DESIGN AND TESTING OF A ZERO-NET RADIAL TAPER, RADIALLY COMPLIANT FACE SEAL

by

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From 1971 to 1987, he served on the faculty of the Mechanical Engineering Department at the University of New Mexico, as Professor and Chairman of the Department and as Director of the Bureau of Engineering Research. During this time, he started a mechanical seal research program under the sponsorship of the National Science Foundation and the US Navy. This work served as the basis for numerous papers, reports, and inventions. A seal test program was started in 1978 and has continued since.

In 1987, he started Mechanical Seal Technology, Inc. (MSTI). MSTI performs research and product development, design software development, and consulting, all in relation to mechanical seals.

In 1991, his book, Principles and Design of Mechanical Face Seals, *was published by John Wiley.*

ABSTRACT

In conventional seal designs, the net radial taper changes with operating conditions so that contact is not always maintained across their entire width and leakage can occur. The "zero-net" face seal combines two ideas to avoid this problem. The first is where the geometry of the cross section is selected so that thermal radial taper of both the primary and mating rings is essentially the same so that the faces remain parallel at various operating conditions. The second is where one of the rings is made axially very short so that it becomes radially self aligning. The final result is a seal design where the faces remain in parallel contact across the face in spite of variable temperature, pressure, and speed.

The concept has been proven in the laboratory. Wear profiles show that the seal readily contacts across the entire face width. Test data show that the zero-net seal design has very low leakage in a water environment. The seal has been tested for more than a thousand hours and has been shown to give reliable and consistent performance. The zero-net face seal is expected to give longer, low-leakage life in field service than conventional rigid designs. The zero-net seal is now ready for field testing.

INTRODUCTION

A mechanical seal must control leakage below some acceptance level for the given application. Otherwise, it has failed, and equipment must be shut down and the seal replaced.

There are at least three points in time during the life of a seal where failure occurs.

- A seal may fail upon initial installation and startup; startup leakage never goes down to an acceptable level.
- A seal has been operating successfully, and the process is upset or restarted, and the seal then leaks excessively.
- A seal has been operating satisfactorily for some time and then under apparently constant sealing conditions begins to leak excessively, either suddenly or over a period of days or weeks.

Poor Radial Contact

There are many potential reasons for these failures, and these have been categorized and summarized by Lebeck [1]. One of the most important and common reasons for seal leakage is poor radial or tangential face contact. When conditions arise that the seal faces do not uniformly touch across their width, or if they tend to touch, say at only one or two spots radially across the face, or if the faces become too rough so that even though they touch uniformly, there are many leakage gaps due to roughness, then excessive leakage can be expected. Some of the contact patterns that can lead to excessive leakage and those that one expects lead to low leakage are shown in Figures 1 and 2.

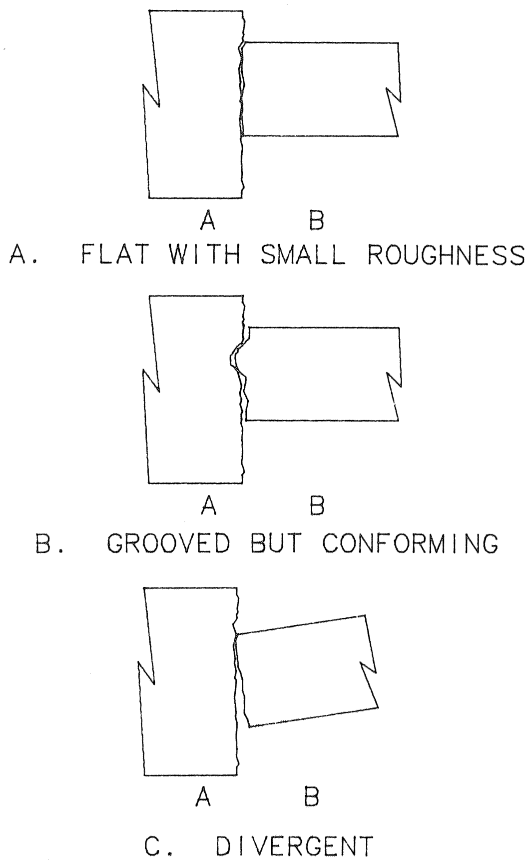
The issue of poor tangential contact or waviness has been addressed by Lebeck [1]. If seals become wavy to the extent that such waves cannot be flatted by the contact component of force on the face, excessive leakage can be expected. It is thought that for most commercial seals, some degree of tangential compliance is intrinsic because of the use of carbon materials and careful axisymmetric design (bellows for example). So, herein, the focus is on radial compliance and misalignment.

Radial Interface Profiles with Low Leakage

Three radial interface profiles or contact patterns that favor low leakage are shown in Figure 1. These are the as-running profiles in the as-worn condition. Seal faces operating essentially parallel and having a very small roughness are shown in Figure 1 (A). The resistance to leakage is high. A common wear profile where there is a deep groove in one face at the outside radius (can also be at the inside radius or both inside and outside) is shown in Figure 1 (B). The point here is that there has been enough wear that even with the deep groove, the faces contact all the way across. For an outside pressurized seal if contact favors the OD, there will be low leakage (Figure 1 (C)). Such divergent faces, where the film thickness increases in the direction of decreasing pressure, permit only a small amount of fluid pressure between the faces. Therefore, the amount of contact load pressing the faces together is high, and even though only a fraction of the radial width is contacting, leakage is low.

Radial Interface Profiles with High Leakage

Unfortunately, it is very difficult to achieve the ideal contact patterns above, particularly over time with variable operating conditions and restarts. Some of the problems encountered in



1. Interface Profiles Favoring Low Leakage.

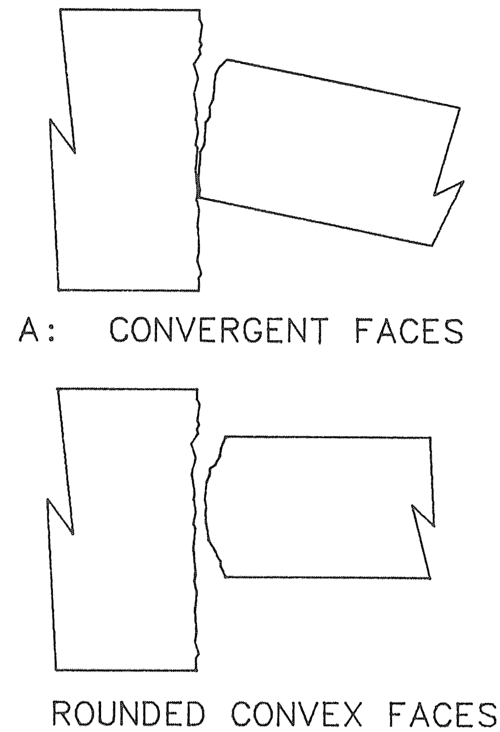
practice are shown in figure 2. If a seal is designed so that it does not develop radial taper due to pressure and is initially lapped flat, it will very often develop a convergent radial taper due to heating at the face on startup like that shown in Figure 2 (A). This contact pattern will often lead to high leakage on startup. If the balance ratio is high enough, this profile will wear parallel over time, and leakage will decrease. One can deliberately design a pressure caused divergence such as is shown in Figure 1 (C) that will offset the thermal radial taper of Figure 2 (A). At one particular operating point, the seal will have parallel faces like Figure 1 (A). At any other operating condition, the seal will be convergent or divergent.

If a seal operates with a pressure caused divergence as shown in Figure 1 (C), then over time the seal will wear flat. Now if the seal is restarted at a lower pressure, it will have a convergent taper such as in Figure 2 (A), and it may leak. If a seal operates with a temperature caused converging radial taper that wears to the conditions at 1 (A) or 1 (B) over time, sometimes the friction in the seal will increase and the radial taper will increase. This leads to reduced friction and the radial taper angle becomes different. In this case, the radial taper angle may be constantly changing and the seal will start to be rounded across the face, as in Figure 2 (B), and this will lead to leakage.

Finally, if a seal operates under variable operating conditions where both the thermal radial taper and the pressure caused radial taper change with these changing conditions, one will get a seal with a rounded face such as in Figure 2 (B). This seal will have relatively high leakage.

ZERO-NET RADIAL TAPER SEAL

Based on the above description of leakage causing behavior, if one were able to somehow zero out the radial taper under all



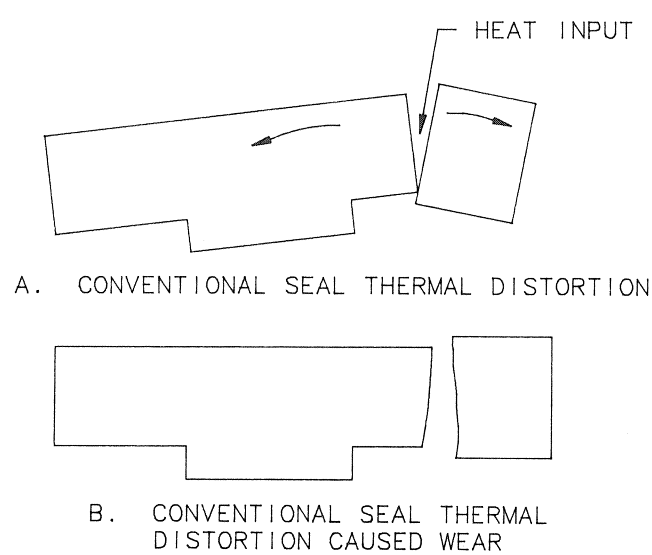
2. Interface Profiles Causing Leakage.

conditions, then the radial profiles of the seal would stay parallel and leakage would be small over time. One would achieve the contact pattern of Figure 1 (A) or 1 (B). Profiles such as in Figure 2 (A) and 2 (B) would be avoided.

The zero-net seal described below is designed to zero out all net radial taper. There are three primary concerns. The first is thermal radial taper, the second is pressure caused radial taper, and the third is residual radial taper, due to pressure and temperature.

Thermal Radial Taper

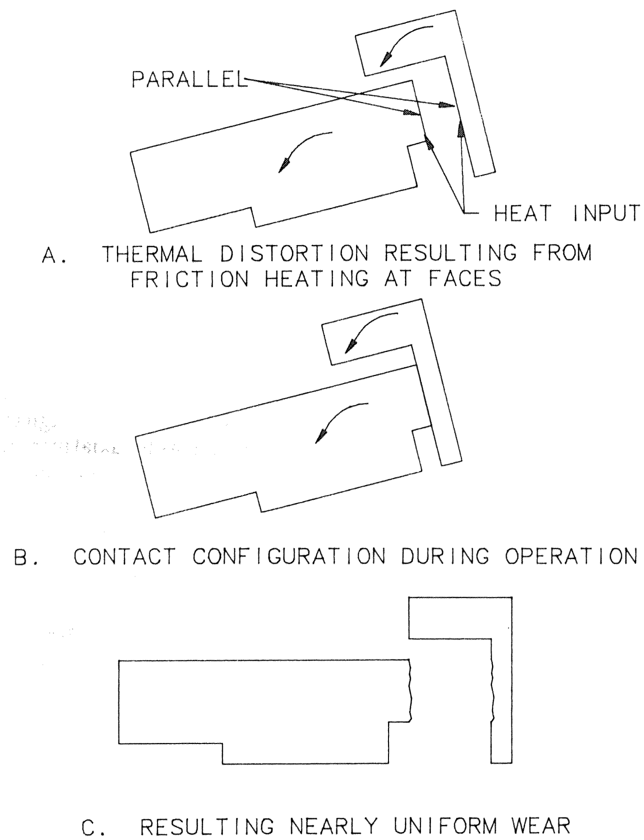
The way a conventional seal design develops a converging radial taper when heated by friction at the interface is shown in Figure 3. The seal nose is hotter than the regions of the ring that are



3. Conventional Seal Thermal Behavior.

further away from the face, and the face region radially expands more, and thus develops a taper. Due to either variable external operating conditions or due to self generated friction variations, thermal radial taper can lead to profiles of the type shown in Figure 3 (B), and leakage can result.

While one can also develop a radial taper due to differential expansion of materials in a composite seal design, and this can be minimized by design [2], the radial taper due to temperature gradient in a monolithic material design would seem to be inevitable. However, close examination of the mechanism of thermal deformation shows that it is not. One way of configuring a face geometric shape that causes the thermal radial taper to become opposite that shown in Figure 3 is shown in Figure 4 [3]. Parmar [4] shows yet another geometric shape that results in a reverse thermal taper.



4. Zero-Net Thermal Taper.

By careful design of the primary and mating rings using heat transfer analysis to obtain the temperature distribution and deflection analysis to find the thermally distorted shape, one can select seal geometries where the primary and mating rings thermally deform the same amount in the same direction as shown in Figure 4 (A). Thus, when the faces operate in the normal mode they will wear parallel as shown in Figure 4 (B). Since thermal radial taper is proportional to frictional heat one would expect that if the radial tapers are matched at one operating condition, they will remain closely matched at somewhat different operating conditions, so that wear is uniform even with variable conditions.

Thus, the concept of zero-net thermal radial taper is to cause, by design, the net thermally caused radial taper to be zero, so that the faces will wear and stay flat relative to each other.

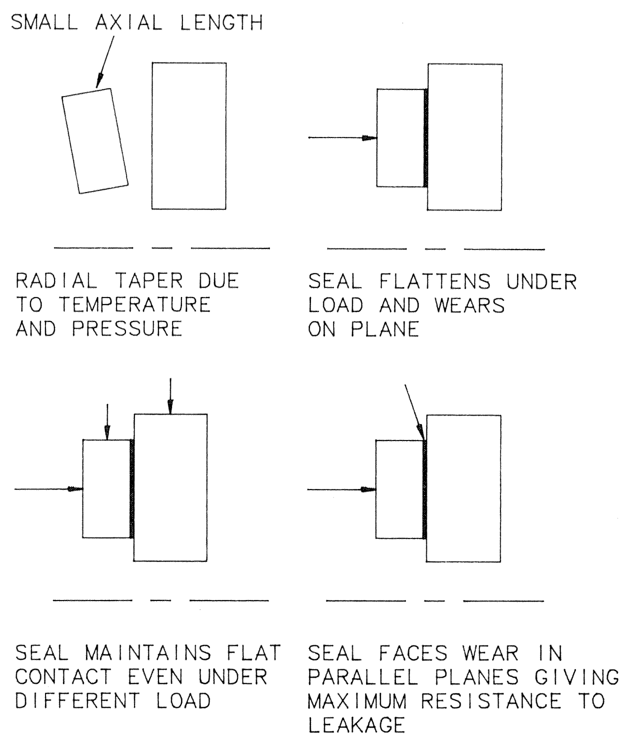
Pressure Caused Radial Taper

As discussed previously, radial taper can also be created by pressure loading. However, as is well known [1], this radial taper can be caused to be zero by again choosing the shape of the cross sections very carefully so that moments due to pressure about the circumferential centroidal axis are near zero. The seal design described here has such pressure caused radial tapers near zero.

Radial Compliance—Residual Taper

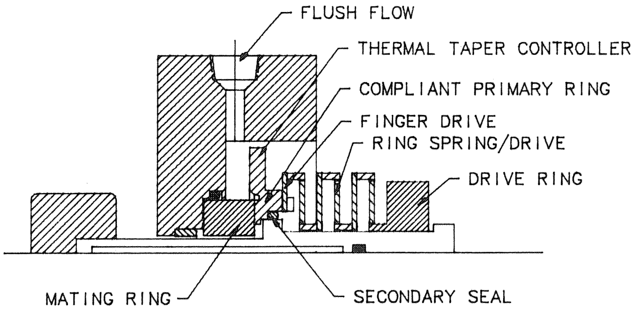
In spite of the efforts described in the preceding two sections, seal faces will not be exactly parallel. First, the net zeroing is not exact. While the largest part of the thermal taper can be zeroed out, there will always remain some significant amount. Second, during transients of startup and operating condition changes, the net radial taper will not remain zero. Third, while pressure moment balancing is exact in theory, such factors as O-ring expansion, geometry changes due to wear, and changing pressure distribution on the face will cause nonzero amounts of pressure caused rotation as well. The point is that no matter how well the radial tapers are designed out of a seal, there will still remain some significant radial taper.

To eliminate this residual radial taper completely, radial compliance is used. By making the axial length of one of the seal rings very small, as shown in Figure 5, it can be shown by deflection analysis that when the faces are brought together under load, the faces will align themselves radially. Thus, given any arbitrary small misalignment due to the above sources mentioned, the seal faces stay in parallel alignment.



5. Radial Compliance.

In order that radial compliance be effective over the life of a seal, there are certain facts that must be carefully considered during design. The first item is how much radial compliance is needed. In the seal shown in Figure 6, one gets on the order of 2000 microradians of radial taper under spring load by shifting the load from the



6. Zero-Net Seal Design.

center of the face to an edge. Thus, under spring load alone, very large misalignments can be accommodated. However, when a seal is made to have such a small cross section, it becomes much more difficult to zero out the pressure caused radial taper as accurately as for a stiffer design. Approximation errors and other factors now create relatively large radial tapers.

In the event that the radial taper due to pressure is not well canceled out, while initially the seal will be flattened anyhow, because very large magnitudes of radial taper can be flattened under pressure loading, over time the faces will wear to the corresponding taper. Now, if the worn taper becomes larger than can be flattened by spring, then one could develop a situation where, under spring contact alone, there will not be complete flattening. This could lead to initial leakage under low pressure.

ZERO-NET DESIGN

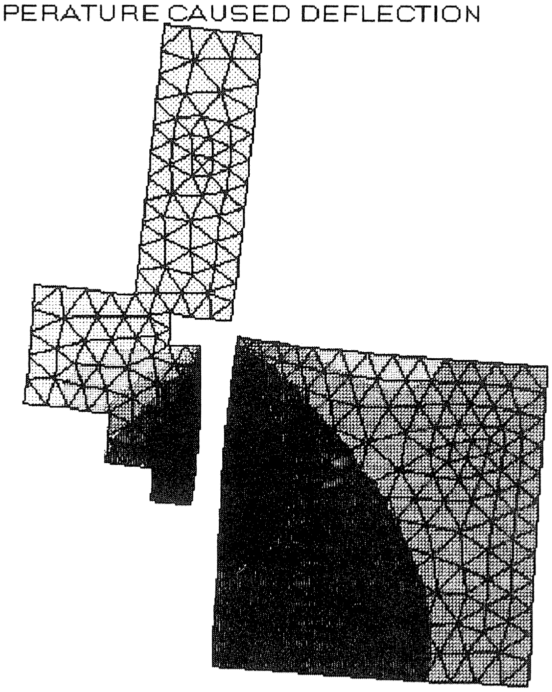
Using the above principles, a test seal has been designed. The final design is shown in Figure 6. The seal shown is for a 2.25 in shaft.

The shape of the primary ring was selected after making a great many iterations of heat transfer and deflection analysis. The analysis was made in an automated fashion using FEA computer programs similar to those described by Lebeck [1]. The analysis is set up so that geometry can be quickly changed, and all boundary conditions are automatically generated so that a complete analysis using a changed shape can be evaluated in just a few minutes.

The shape shown gives a thermal rotation like that shown in Figure 7. The thermal rotation is opposite that of normal. The reason the primary ring rotates as shown is that the thermal taper controller ring stays cooler than the material to the back of the primary ring as the isotherm map shows. Thus, the back of the primary expands radially more than the face, because the face is radially restricted by the cool thermal distortion control ring. As a part of the same design process, the pressure caused rotation was also made to be near zero.

The primary ring is driven by lugs that are placed just opposite the face. The use of 16 lugs (engaging 16 fingers, Figure 7) causes the drive force to be near uniformly distributed around the seal because the fingers provide some degree of tangential compliance, and this minimizes waviness. The drive force is transmitted to the lugs using the drive/spring system shown. The lugs engage a disc type of spring. Each disk has notches at the edge that engage the adjacent spacer. Thus, each disk acts as a spring and transmits the drive torque. The spring force is transmitted to the primary ring also just opposite to the face. Thus, the spring force creates no moment on the primary ring.

The purpose for using the ring spring and drive system is that it produces a very uniform spring force and drive force around the seal. Traditional drive lugs or notches and individual springs cannot be used on the radially compliant ring, because such nonaxisymmetric forces will cause very large wavy distortions in the primary ring.



7. Zero-Net Seal FEA.

The primary ring can be designed using any balance ratio. The seal in Figure 6 has a balance ratio of 0.8. The secondary O-ring seal moves with the primary ring so that moment balance is not upset by axial positioning. Axial travel is limited in this design as shown, but there is enough travel to accommodate the axial motion in most pumping applications. Lapping of the primary ring is accomplished by using a recessed lapping tool.

The thermal distortion control ring extends radially beyond the faces, so that for small bore stuffing boxes, the primary ring can be located in the gland plate as shown. Cooling or flush flow enters as shown and is directed at the mating ring. Most of the heat flows out of the mating ring in the zero-net seal because the primary ring has little of its hot area exposed to the cooling fluid. The mating ring is extended into the fluid as much as possible.

TEST RESULTS

The zero-net seal above has been tested for more than 1,000 hrs in water and oil. The test configuration is essentially equivalent to that shown in Figure 6, except that the cooling flow is directed in at an angle in a radial plane rather than radially as shown in the figure.

The operating conditions tested vary but most tests are at 3600 rpm. Leakage is measured directly by quantifying the flow either by using a tipping device or graduated cylinder. Temperature measurements of the sealed fluid and the mating ring are recorded. The mating ring temperature is measured at a point about 0.050 in back from the interface in the axial direction and at the midpoint of the interface in the radial direction.

Some of the most recent test results are summarized in Table 1. Considering test 1183, this result shows that with a freshly lapped carbon and mating ring, leakage rate is extremely low (assuming that leak rates comparable to these occurred for a zero-net seal sealing light hydrocarbons, 1000 ppm corresponds roughly to 3.0 g/hr and 100 ppm to 0.4 g/hr). After the first 100 hrs, this seal was given a cold restart, and it leaked slightly more. After the second 100 hrs, it was given another cold restart, and it leaked somewhat more, but the levels remained low.

Table 1. Zero-Net Seal Tests.

Fluid	Test	Δp (MPa)	Time (h)	T∞ (C)	ΔT (C)	Leak g/h	B	Materials
Water	1183	1.72	100	37	15	0.02	0.8	CTI101/WC
	restart		100		13	0.18		
	restart		13		14	0.90		
Water	1184	1.72	50	37	17	0.90	0.8	CTI101/worn WC
Water	1185	1.72	3	37	30	1.8	0.8	P658RC/WC
Water	1186	1.72	20	37	32	0.01	0.8	CTI22/WC
	restart		8		37	< 1.		
	restart		28		45	0.05		
	restart		16		40	0.09		
Water	1189	1.72	100	37	10-25	0.1	0.8	CTI101/SiC
Oil	1208	0.14	13	148	8	1.05	0.8	P7465/WC
Oil	1211	0.14	12	166	76	0.84	1.0	P7465/WC
Oil	1214	0.14	22	148	69	0.75	0.8	P7465/WC
Oil	1215	0.14	88	166	24	0.11	1.0	P7465/WC
Oil	1216	0.14	92	166	44	1.25	1.0	P7465/WC
Oil	1217	0.14	82	166	59	0.97	1.0	P10225/WC
Oil	1218	0.14	26	148	105	0.00	0.8	P8412/WC
Oil	1219	0.28	69	130	83	0.78	1.0	P7465/WC

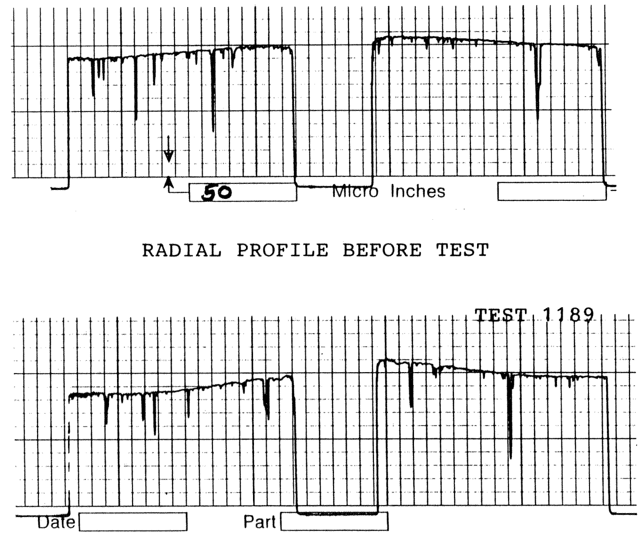
To see if the cause of the increased leakage was the carbon or tungsten carbide (WC), test 1184 is for the same carbon but relapped with the as-worn WC. With the new carbon and worn WC, leakage was higher than in the as-new condition.

Test 1185 uses a different carbon. This carbon does not work well with this seal design. While the reason is not clear, it likely has to do with the fact that this carbon usually develops a surface finish that is so small that it is likely that all fluid is excluded from the interface in this compliant design. In conventional designs, this material works well because fluid is usually present in the interface because of some amount of initial taper.

Test 1186 uses yet a different carbon. Here again, leakage for the first 100 hrs is extremely low. Leakage on the first restart is higher than expected, but it was coming down rapidly before the point where the test was unintentionally halted. Both the second and third restarts resulted in very low leakage, lower than the restarts of test 1183.

In oil, the results vary widely. For the most part, the seal leaks of the order of 1.0 g/hr at this relatively low test pressure. It is thought that parallel sliding hydrodynamic effects cause the seal faces to raise apart at this low pressure [1]. But, test 1218 shows that it is also possible to have zero leakage for this resin impregnated carbon. The explanation for this behavior is that at the very high temperature reached, the viscosity of the oil is so low that a nonsignificant fluid pressure is developed and the seal does not lift off. This seal face is also smoother than most of the other results. The important thing that test 1218 shows is that the compliant seal has the ability to seal very tightly, but that in some applications such as oil, it may not be possible to achieve zero leakage even though the seal faces are closely aligned and parallel, because the faces lift apart in these higher viscosity fluids.

One of the primary concerns about the zero-net seal mentioned above is if the worn radial taper generated will be within the capability of the spring force to maintain aligned faces when the seal is unloaded by pressure. Initial and final radial profiles for the carbon from test 1189 are shown in Figure 8. The initial and final average radial taper are nearly the same amount after this 100 hour test.



RADIAL PROFILE AFTER TEST

8. Radial Wear Profile.

The post test radial profile in Figure 8 has a concave appearance. This is caused by the fact that the surface is bulged out by thermal distortion during operation. The predicted bulge is shown in Figure 7, and the physical evidence (the mating face post test profile is virtually flat) is reflected in Figure 8.

The zero-net seal has been tested for many hundreds of hours beyond what is reported here. It has been tested under a wide range of pressures and temperatures. No seal blowouts, cracking, or breakage has occurred. The seal drive system and spring have worked reliably even for high torque breakout situations. The seal has been hydrostatically tested to 1500 psi with no fracture. In spite of the fact that the axial dimension of the seal is small and the thermal distortion control ring is connected by only a thin section, the seal appears to be robust.

DISCUSSION

The test data for water show that the zero-net seal does control leakage to very low levels. Taking the Nau [5] survey data for liquid seals, one could conclude that a typical low seal leakage might be 0.06 g/hr/mm. Lebeck [1] shows data where a typical low leakage seal might have as little as 0.01 g/h/mm. For the 50 mm seal here, the equivalent range of rates would be from 0.5 to 3.0 g/hr. The zero-net seal shows better performance in most instances with water. In fact, test 1186 and its restarts and test 1189 using the most suitable materials show that results from the zero-net seal show leakage control ten to one hundred times better than these "typical" seals.

The fact that the worn radial taper is very close to the initial radial taper verifies that the net radial taper distortion being imposed on this seal under load by both temperature and pressure are very small as is predicted. Otherwise, a much larger radial taper would wear in to the faces, even within 100 hrs.

Test data also show where a seal is given a cold restart, the average leakage rate will be higher, over the test periods used, than the initial leakage rate. The primary reason is thought to be that given newly lapped faces, there is a break-in period where the seal face goes from flat to the shape shown in Figure 8. That is, the thermal bulge that is naturally generated wears off. When the seal is restarted, a surface such as in Figure 8 mates with a flat surface, so one has contact at the inner and outer edges. This contact pattern will give more leakage than uniform contact all across. It is thought that once the seal faces have cooled, then equilibrium where friction heating would cause the worn bulge to move out to just where it was before, simply cannot become automatically reestablished in the relatively short periods of time tested here. However, over a longer period, one would expect that the leakage would continue to decrease after a restart. The point must be made that even the leakage values under restart conditions are very low.

The test data also show that some materials give much better performance than others. Of the several carbons tried in water, the CTI22 gave the lowest leakage during first run and during restart. The results obtained so far (some not included in the preceding data) show that some materials will work much better than others in the radially compliant seal. The reason is thought to be that the interface is virtually starved of liquid to a greater extent than in conventional seals. Thus, carbons that evolve a definite but small roughness are perhaps better than those that become mirror smooth.

What is not shown in these test data is how well the zero net seal will do under long term field conditions. Many seals will perform well for certain periods of time when new. However, in the case of the zero-net seal one expects that the performance observed on the test stand will typify that in service because the seal faces are well worn in after just a few hours. Contact occurs all the way across, so that there would be no change brought about from further wear in. Conventional seals sometimes have to wear in before a consistent performance is obtained because of the reasons discussed above. Also, unlike conventional seals, one does not expect that the zero net seal performance will degrade because of self induced thermal cycling and radial tapering, because even with changes in friction, the compliance of the seal forces the faces to retain the same alignment. Furthermore, observations on the test stand show that the seal, using the most compatible materials, gives relatively consistent drive torque and temperature rise, unlike some seals observed on the test stand where there is a large variation of friction torque with time.

If one hypothesizes that, under steady running conditions, a conventional seal fails because of self induced frictional changes that lead to variations in radial taper and, therefore uneven wear, the zero-net seal should perform better and last longer. If, on the other hand, seal failure occurs because seal faces become grooved by wear and small radial shifts cause the grooves to misalign and then leak, the zero-net seal may not do any better. However, given the likelihood that seal failures using conventional seals are caused in some part by the inability of seals to radially align under varying conditions, be they self induced or process caused, it is likely that the zero-net seal will perform better.

CONCLUSIONS

The concept of the zero-net seal has been proven by more than a thousand of hours of laboratory testing. Data show that leakage is very low and that the concept works to give self aligning faces. The design has been proven to be robust; there have been no seal blowouts. It is anticipated that because of the small net radial taper and self aligning characteristics, the zero-net seal will last longer in service than some conventional designs. It will adapt to process condition changes and self induced frictional changes without causing radial taper realignment. The zero-net seal is ready for field testing.

NOMENCLATURE

B	balance ratio
Δp	sealed pressure difference
ΔT	face temperature rise above T_∞
T_∞	temperature of sealed fluid

REFERENCES

1. Lebeck, A. O., *Principles and Design of Mechanical Face Seals*, New York: John Wiley (1991).
2. Casucci, D. P. and McCollough, W. J., "Benefits of a Balanced Interference Fit Bellows Seal Design in High Temperature Corrosive Applications," *Proceedings of the Ninth International Pump Users Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1992).
3. Lebeck, A. O. and Young, L. A., "Radially Compliant - Zero Net Thermal Radial Taper Mechanical Face Seal," US Patent 4,792,146 (December 20, 1988).
4. Parmar, A., "Thermal Distortion Control in Mechanical Seals," Paper B4, 12th International Conference on Fluid Sealing, Brighton, England (1989).
5. Nau, B. S., "Rotary Mechanical Seals in Process Duties: An Assessment of the State of the Art," Proceedings of the Institution of Mechanical Engineers, 199 (A1): *17* (31) (1985).