Saving Energy, Saving Water and Saving the Planet through the use of Affordable, Premium Bi-Directional Pumping Rings

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Abstract

This paper deals with the barrier fluid flow characteristics of dual mechanical seals in API682 sealing arrangements. It compares the main conventional integral flow induction / pumping ring devices found in dual mechanical seals on the market today and reports on the results of extensive benchmark testing. It documents the effectiveness on the head vs. flow characteristics of the barrier fluid and how a seal manufacturers take cost from the product, making them affordable. Moreover, it demonstrates, with the aid of case histories, how it is possible to save water and save energy whilst provide the clearances specified in API-682 without having to make design compromises that might risk reducing seal and/or rotating equipment life.

Introduction

Legislation and environmental issues surrounding the containment of pumped fluids dictate that the consequences of leakage and/or emissions can no longer be tolerated for many liquids, including some that were until recently exempt from such protocol. The most convenient method to prevent the leakage of a pumped fluid to atmosphere is to employ a dual mechanical seal where the cavity in between the two sets of seal faces is filled with a neutral liquid that acts as a separation barrier (Fig. 1).

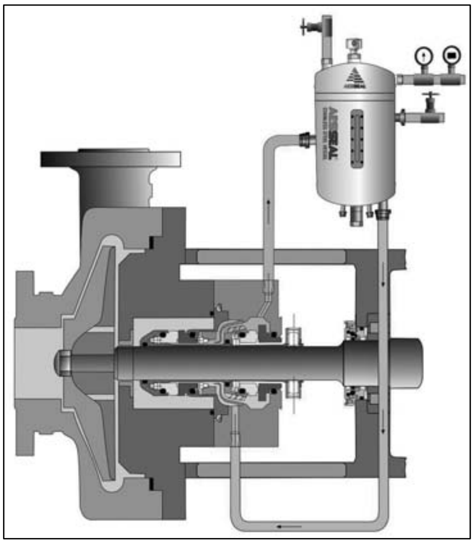


Fig.1 – API Plan 53A

In addition to acting as a neutralizing agent, the barrier fluid also provides lubrication and cooling to the primary seal faces such that it provides a more stable operating environment for the seal thereby increasing longevity.

In order to provide the correct conditions for both containment and providing stable sealing surroundings for the primary seal the barrier fluid needs to be controlled within its own system. This requires the barrier fluid to be circulated via an external pumping device (API Plan 54), an integral pumping device (API Plan 53/52) or through a convective thermosyphon system.

It is widely accepted that an external pumped API Plan 54 system is more expensive to run, consumes energy and is often impractical to install particularly on existing plant upgrades.

Similarly, thermosyphon / convection systems are somewhat unreliable and ineffective at efficiently dissipating the heat within the mechanical seal. These systems are particularly prone to mis-installation, where, for example, sags in the connecting pipework between the seal and system might prevent fluid convection flow leading to the seal overheating.

A more convenient method to circulate barrier fluid is to use an integral API Plan 52/53 pumping device that forms a built-in part of the seal assembly. Such devices rotate with the shaft/seal and cause the barrier fluid to be circulated around the barrier fluid cavity within the dual mechanical seal (Fig. 2).

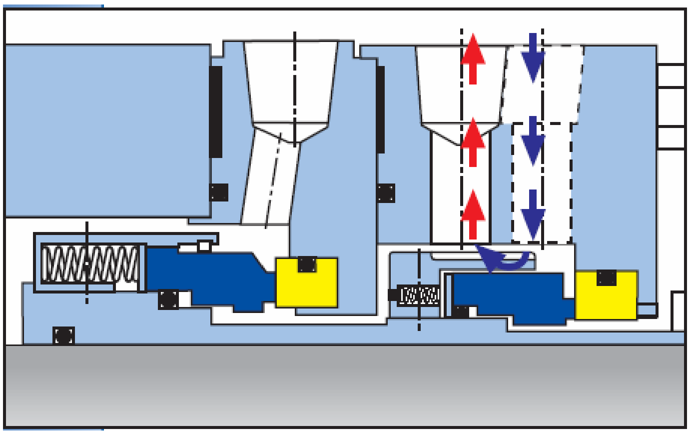


Fig. 2 – Conventional Plan 52/53 Pumping device positioned within a dual mechanical seal

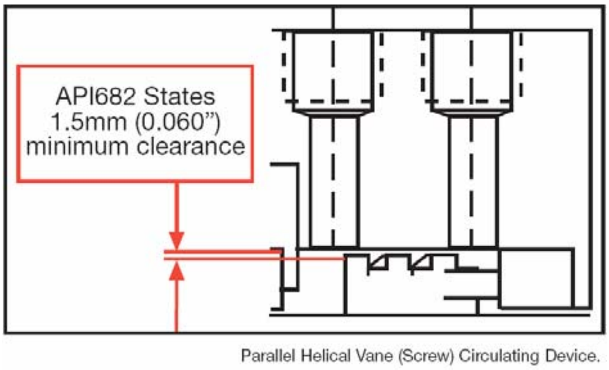
The effectiveness of such pumping ring devices in dissipating heat from mechanical seal faces in dual seals is well documented. Of course, this documentation relates to instances where thus cooling well-lubricated faces tends to increase the mean time between failure (MTBF).

There are, however, quite a number of different pumping ring designs with correspondingly wide range of flow performance characteristics. Some of these designs provide very low flow rates compared to others. It is therefore important that engineers and best-practice facilities understand and note the fundamental differences between each. Understanding these differences will assist in precluding inadvertently compromising a given equipment sealing solution.

It is also important for modern user plants to understand these pumping devices in light of the latest “best practice” reference sources, specifically those created from the wealth of seal expertise in the API-682 Sealing Committee.

Section 8.6.2.3 of the latest API682 specification outlines “best practice” mechanical seal design and defines a radial clearance of 0.060” (1.5mm) between the rotor and stator, as shown in Fig. 3.

Fig. 3 – Radial Clearance requirement between the mechanical seal rotor and stator as outlined by API-682 Section 8.6.2.3



The life expectancy of all dual mechanical seals is strongly influenced by the condition of the fluid in within the sealing cavity. That said, the API-682 standard states a minimum radial clearance of 1.5 mm (0.060”) between the rotating and stationary elements of a mechanical seal for many technically sound reasons.

Close radial clearances between counter-rotating surfaces can lead to component contact and galling. Engineers understand that if a stainless steel rotary component contacts a stainless steel stationary component, galling will occur. Nevertheless, some dual seals applied in industry are designed with radial clearances in the order of 0.010” and 0.020” (0.25 mm and 0.5 mm). This is in clear contradiction to best-practice guidelines and technical common sense. After all, equipment shafts are known to deflect. Pumps operating away from BEP (best efficiency point) and single volute pumps are certainly undergoing a measure of shaft deflection. If, then, the dual seal radial clearance is less than the centrifugal pump bushing clearance, which is going to touch first?

The problem lies in that seal design engineers are often faced with two sets of conflicting requirements. One is to make the radial clearances safe and contact free, and the other is to remove heat from the seal. This requires adequate flow rates and heads produced by the circulation device to overcome the seal and support system resistance. The conflict arises when many circulating devices becoming inefficient as the radial clearances are increased.

In an attempt to quantify the differences in pumping ring performance, this presentation refers to test work conducted on three different types of integral flow inducing devices (so-called pumping rings). The pumping rings were all positioned adjacent to the outboard seal faces and, except for the obvious differences between pumping ring designs, all other seal components used in the test were identical. Here, then, are the particular differences between each of the pumping rings tested.

Parallel helical vane (screw-thread) pumping ring design

Parallel helical vane devices (Fig. 4) are unidirectional components that will only work with the shaft rotating in one direction. Such a device operates similar to a screw and usually requires a radial clearance in the order of 0.010” (0.25 mm) between the rotor and stator.

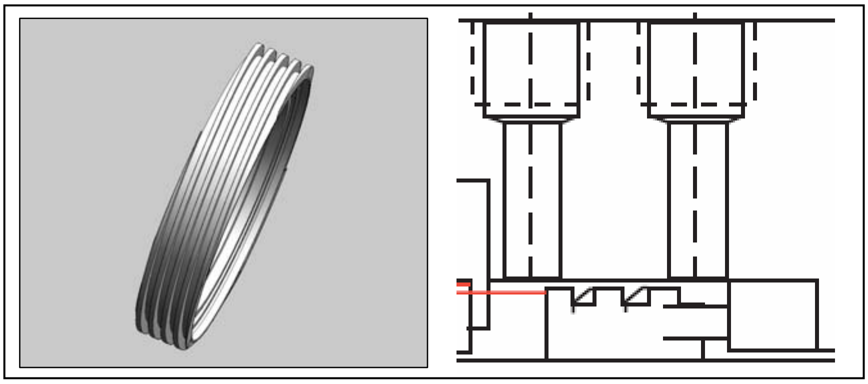


Fig. 4 - View of a typical parallel helical vane pumping device.

Parallel slot or castellated pumping ring design

This device is a series of parallel slots around the circumference of a rotating member, thereby inducing tangential motion to the fluid. Unlike the previous device, a parallel slot configuration works irrespective of shaft rotation. It circulates fluid by expelling it up through the outlet port, using the centrifugal velocity induced from the rotating action of the slots. This means that the circulating device must be positioned adjacent to the barrier outlet orifice. Fig. 5 illustrates a typical parallel slot configuration where the design is such that the fluid is not directly circulated around the inboard seal faces.

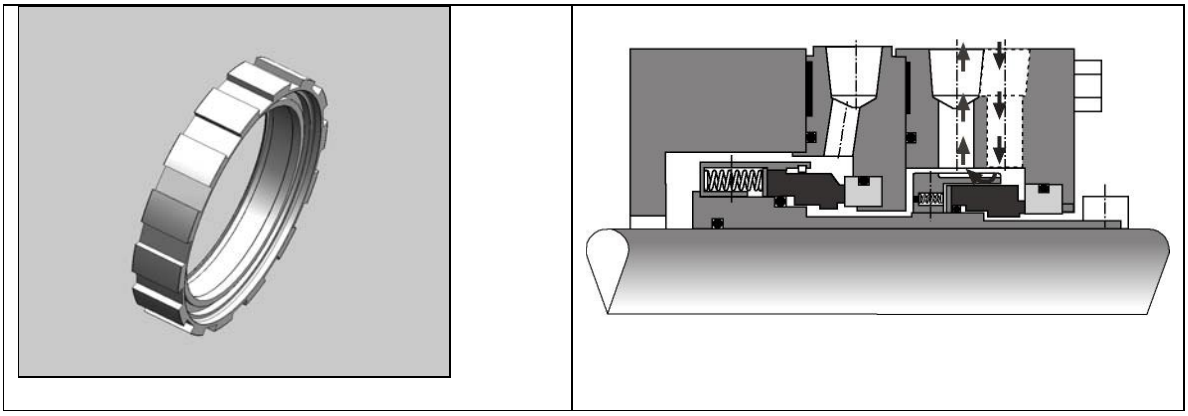


Fig 5. – Left: a view of a typical parallel slot pumping ring; right: the typical position of A parallel slot device within the dual seal barrier/buffer cavity

Tapered vane pumping ring design

Tapered ring devices incorporate a series of vanes around the circumference of the rotor. The vanes are angled in two directions, making the device fully bi-directional and causing fluid to circulate irrespective of shaft direction. Both rotor and stator have an inclined surface that re-directs the fluid. Some of the centrifugal velocity induced by the rotating action of the slots is redirected into an axial flow. This means that the circulation device is not dependent on maintaining a close radial clearance between the rotor and stator. Furthermore, it means that the device can be placed at any axial position between the inboard and outboard seal faces. Fluid can now be effectively replenished at the inboard seal faces as well as the outboard faces. Figure 6 depicts the difference between the designs.

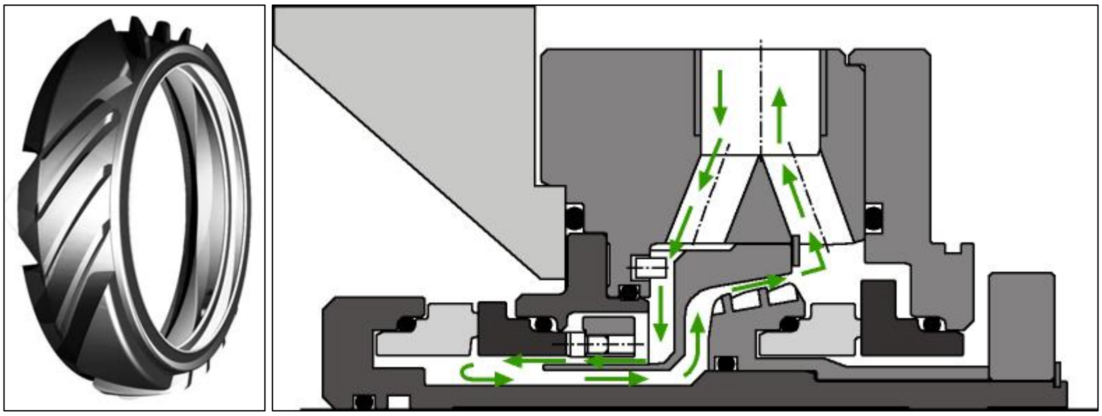


Fig 6. – Left: view of a typical tapered vane pumping ring; right: typical position of a tapered pumping vane device within the dual seal barrier/buffer cavity of a pump

Benchmark test procedure

The test seal was a 100mm (4.000”) API dual pusher seal with Carbon/SiC/Carbon/SiC faces and Viton elastomers throughout. The inlet seal port was positioned at 3 o’clock and seal outlet port at 12 o’clock. All hoses were SS braided with ½” BSP connectors and all tests were conducted on a circuit as shown in Fig. 7.

In each case the device was tested in exactly the same manner, whereby the speeds were set at 3600, 3000, 1800 or 1500 rpm and the head was varied using a flow control valve varying the head from no flow to full flow. This process was repeated for each pumping ring device and conducted with water, oil and diesel fuel. Between each test the test rig was decontaminated to remove all trace of the previous test fluid. The properties of the test fluids are shown in Table 1 and the test equipment in Table 2.

Table 1. Test fluid properties.

	Specific Gravity (SG)	Kinematic Viscosity @ 40^{O} C
Water	1	1 cSt
Thermotec Oil	0.853	16 cSt
Diesel	0.88	4 cSt

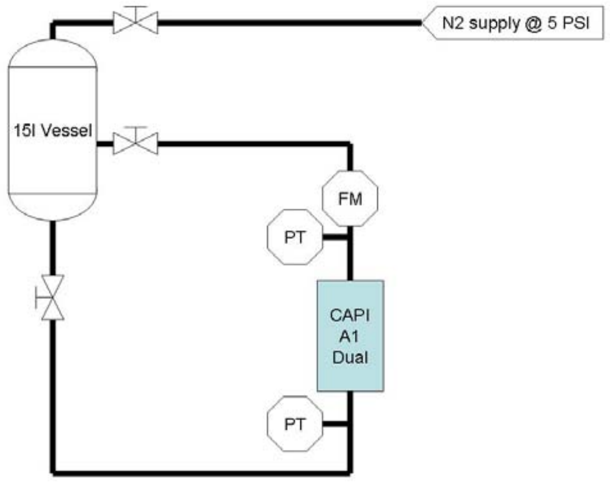


Fig. 7 - The test circuit (schematic left and photograph on right).

Table 2. Test equipment.

Flowmeter– Platon/Roxspur Measurement & Control Ltd (Cache Instrumentation) Type GMTX Vampire (Variable Area Microprocessor Indicator Readout Electronics) totally solid state high performance electronic interface for metal tube VA flowmeters with a 4-20 ma transmitter with comprehensive Liquid Crystal Display. Calibrated on water at 0-10 L/Min with adjusted Flow Rate Multiplier.
Inverter– ALSTOM Alspa MV 3000C inverter controlling a 15 kW 415V 3 phase 2 pole motor. Inverter to Fluke Input channels record speed and rig power consumed.
Data Logger – Fluke 2640A netdac Networked Data Acquisition Unit. The 2640A units are 20 channel front ends that operate in conjunction with netdac logger for Windows to form a data acquisition system and Trend Link Software which is a package that graphs Real Time and historical trending data in the Microsoft Windows operating environment.
Pressure Transducers– 2 off Honeywell Eclipse Model Hirschmann range 0-1 barg. Transducers calibrated against special analogue calibration gauge at high and low values to derive Mx+B values.
System Vessel – AS 15 vessel (Stainless steel 15 litre capacity) with internal cooling coil fed chilled water at 5 Deg C.

Test results

The primary measurements were pressure (head), flow and power; they were recorded. Graphs were generated for the head against flow characteristics and these then used to assess the relative performance for each device. Note that in the case of the parallel helical vane device only the results for the normal direction of rotation are stated. This is because the results in the reverse direction were either so low that flow measurements were too low to be detected or that there was no flow.

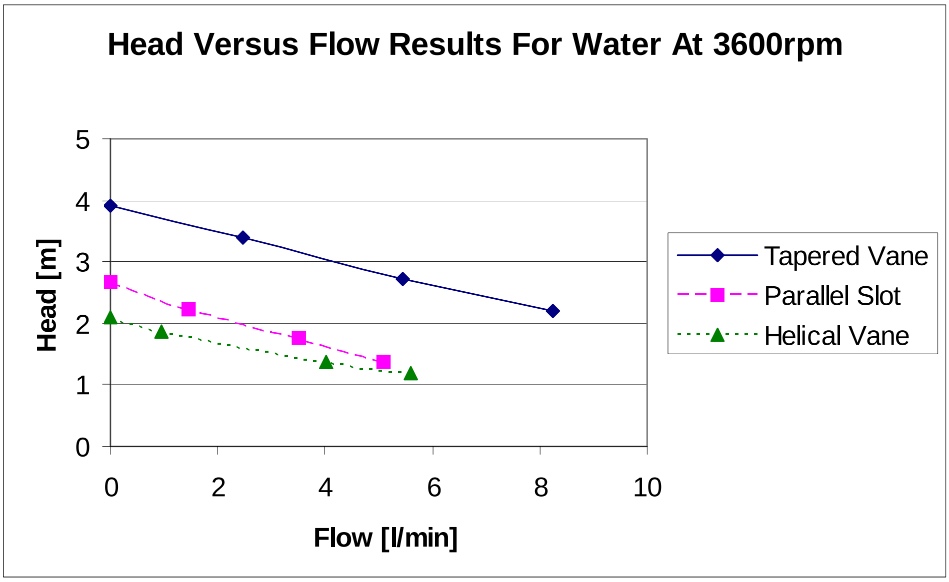


Fig. 8 - Pumping ring results comparison for water at 3,600 rpm.

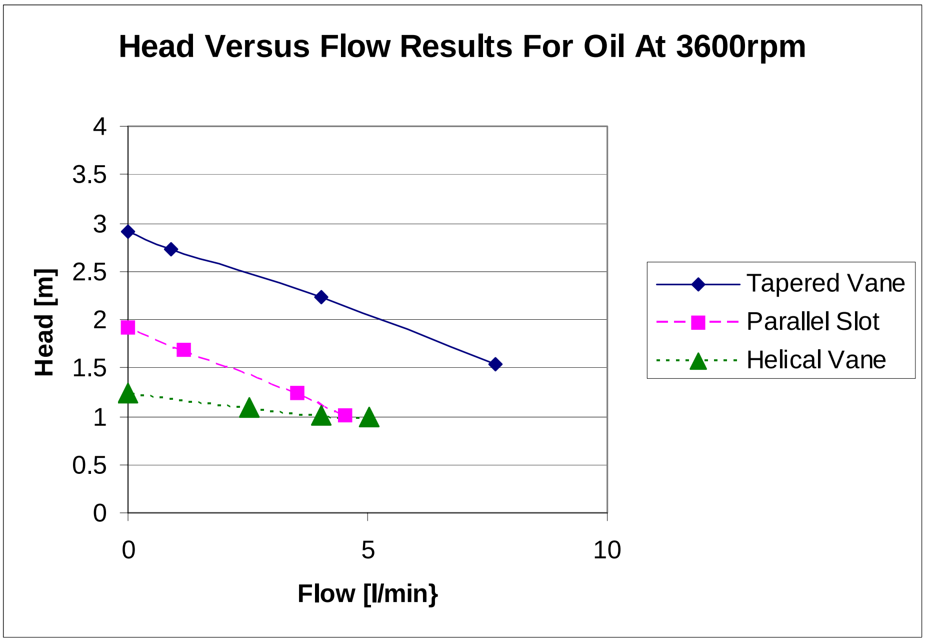


Fig. 9 - Pumping ring results comparison for oil at 3,600 rpm

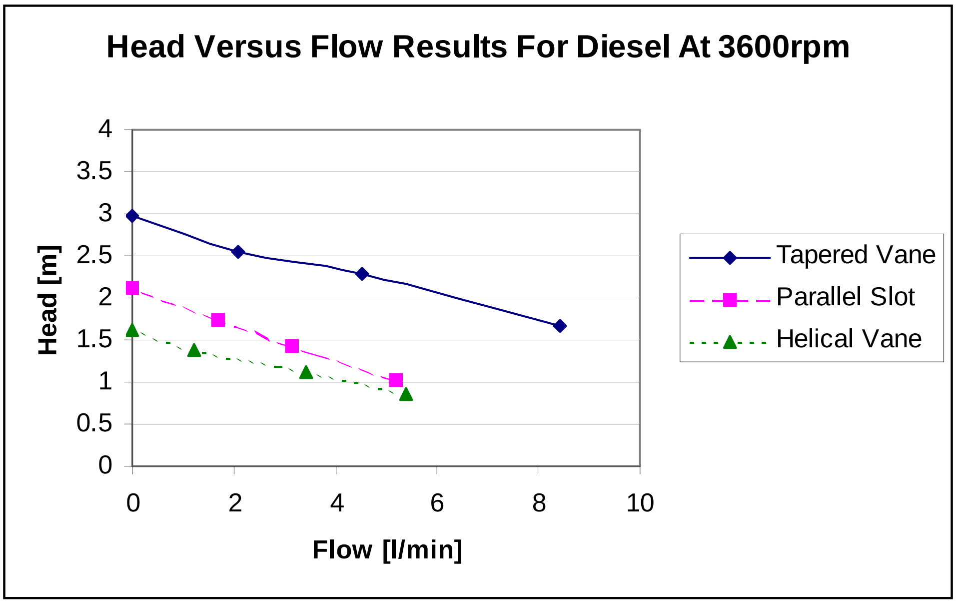


Fig. 10 - Pumping ring results comparison for diesel at 3,600 rpm.

All of the above tests were conducted with devices that had a 100mm (4.000") shaft diameter.

The results show that the tapered vane pumping ring consistently provided better head-flow (H-Q) characteristics than either of the other two devices. It should again be noted that, unlike the parallel screw design, the tapered vane pumping ring design is fully bi-directional and consistently circulates barrier fluid irrespective of shaft rotational direction.

As one would expect, the higher speed tests produced more flow than the lower speed tests. Similarly, the water results tested better than oil and diesel fuel. This is explained by the different specific gravity and viscosity characteristics.

Given that a typical dual mechanical seal operating at speeds over 3,000 rpm requires cooling flows of about two liters/min (0.5 US gal/min) and that a typical system head for a barrier fluid circuit is in the order of two meters (~78"), the results show that the tapered vane device was the only configuration providing sufficient flow for all three fluids tested.

These tests provide evidence that design compromises can be avoided. Indeed, excellent fluid circulation flow rates can be achieved while adhering to the best practice guidance for safe operation, as outlined in API-682. In the interest of brevity and space, our presentation does not list the voluminous test data. Typically, testing at lower speeds of, say, 1800 rpm, produced flows that were about 60% of those presented in this paper and this trend was seen in all three fluids tested.

Further Investigation

From the above results, it is clear that the tapered vane design (Fig. 6) outperforms *all* other known and tested dual seal pumping ring designs available on the market today, and as tested in accordance with the radial clearances stipulated by API-682.

It is worthy of note, again, that the test results presented were for a 100 mm (4.000") seal. It is entirely reasonable to assume that flow/head performance will reduce as the seal diameter reduces fairly equally across all of the designs tested.

It is also reasonable to assume that the average shaft size of hundreds of thousands of centrifugal pumps is around 2.375" (60mm) and rotates at 1,800 rpm or 3,000 rpm. One can therefore anticipate a significant drop-off in the above flow induction performance graphs as both the shaft/seal diameter are reduced, or as speeds are being decreased. To then avoid fluid slippage and so as to gain efficiency, most mechanical seal manufacturers resort to reducing radial clearances. Unfortunately this clearance reduction significantly increases the probability of contact between rotating and stationary flow induction devices. It's an added risk that can be avoided by tapered vane devices similar to the patent sketch of Fig. 11.

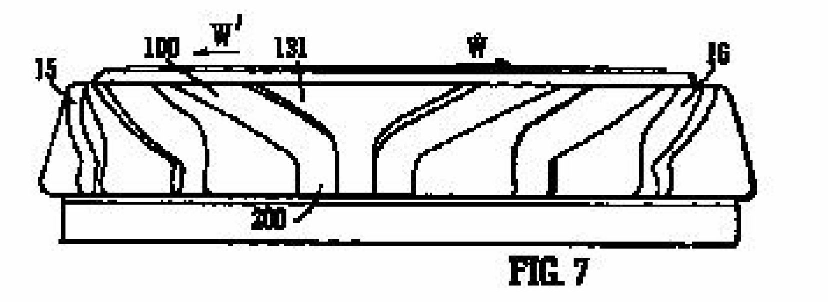


Fig. 11 – "Swan-neck" tapered vane pumping ring

The swan-neck design of Fig. 11 has been tested at all shaft diameters and shaft speeds, in an identical arrangement to the previously discussed test results. This includes radial clearances between the rotor and stator that fully conform to current API-682 standards. The results of the 60mm (2.375") "swan-neck" tapered vane pumping ring design are shown in Figures 12 to 14.

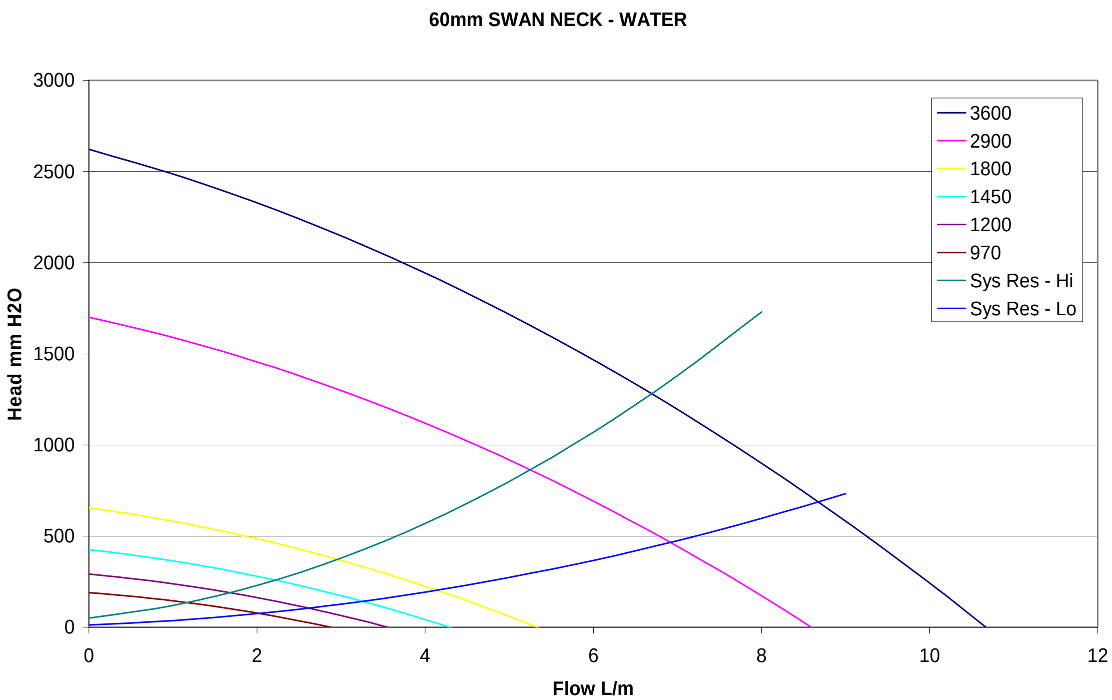


Fig. 12 – 60mm swan-neck tapered vane pumping ring performance on water

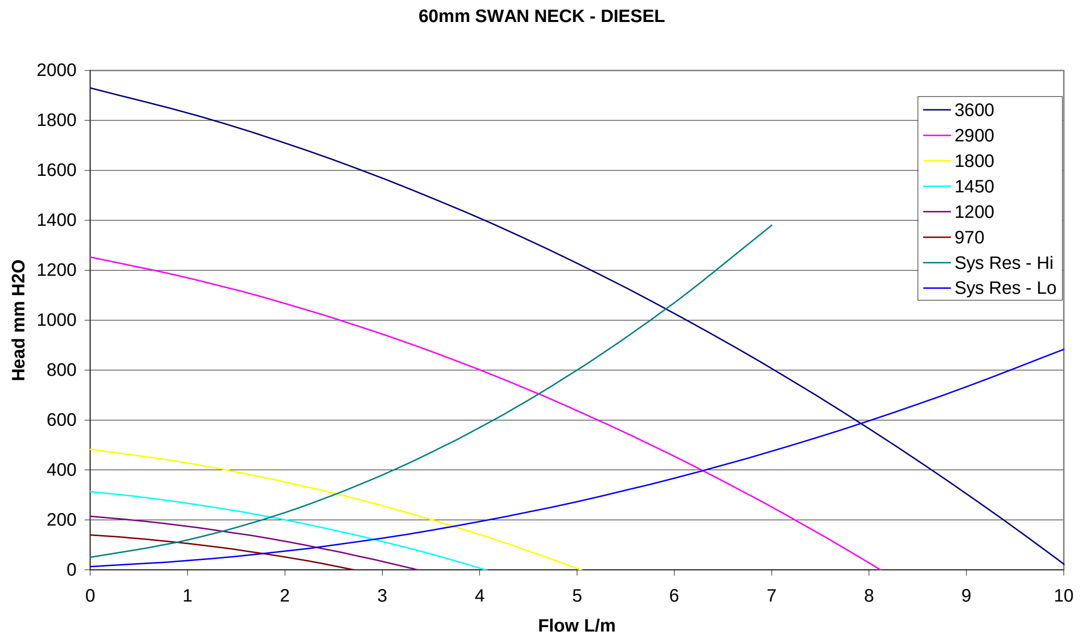


Fig. 13 – 60mm swan-neck tapered vane pumping ring performance on diesel fuel

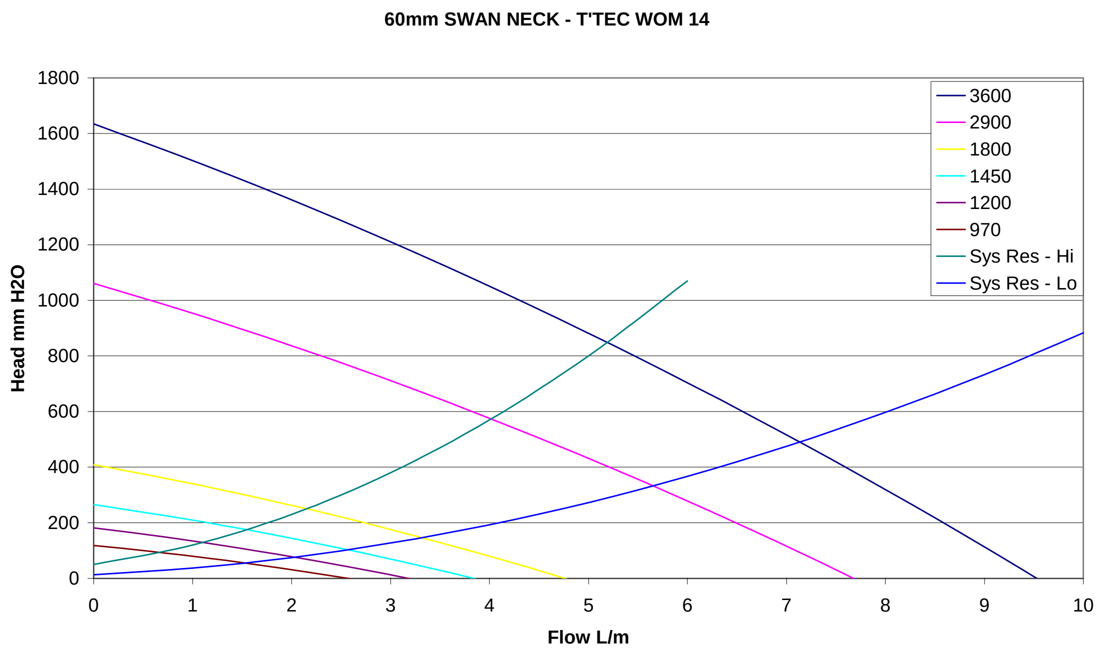


Fig. 14 – 60mm swan-neck tapered vane pumping ring performance on lube oil

The authors believe, from these data, that the design delivers not just cooling flow. By making possible certain flush plan optimisations, it also reduces plant energy consumption. The bi-directional capability of the design means user advantages are realized through product standardization for double ended pumps. This often offers user inventory reduction and reduced likelihood of inadvertent mis-installation.

What about directing the flow to the seal faces?

Unlike parallel slot pumping ring designs that will only work when positioned adjacent to a barrier/buffer fluid inlet/outlet port, the tapered vane design relies on the centrifugal action of the rotating fluid to force the barrier fluid through the port orifice. In contrast, the barrier fluid flow path of Fig. 5 allows circulation around the outboard seal faces only. At the inboard seal faces, the result may well be lack of fluid replenishment and replacement. Stagnation and heating up are likely experiences. Evidence suggests that with stagnating regions surrounding the sealing interface, components tend to fail far more frequently than with more adequate circulation and larger clearances. It follows that if either, or both, of these can be maximized, failure risk would be reduced and seals would be give the chance to live longer.

Comparisons against helical screw devices are quite similar. A distinct advantage with a design such as a parallel screw, taper vane or swan-neck taper vane pumping ring is that it can be positioned at any axial location within the seal barrier/buffer cavity. This means that the flow can be directed to both sets of seal faces, as shown in Fig. 6 and repeated in Fig. 15.

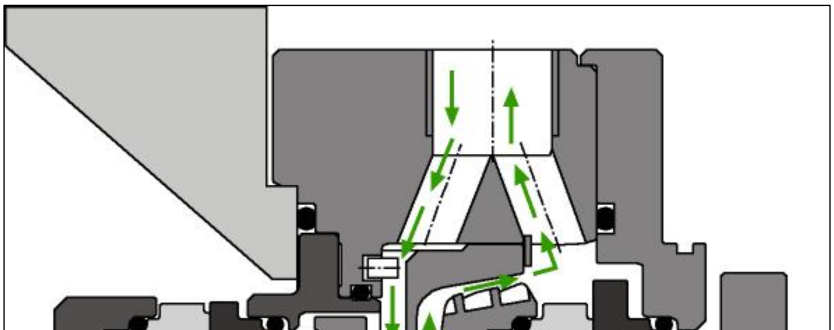


Fig. 15 – Barrier fluid flow path in a seal equipped with tapered vane device

This simple best practice concept of directing cool fluid to where it is needed the most within the mechanical seal has been proven to extend mechanical seal life dramatically.

What can be done to make Pumping rings more economically viable?

Continuous improvement businesses strive to constantly take cost out of their products. Making innovation affordable is the key to delivering real customer benefits. The vanes on the tapered vane pumping ring design need 4 axis milling. In general the milling operation takes a considerable about of machining time compared to turning. Investment castings are therefore one method to eliminate this vast amount of in-cycle machining, given the vanes do not need post casting machining as their function is merely to circulate fluid.

Fig. 16 – Investment Cast Taper Vane Pumping Ring

Figure 16 shows an investment cast tapered vane pumping ring, offered by the product supplier and innovator. This investment cast requires a complex and costly die for each and every size of product produced. The die needs to be designed with a sets of spirally extracting male vanes, to ensure the pattern or wax replica part can be extracted from the die before the liquid ceramic slurry is applied.

What are the Benefits of an Affordable Innovative Bi-directional Pumping Ring?

Plan 53 and 23 systems offer incredible water and energy savings compared to conventional flush systems.

In our technical paper titled,

"Mechanical seal efficiency considerations for pumps in utilities economics of seals vs. packing and savings with best available seal flush arrangements"

presented in the proceedings of the PWR2008, ASME Power, July 22-24, 2008, Orlando, FL, USA, the paper concluded that converting a pump seal from open quench to a barrier system avoided discarding 1577 m^3 of water per year for a large UK effluent treatment plant. This provides a power saving between 236 and 1352 kWh per year and, in the UK, equates to a CO_2 emission reduction between 118 and 676 kg.

There are millions and millions of industrial pumps in operation in the UK alone which still operate with open quench flushing systems. The benefits of efficient bi-directional pumping rings are well documented; - a copy of the above technical paper is available on request.

PRACTICAL APPLICATION EXPERIENCES WITH PUMPING RING TECHNOLOGY

A well-designed barrier system can have a total head of less than two meters (~78”), however in reality the optimum positioning of coolers or vessels is not possible due to plant layout. Under such circumstances the pumping device must be able to provide sufficient head to overcome the resistance in the system. This segment of our presentation focuses on plant experiences. The experimental part shows how the actual flow being generated can also be maximised though careful design of the pumping ring. In this section the work is combined so that enhanced circulation and increased head have been used to provide barrier fluid to difficult applications that had previously had poor MTBF.

The first application is a continuous running chemical slurry process at a chemical manufacturing plant in South Wales, England. The slurry temperature was up to 200^{O}C (392 degs F) with suction and discharge pressures of 0.5barg (7psi) and 2.5barg (36 psi) respectively with a rotational speed of 1500rpm. The application was upgraded to a dual pusher type seal that had SiC/SiC faces with Kalrez O rings inboard and SiC/Carbon faces with Viton O rings outboard. The seal was set up with a Plan 53 arrangement without additional cooling that had dimethyl silicon oil as the barrier fluid set to a pressure of 4barg. The seals had originally been failing due to severe degradation of both sets of faces with less than 6 months running time. The upgraded seal arrangement was removed after 18 months and found to have faces in near perfect condition. This case history demonstrates how improving the flow rates and circulation characteristics of the barrier fluid significantly increase the MTBF.

The second case history installation was on a North Sea Production Platform (Tern Alpha) where the seal was fitted on the closed drains pumps. The layout of the plant equipment was such that the barrier fluid vessel could not be fitted anywhere near to the pump (see Fig. 17).

Fig. 17 - View of the application showing the pump bottom left and the reservoir top right.

This compares with the API-682 extract that gives guidelines for the routing of pipework and the vessel position in reference to the pump (see Fig. 18). Since 2002, several 85 mm API A1 dual cartridge seals with API-682 systems have been successfully running on Warman 3/2 CCHH pumps, pumping produced water with methanol, oil and heavy sand content. The seals are operating in an API Plan 53 system with a seal chamber pressure at 13barg (189psi) and a shaft operating at 2100 rpm. The pipework connecting the seal and system measured over 3 metres (10 feet) horizontally and two meters (~6.5 feet) vertically, with pipe bends in several different directions. This is contrary to ‘best practice’ guidelines. This is one application where having additional head is required to overcome the ‘higher than normal’ losses in the system. Note that this is a common occurrence on upgrades to old equipment where double seals replace single seals due to legislation and environmental issues.

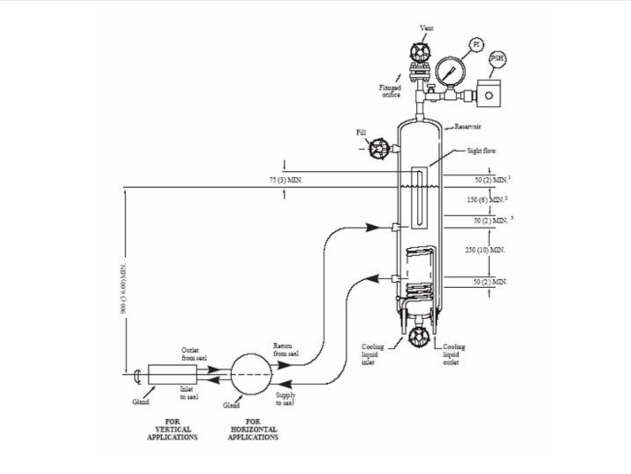


Fig.18 - Barrier fluid vessel layout as suggested by API-682.

The third application is an On-Shore Oil exploration installation in Alaska, USA where a hybrid Plan 53 and Plan 54 system has been installed for more than 3 years, shown in Fig 19.

The innovative Hybrid system embraces the high efficiency of the taper vane bi-directional pumping ring (Plan 53A and 53B) and couples this with a supporting Plan 54 system which generates a localized pressurized barrier fluid circuit.

Fig.19 – Hybrid Plan 53 and 54 system installation in Alaska, USA.

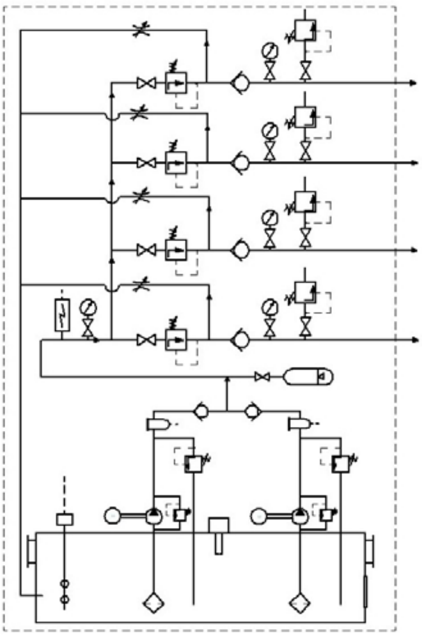


Fig.20 – Hybrid Plan 54/53 system Schematic installation in Alaska, USA.

The single Plan 54 system feeds barrier fluid to a number of Plan 53 seal support systems, which control the environment at the mechanical seal, by the efficient and effective use of taper vane pumping rings.

Localized seal control is enhanced through the ability to supply each seal support system with different pressures due to individual pressure regulators. The hybrid system does not run continuously, only activating when the pressure transmitter detects a drop in circuit pressure *(the circuit contains an accumulator to maintain pressure)*.

SUMMARY

The work highlighted in this presentation shows the differences in dual seal pumping ring performance. These devices are used for the purpose of circulating barrier/buffer fluid.

Ideally, any such pumping ring device should have equal pumping performance in either direction, since this would standardize the design. Pumping rings that are bi-directional, offer the user the best degree failure risk reduction and consistent turnkey performance.

The test results in this presentation show that in identical tests the tapered vane pumping device outperformed the other two devices by a considerable margin. Furthermore, the swan-neck tapered vane pumping ring outperformed the standard taper vane by around 25% in flow and 50% in head developed. This has significant value at smaller shaft diameters and/or slower rotational shaft speeds.

This presentation then also demonstrates that excellent barrier/buffer pumping/flow performance could be achieved while still maintaining sufficient radial clearance. This higher radial clearance is needed if compliance with the latest edition of API-682 is desired.

The presentation reiterates the benefits of efficient pumping rings in Plan 53 and 23 environments and highlights methods to reduce the manufacturing unit cost to make them more readily affordable and available for industrial application deployment.

The three case histories given in our presentation demonstrate how it is possible to increase the MTBF of seals on the most demanding duties and still adopt the ‘best practices’ guidelines recommended by the experienced members of the API-682 sealing committee.

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