



ENVIRONMENTAL TECHNOLOGY

API 682

Contradicting requirements in API 682 dual seal design configurations

Mechanical seal designers face opposing challenges when developing dual seal's. The requirement to provide optimised seal face cooling conflicts with the need to provide a seal that is tolerant of pressure reversal and abrasive or congealing fluids. Poor seal face cooling can potentially reduce reliability. The American Petroleum Institute mechanical seal standard (API682 ISO 21049) offers users a choice of three different configurations. Advantages and disadvantages are discussed of each type. Elegant designs are now available that can satisfy these opposing requirements. The use of CFD and testing has optimised these solutions. Field studies provide verification for these design concepts.

1 Dual Seal Arrangements

Dual seals are increasing in popularity in all aspects of hydrocarbon process industry. Plant hazard safety requirement, reduced fugative emissions and reliability are the main drivers. API 682 3rd edition (1) describes pressurised as arrangement 3, having two seals per cartridge assembly that utilise an externally supplied pressurised barrier fluid. The different configurations are described by API as;

Face-to-back, dual seal in which one mating ring is mounted between the two flexible elements and one flexible element is mounted between the two mating rings or seats (Figure 1a)

Back-to-back, dual seal in which both of the flexible elements are mounted between the mating rings. (Figure 1b)

Face-to-face, dual seal in which both of the mating seal rings are mounted between the flexible elements (Figure 1c)

The description is, in the opinion of the author, incomplete. The principle difference of the face to back configuration is that the pumpage or process fluid is on the outer side of the seal face. The process liquid on the other two configurations is on the inside diameter.

Face to back arrangement is preferred, (2) with back to back and face to face offered as purchasers options. Un-pressurised dual seals API 682 refers to as arrangement 2, face to back is the only option available in the standard. (3)

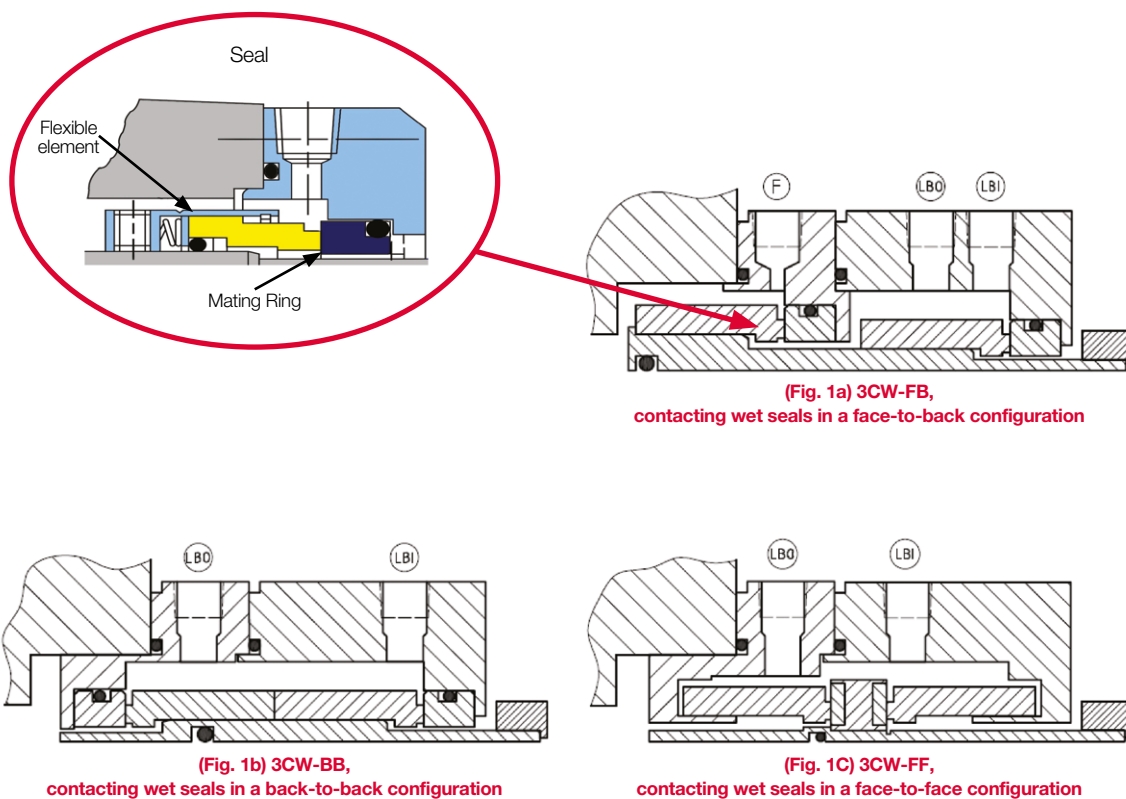


Figure 1

2.1 Advantages of back to back and face to face configurations

In the hydrocarbon process industries these configurations have the higher population (4). They can potentially offer higher levels of performance primarily due to the barrier fluid cooling flowing over both inner and outer seals.

2.2 Disadvantages back to back and face to face configurations

The principal shortcoming is that the process fluid is on the inside diameter of the seal faces. Any entrained abrasive solids will be centrifuged into the seal faces with the potential for damage. The 'dead zone' formed by the small volume of process fluid underneath the inner seal creates susceptibility to fluids congealing or solids accumulation. The secondary O ring will then be striving to advance (Figure 2a) across this fouled area of the sleeve, so called 'hang up'.

The application of reverse balance can be more difficult with back to back designs. Upset conditions such as loss in barrier pressure or increases in process pressure can compromise such configurations. Positive retention of inner seal mating ring can be difficult to realise due to dimensional restriction of associated hardware and seal chamber. Thrust forces in a pressure reversal may cause an unretained ring to be dislodged from its mount.

With some designs reverse pressure loading will apply a hydraulic force to the inner seal spring plate that causes the faces to open.

Over the past two decades the chemical process industry has moved to face to back design and reliability has improved as a result.

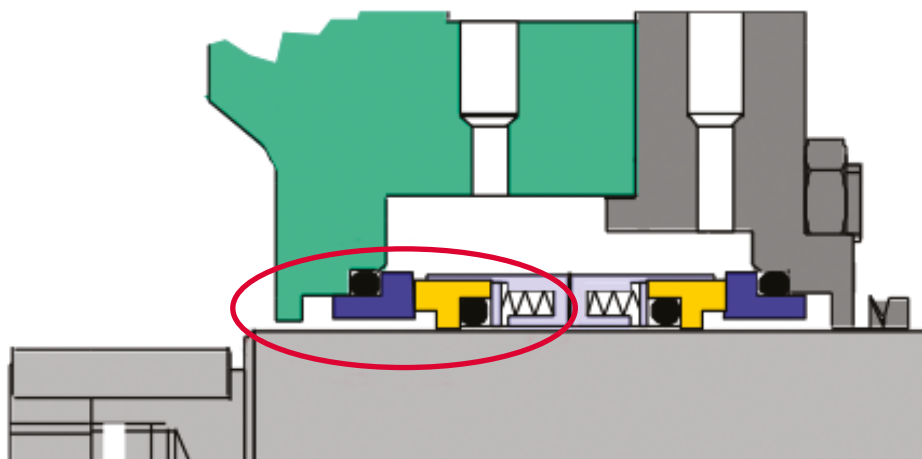


Figure 2a
Back-to-back process fluid on ID of face with potential for hang up inner mating ring un-retained

2.3 Advantages of face to back configurations

These configurations overcome all the weaknesses with other designs. With the pumpage on outside diameter of the seal faces. API 682 preference (5) of this configuration is summarised in a note.

‘The advantages of the series configuration are that abrasive contamination is centrifuged and has less effect on the inner seal.’ A further note supports the case (6). ‘Liquid barrier seal designs arranged such that the process fluid is on the OD of the seal faces will help to minimise solids accumulation on the faces and minimise hang-up.’

Dual balance is easier to incorporate in this configuration. The seal O ring can be configured that allows it to move to the either side of the groove (7) providing a closing force regardless of the direction of pressure. Mating rings can be simply retained either positively or with more modern designs hydraulically. (Figure 2b) Pressure reversal capability provides for greater safety, providing tolerance of process-upset conditions or loss of barrier fluid. API 682 reinforce this point in a note (5),

‘In the event of a loss of barrier fluid pressure, the seal will behave like an Arrangement 2’

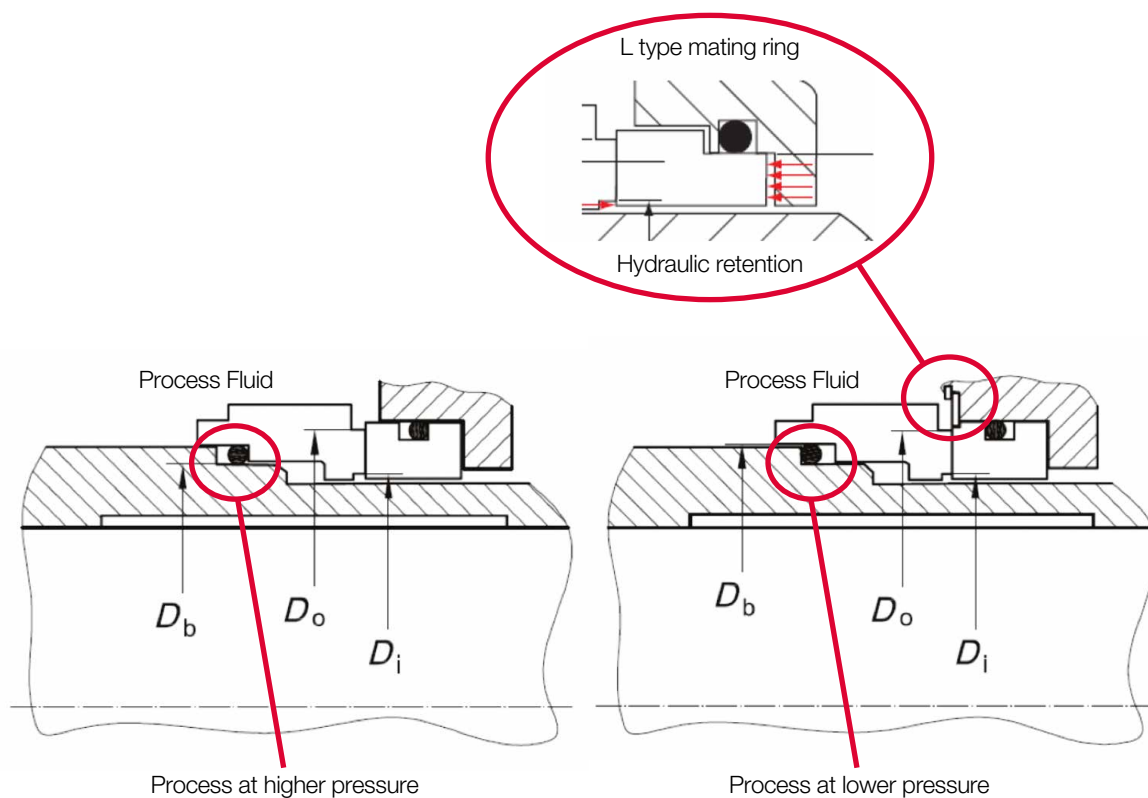


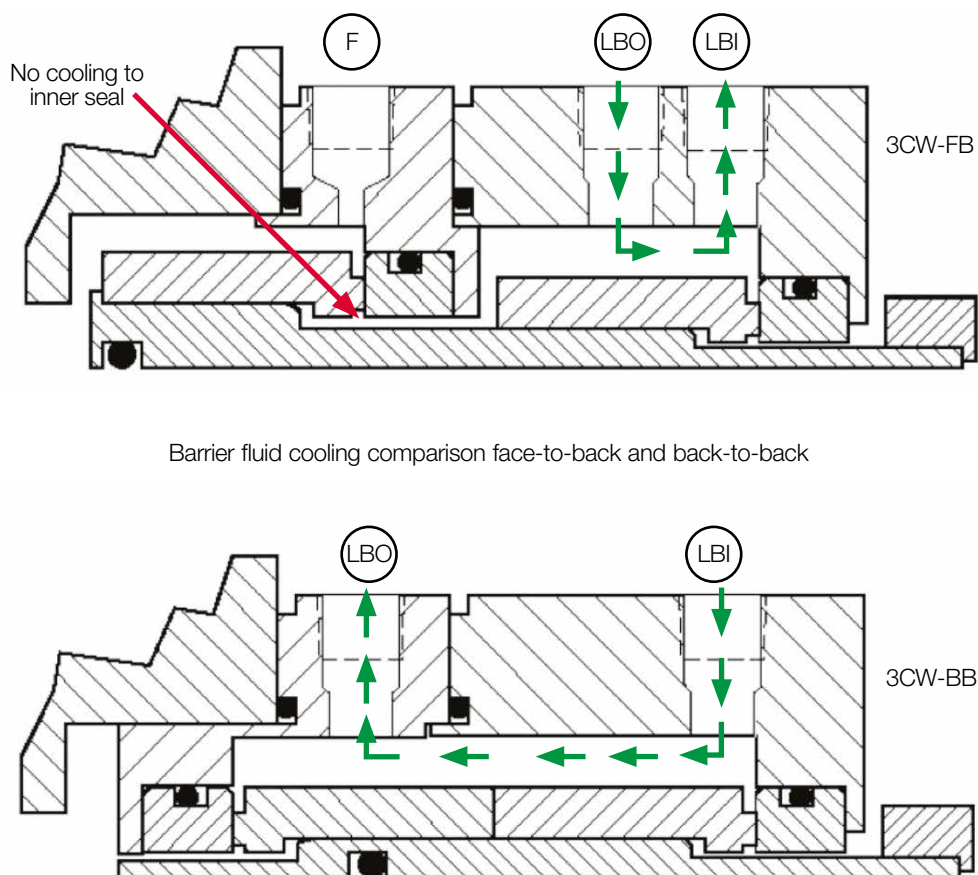
Figure 2b

2.4 Disadvantage's of conventional face to back configurations

The major problem of face to back designs is cooling of inner seal. The conventional approach of seal designers has been to mount component seals with adaptive hardware (sleeve and gland) to form a cartridge. However the barrier fluid flow path to the inner seal is compromised with an area of low or zero flow. The temperature in these stagnant areas will be elevated by heat soak and face generated heat. (Figure 3) The 2nd Edition of API 682 very eloquently provided a (8) warning. 'Restricted seal chamber dimensions and the resulting cartridge hardware construction can affect the ability of the barrier fluid flush to adequately cool the inner seal. Inadequate cooling of the inner seal can result in reduced seal reliability. Selection of a back to back or Face to Face configuration may resolve an inner seal cooling problem'

2.5 Conundrum

The challenge for seal designers is to provide a seal with optimised cooling and simultaneously provide resistance to 'hang up' and pressure reversal.



Barrier fluid cooling comparison face-to-back and back-to-back

Figure 3

3 Developments in face to back technology improving inner seal cooling

Superior cooling can now be achieved by the inclusion of high performance circulating devices and a flow deflector baffle. The deflector baffle diverts the barrier fluid flow to the inner seal providing cooling to the inner seal faces and an elegant solution to the dual seal challenge.

3.1 Deflector Baffle

Deflector baffles have long been employed on single high temperature seals with steam quench. Restrictions in seal chamber dimensions and large cross-section of conventional dual balanced pusher seals have generally prevented their use. Bellows seals with smaller (Figure 4) cross-sections provide sufficient space for use of deflectors in dual arrangements; however the deployment of deflectors has generally been limited to engineered specials rather than catalogue designs.

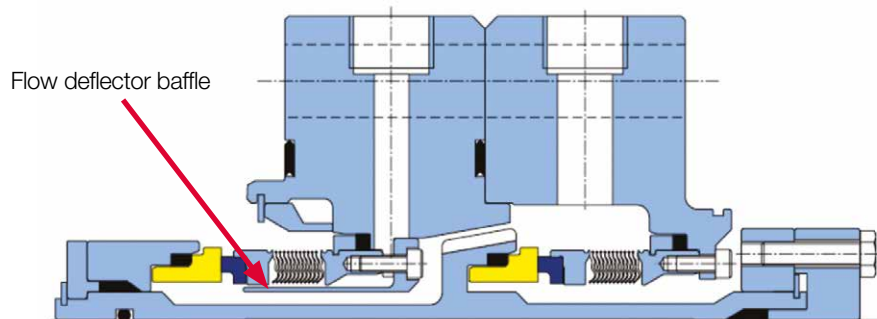


Figure 4

3.2 Developments in Pusher Seal Technology

An alternate method to achieve seal balance is to place O rings mounted on the outside of the seal faces. This is now common practice in the chemical process industries integrating the sleeve as face holder is now possible by utilisation of modern CNC machine tools. These techniques provide a more compact design opening up the inner seal envelope and allow for the provision of the deflector baffle. (Figure 5). Separation between inlet and outlet of barrier flow, results in deposition of cool barrier fluid at the inner seal faces, the most important part of the mechanical seal.

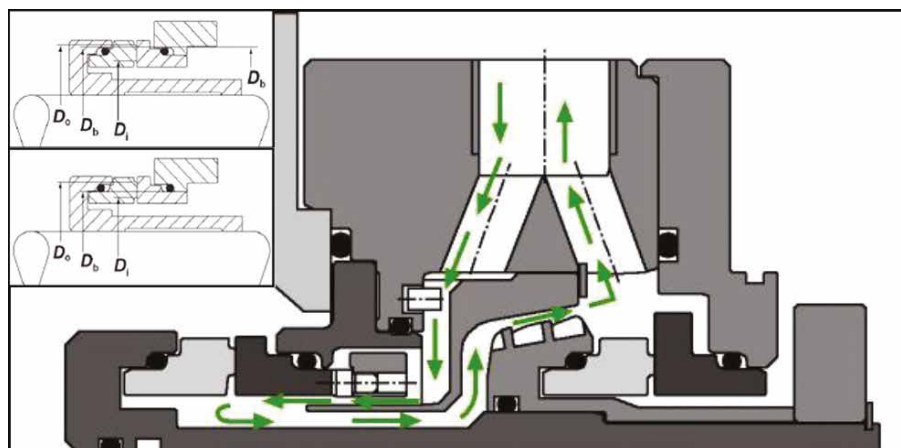


Figure 5 Flow of barrier fluid with deflector baffle

Understanding barrier fluid circulation within a dual seal cavity is a complex issue that requires the use of computational Fluid Dynamics (CFD) to establish flow regimes.

Full 3D CFD models were created of double pusher seal cavities, similar in construction to the one shown in figure 5. The model was generated by creating a cross-section of the seal cavity and sweeping it through 360°. The inlet and outlet ports were properly represented. Typical operating conditions were adopted with barrier flow at 2l/min and speeds of 3600, 2500 and 1800rpm. The fluid was assumed to be water at ambient conditions.

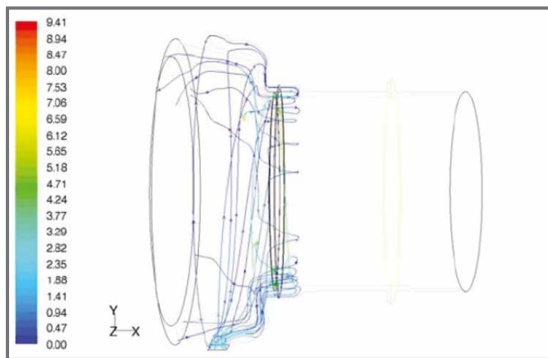


Figure 6a 3D view of Deflector

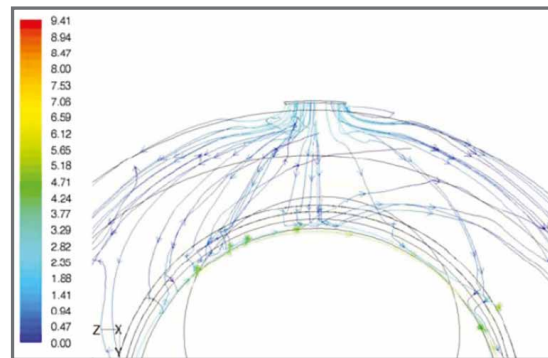


Figure 6b End view plot of deflector

CFD simulations make it easy to examine how the flow occurs in the entire model. Figure 6a provides a 3D view of some particle tracks seeded at the inlet of the model. Figure 6b presents an end view zoom plot close to the inlet region. This figure shows the tracks of particles in the region where the fluid starts to split into two opposite directions providing circumferential flow to the deflector.

Full 3D models were created of double pusher seal cavities, similar in construction to the one shown in figure 5. The model was generated by creating a cross-section of the seal cavity and sweeping it through 360°.

4.1 Deflector flow optimisation

Figure 7a illustrates the velocity vector distribution (for the X and Y components) in the region next to the deflector.

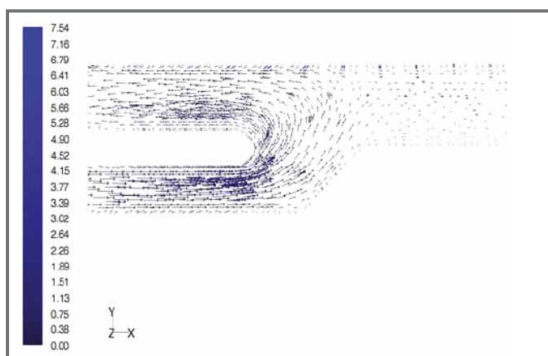


Figure 7a Deflector end

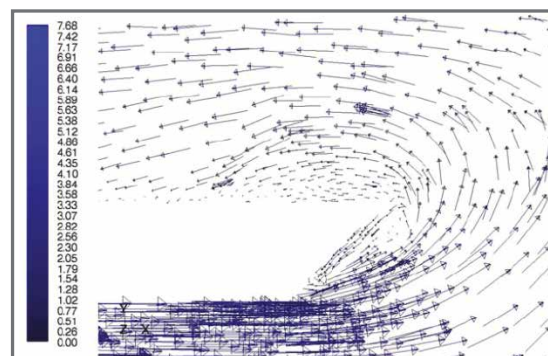


Figure 7b Deflector

It was expected that the flow would separate at the end of the deflector as it turns by 180 degrees. However, with the low velocities and the round profile the flow remains attached. It is desirable to get more flow underneath the inboard seal faces however few particles of fluid seem to travel in the direction. With the end of deflector re-profiled to triangular sharp edge (figure 7b), the analysis indicates there is a radial motion next to the extremity of the deflector. This re-circulation reduces the flow path and prevents some fluid close to the deflector nose escaping directly. Compared with the original round shape the sharper shape is able to redirect more flow to the seal faces due to the vortex motion.

5 Circulation devices performance for API plan 52 & 53 systems

Dual wet seals barrier fluid circulation can be, by external means, API Plan 54 figure 8a, or internal circulating devices as with API plan 53a,b,c, (also API plan 52), figure 8b (8).

Figure 8a

API Plan 53a Pressurised external barrier fluid reservoir. Circulation by internal pumping ring.

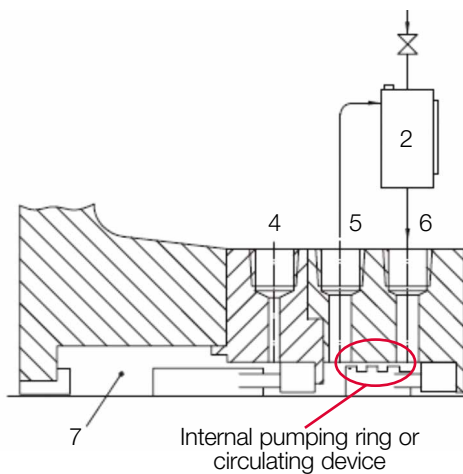
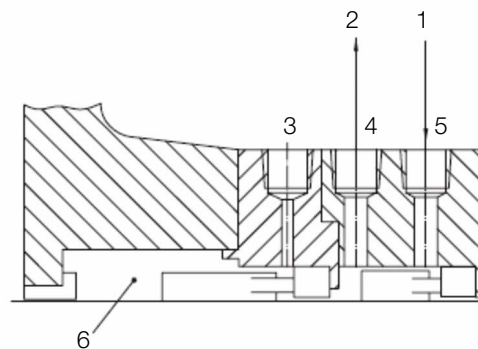


Figure 8b

API Plan 54 Pressurised external barrier fluid system. Circulation is by an external pump.



Effective cooling of the seal depends upon the efficiency of these devices. Internal devices are part of the seal cartridge; their performance can be affected by many factors:

Factors which can affecting circulation flow

- Direction of rotation
- Seal size
- Seal type
- Shaft speed
- Barrier fluid density
- Barrier fluid viscosity
- Circulating or pumping device design
- Barrier fluid containment vessel (or cooler)
- Seal cavity flow path, concentricity, contour
- Gland port orientation ports, (top/bottom, tangential)
- Connecting pipe size & layout (bends/distances)
- Fittings connections roughness pipe bore

Traditional internal circulating devices Figure 9a & b fall into two groups, parallel slot (castellation) & helical vane. Parallel slot devices induce a radial flow and must be positioned adjacent to the barrier outlet orifice, they can be bi-directional only if used with radial ports. Helical vane devices are uni-directional, provide an axial flow and are less dependent on port proximity. The use of multi axis CNC machine tools has provided their designers with far more freedom in sculpturing efficient arrangements. Figure 9c illustrates a modern bi-directional large clearance ‘taper vane’ that provides improved circulation.



Figure 9a

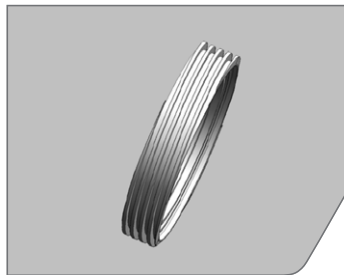


Figure 9b



Figure 9c

5.1 Experimental investigation of pumping ring performance

Back to back testing was conducted on the three devices described using water, oil and diesel(10). The test seal was a 100 mm API 682 dual pusher seal with radial ports. A simple plan 53a circuit was constructed and the devices tested on the same seal. Test rig speed was maintained at 3600, 3000, 1800 or 1500rpm and the system resistance was varied using a flow control valve. Results concluded (Figure 10) that the tapered vane design provided improved performance in both head and flow.

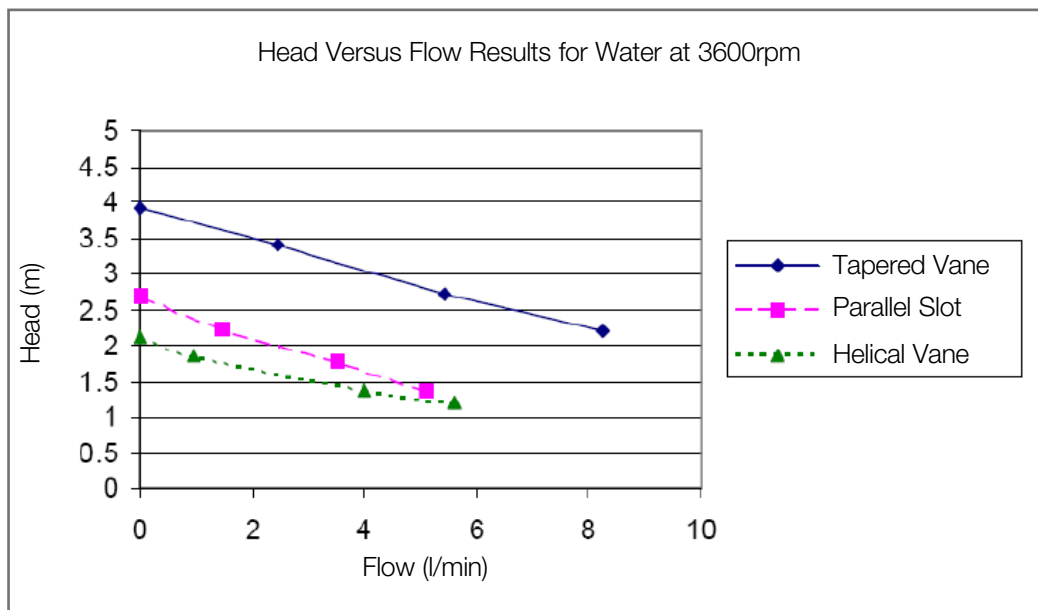


Figure 10
Results for 100mm seal Radial (1/2 NPT) Ports (Concentric Cylindrical Bore)

5.2 Circulating device practical application & limitations

Tangential porting arrangements can offer improved performance on all three types of device, and utilising this feature on large between bearing pumps is often straightforward. On smaller units this can be challenging where gland stud position and pump frame casting will often interfere. The performance of all the three tested devices may also be improved by modifying the internal cavity to include a cut water or eccentric bore. Due to gland plate machining costs these practices on API 610 pumps tend only to be found on engineered specials.

Reduction in internal radial clearance can also improve device performance, however there are safety implications, sufficient clearance is required to prevent contact between rotating and stationary components under fault conditions. API 682 specifies (11) a minimum radial clearance of 1.5mm. Small internal clearances may also prevent adequate thermosyphon action (12) when the shaft is not rotating. This is a required to prevent over heating of the seal just after shut down (when a pump will still be a full temperature) or during warm up. Increase port sizing also improve pump ring performance.

The flow required in service depends to a large degree upon the amount of heat which must be removed from the seal by the barrier system. Formulae for seal face temperature heat generation and heat soak from the process to the seal chamber are widely published (13). On small higher temperature pumps the heat load on the seal barrier system will primarily be heat soak. Cooling requirement does not vary greatly regardless of shaft speed.

As with any pump, two very significant factors affecting circulating device performance are the impeller diameter and the shaft rotation speed. Performance becomes critical to seal reliability on smaller pumps and on pumps operating at 4 pole motor speed or with variable speed drive. Circulating devices that can operate efficiently at lower speed on smaller shaft using larger clearances with large clearances and radial porting will have the widest potential application group. Bi directional designs will eliminate installation error on between bearing pumps and can reduce spares holding.



Figure 11

5.3 Enhancing performance for lower speed or smaller pumps

The original taper vane bi directional device has now been in commercial service since 1999. Recent research into vane profile and angles has provided a further increase of up to 40% in flow. The vanes are contoured into a 'swan neck' shape which prevents reverse flow on the back edge of the vane. The performance of this device is virtually equal to that of a similarly configured unidirectional taper vane.

The swan-neck design of Fig. 11 has been tested at different shaft diameters and shaft speeds, in an identical arrangement to the previously discussed test results. This includes radial clearances of 1.5mm between the rotor and stator, thus fully conforming to current API-682 standards. The results of a (typical medium size pump) 50mm seal 'swan-neck' tapered design are shown in figure 12.

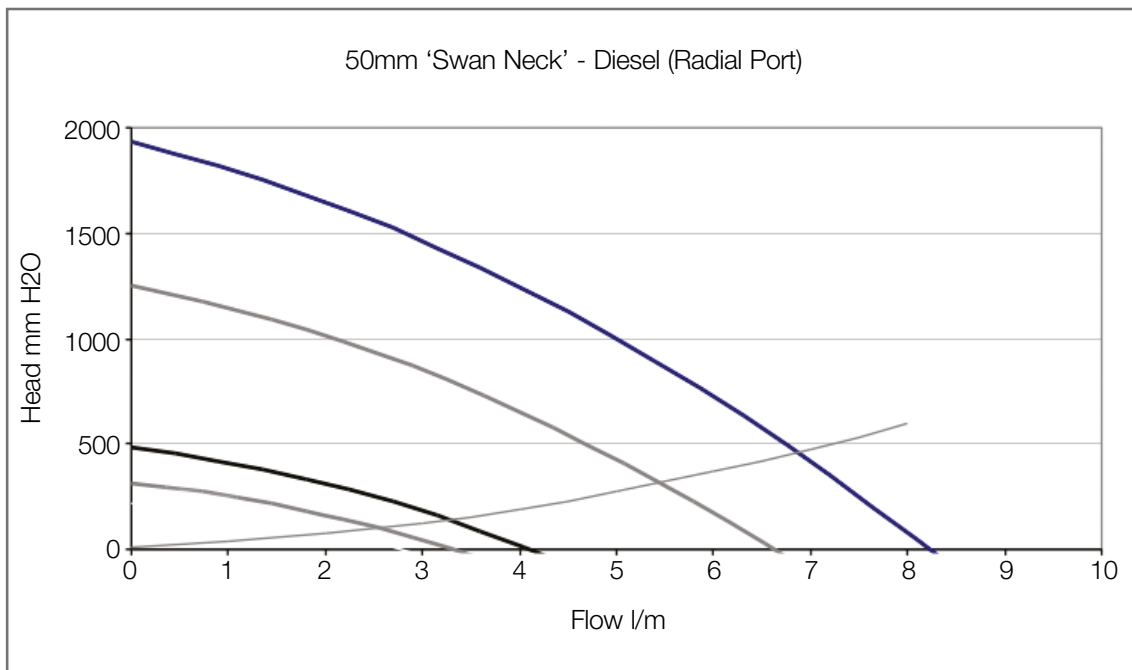


Figure 12

Three case studies are presented where improved reliability has resulted from implementing some of the issues discussed in this paper.

6.1 Face to back replacement of back to back.

There are many case studies published that demonstrate improved reliability where back to back seals have been replaced by face to back. Perhaps one of the most unusual recent examples comes from a relatively benign industry. Back to back seals have been installed in circulation pumps used on industrial laundry machinery packages manufactured in the UK by Thomas Broadbent and sons Ltd.

The duty would be considered 'light' by comparison to most industries. The pumps were ISO type operating at 85°C with a seal chamber pressure 3bar and seal sizes 28mm and 65mm. The pumped fluid however was contaminated with fibre (lint). The fibre would enter the seal chamber and become packed up the inside diameter of the inner seal faces. Resultant seal face hang up caused premature failure. The manufacturer wanted to improve the reliability of the package plant and converted the pumps to face to back arrangements. The new seal arrangement has improved pump MTBF over 6 fold.

6.2 Replacement of a back to back with advanced face to back technology including flow deflector baffle

A complex reactor circulation duty at a chemical plant in Wales, UK, demanded that a twin screw pump (Albany Engineering) be used and fitted with four dual seals. Conditions included; process temperature between 25 -180°C, seal size 54mm, seal chamber pressure vacuum to 3.5 bar, viscosity between 0.5cP -5000cP and shaft speed 180 - 1500 RPM.

Light silicon oil was selected as the barrier fluid. This had proven problematic on other applications due to its poor lubrication problems and heat transfer properties at elevated temperatures but had to be used due to compatibility with the process fluid. An external pumping device Plan 54 was selected due to the use of variable speed drive.

Several variants of back to back bellows cartridge seals were trialed. Failure occurred typically after 6 months. Failures have been attributed predominantly to pressure reversal issues as described in 2.2. Face to back seals with deflector baffles had been successfully used in the plant, replacing both traditional face to back and back to back designs. This technology was adopted for the screw pumps with replacement seals providing a service life in excess of 2 years. The seals removed after this period exhibited seal faces in first class condition.



6.3 Tradition Face to Back Vs Advance face to Back with internal pumping Plan 53 Paint plant

The automotive industry has made significant strides in the last two decades in its coating technologies. Car body corrosion is now virtually unheard of. One of the reasons for such a success is full immersion in primer dip tanks with an electrical charge applied. A difficult sealing application is the sealing of the electro coat primer circulation pumps. Double seals are employed for the sealing of this duty. The paint presents many challenges. Containing sub micron abrasive particles is necessary to use hard-faced materials on the inner seal to provide adequate protection against wear. The paint has to be kept very close to the process temperature of $25^{\circ}\text{C} \pm 10^{\circ}\text{C}$. Elevated temperature excursions cause the paint to rapidly congeal. Compatible barrier fluid selected for these duties is ultra filtrate, which is primarily de-ionised water with a few other chemicals acting as a thinner.

End suction pumps are used for these services typically with a 50mm shaft. Inner seal cooling and poor circulation device efficiency have prevented reliable operation of conventional face to back seals and API plan 53a systems. The paint is tenacious and climbs across the barrier fluid film pressure between the inner seal faces. Heat generated by these faces causes the paint to congeal or polymerise, as the paint particles roll up; they jack the seal faces apart causing excessive leakage. After clean up removed seals would reveal faces in immaculate condition. Increasing the barrier fluid pressure is a natural defence however this increases temperature generation with added heat load accelerating the problem.

Reliability has traditionally been achieved by using the high flow rates of API plan 54. These complex external barrier fluid circuits require instrumentation ensuring that equal flow to each seal in the train of pumps. Seal fault diagnostic across a train of pumps is also difficult. The cost in terms of CAPEX and OPEX of such Plan 54 systems can be very high.

Plant manufacturers and plant operators have sought alternative solutions. Modern face to back cartridge seals with high efficiency circulation devices and barrier fluid flow separation can now operate in this duty with API plan 53a. This system offers many advantages in terms of cost and simplicity & has been installed successfully in 5 automotive manufacturing facilities across Europe with other installations across the world. In an automotive facility in the North West of England, the advanced face to back technology seal is now providing service life exceeding 7 years life.



Mechanical seal technology continues to develop. Sound principles are applicable across all industry sectors. Robust faces to back designs are now available with enhanced cooling features, which will increase the application base.

References

1 ANSI/API Standard 682 3rd edition, September 2004 ISO 21049 (Identical) Pumps-Shaft Sealing Systems for Centrifugal and Rotary Pumps American Petroleum Institute Washington DC

2 API Standard 682 3rd edition, clause 7.3.4.2.1

3 API Standard 682 3rd edition, figures 3 & 4

4 API Standard 682 3rd edition, clause 7.3.4.3

5 API Standard 682 3rd edition, clause 7.3.4.2.1 NOTE

6 API Standard 682 3rd edition, clause 7.3.1.1 NOTE 1

7 API Standard 682 3rd edition, clause 6.1.1.7

8 API 682 2nd Edition JULY 2002 7.3.4.2.1 NOTE

9 API Standard 682 3rd edition Annex G

10 Integral pumping devices that improve mechanical seal longevity.

C Carmody, A Roddis, J Amaral Teixeira, D Schurch 19th International Conference on Fluid Sealing, Poitiers, France: 25 - 26 Sept 2007

11 API Standard 682 3rd edition, clause 8.6.2.3

12 API Standard 682 3rd edition, clause 8.2.3

13 API Standard 682 3rd edition Annex F

Acknowledgements

The author would thank the following for their assistance in the compilation of this paper;

The American Petroleum Institute, for the use of their images.

Mike Munro EEMUA machinery Committee.

Albany Pumps, Gloucestershire, England.

Jaguar Cars, Halewood, England.

Chris Leeper Thomas Broadbent & Sons Limited Huddersfield England.

Dr Chris Carmody & Chris Booth Ba, MBA. AESSEAL plc.

Richard Smith was originally trained in Design Engineering in the Automotive Industry. For the last 28 years Richard has worked in sealing technology, the past 24 years with AESSEAL plc. Now a Director of the company Richard is actively involved with the company's developments in Oil Gas and Petrochem Industries and is the company's representative on the API 682 Task Force currently working on the 4th revision. Richard has had a number of papers published at international conferences journals. These have been predominantly on the practical application of sealing technology.

For further information, visit www.aesseal.co.uk