LEAKAGE, WEAR AND FRICTION IN THE MECHANICAL FACE SEALS ANALYSED BY FEM

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ABSTRACT
The behavior of a face seal is determined by the complex interaction of a number of factors. Advantages are usually attained at the price of disadvantages in the other directions. For example if the roughness is constant, an increase of the contact pressure reduce leakage, but the wear and frictional heat increase. As against this, increasing leakage losses can reduce the friction and the heat production, but the effectiveness of the unit as a seal is reduced. Again, a high friction may not only lead to increased wear but also, due a thermal distortion, to considerable leakage losses, or it may cause the seal to break down because of a thermal stress cracks.

By appropriate seal design, choice of materials and type of seal arrangement, individual requirements such as minimum leakage, maximum life or minimum friction can be met.

The paper presents the FEM (finite element method) analyze of the leakage and friction rates of a mechanical face seal for the automotive industry.

KEYWORDS: face seal, FEM, leakage, wear, friction.

1. INTRODUCTION
A seal (packing) is a combination of components which provides for tightening effect.
The sealing capacity is the main characteristic of the performance of a seal; it is measured by the mass of a substance escaping through the seal (leakage rate).
Tightness specifications are dictated by the reliability and service life requirements for a particular piece of engineering equipment.
The seal performance factor is expressed by the relationship

\[ i = 1 - \frac{m}{m_p} = 1 - \frac{1}{m_p} \int_{t_0}^{t_r} m(t) \, dt \]  

where \( m \) and \( m_p \) are respectively the actual and permissible masses of leaking substance over the rated service life \( t_r \), \( m(t) \) is a function describing the variation of leakage with time, \( t_0 \) is the moment at which the seal begins to leak, determined by a particular tightness checking method.

If the leakage mass \( m < m_p \) over the time \( t = t_r - t_0 \), then the factor \( i > 0 \). In the inoperative-failure conditions when \( m > m_p \), the factor \( i < 0 \).

The performance of seals is characterized by the degree of tightness, service life, power losses, by the extent of damage to the contacting surface in operation, etc. the degree of tightness, wear life \( t_w \), and performance factor \( i \) are the most important characteristics of seal performance. In addition to the above factors, temperatures, whose level is determined by their joint action, also affects the performance of dynamic seals. Whereas temperature has the major influence on the frictional effects in the contact area, the leakage is caused by reduction in the contact pressure and distortions in the geometry of the rubbing surface due to wear, increased thermal deformations, etc. in some instances, these factors are interdependent.

The service conditions of sliding contact seals in machinery, determined by combinations of the above factors, are very diverse.

For transport the general characteristics of seal operating conditions are:

Operating conditions:
- \( p_s \) (0-10), kgf/cm²,
- \( v \) (5-10), m/s,
- \( p_s \cdot v \) (80),
- fluid to be sealed: - non-aggressive; temperatures (– 60 - + 110)°C (oil, kerosene etc.)

Reliability requirements:
Average; low-rate leakage (e.g. 0.005g/h) are admissible.
Required service life, \( h \) (approximately): 10.000.
The static and sliding surface of the traditional shaft is closed by a static seal such as an O-ring. The axial leakage path between the floating ring and rotating floating ring against a fixed counterface.

Require good friction properties: high wear resistance and low coefficient of friction.

All sealing rubber applications must often be combined; some of them are mutually incompatible. Anyhow, all sealing rubber applications require good friction properties: high wear resistance and low coefficient of friction.

For the face seals design calculation methods have been devised for the assessment of fluid pressure (with regard to out-of-squareness of the faces and to pressure in the clearance), behavior of the fluid sealed in face clearances, hydrodynamic effect for the sealing rings, deformations of the rings due to pressure and temperature, and also temperature fields in the rings of the rubbing pair.

Depending on application, sealing rubbers should be strong, heat resistant, cold resistant, or resistant to chemical attack. These characteristics must often be combined; some of them are mutually incompatible. Anyhow, all sealing rubber applications require good friction properties: high wear resistance and low coefficient of friction.

In a face seal (Fig. 1), an axial force presses a rotating floating ring against a fixed counterpart. The axial leakage path between the floating ring and the shaft is closed by a static seal such as an O-ring.

The static and sliding surface of the traditional stuffing box are effectively interchanged, with the least one flexible member such as a diaphragm, bellows, elastomeric seal, or spring (Fig. 1).

However there are various causes for face seal leakage, that normally takes place through the radial seal gap formed by the two sliding surfaces. Leakage appears in most cases due to the distortions caused by stresses. So it is of great importance to know the stress distribution along the interface.

End face type seals have been used as sealing devices of water pumps in cooling systems of automotive engines. From the beginning, various kinds of troublesome problems have occurred in the practical market, i.e. excessive wear of carbon surface, cracks on ceramic surface, spring breakage, bellows breakage, excessive friction, separation at bounding parts, squeaking phenomena and etc.

Through the investigations on causes and counter measures of the problems, face seals of this type have been greatly improved in the sealing performance.

The excessive abrasion of mechanical seals is caused by foreign particles in liquid to be sealed. As to the ringing phenomena, two types were already reported. One was caused by the boiling of liquid to be sealed near rubbing portion and another was due to the stick-slip between the sliding surfaces.

Foreign particles collected from the cooling system of automotive engines are composed of cutting chip of cast metal, aluminum and molding sand including several kinds of crystalline materials, i.e. SiO₂, Al₂O₃, Fe₂O₃, Cu₂O and etc. As the best countermeasures is to reduce the amount of particles damaging the rubbing surface. One can advice pump makers to flush engine blocks completely, improve filters and change liquid periodically. On the other hand, the countermeasures of mechanical seals are considered to establish uniform thin hydrodynamic films by decreasing the thermal distortion of sliding materials.

In particular, carbon materials had been improved positively to have the characteristics of higher thermal conductivity and lower coefficient of thermal expansion.

Only the primary leakage through the seal gap between the faces of the seals will be considered on the assumption that a hydrodynamic film exist in face seals and that the leakage can be calculated in accordance with the known equation for laminar flow through a radial annular gap.

\[
Q = \frac{\pi d_m h_0 (p_1 - p_2)}{12 \eta b}
\]  

Since in practice the liquid film thickness \( h_0 \) is seldom constant and the actual gap form can deviate considerably from the assumed parallel gap because of temperature differences in the ring.

At present manufacturing specifications prescribe polished surface of \( R_a = 0.015-0.5 \mu m \) and a flatness of 2-3 light bands. An optical interference testing instruments with a monochromatic helium light source (1 helium light band = 0.3µm) is used for measuring the flatness.

When a face seal with lapped surface is brought into operation, the face profile alters suddenly,
depending upon the torque applied and the axial and radial forces and temperature gradient set up. The rings distort (Fig. 2) and the gap can become convex, concave or inclined with contact on either the internal or external periphery. If now the operating condition, wear will cause the faces to return gradually to the parallel if sufficient time and contact pressure are available. The necessary running-in period above all depends on the magnitude of the distortion, the net contact pressure and the wear resistance of the material combination. In face seals this time can be from a few minutes to several months. If, however, during the starting process the surface are already damaged, e.g. by thermal stress cracks, seizure or by exceeding the thermal limits of the materials (melting of the metal impregnated carbons) of if the sealing surface are stripped, then the seals cease to function. The distortion occurring in face seals, especially under intermittent operating conditions, make it particularly difficult to calculate the leakage accurately. In all, three main factors influence the distortion of the sealing gap, the axial forces, the radial forces and the temperature gradients.

Mechanical forces which tend to alter the shape of the seal gap are mainly controlled by the area ratio, \( A \), and by the sealed pressure and its direction of application. Due to distortion, face seals tend to make contact on either the outside periphery \( D \) or the inside periphery \( d \) (Fig. 3a and b) which, of necessary, considerably alters the load. By suitable dimensioning and selection of materials it is possible, in theory, to calculate the distortions and keep them within acceptable limits. In practice, however, these calculations are rather complex.

Apart from the geometry of the seal surface also has a considerable influence on the behavior of a face seal. As the roughness increases so does the leakage and in the case of balanced seals \( (A < 1) \) there is a danger of the seal faces parting.

In normal seals, which do not have radial grooves in the sliding faces as do hydrodynamic mechanical seals, no hydrodynamic pressure is generated in the seal interface. However, mechanical and thermal distortion can result in a hydrostatic pressure increase in the gap. With mechanical seals of \( A \leq 1 \) and a parallel gap, solid contact normally exists, hence permeability of the interface to leakage is determined by the magnitude of the surface roughness.

The geometry of the seal gap and roughness of the mating surface can change under operating conditions. This in turn influences the closing force \( P \), the gap pressure \( p \), the leakage losses \( Q \) and even the friction condition. The leakage can thus alter under nominally constant operating conditions. In face seals, the condition of the non-pressurised lubricant in the seal gap (see X zone in figure 4) is of importance because of the stability of this condition and the low leakage loss.

There are a number of factors that can lead to failure of the sliding faces of a seal, such as corrosion, overstressing, thermal overload and wear. Wear, in particular, can reach disastrous proportions if unsuitable combinations of face materials are used. In practice, a badly designed seal that has a good combination of face materials is often preferable to a better design with unsuitable face materials.

It is exceptional to use purely metallic material in couples for face seals, since there is not always a hydrodynamic film of lubricant separating the faces and solid contact must be catered for. All-metal couples are impracticable because of their poor emergency dry running characteristics.

In selecting sliding materials, consideration should be given to operating conditions, ease of manufacture and material costs. The chemical activity as well as the physical and mechanical properties have to be considered.
Characteristics of artificial resins are their low modulus and high wear resistance, even with water, when run against suitable materials such as iron, cast chrome, bronzes and ceramics. The poor thermal characteristics, such as very high thermal expansion coefficients, low conductivity coefficients and restricted range of permissible working temperatures are disadvantages.

The resistance to wear can be strongly influenced by manufacturing processes, such as moulding pressure, heating temperature and heating period.

Carbon materials can be divided into hardened amorphous carbons, carbon graphites and electrographites. A wide range of physical and mechanical properties can be produced depending upon materials, annealing time and temperatures. Hardened amorphous carbons are characterized by their high strength (they can often only be machined by grinding) and their basically low thermal conductivities. By contrast the electro-graphites are of low strength but have considerably greater conductivities. The carbon-graphites have intermediate properties.

After annealing, a volume porosity of 10-30% remains in synthetic carbon. For mechanical seals the residual porosity after impregnation should be below 2%, so that it is sufficiently impervious to liquids. Where gases have to be sealed the value should be below 1%. Hence carbons and graphites are impregnated with synthetic resin, metals, glass, PTFE dispersions and salt solutions.

In general it can be said that for synthetic carbons the hardness, strength, elastic modulus and thermal expansion coefficient are raised by impregnation.

Under working conditions these differing expansion coefficients with the varying conductivity coefficients, lead to the formation of micro-asperities and valleys which constitute lubricant reservoirs and give rise to hydrodynamic effects. These result in a reduction of friction and wear. This is particularly the case with water and water solutions where good heat transfer characteristics and higher temperature gradients result.

Because of the rational manufacturing process they are no more expensive than plastics.

Nickel-based alloys are used with some reservations since groove formation leads to severe wear.

Chrome, cobalt and their alloys, e.g. ferritic cast chrome, stellite and other plating materials, are characterized by high wear resistance and good chemical resistance.

In hardened steel, free carbine material residue gives a marked wear reduction due to improved boundary lubrication and the formation of lubricant pockets. The same applies to cast materials. Non-homogeneous metal alloys, for example lead-bronzes (H > 135 BH) and some cast chromes, can also favor the formation of hydrodynamic lubricating effects because of the mechanical and thermal microdistortions in the seal surface. The strength of metallic sliding materials is better than that of plastics and carbons.

Despite machining difficulties metal oxides are used as sliding face materials for their extreme chemical inertness and wear-resistance. These materials are mostly very sensitive to impact loads and thermal shock.

Tungsten carbides with cobalt binding agents are used most commonly but these can be attacked by chemically pure water and oxidize at a temperature as low as 600ºC. Tungsten carbides have been effective for sealing liquid oxygen and nitric acid. Titanium carbides have a low thermal conductivity coefficient, but can operate at very high temperatures and are resistant to oxidization. Recently hardenable titanium carbide, with iron or Cr-Ni steel binding agents, have appeared on the market and these offer the advantage that the rings can be machined to the required dimensions and hardened later.

Metal carbides are often used as a thin layer formed by the flame-spraying process.

The sealing face wear can be divided up into five groups.
1. Adhesive wear from attractive surface forces.
2. Abrasive wear from the filling effect of two rough surface or foreign bodies in the seal gap. The extent of the abrasive wear is greater than the adhesive wear by several orders of magnitude.
3. The corrosive wear which is initiated by chemical attack and especially by high interface temperatures.
4. Surface wear which can be caused by either thermal stress or fatigue failure in the sliding faces.
5. Radial wear which can be caused by the erosive effect of liquids and gases at high flow rates.
In a seal the hardness, rigidity, sliding characteristics and heat conduction properties have a further influence on the resistance to wear. In the range of the adhesive wear, a decreased wear rate is to expected as rigidity is increased.

Temperature increased often lead to reduction of rigidity, hardness and also to destruction of the impregnation. They can also cause the lubricant film in the seal gap to vaporize.

Efficient heat dissipation must be provided. Excessive temperature rises can not only lead to high wear but also to increased leakage.

By selecting materials with appropriate thermal conductivity coefficients, by additional cooling, lubrication and load ‘balancing’.

The sealing medium also has a considerable influence on the life of a mechanical seal.

A material pairing may show considerably lower wear values when using water than using oil.

It is known that strong axial vibration can cause the sealing surface to break down very rapidly.

Mounting seals on elastomeric rings has a very beneficial effect on wear because of the damping action of the elastomer.

Radial vibrations are more tolerable with grooveless sliding faces than axial vibration, since the latter have a direct effect on the sliding face contact and the wear. With strongly vibrating, e.g. drilling, equipment, better springing or spring loaded construction have greatly improved both sliding and counter rings. With rising amplitude and frequency, wear and leakage will increase. Good design should aim at improved support and balancing of shafts etc., and thereby minimize oscillation of the machines.

The formation of thermal stress cracks on the sliding faces. This can be caused by a momentary overload due to dry running, as a result of failure of the coolant or by a large load or speed variation. The cracks lead to increased wear and with balance of seals can cause the seal faces to part.

Can be largely avoided if the physical and mechanical properties of the faces and the operating conditions are considered carefully enough when the seal is designed.

The face materials must be selected bearing in mind the chemical activity of the medium to be sealed. It should be noted that at higher temperatures the corrosion may be accelerated so that the sliding surface in particular, having considerably higher temperatures than the bulk of the sealed fluid, corrode to a considerable extent.

What life can be expected from a face seal under certain operating conditions? If the wear is mainly adhesive and if operating conditions are constant and known, then wear values $A$ obtained under similar conditions enable the life $F$ to be closely calculated from $L = a/A$.

Often the durability of a seal is determined not by the wear of the seal alone but by the resistance to ageing of any used elastomer.

Intermittent operation as well as increases of contact pressure, friction coefficient, sliding speed and temperature will reduce the life.

Since the effects of adhesive wear, abrasive wear, corrosive wear and erosive wear, let alone vibration, temperature and material effects, can be cumulative, data in respect of seal life are always subject to a considerable uncertainty factor.

3. FRICTION CONTACT PROBLEMS
   BY FEM

A face seal behavior can be simulated by FEM. For each dimension the basic structure is represented by the rotating floating ring and the fixed counterpart.

Both geometrical, mechanical and thermal loading are axi-symmetric.

The floating elastic ring has 5 degrees of freedom and loads are: hydrodynamic, elastic, the inner pressure etc.

The simulation consists in solutioning the moving equations of the floating ring associated with the interaction between the two rings due to the rubbing surface contact. The FEM analysis allows the detection of the instabilities and the determination of the functional factors of a face seal.

The performance factor is leakage.

The axi-symmetric elements can be defined in $xy$ or $xz$ plane of the global coordinates system. The post-processing is performed for the whole volume of the structure so it is enough to study a section that contains the revolution axle of the body (for the entire structure).

Due to symmetry circumferential displacements are not tacking place.

The displacement vector has so only radius components $u$ and axial components $v$. so the degrees of freedom attached to the element are the nodal values of those displacements. The approximation function express these variation of the components on element domain:

$$\{u(r,y)\} = [N]\{\delta\}$$

where $\{\delta\}$ is the nodal displacement vector.

If the natural coordinates $s, t$ are used, the relation becomes:

$$\begin{bmatrix} u(s,t) \\ v(s,t) \end{bmatrix} = \begin{bmatrix} N_1 & 0 & N_2 & 0 & ... & N_n & 0 \\ 0 & N_1 & 0 & N_2 & ... & 0 & N_n \end{bmatrix} \begin{bmatrix} u_1 \\ v_1 \\ ... \\ u_n \end{bmatrix}$$

where $n = 4$ for the quadrilateral three-dimensional elements.
3D, 2D, 1D models in cylindrical coordinates.

The axi-symmetric problem is not alike the plane deformation situation because the normal specific deformation on the working plane of the section $\varepsilon_0$ is not zero and it must be explicitated in the specification vector $\{\varepsilon\}$.

Its value is:

$$\varepsilon_0 = \frac{u}{v} \quad (5)$$

The relation between the specific deformation and displacements is:

$$\begin{bmatrix} \varepsilon_r \\ \varepsilon_y \\ \varepsilon_0 \\ \gamma_{ry} \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 1 & 1 & 0 \end{bmatrix} \begin{bmatrix} \partial u \\ \partial r \\ \partial y \\ u \end{bmatrix} \quad (6)$$

The tension/specific deformation relation is:

$$\begin{bmatrix} \sigma_r \\ \sigma_y \\ \sigma_0 \\ \tau_{ry} \end{bmatrix} = \begin{bmatrix} 1 & -v & v & 0 \\ v & 1-v & v & 0 \\ 0 & v & 1-v & 0 \\ 0 & 0 & 0 & \frac{1}{2} \end{bmatrix} \begin{bmatrix} E \\ v \end{bmatrix} \begin{bmatrix} \varepsilon_r \\ \varepsilon_y \\ \varepsilon_0 \\ \gamma_{ry} \end{bmatrix} \quad (7)$$

If $[J]$ is the Jacobian transformation matrix and $[J]$ its reverse, then the displacement derivatives $u$, $v$ reported to the global system $r$, $y$ are like in relation (8) where derivates vector related to the natural system $s$, $t$ are calculated with relation (4).

Relation (9) has a general form and it is available for linear finite elements and for superior degree ones too.

Particularly for face seals $n = 4$, the element has four nodes in $r$, $y$ plane.

Relations (6), (8) and (9) allow the matrix $[B(s,t)]$ computation from the relation $\{\varepsilon\} = [B] \{\delta\}$.
The [B] matrix has four lines in state of three from the plane problem. The elasticity matrix is 4x4 too (relation (7)).

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The basic nonlinear solution approach involves a series of incremental solutions. The load is applied in increments. During each increment, a solution is „predicted” using the current state (stiffness and load increment). Depending on the type of nonlinearity, a force imbalance or „residual” is created during an iteration where nonlinear behavior occurs. Solution iterations are required to balance equilibrium (correct) for unbalanced forces.

The iterations continue during an increment until the convergence criteria are satisfied. Once convergence is satisfied a solution is obtained for the increment and the solution progresses to the next increment using this „predictor-corrector” method. An advancing scheme is a strategy used to apply loading in a logical and controlled manner. Advancing schemes are used as a part of a strategy to obtain an efficient, numerically stable solution.

The most common advancing scheme is the application of loads or displacement increment is important, especially when small values of force or displacement cause a large change in response.

### 3. FRICTION CONTACT PROBLEMS BY FINITE ELEMENT ANALYSE

When friction is considered, the tangential displacement in the interface implies energy dissipation. The problem is solved by incremental computation. Portions of the structure can have areas of gaps which can open and close or slider in relation to each other. Similarly, boundary conditions can change during a nonlinear analysis.

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### 4. MODELLING, AND ANALYSING BY FEM OF A „EFS 103-S” MECHANICAL SEAL

The face seal has a geometrical axis of symmetry so it was modeled as a structure with
axisymmetric elements. It was meshed in 74 elements. The material of the static ring and the rotating one is a 40C130 stainless steel, the pressure of the fluid (boiled water) is 1 Barr. The motion parameter \( \omega = 150\text{rad/s} \) and the friction coefficient is estimated at 0.15.

With the method described in §2 we were obtaining the distribution configuration of the equivalent stresses under the form isosurfaces (Fig: 9) expressed in [MPa], that reveals the most solicited zones of the mechanical face seal. The configuration of those quantities is an expected one with a maximum to the inner radius of the rings of the mechanical seal.

5. CONCLUSIONS

The FEM analysis is the only way to re-establish, by mathematical means, the rubbing contact pressure distribution from a face seal interface.

This allows the physico-mechanical and functional influence factors evaluation on the sealing performance of a seal.

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