MECHANICAL SEAL DESIGN

FOR HIGH AND VARIABLE PRESSURES

BY

J. PHILLIPS, C.ENG.,M.R.AE.S.
(Flexibox International)

This article is based on a paper presented at the Conference on 'Naval Engineering—Present and Future' held at the University of Bath in September 1983 to mark the centenary of the formation of the Royal Corps of Naval Constructors. It is reproduced by permission of the Institution of Mechanical Engineers.

Introduction

Recent development work with high duty single mechanical seals has demonstrated that significant improvements in their ability to contain high and variable pressures can be achieved by changes to the sealing face geometry. An extension of this work has also led to the design of high duty double mechanical seals which can safely handle the considerable reverse pressures that invariably arise with the loss of barrier fluid pressure.

In many marine and industrial applications, mechanical seals have to withstand not only the distorting effects of constant high pressures but also the variable distorting effects of pressures which fluctuate between wide limits with the extremes still within what might be termed the high pressure envelope.

The aims of this article are to:

(*a*) Examine how these high pressures and pressure fluctuations affect the performance of existing seals.

(*b*) Review the work being done to extend the pressure containment limits of these seals whilst retaining preferred materials.

(*c*) Consider how the improvements achieved with single seal designs can be applied to double seals, not only to increase their pressure containment limits in the forward direction but also in the reverse direction*.

In the course of the article, it will be seen that the changes made to achieve the higher pressure ratings have also enabled the operating limits of seals to be extended in terms of permissible PV values†; this aspect of the work is discussed in some detail.

The Limitation of Single Seals

Mechanical seal designers are extremely loathe to choose face material pairings other than those which include carbon as one of the pair. The main reason for this is that carbon is a forgiving material and will readily tolerate a certain amount of abuse particularly in the way of dry running. It is also less likely to suffer ill effects from thermal shock than its carbide or ceramic alternatives.

Carbon is, however, a relatively weak material having a low modulus of elasticity and hence is prone to distortion when subjected to high pressures. Although seal designs invariably arrange for the carbon component to be in compression, fault conditions in double seals can subject the carbon sealing ring to internal pressures putting it in tension and, in this mode, the carbon is very likely to fracture if the duty is a severe one.

FIG. 1 shows a section through a high duty seal which, during the past few years, has been very successfully used in such high duty applications as pipeline pumps, water injection pumps and boiler feed pumps. Seals of this type have the capability of handling PV values up to 1960 bar metre/second when sealing water and 4200 bar metre/second when the pumped product is hydrocarbon. Many seals of this type are giving reliable service at pressures up to 110 bar and many are running at speeds in excess of 6000 r.p.m.

The maximum operating pressure of the seal is, within the limits of its PV value, dictated by the strength of the carbon stationary seal ring. If the operating pressure is higher than the prescribed limit, then the resulting distortion is greater than can be tolerated and the outcome is either a tight seal with a high rate of wear or a poor seal with a low rate of wear, dependent upon whether the face goes concave or convex. Both these cases are undesirable.

When designing the seal illustrated in FIG. 1 much attention was paid to the geometry of the carbon in order to balance out the hydraulic forces acting about the centroid of the face (FIG. 2). In addition to the usual distortion calculations based upon elastic theory, a series of finite element stress analyses were also performed on a number of possible shapes. The final choice is illustrated in FIG. 3 and shows that the running face would deflect some 10 light bands in a concave direction when subjected to a pressure of 103 bar. In the light of the original design specification and experience at that time, this carbon was deemed to be acceptable both in terms of the amount of deflection and the direction of that deflection. The logic for this was that at start-up, there would be a fairly high specific face load at the outer diameter of the nose of the carbon which would ensure a seal and this would then only require a short running-in period in order to generate two acceptably parallel faces. In fact, when the seal had run in, the specific face load would reduce, the coefficient of friction would decrease,

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*Barrier fluid pressure is considered to act in the forward direction and is usually arranged to be some 2 to 10 bar higher than the maximum pumped product pressure seen by the inner seal which is said to act in the reverse direction.

†PV is, within limits, a useful way of expressing a seal's capability to meet a particular duty; it is the product of the pressure in the seal cavity and the velocity of the seal face.

and from that time on the wear rate would become extremely small. Provided the pressure remained reasonably constant, more than adequate life would be obtained.

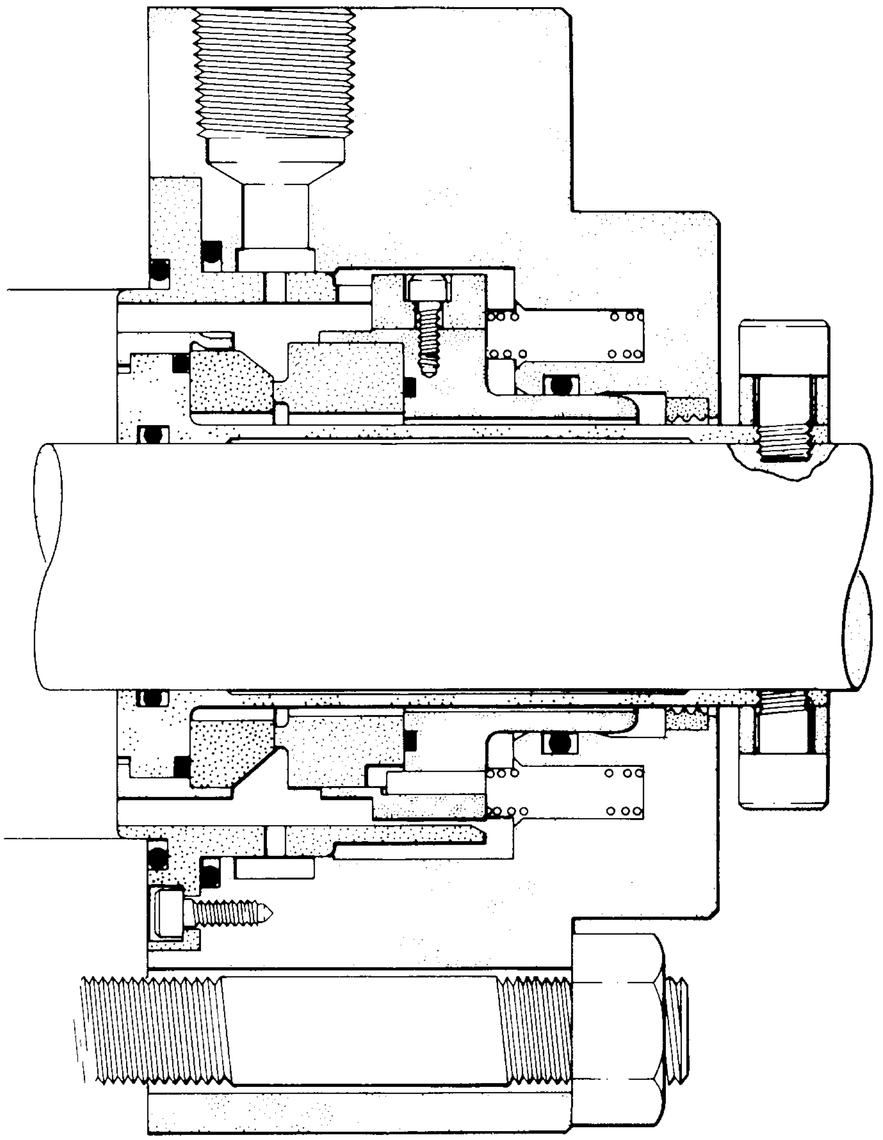

FIG. 1—EXISTING HIGH DUTY SEAL

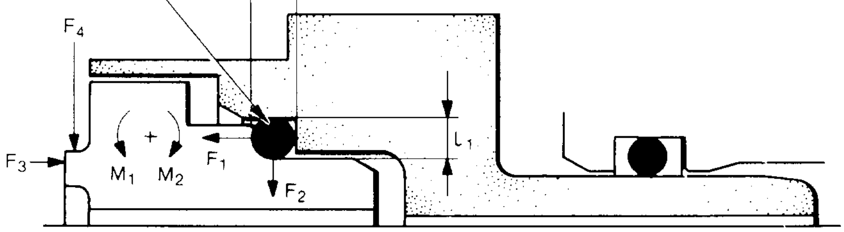

FIG. 2—FORCE BALANCE OF STATIONARY SEAL RING INSERT
p' > p where p = product pressure to be sealed
p' = pressure seen by 'O' ring

Currently, duties are becoming ever more arduous and pressures greater than 150 bar are by no means uncommon, with pressure swings of up to 80 bar having to be tolerated. In reviewing the existing design in relation to these much more difficult conditions, it was seen that not only will the existing carbon deflect more than a tolerable amount, but, even if the application allowed a running-in period, a subsequent reduction in pressure would permit the carbon to relax and the face would go convex. In this mode, there would be a converging fluid face film and the ensuing pressure distribution across the faces would force them to part and the seal would leak. This effect is illustrated in FIG. 4.

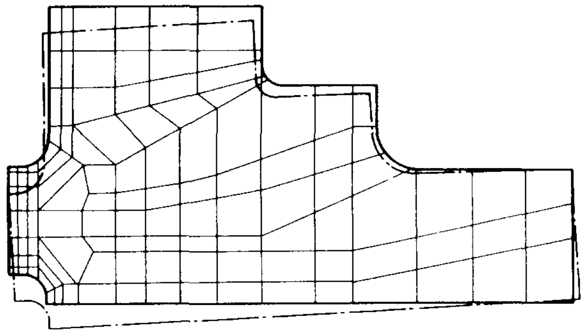

FIG. 3—COMPUTER PREDICTION OF FACE DISTORTION, GREATLY MAGNIFIED

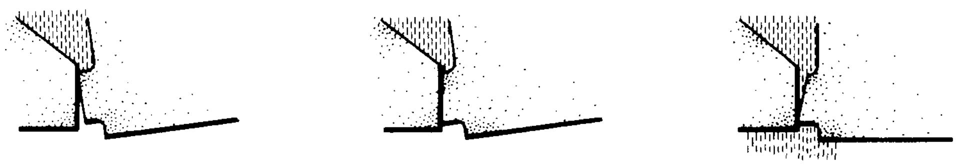

FIG. 4—EFFECT OF CHANGES IN OPERATING PRESSURE

In developing the existing design of carbon, 'O'-ring tolerances were taken into account but under the more difficult conditions of today's applications, the variation in face deformation due to this tolerance becomes untenable and alternative designs had to be looked for if carbon was to be retained as one of the face materials.

An example of the magnitude of variation of distortion due solely to 'O'-ring tolerances is shown in FIG. 5.

Improvements to Single Seals

FIG. 6 shows a revised carbon geometry and a change to the positioning of the 'O'-ring packing. This siting of the 'O'-ring was chosen to give the desired seal between the insert and its carrier and to continue to give the required measure of flexibility in the stationary seal ring assembly.

In this case, the action of the 'O'-ring is markedly different from the existing design and it can be clearly seen that the force balance of the section is not affected by 'O'-ring tolerances. In fact, variation in face twist for a given pressure is only dictated by the 'O'-ring groove dimensions, and these can be toleranced and controlled to such small limits that they can be considered to have negligible effect.

It will also be seen that this new shape is much more robust than the old design, it being more constant in section, and hence more predictable in its behaviour when subjected to both high and variable pressure conditions.

Confirmation of the expected performance of the new carbon design was established by finite element analyses of various shapes and the results pertaining to the final choice are also shown in FIG. 5. This set of curves illustrates beyond doubt the measure of stability achieved by the new design.

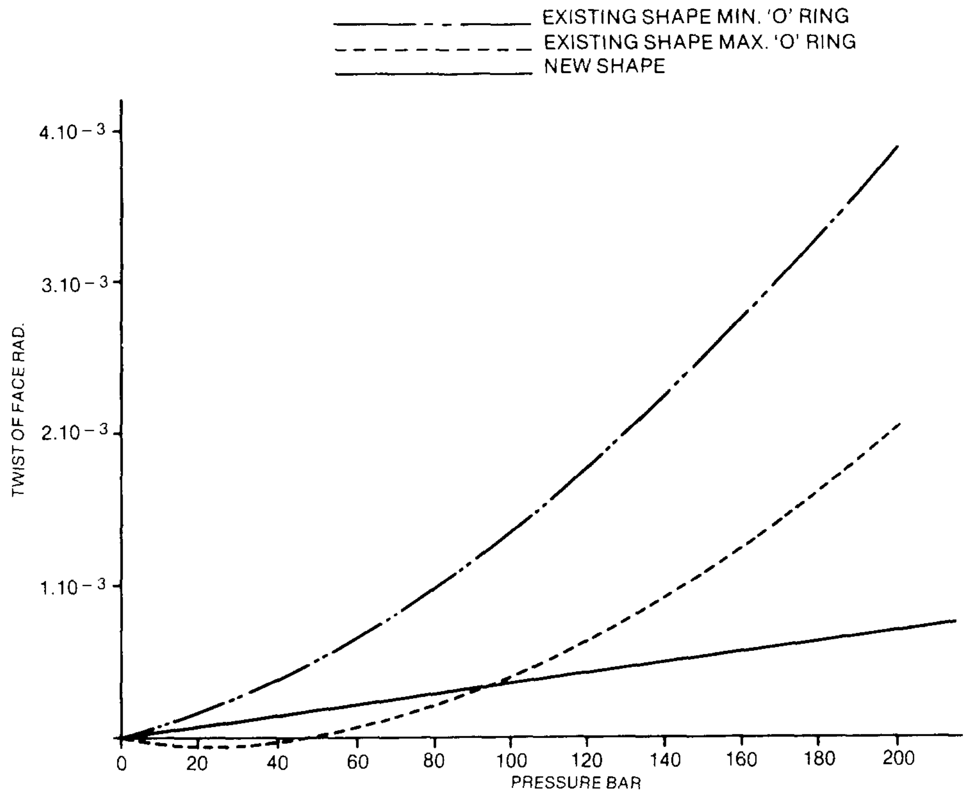

FIG. 5—COMPARISON OF NEW SHAPE CARBON DISTORTION WITH EXISTING SHAPE DISTORTION

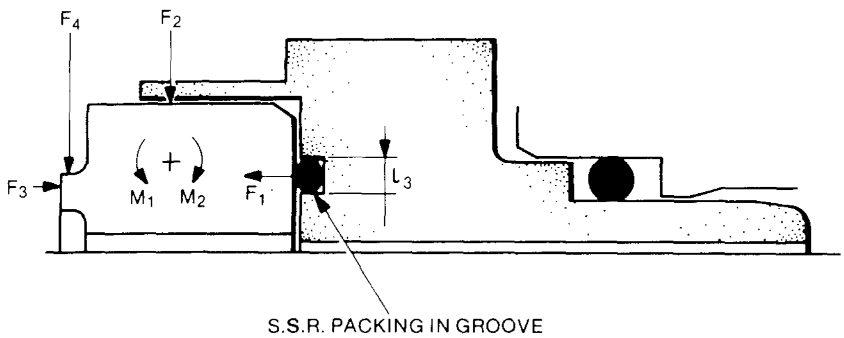

FIG. 6—REVISED CARBON SHAPE

Perhaps a pertinent question to ask at this point is why the work was not taken further and why the twisting moments were not totally balanced out. The answer to this question lies in the result of some earlier work in the laboratory which had shown that perfectly square and non-distorting carbons exhibited a tendency to instability in their running patterns when subjected to pressures approaching 200 bar. This work had also shown that there is a secondary thermal distorting effect which acts in opposition to the hydraulic forces generating nose concavity. With the new design, this force is arranged to be marginally the greater of the two and the face assumes a slightly convex form. This in turn results in a thicker hydrostatic film on which the seal runs and explains why the new seal requires no running-in period and why it is capable of operating at increased PV values.

Obviously the same mechanism takes place in the existing carbon design but, due to the need to have a geometry which will take care of maximum 'O'-ring tolerance in a seal operating on a relatively low pressure, the nose must always be concave when sealing high pressures.

The results illustrated in FIG. 5 show that carbon is still a viable face material for mechanical seals operating at pressures up to 200 bar.

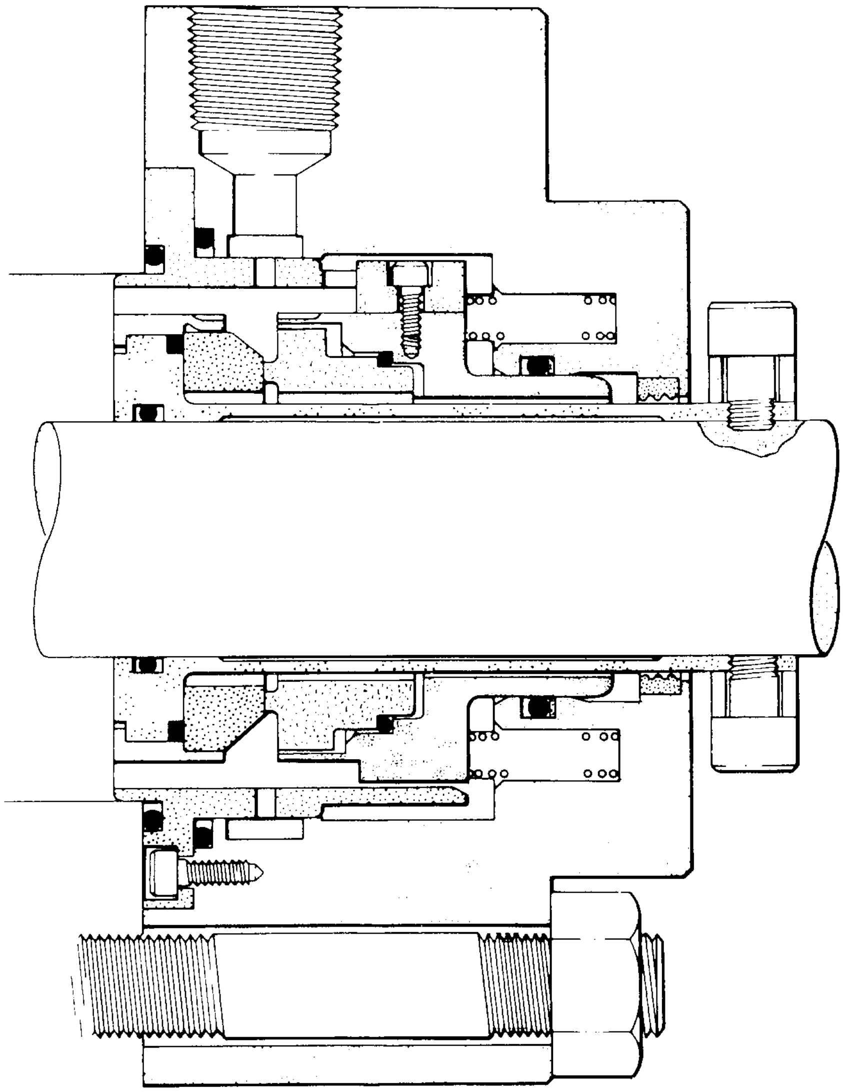

FIG. 7—IMPROVED HIGH PRESSURE SINGLE SEAL

The next step was to test a complete seal, and FIG. 7 shows the seal arrangement incorporating the new carbon, designed to meet the following duty:

Pump shaft diameter:	95 mm
Seal size:	125 mm
Dynamic pressure:	40 to 100 bar
Static pressure:	180 bar
Permitted leakage—dynamic:	30 ml/min
Permitted leakage—static:	0 ml/min

Product:	water
Temperature:	65 °C
PV value:	4385 bar metre/sec

It should be noted here that the PV value for this seal is 2·2 times the maximum permitted PV value of the existing seal for water duties.

Tests have shown that the new seal requires no running-in period and that the seal is absolutely tight at 180 bar pressure. During the dynamic tests the maximum leakage measured never exceeded 1 ml/min at maximum duty. During a stop/start test, it was shown that the new seal could be satisfactorily stopped and started 30 times in 60 minutes with the pressure varying from 0 to 40 bar and the speed varying from 0 to 6700 r.p.m. At the end of this particular test, the leakage rate at 40 bar and 6700 r.p.m. was 0·45 ml/minute and at 100 bar and 6700 r.p.m. was 0·85 ml/minute thus exemplifying the stability of the new design when subjected to varying pressures.

The Limitations of Double Seals

Safety and environmental awareness are today demanding the use of double seals even at the very high pressures applying to single seals and, in addition to the already discussed problem of pressure distortion, the danger of a total loss of barrier fluid on the integrity of the seal must never be forgotten.

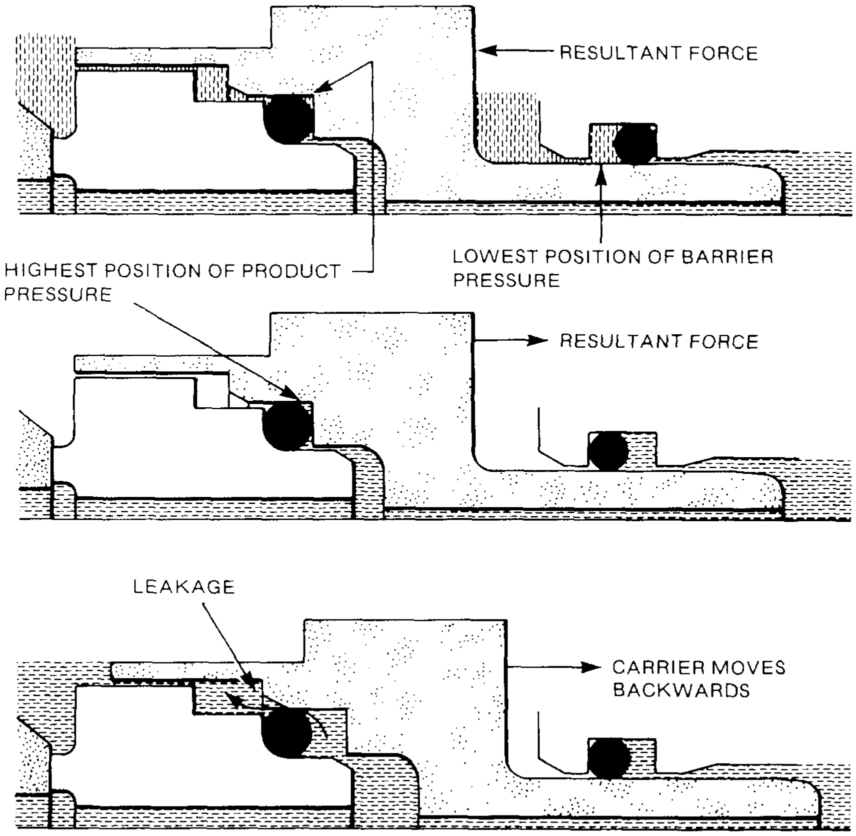

FIG. 8—HYDRAULIC FORCES ON INNER SEAL OF A DOUBLE SEAL

FIG. 8 shows the inner seal of a double seal, based upon the existing single seal, which like the single seal has been very satisfactory in service. It is apparent from the design that, while the barrier pressure is available and higher than the pumped product pressure, the resultant hydraulic forces keep carbon insert and its carrier together and also maintain the carbon in a state of compression.

It is evident from the design that if there is a total loss of barrier fluid pressure, then the carbon of the inner stationary seal ring and its carrier could be forced apart and if the 'O'-ring packing left its land, a leakage path would be created. The current design handles this problem by ensuring that the permitted axial movement under reverse pressure prevents the two seal components from parting by as much of the length of the 'O'-ring land. This measure is reasonably satisfactory except that under these conditions, the carbon is subjected to a reverse pressure, putting it in tension. Typical values of reverse pressure which can be handled by the existing design are:

150 mm seal: 46 bar
95 mm seal: 42·5 bar
67 mm seal: 47·5 bar

Unless the integrity of the barrier seal system can be guaranteed, these values also limit the maximum working pressure of the complete double seal and, even if the seal pressure system can be guaranteed, it is still imperative that the pump start-up procedure ensures that barrier fluid pressure is on before the pump rotates.

Improvements to Double Seals

It was a simple step to realize that the improvements to the performance of the new carbon when applied to single seals could, with advantage, be introduced into double seals and that by providing a double balance line for the inner seal carrier, an elegant solution to the blow-out problem would be achieved. This is illustrated by FIGS. 9 and 10.

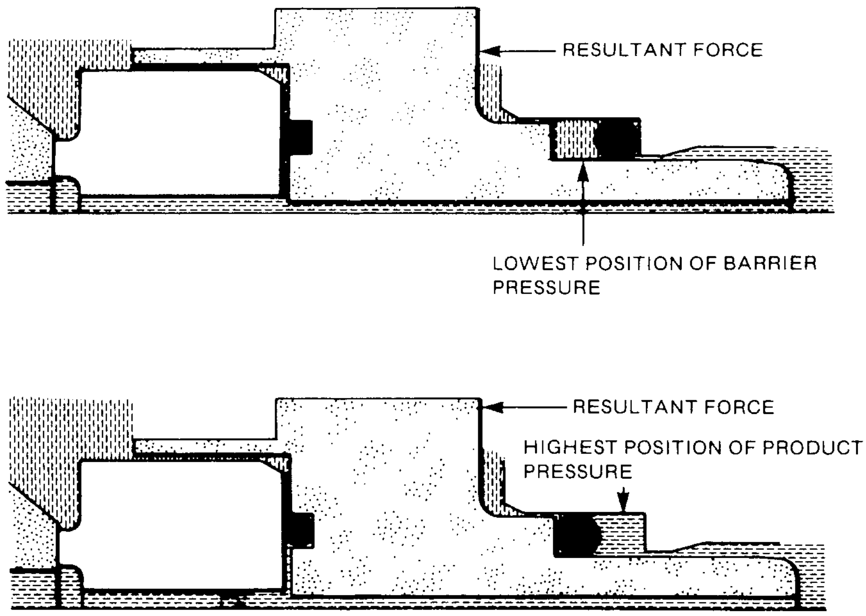

FIG. 9—HYDRAULIC FORCES ON INNER SEAL OF DOUBLE SEAL WITH NEW CARBON SHAPE

Whilst this arrangement considerably enhances the integrity of high pressure double seals, it does not in itself solve the problem of high reverse pressures acting on the carbon. The chosen solution to this problem was to shrink a steel support band around the outside of the carbon. This arrangement has been tested on a number of 67 mm, 95 mm, and 150 mm seals and the results have shown that internal pressure in excess of 160 bar on a 150 mm seal can now be tolerated. This arrangement is, of course, temperature dependent due to the difference in coefficient of expansion between carbon and steel and the working envelope is shown in FIG. 11.

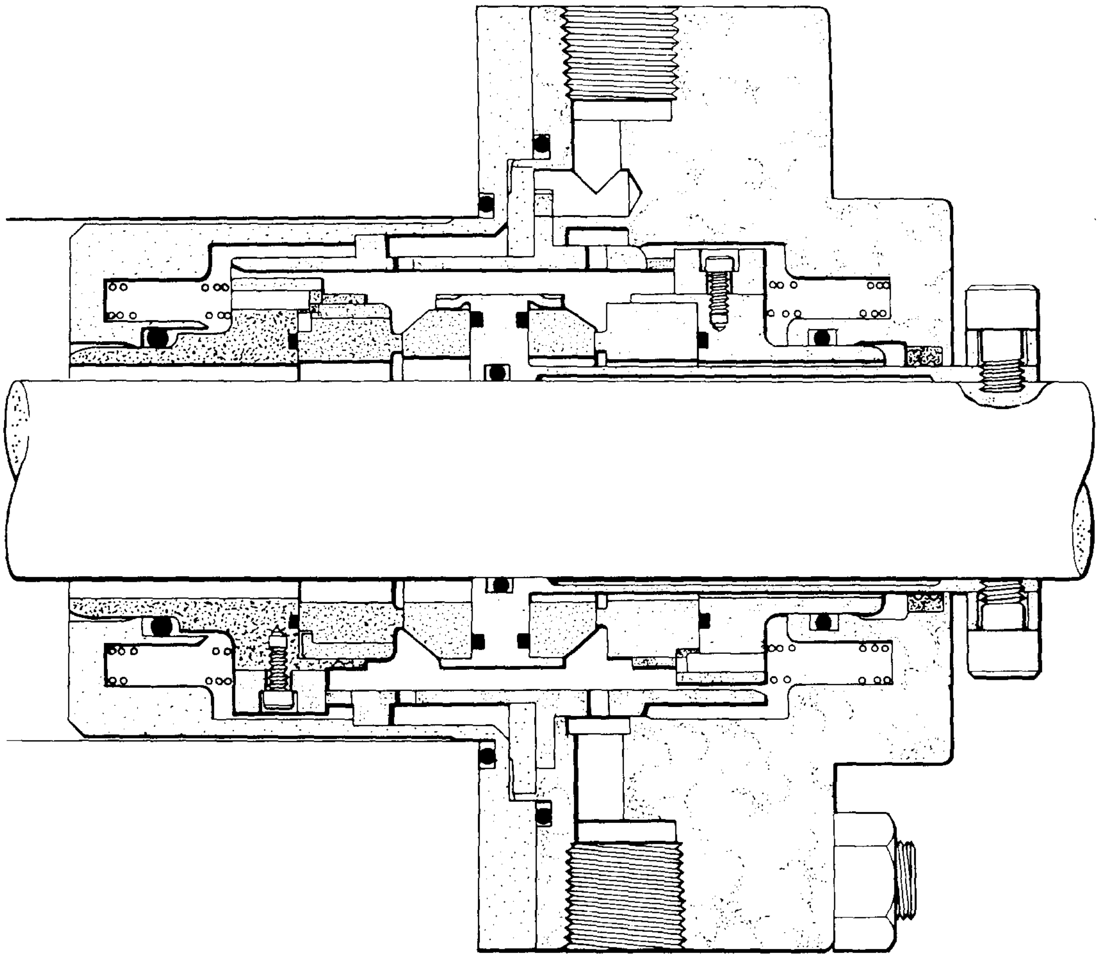

FIG. 10—IMPROVED HIGH PRESSURE DOUBLE SEAL

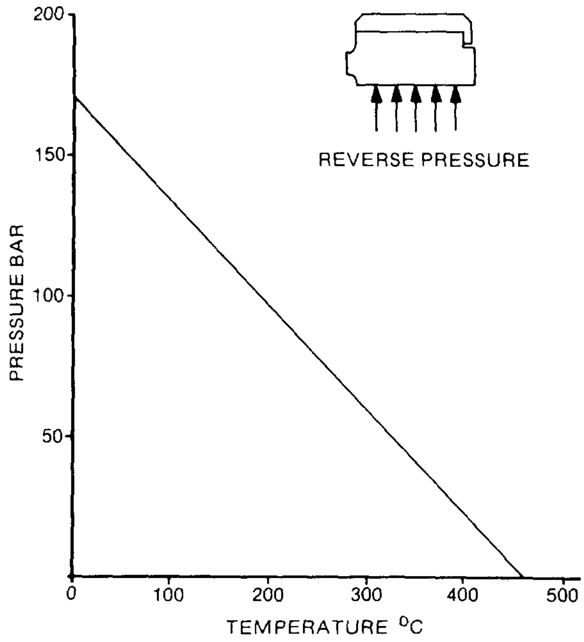

FIG. 11—OPERATING ENVELOPE OF IMPROVED SHROUDED CARBON

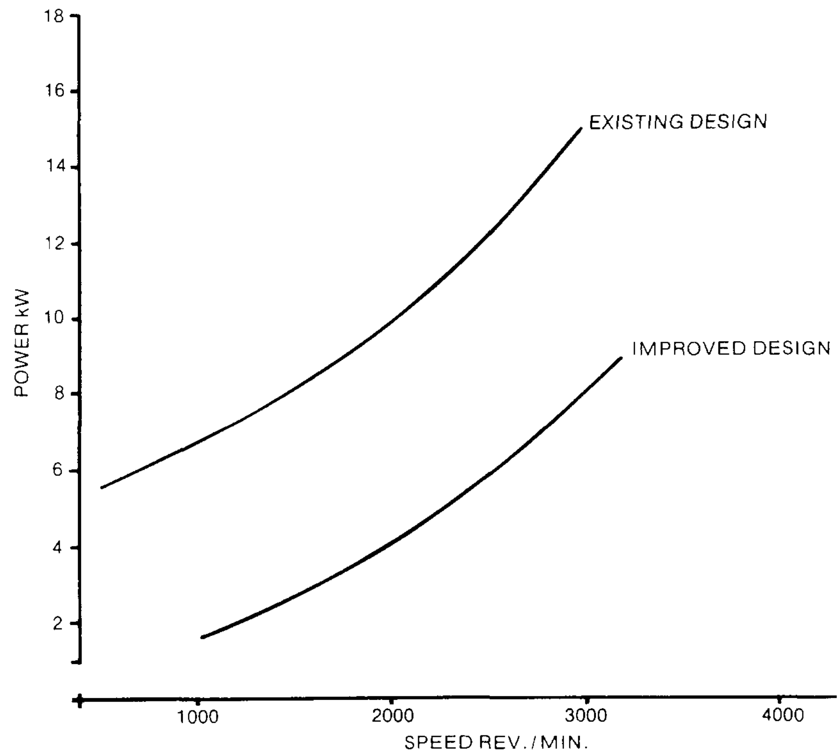

FIG. 12—COMPARISON OF POWER ABSORPTION BETWEEN EXISTING AND IMPROVED DESIGNS

Seal size = 150 mm
Product = water
Pressure = 83 bar

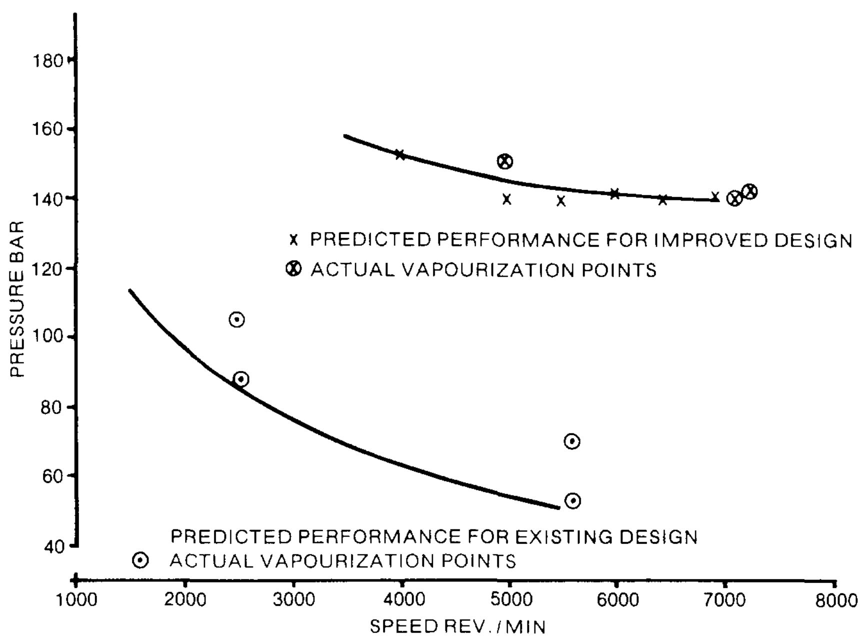

FIG. 13—COMPARISON OF VAPOURIZATION LIMITS BETWEEN EXISTING AND IMPROVED DESIGNS

Seal size = 150 mm
Product = water

Power Consumption

One highly desirable benefit resulting from the reduction in distortion of the seal faces, and the better control over the direction of distortion that is now achievable, has, as stated earlier, led to a considerable improvement in the ability of the seal to create for itself a thicker film on which to run. This is illustrated by a significant reduction in friction between the faces under maximum duty conditions.

FIG. 12 compares power absorbed by the existing design with the new design and FIG. 13 illustrates the higher PV values that can be accepted. These improvements mean that seals can now be operated much closer to the vapour pressure/temperature curve of the pumped product than hitherto without vaporization at the faces taking place.

Conclusions

The new design thus promises, for single and double seals:

(*a*) Markedly reduced and consistent pressure distortion characteristics which will vastly improve the operating pressure range.

(*b*) A significant increase in static pressure sealing capability.

(*c*) An elegant solution to the problem of pressure reversals in double seal applications.

(*d*) An increase in PV limits currently achievable on both aqueous and hydrocarbon duties.

(*e*) The confident selection of mechanical seals for sealing duties where the pressure is both very high and widely variable.