reverse pressure excursions away from the design point (where process fluid pressure exceeds barrier fluid pressure) are also controlled by the balance system.

If a large loss in barrier fluid pressure occurs, the inboard seal rings operate in the optimised

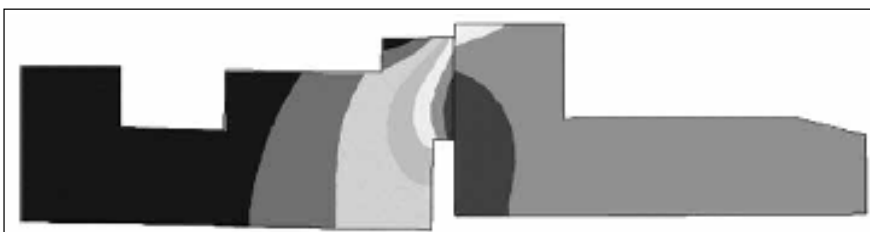


Figure 7. Optimised seal rings.

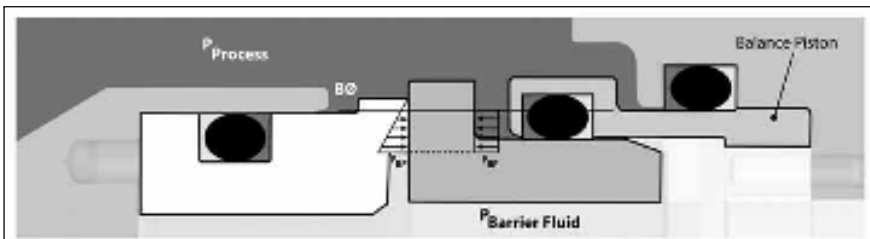


Figure 8. Arrangement 3 balance diameter.

Arrangement 2 mode. These particular seal rings have been optimised for single-seal performance on light hydrocarbons. In all cases, Arrangement 3 – and even sustained operation in Arrangement 2 – are extremely manageable.

In these situations, the process pressure and barrier fluids generate a closing force on the seal faces, keeping them in contact with one other. Operation with a complete loss of barrier fluid pressure has little effect on the sealing integrity.

The graph shown in **Figure 9** illustrates the same mechanical seal cycles from Arrangement 2 to Arrangement 3 without impact on emissions and sealing integrity

A2382 sets higher expectations for API dual seals

The facts and comparisons spell out a clear conclusion: the cutting-edge Chesterton A2382 dual seal is leading the pack in one-seal design.

It has the proven ability to offer superior functionality in both API 682 Arrangements 2 and 3. The innovative design offers consumers an adaptable and functional seal choice that will both perform and last. Its versatility allows for standardisation which previously has not been an option in the industry.

Taking advantage of the A2382 dual seal makes it possible to reduce plant inventory

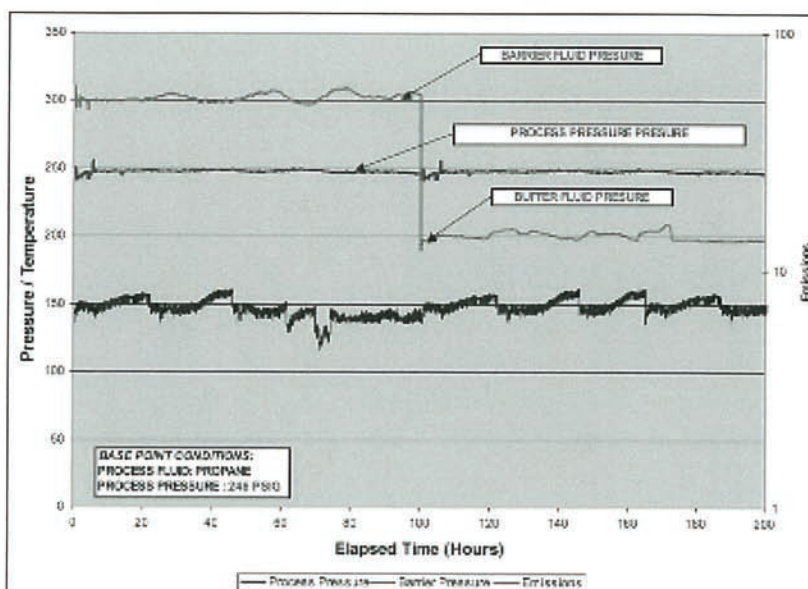


Figure 9. Operation of Arrangements 2 and 3.

devoted to wet-lubricated dual seals by almost half. Moreover, its use improves safety and reliability while simplifying application. The seal accomplishes all of this while achieving impressive passing results within the API 682 (ISO 21049) specification.

The A2382 dual seal raises the bar for API dual seals as it advances the standard's fundamental goals – which is to assist plants in improving reliability and standardisation

when sealing hazardous fluids, while reducing both emissions to the atmosphere and life-cycle sealing costs.

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