

THE DEVELOPMENT OF LOW FRICTION, LOW LEAKAGE MECHANICAL SEALS USING LASER TECHNOLOGY

by

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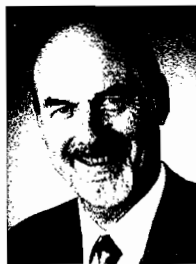


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Mr. Wallace earned his B.Sc. degree at Manchester University (1965), and worked with Renold Limited until he joined Flexibox (1974). He has extensive experience in the field of mechanical seals and power transmission couplings and has presented several

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Heinz Konrad Müller received a Mechanical Engineering Degree (Dipl.-Ing.) from Technische Hochschule in Stuttgart, Germany (1957) and his Dr. Ing in 1962. From 1957 to 1960, he was involved in R&D work in industry on automotive and aircraft control equipment. From 1960 to 1972, Dr. Müller was an Assistant Professor and Lecturer in basic machine design in Mechanical Engineering. He lectured on fluid sealing, and conducted research on seals

and bearings.

Since 1973, Dr. Müller has been a professor of Mechanical Engineering at Universität Stuttgart and Head of the Fluid Sealing Department of the Institute für Maschinenelemente. He has been conducting research on hydraulic seals, mechanical seals, viscosity screw seals, and noncontacting spindle seals, and has written over 70 technical publications and conference papers.

appreciably, however, particularly when the operating conditions, pressure, temperature and speed, vary with time. The reverse flow laser machined seal offers a way to bring mechanical seals closer to the ideal.

INTRODUCTION

There is an increasing demand for reliable low leakage mechanical seals to meet evolving market requirements such as emission legislation.

There is, additionally, a requirement for minimal heat generation, to ensure good seal operation, and to reduce the size of associated equipment in the seal oil systems of double and tandem seals which are being increasingly used to meet environmental and safety requirements.

The above requirements are difficult to achieve simultaneously using currently available technology.

The concept, evaluation and assessment are described of areas for application of a new departure in face technology that is aimed at getting closer to ideal performance.

The concept involves the use of two sets of laser etched patterns in the seal face. One set is designed to introduce fluid between the faces to create stable and repeatable low friction levels and the other set to pump-back fluid from the fluid film to the sealed fluid to ensure low leakage levels.

The concept originated at Stuttgart University and was evaluated there in real seals on test rigs at pressures up to 10 bar. The concept is now being evaluated and developed for commercial use.

INITIAL EVALUATION AT STUTTGART UNIVERSITY

The Laser-Face Concept

The behavior of standard mechanical seals

Standard mechanical seals are passive systems. In operation the lapped radial faces form a very small gap, one micrometer or less. This gap constitutes a high flow resistance against the radial leakage flow which is caused by the pressure differential across the faces. Experience shows that, under favorable conditions, leakage rates are not lower than some tenths of one milliliter per hour, but normally leakage is more likely to be a few milliliters per hour. Higher leakrates occur when the seal rings deform under pressure and the action of temperature gradients in the components. Normally one of the seal rings is made of carbon graphite and can, therefore, compensate for such deformations by wearing. Sometimes, it is desirable to have both rings made of the extremely hard

ABSTRACT

An ideal mechanical seal would operate reliably, for long periods, without wear, with permanently low friction and extremely low leakage. Real mechanical seals often do wear and leak

and highly wear resistant silicon carbide (SiC). The problem with SiC/SiC mechanical seals is, however, that due to high local contact pressure induced by deformation or temporary lack of lubrication, even such extremely hard seal faces may wear and form local asperities or become damaged. This, in fact, may decrease friction, but at the expense of a drastic increase in leakage.

Stabilizing the seal gap

Defining consistency and stability of operation at minimum wear rates as the overriding goal, the seal gap and hence the leakage will be at the minimum if the 'high spots' of the faces do not create appreciable thermal distortion of the rings when they come into contact. In other words, most of the axial load of the seal ring should be born by the interface fluid pressure such that the seal works in a region between mixed lubrication and full film lubrication. In practice, under varying operating parameters, it is difficult to achieve and maintain such conditions. Because the seal user often is primarily interested in zero leakage, the seal supplier opts for a higher gap-closing force, which means increasing the solid contact pressure of the seal faces. But running a mechanical seal at high speed in the boundary lubrication regime almost invariably leads to unreliable operation.

The radial seal faces operate without contact and without wear, if the axial load of the sliding ring is totally compensated by the interface fluid pressure. Many different ways are known to create stable conditions of this kind. Basically, the faces are made or allowed to create circumferential and radial variations of the clearance, for example tangential waviness and radial taper, whereby the hydrodynamic and the hydrostatic fluid pressure in the clearance is arranged to balance the axial gap-closing force. However, such measures inevitably, and sometimes unpredictably, increase the mean thickness of the sealing gap and, hence, increase the leakage of the seal.

Returning liquid from the seal gap

The wish to preserve the advantages of full film lubrication on the one hand but, at the same time, avoid high leakage rates, leads to the idea of creating an interface pumping mechanism which returns fluid from the sealing gap to the pressurized space. Four different sealing concepts designed to create 'return-pumping' are shown in Figure 1. In an early approach [1], a number of angled grooves, much deeper than the thickness of the sealing gap, were installed to improve hydrodynamic lubrication and to create upstream pumping in the sealing gap. Other return-flow mechanisms have been created in a number of complicated seal structures [2, 3, 4]. These were mainly focused on relatively large shaft diameters in high-pressure, high-speed applications. All the systems are based on shear-flow pumping of the sliding face that draws fluid through the gap towards the space to be sealed. Most of the state of the art return seals appear to have geometrically complex, wide-faced seal rings with correspondingly expensive manufacturing processes and return pumping is restricted to only one direction of rotation. A further disadvantage of known return seals is that the pumping grooves are open towards the low pressure space favoring liquid to escape at standstill and eventually allowing foreign liquid to be pumped upstream from the low pressure space to the sealed space.

The reverse flow laser machined seal

To be successful, a return flow-lubricated seal must:

- Comply with the face geometries of modern, narrow faced mechanical seals.
- Separate the faces by approximately one micrometer.
- Be effective for clockwise and anticlockwise rotation.

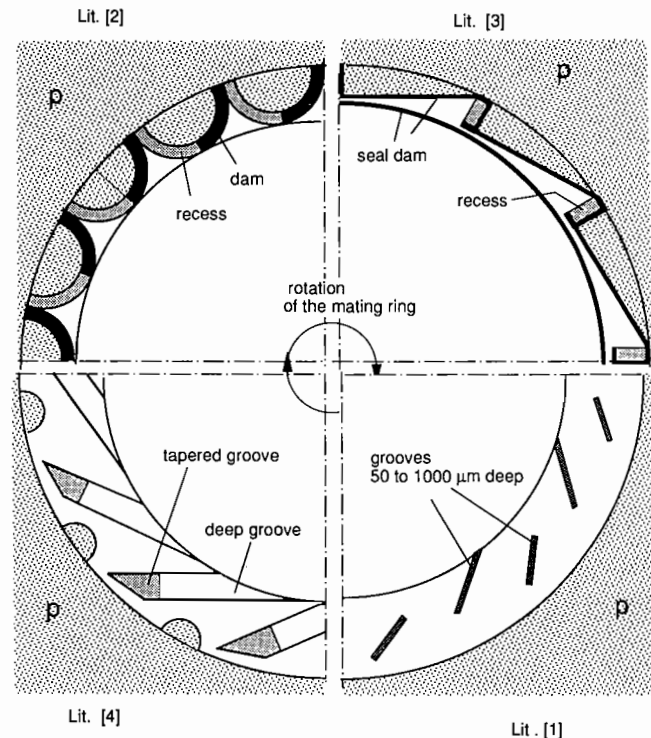


Figure 1. Various Concepts of Mechanical Seal Faces Creating Return-Flow.

- Be manufacturable on a micrometer scale at a reasonable cost.

The proposed solution is the reverse flow laser machined seal. By means of a laser beam, a new type of return flow recess is engraved in the seal faces. The structures are manufactured on the prelapped faces of conventional ceramic sealing rings, preferably silicon carbide. The main features of the structures are:

- the depth of the recesses is between 0.2 and 10 micrometers. (8 and 400 μin)
- the recesses are totally and permanently covered by the mating face.
- the rotating ring draws fluid in and hydrodynamically increases the pressure inside the recesses.
- the recesses are formed to guide the fluid positively towards a discharge region that is placed very close to the sealed space, and where the hydrodynamic pressure inside the recess is at maximum.
- the return-flow is effective regardless of the direction of rotation when the recesses are made to mirror-image with reference to a radius.

The function of a reverse flow laser machined seal is shown in Figure 2. The recesses are symmetrical and, therefore, work regardless of the direction of rotation. The reverse flow (RF) structures collect liquid which is entering the gap via the high pressure edge and via a number of additional entry-flow (EF) structures that are connected to the high pressure space and create hydrodynamic lift of the seal faces, thus stabilizing the sealing gap. Inside the reverse flow (RF)-recess, the sliding ring draws the fluid tangentially and, guides it along the trailing edge of the recess. The fluid finally reaches the discharge-end, which is very close to the high pressure edge of the sealing gap. Here the fluid pressure inside the RF-recess is at maximum that can be much higher than the fluid pressure in the sealed space. Following the path of least flow resistance, a great deal of the liquid leaves the recess towards

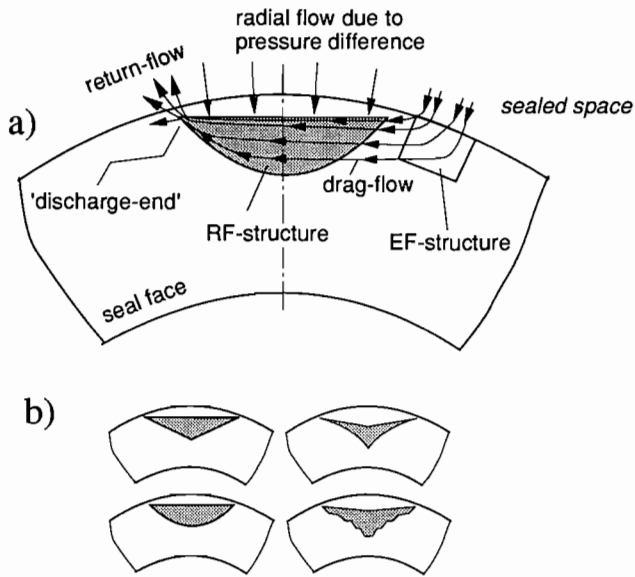


Figure 2. RF-Seal Concepts. a) Recesses and Schematic Flow Field and b) Various Shapes of RF-Structures.

the sealed space. From the great variety of possibilities geometries, four different configurations of RF-recesses are shown in Figure 3.

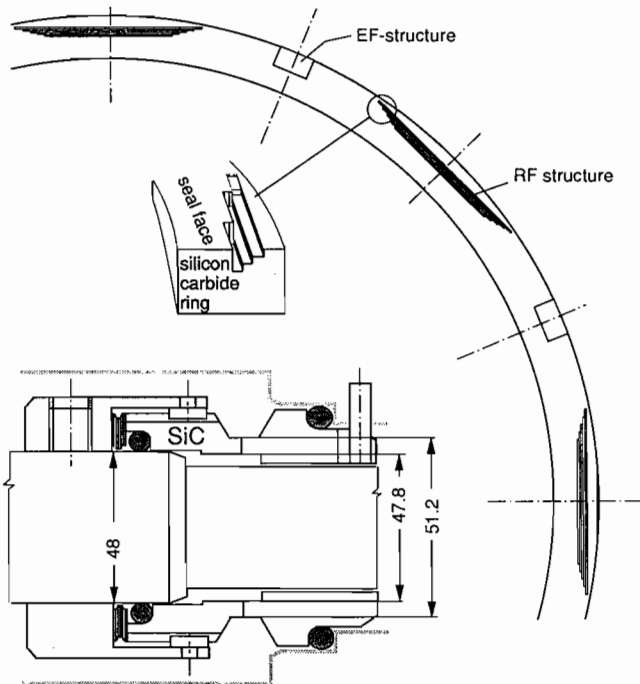


Figure 3. RF-Test-Seal.

The EF-recesses may be placed on the same face as the RF-recesses or, in the case of two hard faces, may also be placed on the mating face. In the latter case, there is a periodic overlapping during which the EF-structures do feed the RF-structures.

While the basic functioning of the structures is easily understood in principle, because of the complex geometry and the fact

that parts of the recesses are cavitated, a theoretical analysis of the interface flow turns out to be rather complex.

A promising way to solve the problem analytically is to apply the finite-element-program FIDAP and such analysis is under way.

Initial Laboratory Evaluations

Preliminary experiments

The basic functioning of reverse flow laser machined seals was tested in the laboratories of the Fluid Sealing Department of the Institut für Maschinenelemente, Universität Stuttgart [7. 8]. The typical shape is shown in Figure 4 of the RF-structures of the test seals which was chosen. The recesses were engraved with an EXCIMER-Laser by the Institut für Strahlwerkzeuge (IFSW), Universität Stuttgart. Specially adapted jigs and software were used to position the SiC-ring on the laser equipment. The actual RF-recess consists of a number of grooves, radially 0.1mm (0.004 in) wide. The longest groove is adjacent to the sealed space and is 1.0 micrometer (40 µin) deep, each following groove is deeper by 1.0 micrometer. The geometry of the test seal is also shown in Figure 4. All tests were run with tap water as the sealed fluid at a pressure difference between 0.05 MPa and 0.5 MPa (7.25 and 72.5 psi). The actual test parameters are shown in the following figures.

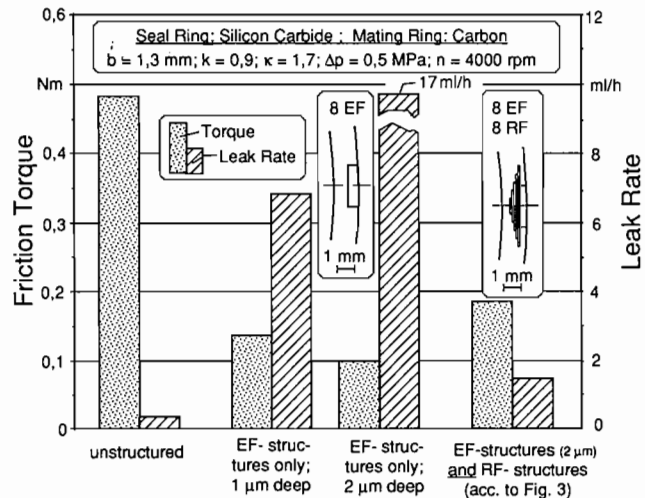


Figure 4. Friction Torque and Leakage Rate of Experimental Seals with Unstructured SiC-Face Compared to EF and RF/EF Structured SiC Face.

Conventional mechanical seal and reverse flow laser machined seals

To assess the behavior of reverse flow laser machined seals under equal conditions in a number of tests, the seal was first run with a plain, unstructured silicon carbide ring. A typical running-in behavior is characterized by with stochastic fluctuations of the gradually decreasing friction torque, accompanied by a decrease in the leakage rate. After an initially high leakage (in the order of 10 ml/hr (0.6 cu in/h)), the leakage rate settled below 1.0 ml/h after a few hours. Thereafter, the friction torque was approximately 0.5 Nm (4.4 lb in) and the friction factor $f = 0.12$. A continuous stochastic variation of the friction torque indicates that the seal was in the mixed or boundary lubrication regime and wear occurred during all operation.

As a further basis for comparison, the seals were tested with EF-structures only. EF-structures are well known measures to relieve

the seal faces from mechanical contact by creating hydrodynamic lift.

However, while conventional structures are often deep and based on thermo-hydrodynamic effects inside the sealing gap beyond the edges of relatively deep grooves, the EF-structures used here are just 1.0 to 2.0 micrometers (40 to 80 μm) deep and, therefore, the pressure generation is inside the EF-structure itself.

The frictional torque and leakage of seals are compared in Figure 5 with different configurations. In these tests the radial width of the sealing dam was $b = 1.3 \text{ mm}$ (0.05 in), the balance ratio $k = 0.9$, the spring pressure $p_f = 0.4 \text{ MPa}$ (58 psi), the water pressure $p = 0.5 \text{ MPa}$ (72.5 psi) and the speed $n = 4000 \text{ rpm}$. After running in, the unstructured seal exhibited boundary lubrication behavior with a relatively high friction torque and low leakage. The seals with EF-structures only were clearly full film lubricated with low friction and high leakage. The seal with the 2.0 micrometer (80 μm) EF-structures leaked as much as 17 ml/h (1.0 cu in/h). Most interesting is the behavior of the seal combining RF-structures with the same 2.0 micrometer EF-structures. A constantly low friction torque indicated that the seal was working with full film lubrication, but the leakage had simultaneously reduced from 17 ml/h down to 1.5 ml/h (1.0 down to 0.1 cu in/h). In later tests, it was shown that, by further increasing the axial load on the seal ring (by increasing the spring load) as high as 1.0 MPa (145 psi), the seal, nevertheless, remains in full film lubrication and the leakage is further reduced.

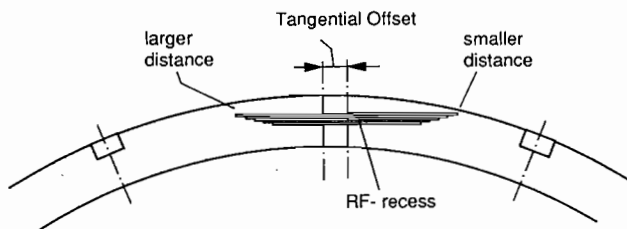


Figure 5. RF-Experimental Seal with Recesses Tangentially Offset.

In order to demonstrate the effect of the distance between the discharge-end of the RF-recess and the high pressure edge of the sealing gap, a seal ring having a radial width of $b = 2.2 \text{ mm}$ (0.09 in) was equipped with RF-structures tangentially offset as shown in Figure 6. Here the ends on one side were at a distance of between 0.1 and 0.2 mm (0.004/0.008 in) from the sealing edge and approximately 0.5 mm (0.020 in) at the other side. The balance ratio was $k = 1$, the spring pressure $p_f = 0.25 \text{ MPa}$ (36 psi), the water pressure $p = 0.5 \text{ MPa}$ (72.5 psi) and the speed $n = 4000 \text{ rpm}$. The friction torques and the leakrates are shown in Figure 6 at clockwise (A) and counterclockwise (B) rotation of a seal. With clockwise rotation, where the discharge-ends of the RF structures is very close to the high pressure edge, the leakage for approximately 70 hr was as low as 0.0016 ml/h (10E-4 cu in/h). During that period, the friction torque was free of fluctuations and climbed slowly from 0.35 up to 0.4 Nm (3.1 up to 3.5 lb/in), until a sudden friction peak occurred. Then the friction suddenly decreased to 0.3 Nm (2.6 lb/in) and then remained smooth and stable while the leakage rate settled at 0.15 ml/h (0.01 cu in/h). After 100 hr, rotation was reversed to counterclockwise. From then on, the leakage rate was approximately 1.0 ml/h (0.06 cu in/h) but the friction torque, again smooth and stable, and almost the same level as on clockwise rotation. The behavior was reproduced with subsequent changes of the direction of rotation, but on clockwise (A) rotation the seal later ran approximately at the conditions after the friction peak.

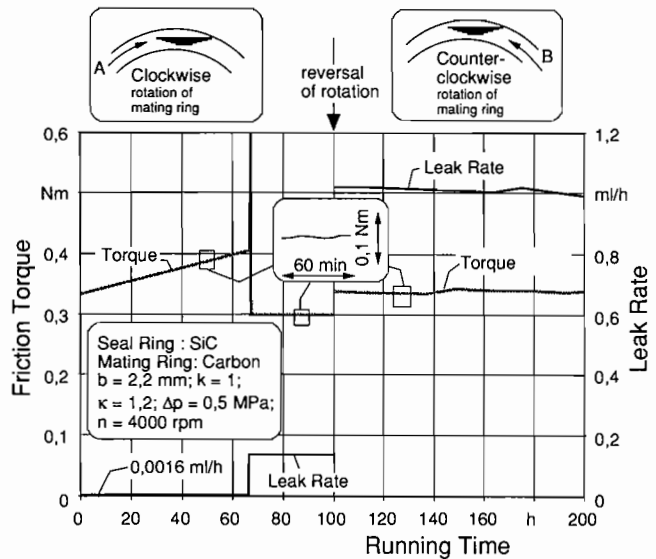


Figure 6. Friction Torque and Leakage Rate of RF-Experimental Seal with Recesses Tangentially Offset as a Function of Running Time at Clockwise and Counterclockwise Rotation.

Conclusion

No detrimental effects could be observed on water-sealing reverse flow laser machined seals running for more than 1000 hr at varying conditions of pressure (0.05 ... 1.0 MPa (7.25 ... 145 psi)) and speed (up to 5000 rpm). An overview is given in Table 1 of the parameters tested so far with seals with dimensions $b = 1.8 \text{ mm}$ (0.07 in) and $K = 0.95$. The load factor K means the sum of the balance factor k plus the ratio of spring pressure to sealed pressure difference ($K = k + (p_f/p)$). Under all conditions, the friction factor of reverse flow laser machined seals was appreciably lower than of the conventional seal. In order to achieve further leakage reductions, it should be emphasized that increasing the axial load on the seal ring of a reverse flow laser machined seal is an effective method to control leakage to very low levels without sacrificing the regime of stable lubrication. A positive side-effect

Table 1. Tribological Parameters for an RF-Test Seal at Various Operating Conditions.

speed rpm	pressure p MPa	spring- pressure p_f MPa	load factor κ -	Friction Torque Nm	Friction Factor f -	G- Number $\cdot 10^7$ -	Leak Rate ml/h
Face width $b = 1,8 \text{ mm}$; Balance ratio $k = 0,95$							
8 EF ($1 \mu\text{m}$ deep), 8 RF							
4000	0,5	0,27	1,5	0,28	0,051	2,8	0,25
4000	0,5	0,54	2,0	0,28	0,038	2,1	0,2
4000	0,5	0,81	2,5	0,3	0,03	1,7	0,18
4000	0,5	1,08	3,1	0,43	0,038	1,4	0,12
4000	0,2	1,08	6,4	0,31	0,034	1,7	0,41
4000	0,8	1,08	2,3	0,41	0,031	1,1	0,18
3000	0,5	1,08	3,2	0,37	0,037	1,0	0,13
3000	0,1	1,08	11,8	0,2	0,027	1,3	0,03
3000	0,05	1,08	22,6	0,22	0,03	1,4	< 0,01
Conventional unstructured seal							
4000	0,5	0,27	1,5	0,53	0,117	2,8	0,25

of the higher load-bearing-capacity of reverse flow laser machined seals is that the spring force can be increased to a level which, under all circumstances, will avoid an O-ring induced axial hang-up of the seal ring. This would sensibly enhance the behavior of pusher seals while retaining all of their advantages.

Summary conclusions from the work at Stuttgart University

The reverse flow laser machined seal is characterized by low leakage rates and full film lubrication with simultaneous low friction factors. The RF concept is applicable to the geometrical proportions of proven conventional mechanical seals. Preliminary tests with carbon graphite/silicon carbide reverse flow laser machined seals convincingly confirmed the lubricating and return-flow mechanisms. Under laboratory conditions, the reverse flow laser machined seals proved to run stably at varying conditions of pressure and speed. The friction factor was found predominantly in the range $f = 0.03$ to 0.05 , but in any case below 0.07 . The results suggest that there should be significant benefit in the application of the RF concept to SiC/SiC mechanical seals.

FURTHER DEVELOPMENT FOR USE IN SEALS IN THE PROCESS INDUSTRIES

Appraisal of the Principles for Industrial Use

The above test work suggested that this new concept offered a number of distinct potential advantages.

These are evaluated here in the context of commercial seals.

- Low leakage levels
 - Are desirable in most applications.
 - Low and *reliable* leakages are particularly desirable in seals for volatile organic compounds (VOCs) to meet legislation.
 - They are very important in double and tandem seal applications to reduce top-up requirements and reservoir capacities.
- Low friction levels
 - Lead to good seal operation through lower face temperatures and extended operating margins.
 - Are important in seals where the temperatures can cause the breakdown of the fluid film including light hydrocarbon seals, hot water seals and high pressure seals on oil where the oil can degrade at high temperature.
 - Reduced friction levels in double and tandem seals mean reduced cooling requirements.
- Repeatable performance
 - Is important for predictable operation.
- Stable operation with good margins to failure.
 - Gives tolerance to varying operating conditions.
 - Gives tolerance to upset conditions which frequently occur in real installations.
 - Is important for long times between failures.
- Applicability to existing seal designs and existing installations
 - The principle must fit in with current seal design practice.
 - Allows existing seals to be easily converted offering full benefit at minimal cost.
- Use in hard faced seals
 - Extends the operating envelope of hard faced seals by increasing their tolerance and avoiding premature failure. Example : SiC/SiC seals on HF alkylation service sealing alkylate at 30/40 bar (435/580 psi).

• Limitations

The testing described was exclusively on tap water at pressures up to 10 bar (145 psi). While the potential for use with other fluids such as oil and clean hydrocarbons seems good, there are many questions to resolve by further evaluation.

The principle limitations currently envisaged are with abrasive fluids or fluids, which could potentially foul the surface depressions due to coking, crystallization, or deposition in that area.

Laboratory Tests in Process Industry Seals

General Purpose Cartridge Seal

The first brief tests conducted in the company laboratories were on a general purpose cartridge seal, as illustrated in Figure 7. The objective was to check that the principle was effective at pressures up to 20 bar (290 psi) on water; previous testing having been limited to 10 bar (145 psi).

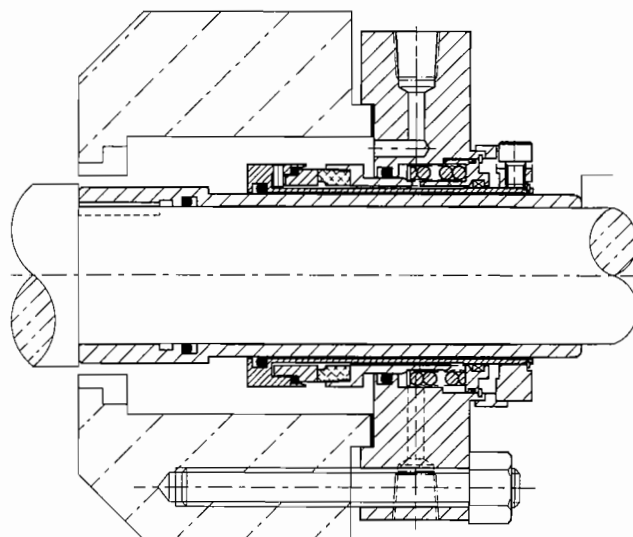


Figure 7. General Purpose Cartridge Seal.

Seal Details

Seal size 55 mm (2.165 in)
Shaft speed 3000 rpm
Stationary face : carbon
Rotating face : SiC, laser etched

test duration hrs.	Sealed pressure bar/psi	Relative friction factor
22	4.5 / 65.25	1.00
25	10 / 145	0.57
25	15 / 217.5	0.37
30	20 / 290	0.34

External liquid leakage over the 72 hour test was zero (vapor leakages were not measured) and, although accurate friction factors were not established, it was possible to look at the variation of friction factor with pressure by comparing the temperature rise of the cooling flow across the seal at each pressure.

This result is plotted in Figure 8 and illustrates how the friction factor varies with sealed pressure in a way which is characteristic of a high performance seal (Figure 9) where the actual friction

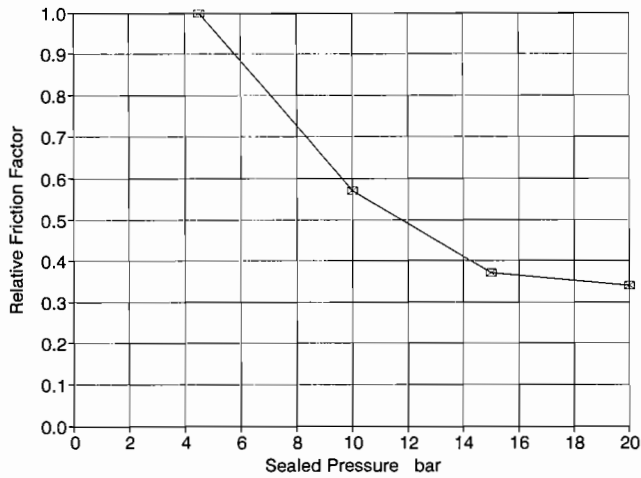


Figure 8. Relative Friction Factor Vs Pressure. General purpose cartridge seal with laser etched faces.

torque does not vary significantly with pressure and, hence, the friction factor decreases with pressure.

The seal was subsequently run in parallel with a standard seal on BP Energol HLP 10 oil (an ISO VG 10 oil with additives) at pressures between 2.0 and 20 bar (29/290 psi). The laser etched seal ran with no leakage and substantially lower friction than the standard seal. For that test, the laser etched seal was otherwise standard. Future testing will include seals with much narrower faces, higher spring loads and lower degrees of balance to increase the specific closing forces in line with the experiences at Stuttgart University.

Formed Bellows Seal

The formed bellows seal shown in Figure 10 is a general purpose bellows seal for dynamic operation to 30 bar (450 psi).

were tested on mineral oil between 3 and 30 bar (43.5 and 435 psi), monitoring heat generated, face temperature, and leakage.

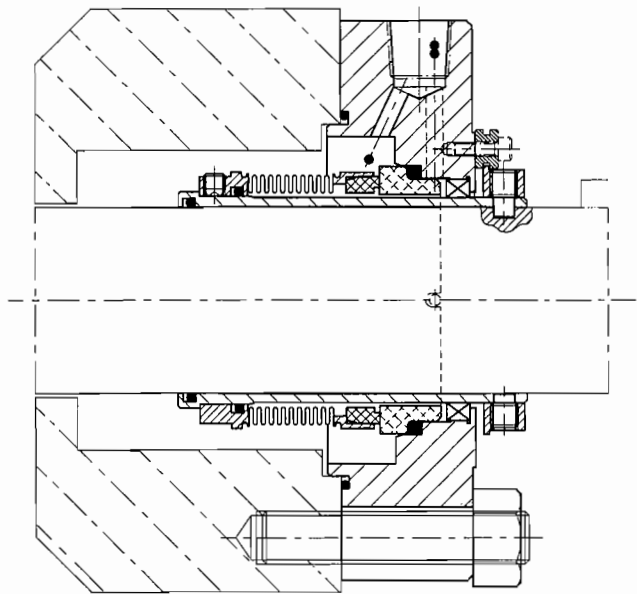


Figure 10. Formed General Purpose Bellows Seal.

Seal Details

Seal size 76.2 mm (3.00")	Oil type : BP Energol HLP 10
Face dimensions etc.:-	ISO VG 10
Outside diameter : 92 mm (3.62")	Circuit temperature : 20/40 C
Inside diameter : 84 mm (3.31")	Shaft speed 3000 rev/min
Balance diameter : 87 mm (3.42")	
Bellows spring load : 243 N (55 lbf)	
Balance ratio : 0.64	
Spring specific face pressure 0.22 mPa (32 psi)	

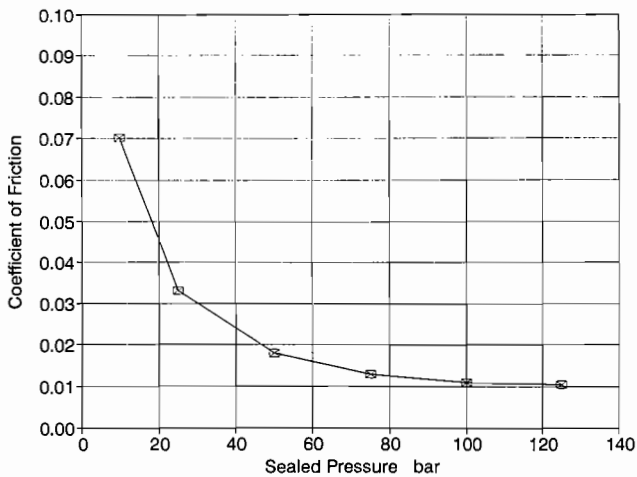


Figure 9. Friction Factor Vs Pressure/High Performance Seals on Mineral Oil.

A test was performed on a double ended test rig with a standard seal at one end and a Reverse flow laser machined seal at the other. Apart from the laser etching, the seals were identical. The seals

The measured values of heat generated and the derived coefficients of friction are plotted on Figures 11 and 12.

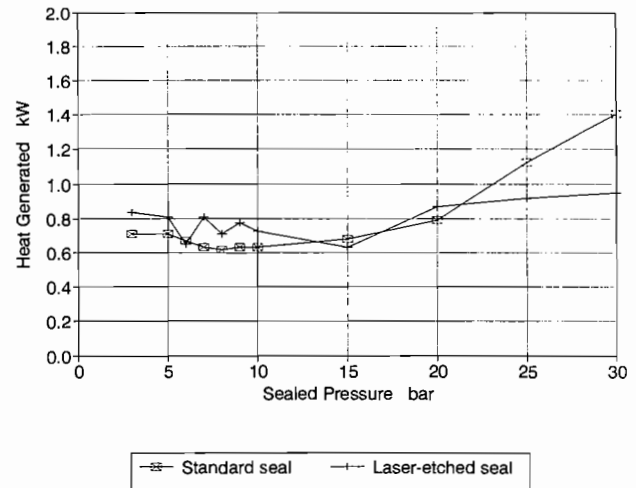


Figure 11. Bellows Seal Heat Generated Vs Pressure for Standard and RF faces.

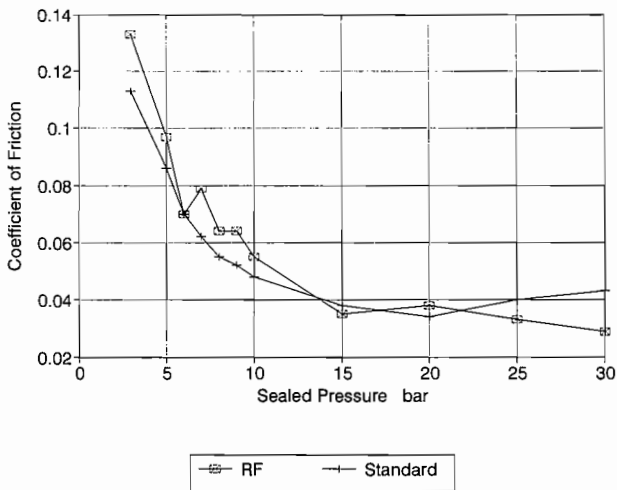


Figure 12. Bellows Seal: Coefficient of friction vs pressure for standard and RF faces.

The values are similar up to 20 bar (290 psi) when the reverse flow laser machined seal shows a continuously reducing value and that of the standard seal starts to increase. Again, the friction factor varies like the high performance seal with pressure (Figure 9) with an almost identical value of 0.25 @ 30 bar. The rising value of friction coefficient in the standard seal is indicative of a move toward increased asperity contact as the closing pressure increases and is not fully supported hydraulically.

A welcome result was that the face temperature rise of the RF laser machined seal (Figure 13) was consistently lower than the standard seal at 75.5°C (136°F) compared to 114.5°C (206°F) at 30 bar. This behavior has been observed in a number of other tests and may be a result of increased heat transfer from the faces due to the flow through the RF structures.

No leakage was observed during the tests until 25 bar, when both seals started to leak at about 5.0 ml/h, which continued to 30 bar. This level of leakage is not abnormal with seals on mineral oil.

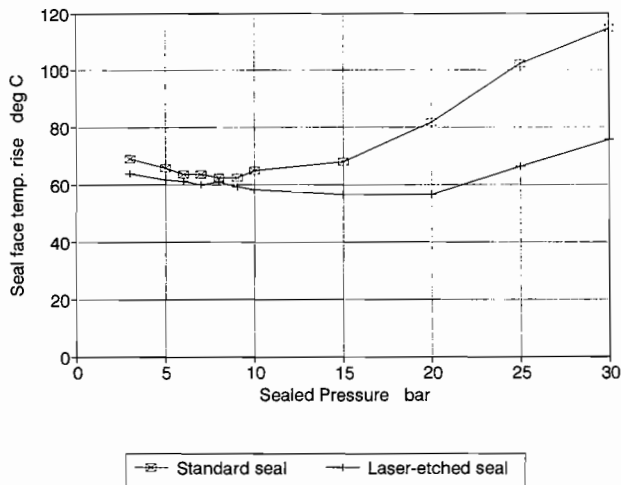


Figure 13. Bellows Seal: Seal face temperature rise vs pressure for standard and RF faces.

It should be emphasized, however, that the seals tested were again to normal geometries with normal balance ratios, face widths and spring loads. Further tests are in hand with the same seals, with

narrower faces and lower degrees of balance, where the authors hope to increase the tightness of the seal without compromising its low friction properties.

The test indicated that the RF structures had extended the low friction properties of the seal to the full pressure limit.

The reduction in face temperature was a very welcome benefit.

Double Seals

Double seals are being increasingly specified for safety and environmental reasons. The RF laser machined seal is seen as being important for double seals as indicated under *Appraisal of the Principles for Industrial Use*.

Tests are currently in hand on high performance double seals at speeds to 3,600 rpm and pressures to 125 bar (1800 psi).

The friction factors in Figure 9 are typical of the levels achieved with this type of seal.

High Duty Water Seals

The seal illustrated in Figure 14 was specially developed for service in boiler circulator pumps where the duty conditions are arduous and include:

- High sealed pressures—over 100 bar, (1450 psi).
- High pumped temperatures—of the order 300°C (572°F).
- The presence of abrasives including magnetite (magnetic iron oxide).

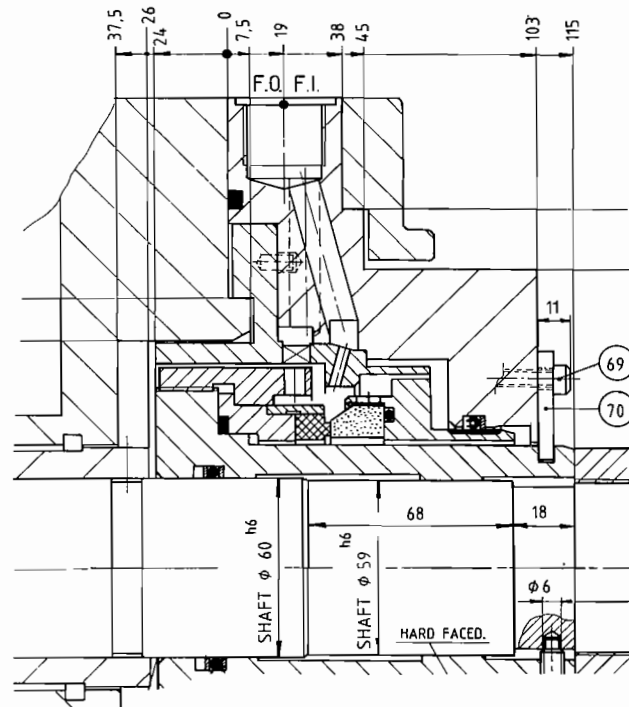


Figure 14. Boiler Circulator Seal.

The use of mechanical seals in this application is not common although we have good experience over several years.

Early seals featured carbon/silicon carbide faces but, recently, we have been using hard face combinations with either silicon carbide or tungsten carbide as the stationary seal ring and a silicon carbide/graphite composite material in the rotary.

The seal is fitted with an integral pumping ring that drives the sealed liquid through a cooler in a closed loop (API Plan 23).

With this arrangement, the prevailing water temperature in the region of the seal is normally 80°C (176°F) or less. A sister company that had developed the boiler circulator pump seal decided to evaluate the new principle in the context of this seal and were looking for reductions in seal face friction and increased tolerance to variations in operating conditions with a view to longer life.

A number of short term tests were conducted at pressures up to 160 bar (2320) psi.

Seal size 80 mm (3.15")

Face dimensions etc

Outside diameter	- 96.1 mm (3.783")
Inside diameter	- 90.5 mm (3.563")
Balance diameter	- 91.95 mm (3.620")
Spring load	- 325 newtons (73Lbf)
Spring specific face pressure	0.356 mPa (51.6 psi)

Operating conditions

Shaft speed	- 2950 rev/min
Seal fluid	- town's water
Fluid temperature	- of the order of 25°C (77°F)
Test pressures	- 115,150,160 bar (1667, 2175, 2320 psi)
Circulator flow rate	- 500/650 l/h (132/172) US gal/hr

Face materials

	Stationary seal ring	Rotary seal ring
SEAL A	SiC reaction bonded + RF structures	SiC/graphite composite plain face
SEAL B	Tungsten carbide + RF structures	SiC/graphite composite plain face
SEALS C/D	Tungsten carbide plain face	SiC/graphite composite plain face

A number of short runs were conducted and measurements were made of heat generated and leakage rate. The results are listed in Table 2.

Discussion of results

Test 1 is directly comparable with Tests 2.1 and 2.2 where the sealed pressure and lapping standard were identical. By averaging the results, the following comparison could be made:

AVERAGE	PLAIN FACES	WITH RF STRUCTURES
Friction factor AT 150 bar	0.018	0.0083
6 light bands		
Convexity		

In view of the above, it was surprising to find that, after the tests, the plain face seals were in absolutely excellent condition while the seals with RF Structures had changed during the tests.

The tungsten carbide seals had fine thermal cracks around the plain area inside the area containing the structures while the silicon carbide seal faces had become rough in that area. In both cases, the area containing the structures was absolutely excellent (and in all cases, the condition of the composite rotary seal ring was excellent).

The structures had been added to standard seals faces that left a significant plain area inboard of the structures.

The conclusion from this series of tests is that the RF structures have shown great promise in halving the face friction levels although further evaluation with alternative face configurations is now required to eliminate the face marking observed.

Plans for Further Evaluation

In the immediate future, testing will concentrate on seals for VOCs and double and tandem seals. The objective is to map performance across a range of seals sizes, at pressures between zero and 150 bar, speeds between 1500 and 6000 rpm, and using a range of fluids including light hydrocarbons, water, and oils.

A theoretical model for the principle is currently being developed and will be used in the future for engineering the optimum configuration for general use and also for specific high duty applications.

From the work done so far, however, it is clear that the configuration can be tuned to give a desired compromise between

Table 2. Test Results, Boiler Circulator Pump Seal.

Test Results

TEST NO.	TEST PRESSURE	SEAL	MUTUAL FACE FLATNESS (Lightbands)	STATIONARY SEAL RING MATERIAL	RF STRUCTURES	TEST DURATION (Hours)	HEAT GENERATED (Watts)	FRICTION FACTOR	LEAKAGE RATE (ml/h)
1	150	C D	6 cx	WC WC	N N	2	2186 2751	0.016 0.020	12 12
2.1	150	A B	6 cx	SiC WC	Y Y	2	1151 983	0.0084 0.0072	11 18
2.2	150	A B	6 cx	SiC WC	Y Y	2	1041 1355	0.0076 0.0099	18 2.5
2.3	115	A B	6 cx	SiC WC	Y Y	3.5	907 1000	0.0085 0.0094	25 4.3
2.4	160	A B	6 cx	SiC WC	Y Y	2	1035 930	0.0071 0.0064	2.5 17.5
2.5	150	A B	FLAT	SiC WC	Y Y	5	1320 1610	0.0096 0.012	6 16

NOTE: cx - convex

leakage and friction which offers a lot of potential for improved performance in given applications. (This is illustrated in Figure 15 which uses data listed in Table 1. By varying the spring pressure Pf, for example, the compromise between leakage and friction torque may be changed.)

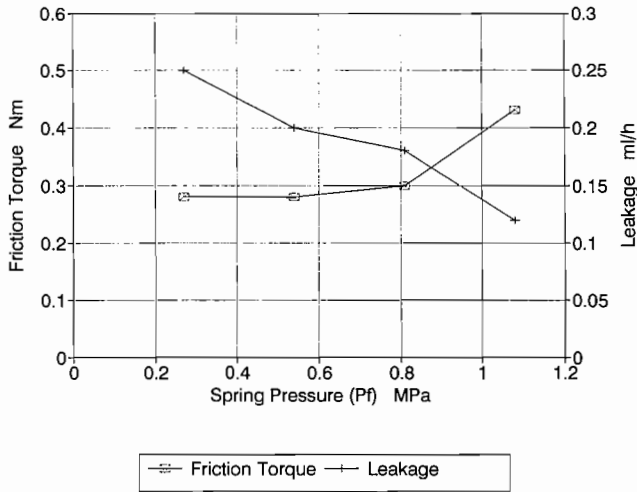


Figure 15. RF Laser Seal: Friction torque and leakage vs spring pressure.

APPLICABLE EXPERIENCES

Field tests are a vital part of the development process. The authors are planning to fit reverse flow laser machined seals into a number of applications including light hydrocarbon service, double seals, general process, and hot water applications. Initial results will be reported when the paper is presented.

CONCLUSIONS

As a result of the initial evaluations, there appear to be substantial potential benefits for the mechanical seal user. Those include

reduced leakage and reduced face friction, together with improved tolerance and reliability.

The increased heat transfer from the face observed with the reverse flow laser machined seal offers further reductions in face temperatures, which could prove very important in a number of applications.

An important prospect for this new technology is in existing seal installations where improved performance could be available economically simply by modifying existing components.

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