pressure fluid due to the circumferential pumping grooves. Since the stiffness of the lubricating film is high due to the Rayleigh-steps, this seal is very reliable not only on noncontacting operation but also on sealing.

References

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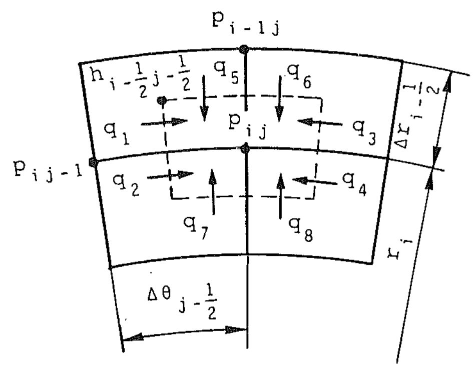

Fig. A1 Grid lines for numerical analysis

APPENDIX

Figure A1 shows grid lines for cell method [4] [5]. Mass conservation equation for incompressible fluid is

$$\sum_{k=1}^{8} q_k = 0 \tag{A1}$$

Sample expressions of local flow rates are

$$q_1 = \frac{h_{i-\frac{1}{2}\,j-\frac{1}{2}}\,\Delta r_{i-\frac{1}{2}}/2}{12\mu r_i \Delta\theta_{j-\frac{1}{2}}}(p_{i\,j-1}-p_{i\,j}) + \frac{h_{i-\frac{1}{2}\,j-\frac{1}{2}}\,\omega r_i \Delta r_{i-\frac{1}{2}}/2}{2}$$

$$q_5 = \frac{h_{i-\frac{1}{2}\,j-\frac{1}{2}}\,r_{i-\frac{1}{2}}\,\Delta\theta_{j-\frac{1}{2}}/2}{12\mu\Delta r_{i-\frac{1}{2}}}(p_{i-1\,j}-p_{i\,j})$$

Where $p_{i\,j}$ etc. are gauge pressure in this Appendix for simple expression. Substitution of $q_1 - q_8$ into equation (A1) leads to a linear function of nodal pressure with constant terms which are proportional to ω. This equation corresponds to Reynolds equation.

Equations of boundary condition

$$p_{i\,j}(r=r_o)=p_s$$

$$p_{i\,j}(r=r_i)=0$$

are linear functions of $p_{i\,j}$ with a constant term proportional to p_s. Then a set of Reynolds equations and the equations of boundary condition composes simultaneous equations of 1st order. Their constant terms are proportional to either ω or p_s, hence the solution $p_{i\,j}$ can be expressed by

$$p_{i\,j}=p_s \times p_{i\,j}(p_s=1,\ \omega=0)+\omega \times p_{i\,j}(p_s=0,\ \omega=1) \tag{A2}$$

$$f=\sum_i \sum_j (p_{i-1\,j-1}+p_{i-1\,j}+p_{i\,j-1}+p_{i\,j})\Delta r_{i-\frac{1}{2}}\, r_{i-\frac{1}{2}}\, \Delta\theta_{j-\frac{1}{2}}/4 \tag{A3}$$

is a linear function of the nodal pressure without constant term. Substitution of equation (A2) into equation (A3) leads to equation (2).

Leakage flow rate is calculated by

$$q=\sum_j (q_5+q_6)^{i=\text{const.}}$$

hence equation (3) is obtained in the same manner.

DISCUSSION

A. Lipschitz[1]

The authors of this paper are to be congratulated on providing additional test information on the behavior of "zero leakage" pumping seals. This work again proves that film riding seals may be designed for net zero leakage in spite of their non-contacting mode of operation. However, some points of qualification seem to be in place:

1. The pumping principle described herein has been suggested before by Etsion [2] and should have been treated as such.

2. To add the necessary film stiffness to the seal, Rayleigh-Pads were coupled into the pumping cycle, therefore reducing the pumping efficiency. Why not just decouple them from this cycle, having thereby high pumping as well as high film stiffness?

3. When both pumping and film stiffness are sought, the two-in-one mechanisms suggested in reference 3 may be used.

4. An interesting and very important feature of such seals is depicted in Fig. 9. At a certain speed (here about 1450 rpm) the seal loses its "zero leakage" characteristics. Apparently, under such speed, the Rayleigh-Pads open the operating gap to a magnitude where the pumping mechanism no longer overcomes the pressure drop. Designers of such seals should try not to lose the seal's pumping effectiveness by specifying large thrust bearing pads.

These pads, which provide excessive load carrying capacity, open the operating clearances to values where the pumping effectiveness drops drastically.

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J. A. Findlay

It is refreshing to review a paper that contains experimental measurements to support the analysis presented. Too many papers on seals seem to be mathematical exercises with neither verifying data nor recommendations as to how the results are to be applied by the seal designer. The authors are to be congratulated on this aspect of their paper.

A number of papers have been published on the application of Rayleigh steps to face seals. A comparison of the current work to these previous publications would enhance this paper.

The new variation which the authors present here is the addition of pumping grooves. The hydrodynamic action of these grooves pumps fluid from the low pressure side to the high pressure side when there is liquid on the low pressure side. This partially offsets the hydrostatic leakage through the seal.

Unfortunately these grooves allow leakage through the seal under static conditions. The alternate design presented in Fig. 10 has the added complication of a valve and a pipe that is not very practical. The designs presented by the authors have some definite disadvantages and other alternatives need to be investigated. The design presented is in some respects a variation of the spiral groove face seal design. Additional ideas may be developed by reviewing these concepts, particularly the spiral groove designs having static sealing dams at both the inside and outside diameters.

In many seal applications the same fluid is not present on both sides of the seal. The present design would cause an interchange of the two fluids to occur that may be undesirable. This potential application problem should be discussed. Application to situations where a gas is present on one side of the seal and a liquid on the other side should also be discussed.

I. Etsion[2]

The concept of a noncontacting pumping seal with means for hydrodynamic lift was described by this discusser in reference [2] several years ago. It is pleasing therefore to see a continuing interest by others and especially an effort to test the concept, for which the authors are to be congratulated.

Regarding the balance ratio, K, defined in equation (4), it should be noted that this is not the same balance ratio commonly used in seal design practice. Would the authors please clarify how their K ratio relates to a certain seal geometry and how can a constant K value be set and maintained in the experiment as shown for example in Figs. 8 and 9. Also, considering the results in Fig. 5, why was a value of $K = 0.598$ selected for the experiment if higher values seem to give better pumping.

Did the authors test their seal under different conditions of viscosity, speeds, and pressure than those indicated in the paper? The seal described in reference [2], for example, was successfully tested in water up to a speed of 10.000 rpm and pressure of 6 MPa. It seems that the full benefit of using noncontacting pumping seals will be obtained in high pressure high speed applications where conventional contacting seals give high friction torque.

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M. Kaneta[3]

The authors are to be congratulated for presenting, through their numerical and experimental investigations, the basic characteristics of a new type of noncontacting face seal.

The discusser believes that the sealing and lubricating performances are affected remarkably by the seal face configuration, i.e., the number of grooves, the depth and width of pumping groove, the depth of Rayleigh step, the outside and inside diameter of seal ring, etc. With respect to this point, the authors describe that dimensions, except the ratio of θ_s to θ_p, of the seal ring face used in their analysis are nearly optimized for sealing performance by trial and error method through numerical analysis. Could the authors find any relations or tendencies between the sealing and lubricating performances and the seal face configuration in the course of their analysis?

The values of α_s and β_s differ with each combination of θ_s and θ_p. Could the authors explain from physical viewpoint the reason for this matter?

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Authors' Closure

The authors would like to thank the discussers for their comments and interest in this paper.

Referring to Dr. Lipschitz's comment, the authors would like to point out that this paper presents the first attempt to predict sealing gap in a zero leakage seal. Full understanding of sealing performance is impossible unless the sealing gap is known.

1. It is described in the paper.
2. The area of a seal ring face is limited by its outside and inside radii. Under such limitation, this location seems to be best for sealing and lubricating performances. It is pointed out in the paper that the loss of pumping ability is typically only 3 percent, and it can be easily suppressed further if necessary.
3. It will be interesting to compare this seal with the one by Dr. Lipschitz. A seal designer will be able to select a type of seal according to operating condition.
4. The limit of shaft speed for zero leakage operation can be predicted and controlled by seal face design and balance ratio. If shaft speed exceeds its design value, the gap becomes wide and leakage appears. After that, the seal will operate safely due to increased cooling effect. The authors do not think that the characteristic is a weak point of this seal, because fail safe operation is most important for every seal.

As pointed out by Dr. Findlay, some additional dams will be necessary if static leakage must be suppressed. Before the application of such dams, one problem seems to be resolved. At static state, the gap at the dams is the same order width with surface roughness. If the clearance at the outside dam is slightly increased by thermal distortion or wear, the high pressure region extends to the low pressure side dam across the seal face and the gap opens widely. Then the double dam system loses its function.

If liquid-liquid interface exists in a seal, mixture seems to be inevitable due to circulating flow. Hence the interface should be kept out of the sealing gap.

In many cases, the sealing performance of a spiral grooved pumping seal surrounded with gas is limited by gas injestion or seal breakdown. On the other hand, liquid-gas interface is usually stable in this seal, because no diversing gap exists except in the deep radial grooves.

Dr. Etsion points out that equation (4) is different expression from the definition of K for a conventional mechanical seal. In a noncontacting seal, since film force f is balanced with closing force, the next equation is obtained:

$$f = \pi (r_0^2 - r_b^2) p_s + f_s$$

where r_b is balance radius and f_s is spring force. Substitution of this relation to equation (4) leads to the next well-known expression.

$$K = \{ (r_0^2 - r_b^2) + f_s/\pi \} / (r_0^2 - r_i^2)$$

Equation (4) may be regarded as a general expression of K for a noncontacting mechanical seal.

If the balance ratio is too high, wear rate may be excessive, since lift off speed is high and lubricating film is thin. So K should be chosen in order for the sealing ability to be compatible with noncontacting operation.

Since the main purpose of the experiment is to verify the theoretical prediction, fluid pressure and speed were not extended to extreme values. However, the authors believe that this seal has high potential for high PV operation.

In reply to Dr. Kaneta's question, if h is fixed, the following relations give the optimum values for sealing performance:

$$l_g/(r_0 - r_i) = 2.5, \quad a/(r_0 - r_i) = 0.6, \quad g/h = 3$$

where l_g is the length of pumping groove and a is the width of the groove. A value of $g/h = 1$ is recommended in reference [4] for lubrication and sealing performances. On the other hand, reference [A1] shows that $g/h = 2.5$ is the optimum value for load capacity. According to these results, optimum depth of a pumping groove is somewhat larger than that of a Rayleigh-step recess. However, the depth of both grooves were set to be equal for productivity.

A Rayleigh-step recess is not connected with low pressure side, while a pumping groove is connected with both sides. The average pressure in the recess is hence higher than that in the pumping groove. Accordingly, if θ_s is large and θ_p is small, α_s is large. The fluid leaks directly through the groove, while it is interrupted in the recess. So β_s is large, if θ_p is small and θ_s is large.

Additional Reference

A1 Gross, W. A., et al., *Fluid Film Lubrication*, John Wiley & Sons, Inc., 1980, pp. 225–241.