speed and viscosity, the clearance \( h \) of the test seal is about 3 micro-meters. For comparison, a seal with a smooth sealing ring having the same I.D. and O.D. would leak about 40 cc/hour under the same conditions. The seal was tested at higher pressures (35 and 40 bars) for very short periods of time, but was not allowed to "run-in." As expected, the leakage became higher, but was still a small fraction of the leakage of a conventional non-contacting seal. At pressures high above the design pressure, the seal is expected to approach the leakage rate of a conventional non-contacting seal as its pumping effect dwindles against the pressure differentials.

Figure 9 depicts the total power loss measured during the same tests shown in Fig. 8. In this case, the rate of drop of power consumption does not lag behind the rate of drop in viscosity. Both viscosity and friction power stabilize after a period of 2-3 hours of running time. For comparison, the calculated interface friction of a smooth ring operating at a clearance of 4 micro-meters under the same conditions is shown. The qualitative behavior is identical, quantitatively different due to the contribution of additional losses to the power consumption. Because the power loss is proportional to \( h^{-1} \), it is not as sensitive to the flatness of the sliding surfaces as the pumping effect which is proportional to \( h^{-2} \). This is perhaps the explanation to the lag in sealing effectiveness behind the rate of drop in viscosity, while the power consumption does not exhibit such a lag.

After proper running-in period where minor deformations are "lapped-off," the sealing capabilities come into effect.

Figure 10 shows the leakage during 12 test runs which were conducted under pressure of 25 bars and 1300 rpm. Each test started after an overnight stand still of the system and lasted 9-11 hours. Here, the sealing effectiveness appears to improve at the same rate as the oil viscosity; a behavior which is expected to show up once the running-in process is completed.

After 200 accumulated hours of run time, and about 50 complete cycles of start-up, run and shut down, the seal was removed from the test rig and inspected. No change has been found in the geometrical dimensions of the hardened stainless steel step bearings. The only finding was concentric score marks with mean roughness of about 1 micro-meter, which corresponds to peak-to-peak height of 4 micro-meters. The bronze rotor showed the same score marks with slight wear of a few microns. The exact amount of wear was not determined because of the lack of appropriate measuring equipment.

The above findings strongly suggest that the seal indeed operated under no contact between the seal ring and seat, but that some wear takes place, probably of the abrasive type, due to contamination of solid particles which score the sealing surfaces.

Conclusions

1. The results show that the noncontacting "zero-leakage" face seal is feasible.
2. The sealing effectiveness of the proposed seal is found to be superior to the older viscoseals while the friction characteristics are similar.
3. The results demonstrate that some abrasive wear takes place in non-contacting seals—a phenomenon which can be used advantageously during the running-in period—provided that a proper material combination is used.
4. Further testing at different speeds and fluids is necessary for establishment of a reliable data base.
5. A numerical analysis of the seal performances is required for better understanding of the seal behavior, particularly during unsteady periods of operation.

References


Discussion

I. Etsion

The seal described in this paper is claimed to present a new concept of a zero-leakage noncontacting seal, different from the previous one described in references [8], and [11]. Referring to Figs. 2 and 3 the concept described in [8] is such that the large gap \( H \) prevails over the entire width \( L_1 + L_2 \) of the inlet to the pressure cell. In this case the shear induced flow into the cell is

\[
Q_{sl} = a \frac{U}{2} H \tag{1}
\]

while the shear induced flow out of the cell is

\[
Q_{so} = a \frac{U}{2} h \tag{2}
\]

The net flow pumped into the cell is therefore

\[
Q_s = Q_{sl} - Q_{so} = a \frac{U}{2} (H - h) \tag{3}
\]

The flow given in (3) is the one available for offsetting pressure induced flow leaking out of the pressure cell. By adding the step at the inlet to the pressure cell the concept is not changed, only the incoming shear induced flow is reduced. Due to flow continuity at the step the incoming shear induced flow \( Q_s \) can be expressed (assuming ambient zero pressure inside the cell) either in the form

\[
Q_s = \left( \frac{U H}{2} + \frac{h^3}{12 \mu} \frac{p_r}{L_2} \right) a \tag{4}
\]

or

\[
Q_s = \left( \frac{U H}{2} - \frac{h^3}{12 \mu} \frac{p_r}{L_1} \right) a \tag{5}
\]

Depending on which of the above two forms is used for \( Q_s \) in equation (3) the net flow into the cell is either

\[
Q_n = \frac{h^3}{12 \mu} \frac{p_r}{L_2} a \tag{6}
\]
or its equivalent
\[ Q_s = a \left( \frac{U}{2} (H-h) - \frac{h^3}{12\mu L_1} p_s \right) \]  

Unfortunately, the author chose to use the form given in equation (4) thereby concluding erroneously that the hydrodynamic step pressure \( p_s \) is the only mechanism for achieving the offsetting flow \( Q_s \) as given in (6). In fact from equations (3) and (7) it is clear that the step pressure \( p_s \) is unfavorable and reduces the pumped flow \( Q_s \) that could be obtained in the absence of the hydrodynamic bearing.

One can only conclude that the so called “different new concept” described in the present paper is not new but rather a less efficient form of the basic concept described in [8].

Additional Reference

A. O. Lebeck

The author has made an interesting contribution to the literature on “zero leak” seal concepts. A complete analysis which predicts leakage over the full range of possible operating conditions would be useful in evaluating the potential of this seal. Thermal coning could have a strong effect on the operation of a wide face seal such as this.

While the effect of pressure on leakage has been measured and is shown in the paper, I would like to know the effect of speed on leakage. What is the leakage rate at 0 speed, 50 percent, and 200 percent of the design speed. This would be valuable to get some idea how the seal works at off design conditions. Also, what happens if the seal is operated at zero pressure. Does it pump itself dry? Does the author have some specific applications in mind where this seal concept has some advantage over present designs?

R. F. Salant

This paper presents a very interesting and innovative seal concept, which may prove to be highly effective for certain special sealing applications in which it is critical to minimize liquid leakage into a gaseous environment.

It is important to note, however, that although this concept is referred to as a “zero-leakage” seal, it is more accurately described as a “zero-net leakage” seal, since liquid does leak from the high pressure side of the seal to the low pressure side, but is then pumped back to the high pressure side through the action of hydrodynamic forces. This distinction is significant because there are many applications in which the temporary residence of the sealed fluid on the low pressure side of the seal destroys the effectiveness of the seal. For example, if this seal were to be used in light hydrocarbon service (where there is a definite need for a zero-leakage seal), before the liquid which has leaked across the weir can be pumped back into the pressure cell, significant amounts of gaseous emissions could escape into the environment. Similarly, in the application of such a seal to a submersible motor, where it is critical to prevent contamination of the motor oil by water in the surrounding environment, even temporary residence of the water on the oil side of the seal will be sufficient to produce significant contamination.

It is also important to note that the dimension of the Rayleigh step \( (H-h) \) must be of the same order of magnitude as the desired film thickness \( h \) which is in the micron range. While this may present some manufacturing difficulties, a more serious consequence is the need to maintain the integrity of such a small-scale geometry over the life of the seal. It is likely that in some services the Rayleigh step will be destroyed by the buildup of deposits in the bearing area, by erosion, or by wear due to face contact during transient operation.

However, within the limitations imposed by the phenomena discussed above, it is likely that there are applications in which this new seal concept would be quite useful. Thus, further development of the concept should be encouraged. In this regard, it is suggested that the author develop the steady state operating characteristics of the seal, in the form of plots of film thickness \( h \) versus sealed pressure \( p_s \) for various operating conditions (speed, fluid) and seal designs. The development of such plots would also lead to the determination of the maximum sealed pressure without leakage, which could be compared with experimental results. The generation of such plots would be comparatively simple since the film thickness is determined by a force balance on the floating seal assembly, and the author has already developed expressions for the pressure distribution with the seal gap.

Finally, the author deserves congratulations for a stimulating piece of work which deserves further development.

Author’s Closure

I would like to thank the discussers for their interest in this paper.

1. Referring to Dr. Salant’s discussion, I would like to comment as follows:

1.1 The “Zero-Net-Leakage” term is appropriate as far as the bearing-weir barrier is concerned. Moving downstream to the static seal dam (see Fig. 6), no leakage of fluid to the ambient should occur since no pressure differential to drive it out is present, irrespective of whether the fluid is in the gas or liquid phase. However, the possibility of mixing the sealed liquid with an external liquid always exists. This is a “by-definition” characteristic of a non-contacting seal.

1.2 As pointed out in the paper, the stator was made of SST 440c hardened to 55 Rc which is not a highly wear-resistant material. It did, however, withstand some hundreds of hours of operation and tens of shutdowns and restarts under various conditions. No wear could be measured on the stator. This suggests that good wear-resistant material as hard tool steel (for example, M-42 hardened to 68-70 Rc) or tungsten carbide will endure a prolonged service. The rotor material should probably be made of a wearable material to absorb the ablation which results from transient face contact or abrasive operating conditions.

1.3 Figure 4 depicts the sealing performance of the seal as a function of the gap for typical oil, water, and two speeds. The curves are shown for optimum condition under particular design (\( L = 2 \text{mm}, s/a = 1.5, T = 0.1 \) and \( M = 10 \)) but can be plotted for any other choice from equation (3).
2. My answers to Dr. Lebeck's comments are:

2.1 The need for a complete numerical analysis which covers a wide range of operating conditions is recognized and has been referred to in the conclusions.

2.2 Wide faces typical of film riding seals (spiral groove, shrouded step or this seal) may develop thermal coning which harms proper operation. This problem could possibly be overstepped by adapting the following suggestions:

a. Make the rotor from an ablative material which will allow the surfaces to conform after a "running-in" period.

b. Have the stator face made thin enough to act like a membrane to compensate for coning. Heat transfer characteristics may also improve but manufacturing difficulties are likely to arise.

2.3 Sealing effectiveness goes up linearly with speed for a given gap. The gap magnitude will, however, increase (for a given balance diameter) as the opening force of the hydrodynamic bearings also increases linearly with speed. The balance between the two effects will depend upon the particular design parameters and should be carefully analyzed. At zero speed where no hydrodynamic action is present, the static forces should bring the faces to a "floating" contact; complete hydrostatic balance.

2.4 At zero pressure the seal will probably pump itself dry for the most of its interface area. The sealed fluid, if present at all, should just wet the innermost end of the bearings to lift the stator away from the rotor. Completely dry running capability will transform to a question of material combination. The described prototype was tested under almost zero pressure differential for short periods of time with no apparent damage.

2.5 Such seals may be employed in the sealing of high speed machinery where low leakage, minimum friction, and low maintenance are desired.

3. Dr. Etsion in his discussion, questions the validity of this concept as being "new". The claim is substantiated on the fact that the inlet flow to the pressure cell is deterred by the pressure effect compared to a pure shear driven flow which develops when no step is present—a case which is brought in reference [8]. His conclusion is that the pressure induced flow is not a basic mechanism but is rather a phenomenon which comes about solely due to the shear induced effect of hydrodynamic bearings.

3.1 First, I would like to point out that the concept described herein is the general case of the concept described in reference [8]; at the limit as $L_2 \rightarrow 0$ and $P_h - P_{\text{step}}$, the two configurations become identical. Hence reference [8] describes a particular case of this concept.

3.2 The deterioration in the inlet flow is not for vain. Rather, it serves a distinct purpose; the creation of an inherent stiffness effect, the importance of which is unquestionable.

3.3 A different way to describe the sealing capacity of a two dimensional pressure cell or seal (see Fig. 2) in terms of pressure induced flow is

$$P_h = \frac{L}{L + L_2} P_{\text{step}} \quad (P_a = 0)$$

which is most simple and certainly is not unfortunate or erroneous. In fact, it is only a question of the control volume over which one chooses to perform the flow balance that exhibits the particular form of solution.

3.4 Finally, to strengthen the notion that this type of flow is pressure induced, three different ways to achieve the same pressure profiles and hence, solutions, are shown in Fig. 11. The controversial hydrodynamic way of producing $P_a$ is shown in 11(a). The hydrostatic way is shown in 11(b) and a thermal way for compressible and otherwise adiabatic flow, is shown in 11(c). It is clearly depicted that the "step solution" is a way of achieving the pressure induced sealing effect but certainly is not the only one.