PERFORMANCE ANALYSIS OF
AXISYMMETRIC FLAT FACE MECHANICAL SEALS

by

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SUMMARY

Hydrostatic lubrication resulting from axisymmetric mechanical and thermal distortion of initially flat mechanical seal components is investigated. It is believed that this effect is responsible for successful operation of many conventional seals.

Interaction between mechanical and thermal distortions is discussed. It is shown that for an outside pressurized seal, full film hydrostatic lubrication is possible even if mechanical distortion under pressure is unfavourable, since favourable thermal distortion is then able to predominate. The effect of wear on this proposed “hydrostatic” mode of sealing is considered.

Accurate analyses of fluid mechanics, heat transfer and elastic distortion aspects of the hydrostatic sealing mechanism are described. These in turn enable iteration (by computer) to a full solution. Capabilities of the developed computer program in analyzing any axisymmetric flat face seal design and predicting performance, providing boundary conditions can be specified, are demonstrated. It must be determined from each result whether the initial assumption of full fluid film lubrication is indeed justified.

Performance correlation of reported experimental results with those computed for the same seal design is shown to be good, particularly when leakage and friction contributed by actual seal face imperfections are accounted for. It is concluded that the program will be of substantial benefit in predicting, explaining, and designing for performance.

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Analyse de la performance des joints mécaniques à face plate axisymétrique

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Résumé

La lubrification hydrostatique résultant de la distorsion thermique, mécanique et axisymétrique de composants de joints mécaniques initialement plats est étudiée. On croit que c'est grâce à cet effet qu'un grand nombre de joints classiques fonctionnent avec succès.

L'interaction entre les distorsions mécaniques et thermiques fait l'objet de commentaires. On montre que pour un joint extérieur pressurisé la lubrification hydrostatique complète du film est possible, même si la distorsion mécanique sous pression est défavorable, étant donné qu'une distorsion thermique favorable peut alors prédominer. L'effet de l'usure sur le mode de joint "hydrostatique" proposé est pris en considération.

On décrit les analyses précises ayant été effectuées sur les aspects "mécanique des fluides" "transfert thermique" et "distorsion élastique" du mécanisme de scellement hydrostatique. Elles permettent l'itération (par ordinateur) jusqu'à la solution complète. On démontre la capacité du programme d'informatique développé pour analyser n'importe quel concept de joint à face plate axisymétrique et pour prévoir sa performance, pourvu que les conditions limites soient spécifiées. On doit déterminer pour chaque résultat si l'hypothèse initiale d'une lubrification complète de film de fluide est de fait justifiée.

La mise en corrélation des performances des résultats expérimentaux et de celles calculées pour le même concept de joint, s'avère être bonne, particulièrement lorsque les fuites et la friction causées par des imperfections sur la face du joint sont prises en considération. On montre pour conclure, que ce programme présente un intérêt substantiel pour prédire, expliquer et concevoir la performance.

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# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>2. PRINCIPLES OF OPERATION</td>
<td>1</td>
</tr>
<tr>
<td>2.1 Theories of Sealing</td>
<td>1</td>
</tr>
<tr>
<td>2.2 Hydrostatic Effects of Distortion</td>
<td>1</td>
</tr>
<tr>
<td>2.3 Wear of Seal Faces</td>
<td>4</td>
</tr>
<tr>
<td>3. ANALYSIS OF PERFORMANCE</td>
<td>4</td>
</tr>
<tr>
<td>3.1 Assumptions</td>
<td>4</td>
</tr>
<tr>
<td>3.2 Fluid Mechanics Analysis</td>
<td>4</td>
</tr>
<tr>
<td>3.3 Heat Transfer Analysis</td>
<td>5</td>
</tr>
<tr>
<td>3.4 Elastic Distortion Analysis</td>
<td>6</td>
</tr>
<tr>
<td>3.5 Procedure for Seal Performance Solution</td>
<td>6</td>
</tr>
<tr>
<td>4. PERFORMANCE CORRELATION</td>
<td>6</td>
</tr>
<tr>
<td>4.1 Experimental Results</td>
<td>6</td>
</tr>
<tr>
<td>4.2 Computed Results</td>
<td>7</td>
</tr>
<tr>
<td>4.3 Correlation and Discussion</td>
<td>8</td>
</tr>
<tr>
<td>5. CONCLUSIONS</td>
<td>8</td>
</tr>
</tbody>
</table>
NOMECLATURE

A, B

seal shape parameters

h

face separation (fluid gap width)

\( h_{m} \)

face separation of fluid element, m

\( H_{i} \)

face separation at inside edge of seal

\( H'_{im} \)

face separation at inside edge of fluid element

M

balance ratio

p

pressure

\( P_{i}, P_{o} \)

inside and outside edge pressures

\( P'_{im}, P'_{om} \)

inside and outside edge pressures of fluid element

Q

leakage

r

radial coordinate

\( R \)

radius ratio (\( R_{i}/R_{o} \))

\( R_{i}, R_{o} \)

inside and outside radii

\( R_{m} \)

radius ratio of fluid element

\( R'_{im}, R'_{om} \)

inside and outside radii of fluid element

\( R_{s} \)

static seal radius (seal balancing radius)

W

heat generation

\( \mu \)

viscosity coefficient

\( \mu_{m} \)

viscosity coefficient of fluid element

\( \omega \)

angular velocity
PERFORMANCE ANALYSIS OF AXISYMMETRIC FLAT FACE MECHANICAL SEALS

1. INTRODUCTION

Mechanical face seals are used extensively to restrict leakage of pressurized fluids along rotating shafts. A conventional mechanical seal consists essentially of a pair of discs coaxial with the shaft, one of which rotates with the shaft while the other remains stationary in the housing. Lapped flat sealing surfaces in relative rotation contain the fluid whilst being lubricated by it (Fig. 1).

The stationary seal disc (stator) is usually supported rigidly. An axially floating rotating disc (rotor) is then relied upon to maintain a stable fluid film independent of shaft vibrations or movements. For fluid film stability a suitable non-linear radial pressure profile must be generated between the seal faces. The rotor must then be hydraulically balanced by appropriate static seal placement (Fig. 1).

2. PRINCIPLES OF OPERATION

2.1 Theories of Sealing

Several theories to explain sealing and lubricating performance of flat face mechanical seals have been proposed. Effectively these may be classified as hydrostatic or hydrodynamic. Physical principles underlying the major theories are listed below.

Hydrostatic:
(a) Axisymmetric radial taper, due to distortion.
(b) Surface tension effects (Ref. 1).
(c) Molecular adhesion.

Hydrodynamic:
(d) Non-Newtonian fluid effects (2).
(e) Thermal and pressure wedge effects (2).
(f) Surface waviness, possibly due to distortion (3).
(g) Viscous shearing of surface waves (4).
(h) Vibration, misalignment, and eccentricity (5,6).

Since commercial flat face mechanical seals are generally designed without regard to possible hydrodynamic effects, it is believed that only hydrostatic effects need be considered. Mechanical seals relying on rotation for lubrication should be designed deliberately to optimize the hydrodynamics, not to depend on possibly fortuitous but unpredictable effects. Of hydrostatic effects, molecular adhesion has been shown to be beyond possibility in seals of practical separation and surface finish (1). Surface tension may also be discounted in many practical seals because of the absence of liquid-gas interfaces either between the seal faces or at exit from the gap. Even with such interfaces surface tension can explain only “zero” leakage seals, yet for severe seal applications (high surface speed, large size, high pressure, and low viscosity sealed fluid) considerable leakage is often allowed in promotion of reliability and life of the seal. The axisymmetric radial taper effect remains the most definite, predictable and demonstrable theory of sealing for flat face mechanical seals (7).

2.2 Hydrostatic Effects of Distortion

Assuming then that flat face mechanical seals are favourably distorted in operation to create an axisymmetric sealing gap convergent in the direction of leakage (positive distortion) it remains to study the means of distortion and to analyze seal performance on this physical basis (Fig. 2). Both mechanical and thermal distortions will be experienced.
Balanced Seal

Axial Equilibrium:
\[ \int_{R_i}^{R_o} p \cdot r \, dr = \int_{R_i}^{R_o} P_o \cdot r \, dr \]
Balance Ratio:
\[ \left| \frac{\int_{P_o \cdot r \cdot dr}{R_o}}{\int_{P_o \cdot r \cdot dr}{R_i}} - \frac{R_o^2 - R_i^2}{R_o^2} \right| \]

Unbalanced Seal

Contact Pressure

Axial Equilibrium:
\[ \int_{R_i}^{R_o} p \cdot r \, dr = \int_{R_i}^{R_o} P_o \cdot r \, dr \]
Balance Ratio:
\[ \left| \frac{\int_{P_o \cdot r \cdot dr}{R_o}}{\int_{P_o \cdot r \cdot dr}{R_i}} - \frac{R_o^2 - R_i^2}{R_o^2} \right| = 1 \]

Figure 1 — Hydraulically Balanced and Unbalanced Mechanical Face Seals with Fixed Stators and Axially Floating Rotors
MECHANICAL DISTORTION

THERMAL DISTORTION

Figure 2 — Means of Axisymmetric Distortion of “Flat Face” Mechanical Seals

Initially flat surfaces of a mechanical seal subjected to pressure are known to distort, the amount being determined by the dimensions of each particular seal component, its means of support, and the magnitude of the pressure. Many seals are designed to minimize this mechanical distortion, but thermal distortion is often overlooked. It is believed here that both distortions are significant, and in fact that thermal effects may be the principal reason for continued success in severe applications of most flat face mechanical seal designs.

The proposed mode of operation of an outside pressurized flat face seal is described as follows:

a) If mechanical distortion is positive (see Fig. 3a) then hydrostatic operation with finite leakage will be demonstrated with the seal stationary. Subsequent relative rotation of the seal faces will cause some viscous heat generation in the lubricating film, and hence further positive distortion with increased leakage. The seal will reach a situation of stable equilibrium.

b) If mechanical distortion is negative (see Fig. 3b) the seal will close at the high pressure edge and no leakage will occur while stationary. On starting the surfaces will rub, generating frictional heat. Heating of the surfaces above ambient, giving favourable positive distortion, will cause heat generation rate to decrease until a balance is reached. Since pure mechanical friction between

even the most compatible materials at moderate sealed pressure must be many times more than is acceptable for practical seals, we may assume that full fluid film lubrication is often achieved and that friction may become purely viscous. Although dynamically induced, this will be classed also as hydrostatic lubrication. If the minimum face separation predicted on this basis is unreasonably small (say less than the initial flatness or surface finish of the seal faces, i.e., ~1 μm), then full lubrication will be shown to be impossible. The effects of boundary lubrication with some surface contact, i.e., increased friction and accelerated wear, will be evident, making the particular seal design generally less acceptable.

Turning now to the less common inside pressurized flat face mechanical seal, it is clear that any thermal distortion will be unfavourable for hydrostatic lubrication, tending to cause divergence of the sealing gap in the direction of leakage (negative distortion). For this reason all inside pressurized seals may be considered to be unsatisfactory unless pressure distortion is sufficiently positive to predominate.

Figure 3 — Positive and Negative Axisymmetric Mechanical Face Seal Distortion
2.3 Wear of Seal Faces

Loss of material of seal faces, loosely here termed wear, has not yet been considered. Only initial operation of the seal has been described. It is possible that an initially unsatisfactory seal may quickly wear to a satisfactory configuration, although this cannot justify its original design unless no alternative was possible.

The important aspect of wear is its effect on subsequent performance of an initially satisfactory seal. A radially tapered hydrostatic seal requires that sufficient taper and hence face separation be maintained. Being a complex phenomenon, wear will not be discussed in depth here. However, it appears reasonable that it is affected by both clearance and relative speed between the seal surfaces. Maximum wear at minimum clearance mitigates against maintaining the desired taper. Maximum wear for greatest relative speed favours wear at the seal outside diameter, desirable only for outside pressurized seals. Misalignment and tilting of the seal must also promote wear at the outside. A favourably tapered inside pressurized seal must therefore deteriorate with wear. However the outside pressurized seal should achieve an equilibrium tapered condition with uniform rate of wear across the sealing surface, even though the taper may be very small if percentage difference between the inside and outside radi of the sealing surface is small.

Seal faces may be purposely designed to wear differentially in the radial direction, either by material inhomogeneity or, more easily, by strategic reduction of surface area by mechanical means (drilling holes, etc.). Some effects of this are being tested by the author.

3. ANALYSIS OF PERFORMANCE

3.1 Assumptions

Considering an axisymmetric, outside pressurized mechanical seal with initially flat faces, and accepting the mode of operation just described, then several further assumptions will enable the fluid mechanics, heat transfer and elastic distortion problems of seal performance to be solved.

The usual assumptions of single-phase, viscous flow of an incompressible Newtonian fluid in a thin film will be made for the sealing gap. Validity of these has been demonstrated previously for practical seals in water or other essentially incompressible liquids of similar or greater viscosity (8).

For heat transfer analysis, boundary condition estimates for either heat transfer coefficient or temperature will be made for each seal component. The component will then be assumed to be comprised of a large but finite number of individual elements. Since the thickness and flow rate of the liquid are invariably very small, conduction and convection in this medium will be ignored. But changes in viscosity, caused by radial thermal gradients of the liquid in the sealing gap, will be considered between radial elements, since common liquids such as water show considerable viscosity variation with temperature.

Finite elements will also be assumed for elastic distortion analysis. This method is well suited to high speed digital computation. Further assumptions need only be made if the seal components are loaded (or supported) indeterminately, i.e. if the load distribution cannot be specified precisely.

3.2 Fluid Mechanics Analysis

Work has been reported previously for thin film, single-phase, laminar flow of a Newtonian fluid with constant density and viscosity in the axisymmetric, but radially non-uniform, sealing gap of flat face mechanical seals (9). Equations for pressure profile, leakage and viscous heat generation have been derived in terms of constants describing the sealing gap as follows,

\[
\text{Pressure profile, } p = P_i - \frac{2\mu Q}{\pi BH_i^3} \left[ 1 - \left( \frac{R}{r} \right)^{3B} \right] \quad \ldots 3.2.1
\]

\[
\text{Leakage, } Q = \left( \frac{P_o - P_i}{1 - R^{3B}} \right) \cdot \frac{\pi BH_i^3}{2\mu} \quad \ldots 3.2.2
\]

\[
\text{Heat generation, } W = \frac{2\pi\mu\omega^2}{H_i(B-4)} \cdot \frac{R_i^4}{R} \left[ 1 - \frac{R^{(B-4)}}{R_i} \right] \quad \ldots 3.2.3
\]
The sealing gap is defined by two unknowns, $H_j$ and $B$, ($R_j$ is known for a given seal) shown in Fig. 4, such that face separation $h$ at radius $r$ is given by

$$h = H_j \left( \frac{r}{R_j} \right)^B$$  ..3.2.4

where $H_j$, $R_j$ are separation and radius respectively at the inside edge of the sealing face, and $B$ is an index for the gap. It has been shown (9) that balance ratio $M$ of the seal is given only by $B$ and the radius ratio $R = R_i / R_o$ of the seal; hence $B$ can be determined from $M$ for a given seal,

$$M = \frac{2R^3B + 3B (1 - R^2) - 2}{(1 - R^3B) (3B - 2) (1 - R^2)}$$  ..3.2.5

As a simple extension for radially variable viscosity, the sealing gap may be split into a number of elements (Fig. 5), each liquid element being considered of constant viscosity. Now, representing the face separation $h_m$ in element $m$ by $h_m = h_{im}(r/R_{im})^B$, and substituting $r_{im}$, $B_m$, $R_{im}$, $R_m$, $\mu_m$, $\rho_{im}$, $P_{im}$ for $H_j$, $B$, $R_j$, $R$, $\mu$, $\rho$, $P_o$, equations 3.2.1, 3.2.2 and 3.2.3 can be used for pressure profile, leakage and heat generation in each element.

Figure 4 — Types of Axisymmetric Sealing Gaps
Described by $h = Ar^B = H_j (r/R_j)^B$

Figure 5 — Elemental Mathematical Description of a General Sealing Gap with Viscosity Variation.

To solve for a given distorted gap boundary but unknown minimum face separation, a value $H_j$ for the seal must be assumed. Using this, then matching separation and pressure between elements and equalizing leakage, a balance ratio for the complete seal may be calculated for comparison with the true value. Iteration enables the true minimum face separation to be found, giving true pressure profile, leakage and elemental heat generation for the given distortion.

3.3 Heat Transfer Analysis

Heat generated by viscous shear in the fluid film is conducted proportionally into and through the two seal components. Splitting each component into finite elements and assigning appropriate boundary
conditions for all surfaces but the sealing face enables the conduction problem to be solved numerically for any given input heat radial distribution.

Using the radial heat distribution found from fluid mechanics, the difficulty is to distribute the total generated heat from each fluid element correctly between the two seal components. Taking initial estimates of the proportions of generated heat conducted to each component from each fluid element, temperatures on the seal faces of both components may be calculated. Comparing these, new proportions may be estimated, and hence the correct proportions may be found by iteration, giving identical temperatures at opposing points on the two seal component faces. Full temperature distributions in each seal component will then be known for the given heat generation in each fluid element.

3.4 Elastic Distortion Analysis

Using finite elements again, together with mechanical (pressure distribution on seal face as found by fluid mechanics analysis) thermal (temperature distribution as found by thermal analysis), and boundary conditions, elastic distortion of each seal component can be found numerically. Combining the two distortions gives the resultant distorted gap boundary, as required for fluid mechanics analysis.

Fixed components of conventional seals are often supported indeterminately, as shown by the stators of Fig. 1. Distribution of reaction loads is then a function of distortion and flatness of both the component and its support, and further calculations and assumptions would be necessary for solution.

3.5 Procedure for Seal Performance Solution

For full solution of a seal problem, iterations must be made around the three basic analyses. Using an initial estimate of the distorted gap boundary, fluid mechanics, heat transfer, and elastic distortion may be considered in turn, generating new distortion. Repeating this cycle and testing for convergence to a solution enables all seal performance criteria to be calculated. Unrealistically small minimum face separation indicates that full lubrication will not be attained because of actual seal face imperfections.

All parts of the procedure are well suited for digital computation. A computer program has been developed accordingly, and is being used both to check existing seals and to guide the development of new seal designs.

4. PERFORMANCE CORRELATION

4.1 Experimental Results

Performance characteristics of a flat face mechanical seal have been reported in some detail by Watson and Roche (10). The design (Fig. 6) ensured that loads on the hydraulically supported rotor and stator were fully determinate, and also allowed balance ratio to be varied on test. Balance ratios from .73 to .79, speeds from 75 to 200 rad s⁻¹, pressures from 2.1 to 5.5 MN m⁻², and ambient temperatures from 25 to 72°C were tested.

![Figure 6 - Flat Face Seal and Test Rig Design for Experimental Correlation](image-url)
An average static leakage approximately equal to the leak of 48.3 mm$^3$ s$^{-1}$ at 74 rad s$^{-1}$ was recorded. For .79 balance ratio the experimental points are as shown. Static leakage was again close to that at 74 rad s$^{-1}$.

4.2 Computed Results

Input to the computer program includes material properties (mechanical and thermal), geometrical, loading, and thermal boundary conditions for both rotor and stator, together with fluid properties (viscosity-temperature), rotational speed and balance ratio. Centrifugal distortions are accounted for but are insignificant for normal speeds and sizes.

Heat lost by heating of leakage is assumed negligible in thermal calculations; justifiably because if significant heat is generated leakage must be small, implying little heat loss. (If little heat is generated, leakage will be large but total thermal effect will then be negligible.)

Three computed performance results for .76 balance ratio, and two results for .79 balance ratio are shown, joined by dotted lines in Fig. 7.

Doubts concerning the effect of roughly estimated thermal boundary conditions were allayed by repeating two results using estimated limiting values of heat transfer coefficients. Computed leakage and friction agreed with original results within 1% and 5% for the two cases.

Further doubts regarding imprecise material properties were investigated by repeating one result using material properties 10% different from those originally used for rotor and stator. Less than 1% change in performance was computed.

Additional information computed for each result includes full temperature distributions and deflections of both rotor and stator, temperature and pressure distributions of the fluid in the sealing gap, and gap width distribution including minimum face separation (Fig. 8).
Attempts to compute results for an unsuccessfully tested (10) material combination of alumina against Keewatin steel showed the tendency for unacceptable distortion of the components and the impossibility of full film hydrostatic lubrication. This agreed with and explained the measured pressure distributions and other test findings.

4.3 Correlation and Discussion

Comparing computed to actual results of Fig. 7 for each balance ratio, the performance curves of friction and leakage against angular velocity are seen to be closely parallel but considerably separated. This separation is believed to be due to inherent leakage of non-ideal seal faces, a conclusion reinforced by static leakage measurements recorded after each test.

Extrapolating computed leakage curves to zero angular velocity would indicate zero leakage. Subtraction of the similarly extrapolated zero speed test leakages from each experimental curve would then give very close agreement with computed curve over the entire speed range.

Lesser difference between experimental and computed friction is seen (Fig. 7) and expected, due to the reciprocal (rather than cubic, as is the case for leakage) dependence of friction on fluid gap width. Low test friction results, particularly at low speed, clearly correlate with observations of correspondingly high test leakages.

To explain “high” static leakage (~50 mm$^3$ s$^{-1}$) on test, it should be emphasized that a uniform gap width of only 1.8 μm is sufficient cause at 3.45 MN m$^{-2}$ pressure. Seal face imperfections may well account for discrepancies of this magnitude.

5. CONCLUSIONS

Full film hydrostatic lubrication by axisymmetric mechanical and thermal distortion of components has been shown to predict performance characteristics for flat face mechanical seals closely resembling those reported from experiment. The developed computer program is simple to use and to apply, provided that boundary conditions are determinate.

Computed results for any flat face mechanical seal design of interest may be used first to predict whether full film lubrication is possible, then, if so, to indicate probable performance. It has been shown that because of face imperfections, actual leakage is likely to be somewhat greater and friction less than that predicted.

The importance of thermal effects should not be overlooked. Thermal properties of the component materials, cooling, the effects of heat shock or lack of lubrication should all be considered. Modern coating techniques allow base material and face material properties to be optimized independently if desired.

Seal manufacturers should realize that although analysis is not the solution to many problems, it is becoming indispensable in such critical applications as nuclear power stations, where penalties for unreliability and rewards for improvement are so immense. Quality seals must be designed determinately.
REFERENCES


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