A mechanical face seal model with numerical simulation of stick-slip.

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SUMMARY. A mechanical face seal for automotive applications that under particular conditions generates undesired acoustic emissions is studied. The work has the aim to demonstrate the reason of this noise.

1 INTRODUCTION

Whenever a rotating shaft must pass between two regions containing different fluids and it is important to keep the fluids separated, a rotating shaft seal is needed. The rotating shaft seal has the function of fitting around the shaft in a way such that the leakage between the two regions is acceptably small under all circumstances of operation. The mechanical face seal forms a barrier in the shape of annulus. Leakage is blocked or slowed either by actual contact of the surfaces or by a very thin gap between the surfaces.

Figure 1: Mechanical face seal.

A mechanical seal contains four functional components: 1) Primary sealing surfaces, 2) Secondary sealing surfaces, 3) a means of actuation and 4) a means of drive. The primary sealing surfaces are the heart of the device. A common combination consists in a hard material, such as silicon carbide or tungsten carbide, embedded in the pump casing and a softer material, such as carbon in the rotating seal assembly. Many other materials can be used depending on the liquid's chemical properties, pressure, and temperature. These two rings are in intimate contact, one ring rotates with the shaft, the other ring is stationary. The secondary sealing surfaces (there may be a number of them) are those other points in the seal that require a fluid barrier but are not rotating relative to one another. In order to keep the sealing surfaces in intimate contact, a means of actuation must be provided.

In this paper, a single mechanical face seal for automotive water pumps outside pressurized with the primary ring rotating is studied (see Figure 1).

A spring mechanism holds the annular surfaces together; the cooling liquid is a mixture of water an ethylene glycol and the materials of seal are silicon carbite and Carbon-graphite. Between the faces are assured two movements: one along the axis and the other angular [1]. The studied seal has the problem that under particular conditions generates undesired and noised vibrations.

The aim of this work is to demonstrate that the acoustic emissions are a consequence of insurgence of stick-slip phenomena [2], [3].

In order to investigate the possibility to reproduce the stick-slip conditions for different values of speed a lumped paremeters model is proposed and simluations are carryed on.

2 POSITION OF PROBLEM

The experience of firm, the data from costumers and the scientific literature show that the noise is influenced by temperature of sealed liquid and by shaft speed (see Figure 2) and in particular it appears for low shaft speed. In the diagram Temperature-Speed is possible to locate a zone (area "A"), where there are gathered the most part of noised (ringing) seals.

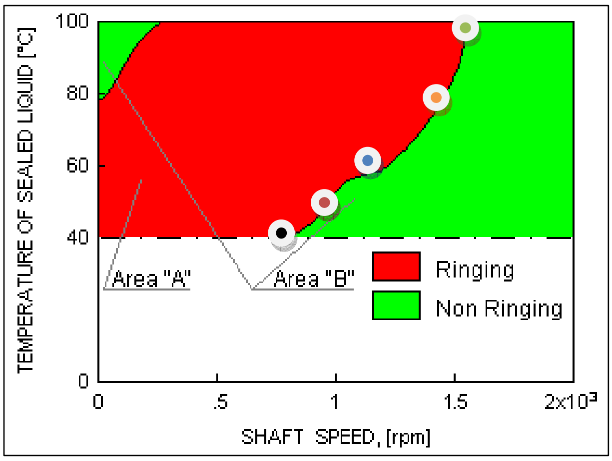

Figure 2: Diagram Temperature-shaft speed

For a fixed temperature (T<80°), below a certain shaft speed, the seal rings, that is compatible with the hypothesis that the stick-slip motion occurs. Stick-slip is the phenomenon of intermittent motion caused by velocity dependent friction force in combination with elasticity of the mechanical system of which the friction interface is part; it can occur during deceleration from the critical velocity above which sick-slip does not occur [2], [3].

3 THE PROPOSED MODEL

3.1 *Lumped parameters model*

In figure 3 a simple lumped parameters model is proposed: a mass *m* with inertia *J* of the mating ring is connected to the frame by a spring and damper device (where: K_h and K_θ are translational and torsional stiffness respectively; C and C_t are translation and torsional coefficient of damper respectively).

The primary ring rotates together with the shaft and slides against the mass; $F(h_1)$ and $M(h_1,\omega)$ are the actions in the seal with ω the relative angular speed between rings.

The model ha two degrees of freedom: the minimum distance between surfaces $h_1(t)$ and the other angular $\theta(t)$ that is the rotation of secondary ring .

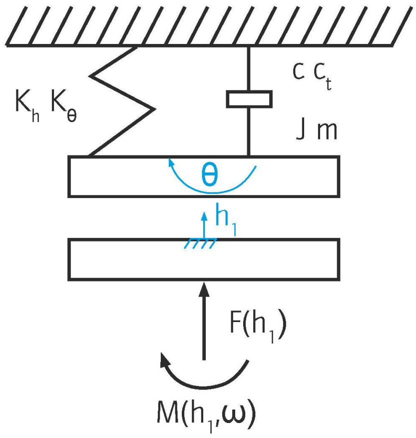

Figure 3: Lumped parameters model

The volume between the annular surfaces is assumed convergent radial taper with a axisymmetric geometry with the distance between the rings described by the following linear function with the radius *r* and where h_2 is maximum of film thickness.

$$h(r) = h_1 + \frac{(h_2 - h_1)}{(r_2 - r_1)}(r - r_1) \tag{1}$$

The dynamic behaviour of the model can be expressed by the following equations:

$$\begin{cases} m\ddot{h}_1 + c\dot{h}_1 + K_h h_1 = F(h_1) \\ J\ddot{\theta} + c_t\dot{\theta} + K_t\theta = M(h_1,\omega) \end{cases} \tag{2}$$

3.2 *Tribological model*

The force $F(h_1)$ and the moment $M(h_1,\omega)$ in the seal are evaluated by a mixed friction tribological model by *Lebeck*'s theory: the theory combines the model of hydrostatic lubrication of *Reynolds* with a model of sliding contact in boundary conditions [1], [4].

And one contribution prevails on the other when the pressure in the lubricant is negligible and vice versa.

In particular, in the model of sliding contact in boundary conditions, the distributions of the roughness is considered to remain the same even though they may be modified by contact itself. The contact pressure, function of h_1, depends on the fractional area on which, statistically, for a given roughness, the contact and/or the conpenetration between asperity of surfaces occurs. Where no contact occurs, the contact pressure is zero, in the other area it is assumed that the normal stress on the asperities is equal to compressive strength of the weaker material S_c:

$$p_c = b_m(h) S_c \tag{3}$$

where $b_m = b_m(h)$ is the fraction of area in contact evaluated assuming a Gaussian distribution.

In the model of lubrication, the fluid is assumed Newtonian with constant viscosity and constant density and the governing equation is:

$$\frac{1}{r}\frac{\partial}{\partial\vartheta}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial\vartheta}\right)+\frac{\partial}{\partial r}\left(\frac{rh^3}{12\mu}\frac{\partial p}{\partial r}\right)=\frac{r\omega}{2}\frac{\partial h}{\partial\vartheta} \tag{4}$$

Because the geometry is symmetric the fluid film is not depending on θ and the pressure can be evaluated by:

$$p(r)=\frac{(p_2+p_1)}{\int_{r_1}^{r_2}\frac{1}{rh^3dr}}\int_{r_1}^{r_2}\frac{1}{rh^3}dr+p_1 \tag{5}$$

The force $F(h_1)$ is evaluated by the integration of pressure and according to:

$$F(h_1) = F_c(h_1) + F_l(h_1) + F_p \tag{6}$$

Where the $F_c(h_1)$ is the resultant of contact pressure in boundary conditions, $F(h_1)$ is the resultant of lubricant pressure in full film lubrication regime and F_p is the spring preload.

The moment is the resultant of the moments in the two regimes:

$$M(h_1,\omega) = M_c(h_1,\omega) + M_l(h_1,\omega) \tag{7}$$

In particular the moment $M_c(h_1,\omega)$ depends on ω as the friction coefficient and on h_1 as the contact force in the boundary conditions according to $M_c(h_1,\omega) = f(\omega) \cdot F_c(h_1)$.

The moment M_l is a function of h_1 and ω according to *Reynolds* equations that assumes Newtonian fluid [5].

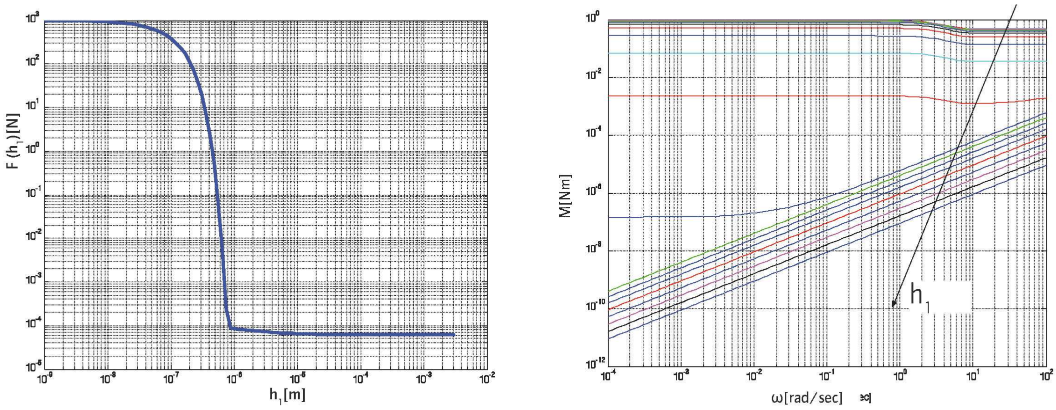

Figure 4: Force and moment in the seal

4 NUMERICAL RESULTS

The equations are solved numerically and the stick slip is simulated with the shaft speed decreasing according to figure 5.

Figure 6 shows the angular dispacement and the angular speed of the secondary ring obtained by simulation. The results show that the stick slip occurs under the critical speed of 150 rad/sec.

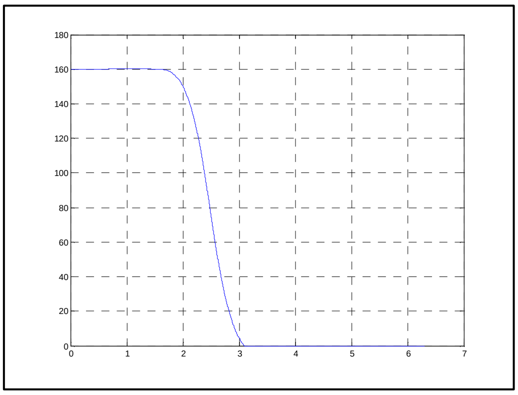

Figure 5: Shaft speed.

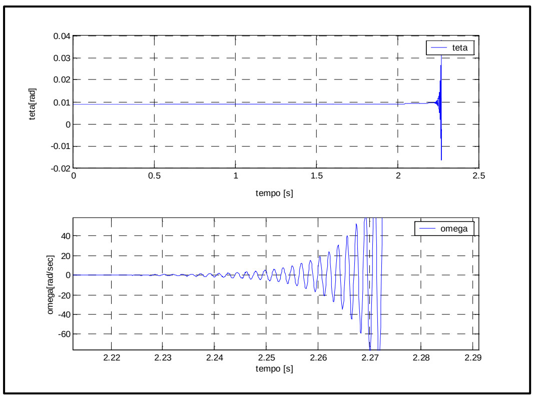

Figure 6: Stick-slip at 150 rad/sec.

Changing the frictional coefficient (figure 7) other simluations are carried on and the stick-slip vibration is simulated with different cirtical speed marked in figure 2 (figures 8-10).

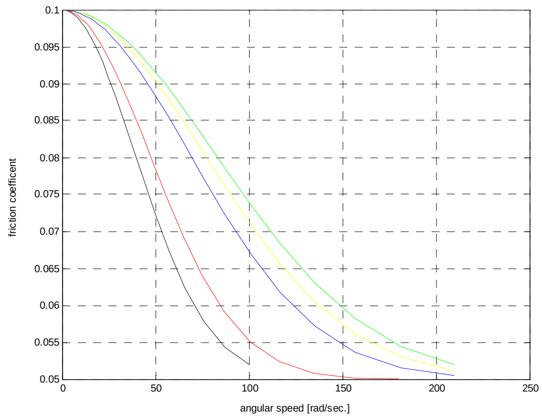

Figure 7: Frictional coefficient function of angular speed

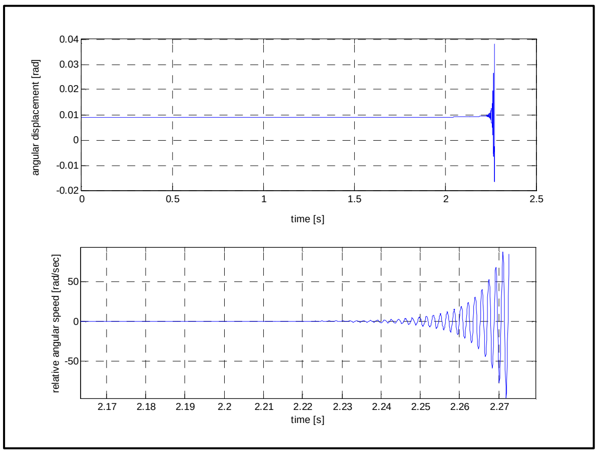

Figure 8: Stick-slip at 93 rad/sec.

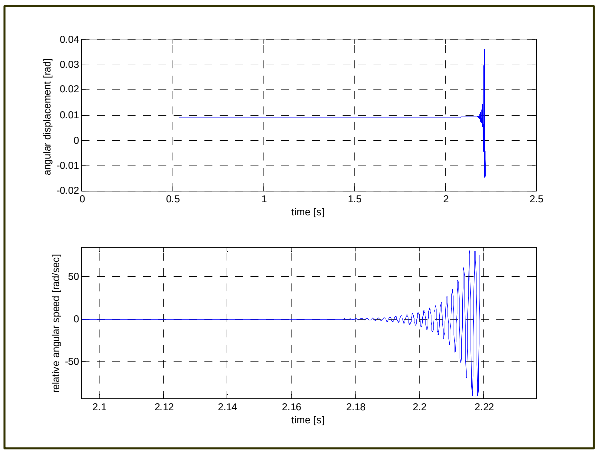

Figure 9: Stick-slip at 83 rad/sec

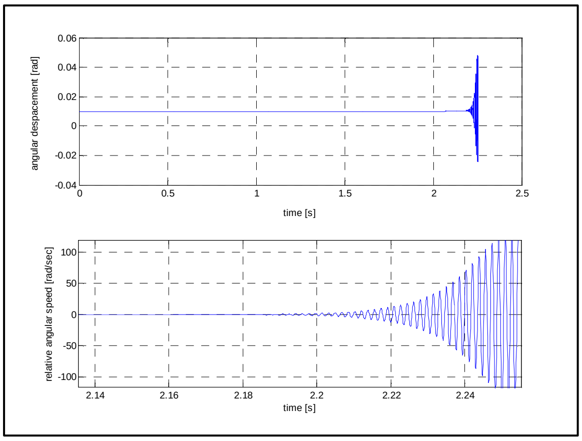

Figure 10: Stick-slip at 118 rad/sec.

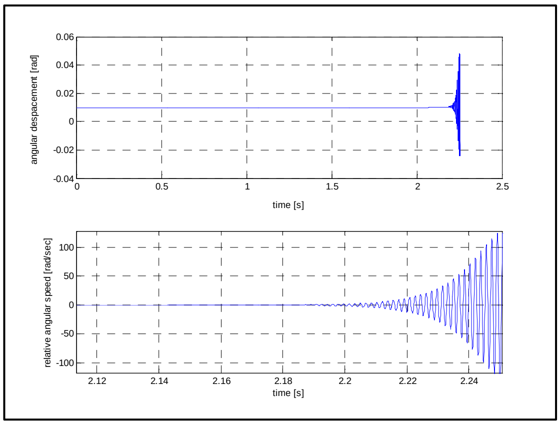

Figure 11: Stick-slip at 130 rad/sec

In order to analyze the influence of stiffness parameters results other simulations with different values are shown. In figure 12 the critical speed at which the stick slip occurs for different value of translational and torsional stiffness is shown (referring to simulation of figure 8).

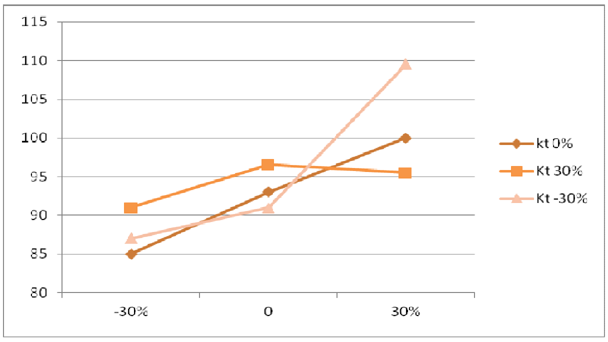

Figure 12- Influence of stiffness parameters on critical speed.

CONCLUSIONS

In this paper a mechanical face seal that under particular conditions generates acoustic emissions was investigated. A lumped parameter model together with a tribological model has been proposed and simulations carried on with the aim to show that the phenomenon is due to stick slip vibrations. A sensibility analysis regarding the stiffness parameters shows that the critical speed increases with the stiffness.

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