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APPLYING HYDROSTATIC LUBRICATION AT MECHANICAL FACE SEALS

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Abstract: *From the engineering point of view, the mechanical face seal has two main constituent parts: one of them is fixed by the shaft and it's rotating at the same time with it and the second one is the stator, which is fixed in the carcass. The two parts together form the primary mechanical seal. They can be rigidly tied by their support, but most of the time they allow one of more liberty degrees. These ties represent one or more secondary mechanical seals and also elastic elements.*

The optimization of a mechanical face seal means the satisfaction of two antagonistic conditions. One is trying to reduce to the minimum the leakages, even to make them vanish and also trying to limit the friction and the wear. The second one presumes that a lubrication normal operation of the HS, HD, THD, EHD and/or M surfaces is ensured.

This paper takes into consideration just the hydrostatic lubrication from the primary mechanical seal, along with the purpose to place at projectors disposal different relations for the functional parameters (the opening axial force, the loss debit).

Key words: *Face seals, hydrostatic lubrication, axial force, loss debit, pressure.*

1. INTRODUCTION

The summary description made previously is common for the majority of practical applications. The mechanical face seals are used in all activity sectors: extractive industry, energetic industry, transportation industry, pharmaceutical industry, biochemical industry, etc. The mechanical face seals are the essential element of sealing for all the performances rotary machines (pumps, compressors, turbines, agitators, dryers, etc.). The exploitation of these machines imposes very severe conditions.

Since the first mechanical face seals design in 1983 until now these mechanical seals have developed at stress factors pv > 5000 bar m/s.

The mechanical seal contact surfaces (rotor-stator) are called active surfaces or friction surfaces and form together the mechanical seal clearance. The rotor is flexible fitted in its support and it has an axial and angular mobility in order to assure a contact with the stator and to balance some manufacture and assembling errors. The elastic element can be a spiral or annular spring which is concentric disposed with the shaft or more spiral springs which are uniformly distributed on the circumference. The secondary mechanical seals have as main function a seal of the respectively elements and, alongside the spring, the rotor fitting flexibility.

The normal working of a mechanical face seal supposes the presence of a film between the rotor and the stator. As regards of the fluid film thickness, there can be talking about three possible situations in mechanical seal clearance:

- a continuous fluid film on the whole sealing surface;
- a frontal contact, fluid free;
- a partial fluid film and a directly contact.

These three cases may appear during a mechanical face seal operation, because of the occurring of some speed and pressure variations, operation conditions stopping, running and changes.

At any time of the mentioned situations, two main forces interfere and have to balance themselves so the mechanical seal is efficient:

- a force which chase to decrease the surfaces clearance; it comes from the hydrostatic pressures resultant of the sealed fluid on the rotor surface on the axial direction, plus the elastic element force, called <u>closing force</u> or <u>external force</u>;
- a force given by the pressures resultant from the fluid film witch exists in the clearance, called <u>opening force</u>.

Besides these two forces, it usually appear the secondary mechanical seal friction force, the centrifugal force created by the liquid rotation inside the clearance and the friction force from the fluid film. In order to simplify the numerical calculations, these can be disregard.

In order to balance the rotor it’s necessary to respect the following condition:

$$F_{ext} = \int_S p ds \,, \tag{1}$$

The nature of the film pressure between primary mechanical seal surfaces has been the subject of lots of researches [1], [2] and most of the theories sustain the following bearing ways: hydrostatic, hydrodynamic, elasto-hydrodinamic, blended.

It hasn’t been yet reached to a unanimous conclusion regarding a single bearing mechanism, although it has been studied some hypothesis. Most of the theoretical studies consider that the film thickness on the axial direction remains changeless. That’s way we consider that the prevailing opening force has a hydrostatic nature and that’s a good reason for it’s correctly calculation and taken into consideration in design.

2. THE FUNCTIONAL PARAMETERS CALCULATION FOR A PRIMARY MECHANICAL SEAL WITH A CONICAL CLEARANCE

In field work is less probable for the rotor and stator mechanical seal surfaces to be perfectly plane and parallel. We will take into consideration the general case of a conical clearance. This mechanical seal space profile may be convergent or divergent (fig.1). A convergent clearance in the sealed fluid flow direction has positive effects on the mechanical face seal durability. The radial coning comes from the following sources: the initial coning, wear, thermal deformation or sometimes even from the design.

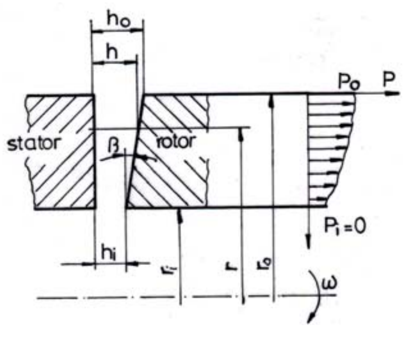

Fig.1 The mechanical face seals with radial coning.

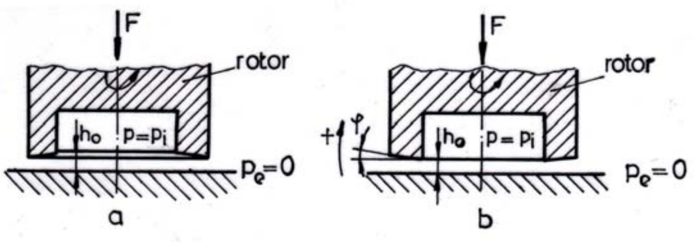

Fig.2 Analytical model for the functional parameters application.

The analytical model of the functional parameters calculation is shown in the figure 2.

The angle which determines the radial coning is made from the following component parts: the initial coning (β_i), the thermal coning (β_t) and the coning due to the pressure (β_p). It results the following relation:

$$\beta = \beta_i + \beta_t + \beta_p \,, \tag{2}$$

According to the film thickness and trying to approximating tgβ ≈ sin β ≈ β_1 it results:

$$\beta = \frac{h_0 - h_i}{r_0 - r_i} \,, \tag{3}$$

The film thickness to the r radius is described by the relationship:

$$h = h_i + \beta(r - ri) \,, \tag{4}$$

or in non-dimensional form, introducing relation (3) in (4):

$$\bar{h} = \frac{h}{h_i} = 1 + \frac{H-1}{1-R}\left(\bar{r} - R\right) \,, \tag{5}$$

where: H = h_0/h_i – film parameter;

R = r_i/r_0 – radius ratio;

$\bar{r}$ = r/r_0 – non-dimensional radius.

Using the hypothesis that the mechanical seal surface’s widthis relatively narrow comparing with the exterior radius, meaning $\frac{r_0 - r_i}{r_e} \approx 0{,}1$, it may be neglected the surface arch, so the Reynolds equation in this hypothesis can be symplified to the following form:

$$\frac{d}{d_r}\left(h^3 \frac{d_p}{d_r}\right) = 0 \,, \tag{6}$$

Using equation (4), it results $dh = \beta d_r$. Changing variable r in variable h, equation (6) becomes:

$$\beta^2 \frac{d}{dh}\left(h^3 \frac{d_p}{dh}\right) = 0 \,, \tag{7}$$

Through integration, it results:

$$p = -\frac{C_1}{2h^2} + C_2 \,, \tag{8}$$

The conditions to the limit for a primary mechanical seal with the model resulting from fig.2 are:

$$\begin{gathered} p = 0,\ r = r_i,\ h = h_i \\ p = p_0,\ r = r_0,\ h = h_0 \,, \end{gathered} \tag{9}$$

It’s calculating the integration constants and finally it results the pressure distribution under the following form:

$$p = p_0 \frac{h_0^2}{h_0^2 - h_i^2}\left[1 - \left(\frac{h_i}{h}\right)^2\right] \,, \tag{10}$$

The opening axial force, given by the clearance pressure which variation is given by the relation nr 10 can be calculated with the relation:

$$F = 2\pi r_m \int_{r_i}^{r_0} p\,dr \,, \tag{11}$$

where $r_m = \frac{r_i + r_0}{2}$ is the average radius of the seal surface. Replacing the relation (10) and changing the variable $d_r = \frac{dh}{\beta}$, it results:

$$F=\frac{2\pi r_m p_0 h_0^2}{\beta\left(h_0^2-h_i^2\right)}\int_{ri}^{r0}\left[1-\left(\frac{h_i}{h}\right)^2\right]dh\ , \tag{12}$$

$$F=\frac{2\pi r_m p_0}{\beta}\frac{h_0\left(h_0-h_i\right)}{h_0+h_i}, \tag{13}$$

Replacing inside the relation (13) the relation (3) and r_m, we obtain in the end:

$$F=\pi\left(r_0^2-r_i^2\right)p_0\frac{h_0}{h_0+h_i}\ , \tag{14}$$

This relation cand be singularize for the ideal case of the parallel surfaces, meaning $h_0=h_i=h$. The HS force in this case is:

$$F_{HS}=\frac{1}{2}\pi\left(r_0^2-r_i^2\right)p_0\ , \tag{15}$$

We can observe that in this case the pressure from the clearance varies in line from p_0 to *0*.

Using relations (14) and (15) we may write the opening force at a conical clearance according to F_{HS}.

$$F=F_{HS}\frac{2h_0}{h_0+h_i}\ , \tag{16}$$

The elementary damages debit according to the radius *r* direction and the unit width is:

$$q_r=\frac{h^3}{12\eta}\frac{d_p}{d_r}\ , \tag{17}$$

The total debit in the radial direction is:

$$Q=-\frac{2\pi r_0 h_0^3}{12\eta}\frac{d_p}{dr}\bigg|_{r=r_0}\ , \tag{18}$$

The pressure gradient is obtained throuh Reynolds equation integration in form (7), which is:

$$\frac{dp}{dr}\bigg|_{r=r_0}=\frac{2\,p_0 h_i^2}{\left(r_0-r_i\right)\left(h_0+h_i\right)h_0}\ , \tag{19}$$

After introducing expression (19) in (18) and making all the numerical calculations, it results:

$$Q=-\frac{\pi r_0 p_0 h_i^2 h_0^2}{3\eta\left(r_0-r_i\right)\left(h_0+h_i\right)}\ \text{or}\ Q=-\frac{\pi r_0 p_0\beta h_i^2 h_0^2}{3\eta\left(h_0^2-h_i^2\right)}\ , \tag{20}$$

3. CONCLUSIONS

According to the analyses of the relations which display the opening force given by the pressure from din clearance and the damages debit, the conclusion we can draw is that these are depending by the coning.

A controlled coning may lead to an opening force which can preserve a minimum thickness hi, necessary for a continuous film fluid with a damages debit imposed by the sealed fluid.

4. REFERENCES

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