Principal research areas on mechanical face-seals

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A critical review of world literature on radial face-seals is presented in order to further understanding of seal operation and to obtain some elements of a design method for optimization of their performance. The principal areas examined are hydrostatic, hydrodynamic and boundary lubrication of the faces, and in particular cavitation and the influence of different constructive parameters on the dynamic behaviour of these seals.

Keywords: radial seals, design + performance, literature reviews, lubrication, cavitation

Mechanical face-seals, shown schematically in Fig 1, are employed to control fluid leakage in a wide range of devices: pumps, gearboxes, bearings, turbines. The fluids to be sealed could be a variety of liquids or vapours and the seals must operate in a diversity of pressure, velocity and temperature conditions. Generally mechanical face-seals fail by excessive leakage, and although the science of face-seals has made tremendous improvements since the early 1960's the performances and the life of these seals cannot always be predicted with any accuracy.

It is not possible to present in this study all the current research on mechanical face-seals. As a result the review is restricted to liquid face seals and not gas or vapour seals, or the materials and lubricant studies. A number of reviews have been published on this subject and we mention the more recent of them by Nau I and Etsion 2 and the general handbook of Mayer 3.

The mechanical face-seal design must determine the seal's characteristics in order to obtain the required performances like leakage, friction and wear. There are, however, several influences on the mechanics of seal operation. Hydrostatic, hydrodynamic and boundary lubrication regimes can set up in the interface region with the complicating problems of vapour transition and cavitation, in addition different operating regimes must be superimposed: motions, deflections and distortions of seal faces, thermal instabilities, etc.

Lubrication in liquid face-seals

The purpose of the initial works published on this subject was to explain the mechanism of generation of pressure and load in mechanical face-seals. Denny 4, Batch and Iny 5 and Nau 6,7 have shown that hydrodynamic pressure larger than hydrostatic is generated in the fluid film due to the effects of waviness, misalignment and vibration of the faces. Salama 8 demonstrated earlier (1949) that the hydrodynamic theory of thin films leads to the prediction of generated load and cavitation regions in films. Summers-Smith 9 has shown that under certain operating conditions face-seals do not operate with complete separation but in mixed friction.

For fully lubricated seals the thin fluid mechanics equation is well known 10-14. In general the seals require one floating element and the faces are not perfectly flat so the film thickness h is a function of radius r, circumferential coordinate θ, and time t. The solution of the complete equation provides:

- hydrostatic component due to radial taper of the sealing gap
- hydrodynamic and squeeze components due to distortion of the faces and motion of the floating element.

This complex equation needs to be solved numerically 11,15. However, many researchers have used assumptions which

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Fig 1 The two principal arrangements of mechanical face-seals
simplify the film thickness expression or Reynolds equation (Fig 2).

Metcalfe has considered the cases of axisymmetric configurations of sealing gap and dynamic tracking of floating element misalignment in a permanent regime according to experimental observations. Other authors have presented mathematical analysis which takes into account angular misalignment and waviness in the film pressure calculation. They use Reynolds equation in a permanent regime. Haardt and Etsion consider transient effects in a flat misaligned face-seal and solve the complete Reynolds equation with a numerical method using finite differences. They also solve analytically a simplified Reynolds equation with narrow seal approximation. The agreement between approximate and accurate solutions is quite reasonable for radius ratios greater than 0.8. Ruddy, Dowson and Taylor consider a seal with well aligned faces experiencing waviness, the floating element being able to vibrate axially. They solve the complete Reynolds equation in a transient regime by means of finite differences. In the study of performance of end face-seals with diametral tilt and coning, Etsion solves Reynolds equation separately for hydrostatic, hydrodynamic and squeeze effects. Because of the linear nature of Reynolds equation with narrow seal approximation the solution of the complete equation is the sum of the partial components.

Summers-Smith studied experimentally in 1961 the transition from hydrodynamic to boundary lubrication but theoretical studies on asperity contact for rough bearing surfaces were first carried out in 1969. Christensen and Tonder have proposed a model in which the part of film thickness due to roughness is taken into account in solving Reynolds equation. Patir and Cheng have introduced surface characterization by means of random process analysis in resolving Reynolds equation. Their developments led to the definition of flow factors which depend on roughness. The principal author who has applied these theories in the case of radial face-seals is Lebeck. The parts of the load supported by liquid film and by mechanical contact area are evaluated using a polynomial distribution to determine the asperity contact area. Models where the centre-line clearance is determined by the equilibrium between the load and the closing forces in a steady-state regime are defined.

In hydrodynamic mode, in diverging regions of a sealing gap, cavitation occurs as the pressure tends towards a critical value. This cavitation or phase change phenomenon was observed in mechanical face-seals by Orcutt, Nau, and Lohou who have used apparatus with a carbon seal ring mounted on a rotor and static plate-glass counter-face (Fig 3). Nau admits a distinction between vapour transition and cavitation. For vapour transition, the cavity extends to the edge of the film whereas the cavitation zone is restricted to the interior of the film. It is possible for the cavity to extend circumferentially beyond the limit of the divergent part of the film, well into the convergent region. Orcutt found that vaporization can lead to a greater load than a fully liquid lubricated seal. It is found that there is no radial flow in either direction when the film contains a continuous circumferential band of vapour and that the cavity can play an important role in the phenomena of inward or outward pumping.

Haardt has shown by means of experimental measurements that just before breakdown, in a local area, the thin film is subjected to tension (negative pressure). Hugues et al. have studied heat dissipation in fluid films and consequent change of phase. They determine pressure and temperature versus radius in the fluid film for flat and parallel faces and for conical faces in a permanent regime. Isothermal and adiabatic models are developed adapted to low or high leakage. They point out the possibility of stable or unstable values of the clearance.

In computing programs different boundary conditions can be used. Gämbel (or Half–Sommerfeld) condition and Swift–Stieber (or Reynolds) condition give an acceptable approximation on the pressure and the load fields but cannot provide a good evaluation of leakage. Boundary
conditions are satisfied when the principle of mass conservation is applied in the cavitation zone. Findlay\textsuperscript{13} carried out a theoretical analysis of the cavitating film in which the flow rates are equated between the upstream and downstream cavity boundaries. Nau\textsuperscript{1} has presented a study which ensures flow continuity both circumferentially and transversely on each elementary surface situated at the boundary of the film cavitation region. The boundary conditions of Jakobson and Floberg\textsuperscript{44} and the algorithm of Elrod\textsuperscript{45} are used by Lebeck\textsuperscript{22,23} to solve Reynolds equation in a permanent regime for wavy contacting face-seals. The conditions of Jakobson and Floberg introduce the proportion $\beta$ of cavity width which is liquid filled at the cavity boundary. Elrod proposes an algorithm based on the evaluation of convective pressure-gradient contributions to mass flow. In order to employ a differential equation valid throughout the lubrication region he introduces a cavitation index. After that he determines the free line of film reformation by resolving the differential equation which assumes mass conservation in the cavity cell.

In a recent paper\textsuperscript{46} Bayada presents a new variational form (taking into account the inlet flow parameter) for the journal bearing which allows the calculation of rupture and the reformation boundary with the help of the finite element method.

Dynamic behaviour of radial face-seals

Considered here are the principal phenomena dependent on shaft rotation; thermal and mechanical deflection of seal elements, motion of the floating element, coupling of deflection and motion with pressure generation in the thin film, phase change and cavitation and thermal dissipation. In addition, the effects of design parameters on seal performances are examined.

Denny\textsuperscript{4} has shown the associated variations of film thickness and fluid film pressure while Batch and Iny\textsuperscript{5} found that vibrations of seal faces can generate hydrodynamic pressure in the film. Metcalfe\textsuperscript{18} describes four modes of fully lubricated operation, the normal mode being tracking by the floating element of the fixed face (Fig 2(a), (b)). In order to study such phenomena Haardt\textsuperscript{10,11} proposes a model with hydrodynamic pressure generation in thin viscous fluid. He attempts to predict the behaviour of a seal under constant closing load, but with both fixed angular misalignment of the rotor face to its axis and fixed angular misalignment of the stator face to the same axis (Fig 2(c)). No account is taken of seal inertia or such factors as mount and spring. Using this approach he is able to calculate movement of the seal faces as the rotor turns, indeed axial vibration of the floating seal ring. He finds that this vibration is at shaft rotation frequency. The influence of angular misalignment on leakage, friction torque and vibration amplitude are shown. Experimental measurements are also made and show general agreement, the values being of the same order and showing the same trends as predicted.

Ruddy, Dowson and Taylor have published similar work\textsuperscript{23} in which waviness was accounted for and the floating element was vibrated axially with neglected inertia. The authors show the importance of the squeeze film effect and calculate the optimum amplitude of the waviness which assures sufficient load support; hence avoiding face contact while producing acceptable leakage. Their calculations are applied in the case of process pump seals studied by Summers-Smith\textsuperscript{47}. More recently Etsion published a paper\textsuperscript{24} in which dynamic behaviour of a non-contacting coned face-mounted stationary ring is studied, taking into account design parameters and operating conditions. The primary seal ring motion is expressed by a set of nonlinear equations for three degrees of freedom; axial and tilting motions (Fig 2(d)). In order to examine the dynamic behaviour of the seal, a flexibly mounted ring is disturbed from its aligned position and these equations are solved numerically at each time step. They allow identification of two dimensional groups of parameters that affect the dynamic seal behaviour (Fig 4). Stability maps for various seals are presented. The effects of various parameters are discussed like dimensions of secondary seal and rear face of seal exposed to hydrostatic pressure, stiffness and dimension of spring, inner and outer pressure, inertia of seal ring and angular velocity of shaft.

In the studies of dynamic behaviour of face-seals one can observe certain important ideas which have formed the subject of more specific works.

Balancing of the seal

The balancing of mechanical face-seals is characterized by the balance-ratio $B$:

$$B = \frac{R_o^2 - R_b^2}{R_e^2 - R_i^2}$$

where $R_b$, $R_e$ and $R_i$ are respectively the secondary seal, outer and inner radii (Fig 5).

Metcalfe has shown the importance of this coefficient for high pressure hydrostatic seals. He demonstrated theoretically\textsuperscript{16} and experimentally\textsuperscript{48} the existence of a critical value of $B$ below which the floating element is subjected to unstable vibrations and oscillations associated with a high leakage. This minimum coefficient value $B$ has also been mentioned by Mayer\textsuperscript{4} and Etsion\textsuperscript{24}. Meanwhile this coefficient is only defined for stable hydrostatic regimes.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig4.png}
\caption{Stability map for a coned rotor subject to an angular disturbance about a diameter (from Etsion\textsuperscript{24} reported by Nau\textsuperscript{19})}
\end{figure}
Distortion of faces and sealing gap profile

Watson⁴⁹ and Metcalfe¹⁶,¹⁷,⁴⁶,⁵⁰ have shown for high pressure hydrostatic seals that the radial profile plays a role in balancing (Fig 6). They admit in general that a divergent radial profile in the flow direction gives rise to a state of unstable vibrations and leads to contact of faces; while a convergent radial profile in the flow direction increases the functioning stability of the seal and reduces the leakage. Etsion⁵¹ also showed the role of the orientation of the face coning angles on the stability of the seals. The stability maps he has established²⁴ clearly indicate the influence of the coning angles but one has to remember that other parameters have equally important roles to play.

Since the mechanical or thermal distortions can modify the sealing gap profile and consequently the dynamic behaviour of the seals, the designer has to take account of these effects. In the early 60’s, Nau⁵² found that elastic distortions of the faces could lead to their damage. Metcalfe⁵⁰ has evaluated by means of numerical programs the deformations of the seal under pressure load and heat dissipation, taking into account the connections between the various components.

The phenomenon referred to as thermoelastic instability, has been investigated by Barnerjee and Burton⁵³,⁵⁴ for fully lubricated face-seals and for a thermal conductor sliding on an insulator. In the proposed model the rotating element has both fixed tilt and waviness. The stator is assumed to be gimbal mounted and to have inertial mass. Its motion is expressed for the case of small perturbations, Only the restrictive case where the stator precesses in synchronism with rotor rotation is considered in association with experimental observations⁵⁵. Hydrodynamic lubrication of a non-contacting face-seal is assumed with the narrow-seal approximation. In the regions of negative pressure, cavitation is assumed to occur and \( p = 0 \). Thermoelastic deflections are considered, the proposed objective of the study is to explain a coupled inertial-thermoelastic phenomenon which may lead to instability as observed in experiments.

Watson⁴⁹ and Etsion and Constantinescu⁵⁶ observed traces of wear on the faces leading to a modification of the radial profile of the interface and a better balance of the floating element. Lebeck²² has studied the case where asperity contacts of the faces appear in certain zones while the other interface zones are hydrodynamically lubricated. He showed that certain combinations between the waviness and tilt give rise to an optimal friction and wear reduction for a given leakage flow. These theoretical results are in good agreement with experimental observations⁵²,⁵³.

In a more recent paper Lebeck⁵⁷ describes various sources of waviness. He develops a geometrical analysis of measured waviness profiles (Fig 7) and a method of calculating the net waviness after the seal has been loaded. A simple calculation of the influence of waviness on hydrodynamic effect and leakage is presented.

Inertia, stiffness and damping

The elements of the seal form a vibrating system and one therefore has to take account of the damping and the stiffness of the O-ring secondary seals as pointed out by Metcalfe¹⁸, Kittmer⁵⁹ and Nau⁵² and also the spring stiffness and the inertia of the floating element as shown by Etsion²⁴ in his numerical model. This author has measured stiffness and damping coefficients for various elastomer O-rings in tests which simulate forced vibration conditions of a secondary seal in a mechanical face-seal⁶⁰.

To analyse simply the dynamic behaviour of mechanical fitting, Nau and Rowles⁶¹ as well as Green and Etsion⁶² have calculated the stiffness and damping of the film and related this to their mobility and coupling terms. Meanwhile the validity of these studies are limited to cases where no cavitation zone appears and does not destroy the symmetry of the pressure field in the thin film.

Other parameters

This essentially concerns the dimensions and the shape of the floating element: the works of Metcalfe¹⁶,⁴⁸ and Thomas and Will⁶⁶ show that a narrow seal leads to minimum heat loss and minimum leakage flow. On the other hand the studies of Etsion²⁴ show precisely that a small radius ratio improves the dynamic stability of the seal. In this last study the role of the spring position with respect to the system axis has been made clear since it contributes to the action of the restoring moments which oppose the oscillations of the floating element. The other important parameters on the hydrodynamic effect are the fluid viscosity and the shaft rotating speed.

Conclusion

The works examined give a set of different results and not a single theory of general seal design. The review relates principally to scientific works but does not contain much on the industrial tests and surveys. It is possible, however, to sort out the following principal points.

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Fig 5 Balanced and unbalanced mechanical face-seals with fixed stators and axially floating rotors (from Metcalfe¹⁶)
In the case of fully lubricated seals it is well known how to determine the pressure field in the thin film. In the case where there is contact of asperities, however, little research has been done and the modelling needs to be more precise. Also, there is no work that clearly describes whether the mechanical face-seals produce optimum performance in one or other of these modes of lubrication and whether the transition - functioning with contact — functioning without contact - can occur during seal operation. Finally, the correct solution of the leakage flow problem leads to the difficult determination of the rupture and reformation zones of the film.

In relation to the dynamic behaviour of face-seals and their performance, the research has clarified the important role of certain constructive or functional parameters such as: the balancing of the floating element, the shape of the interface, the thermal, mechanical and wear distortions of the faces, the damping and stiffness of the spring and of secondary seal the seal geometry.

The functioning of radial face seals involves several complex interacting phenomena and is governed by very many parameters. It appears necessary that experimental research be developed to enable us to measure more accurately the real functioning of these seals. Also, the establishment of a satisfactory method of design calculation of mechanical face-seals requires the use of computer programs to closely model seal operation.

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Fig 6 Distortion of seal-faces in the radial direction (from Metcalfe 16)

Fig 7 Surface profile and seal gap in the tangential direction (from Lebeck 20)
Tourniere and Frene – research areas on mechanical face-seals

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