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FILM THICKNESS PREDICTION OF RADIAL LIP SEAL

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Abstract: *A computational approach has been developed to predict the lubricated conditions of radial lip seals at the vicinity of the lip. The approach follows the procedure proposed by Salant [10]. The procedure developed takes into account both normal and tangential deformations of the surface. In a first step, the deformation of the seal is calculated using a FEA procedure. From these results, elastic influence coefficients are determined which are then used to correct the surface deformation due to the lubricated film and hydrodynamic pressures. The film thickness under the lip is finally obtained by solving the Reynolds equations with consideration of the elastic deformations of the surface.*

Keywords: *oil film thickness, radial shaft seal, tribology*

1. INTRODUCTION

Seals are recognized as a critical element to ensure the life of machine elements. Their main function is to prevent leakage but also to form a reliable barrier to protect the mechanical elements from outside contamination. When the shaft is in rotation, the relative motion of the surfaces naturally promotes the flow of the lubricant underneath the lip, passing from one side to the other. This flow, which is defined as the pumping flow, is critical in the performance of the seal, preventing leakage and providing the lubricated conditions of the lip

The behavior of radial lip seals was extensively studied by Jagger in the late 60's. His study revealed for the first time the existence of the lubricating film under the lip of the seal (Jagger, 1957). In some later works, several researchers have suggested that the micro texture of the rubber surface might play an important role in the film formation (Kawara et a.l., 1980; Nakamura and Kawara, 1984; Nakamura 1987; Horve, 1996). It is assumed in these considerations that the micro-texture is developed due to the rubbing of the surfaces in contact. Different types of micro-textures have been observed; small scale asperities were identified by Jagger, (1966), Poll and Gabelli, (1992) and Müller, (1987). Large scale asperities were observed by Nakamura and Kawara in 1984 and Nakamura in (1987) whilst micro-undulations have been reported in Müller and Ott's work (1984).

In this paper, a model is proposed to predict the lubricated condition, considering the effect of the surface roughness of the rubber.

The prediction is based on an FEA approach to calculate the contact condition at the lip. A linear perturbation method is then used to predict the changes in deformation due the changes in pressures caused by the hydrodynamic effects. The linear perturbation procedure is coupled to the solution of the Reynolds equation to obtain the film thickness and pressure between the surfaces. The approach is similar to the one proposed by Salant et al. (1995-2004)

2. FEA ANALYSIS

The FEA analysis is used to calculate the response of the seal at different interferences under dry conditions. During the calculations, it is assumed that the geometry is axisymetrical. The rubber material is considered to follow the standard Yeoh model.

2.1 Seal Description

The seal to be considered is depicted in Figure 1. The seal is representative of a Rsafe seal which is in use in both automotive and industrial applications under different designs.

The Rsafe seal has two important features: a primary lip, which acts in radial direction and which main function is to prevent oil leakage and a secondary lip which acts in the axial direction. The role of the secondary lip is to avoid ingress and contamination. The action of the radial lip is helped by a Garter spring, which provides a constant force on the radial lip over the life of the seal. The paper only focuses on the radial lip and it will be assumed that the axial lip is free to move at its extremity.

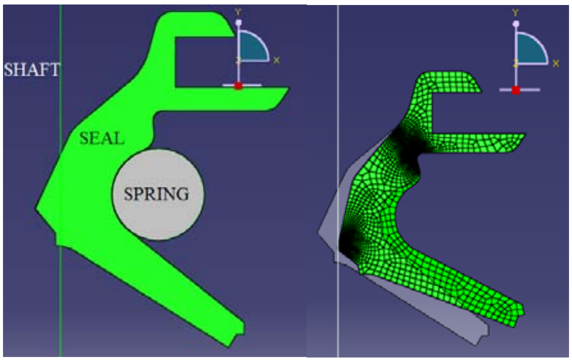

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Figure 1. Seal design and mesh

2.2 Results with no Garter Spring

In these series of results, the calculations were performed without the Garter spring. Figure 2 shows the effects of the interference on the pressure distribution at the contact between the lip and the shaft. It is interesting to observe that the position at which pressure reaches its maximum is almost independent of the level of interference. The maximum pressure also barely changes with the increase in the interference. By contrast, the width of the contact increases rapidly with the interference.

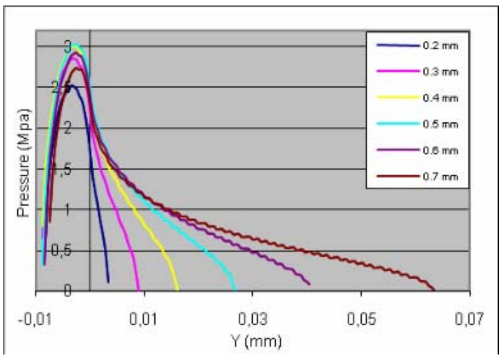

Figure 2. Pressure Distribution for different shat interference

The lip force is defined as the integral of the pressure in the contact area. Its variation as a function of the interference is depicted in Figure 3. It is clear from the graph that the lip force varies almost linearly with the interference. This linear response is caused by the generation of bending and hoop stress in the lip; both effects increase almost linearly with the interference.

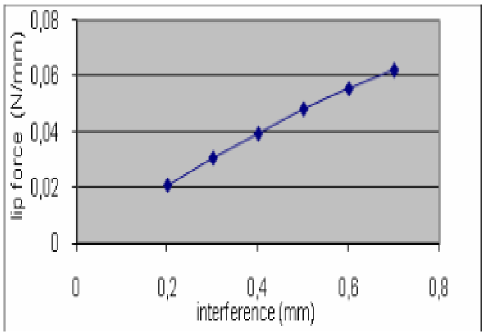

Figure 3. Lip force variation with interference

2.3 Results with Garter Spring

The calculations show similar variations when the Garter spring is inserted. The pressure distribution with and without Garter spring is described in Figure 4. The Garter spring bring an additional load contribution, which leads to a rise in the maximum pressure and a widening of the contact area.

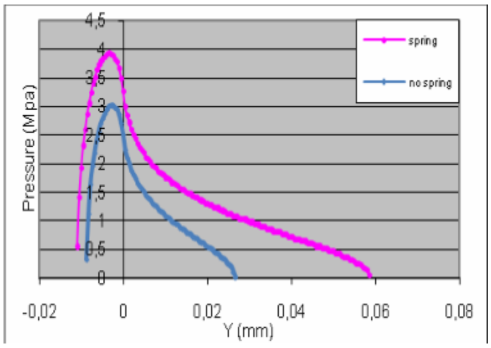

Figure 4. Pressure Distribution with and without the Garter spring

2.4 Contact Stress

When the lip is brought into contact with the shaft, stress develops in the sub-surface of the rubber. Figure 5 shows the variation of the Von-Mises stress below the contact. As expected, the stress reaches it maximum value at the position below the surface, at a depth which is about half the contact width.

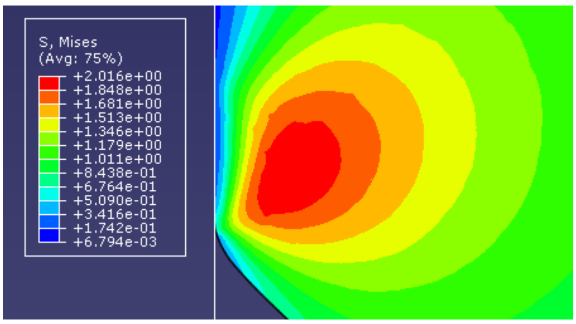

Figure 5. Von-Mises Subsurface stress at the contact between the lip and the shaft

3. LUBRICANT FILM THICKNESS PREDICTION

The prediction of the lubricated film thickness requires the resolution of a coupled problem with consideration of both the surface deformations and the Reynolds equation. There is no direct mathematical solution and an iterative numerical scheme has to be developed to predict the film thickness.

The use of a FEA approach at each stage of the iterative process is cumbersome and CPU time-consuming. A more efficient procedure has been proposed by Salant et al.(1995-2004) based on a linear perturbation of the surface deformation. In this manner, only a single FEA calculation is required, which reduces drastically the solution time. A similar approach is used in this paper. The linear perturbation calls for the calculation of an influence matrix to relate the change in deformation to the change in pressure. The influence coefficient matrix is calculated considering the dry contact conditions between the lip and the shaft.

4. EXAMPLE OF COMPUTATION

The pressure and film thickness distribution generated by an array of asperities on the rubber surface is analyzed. The domain of study is represented by a square domain as indicated in Figure 6 below.

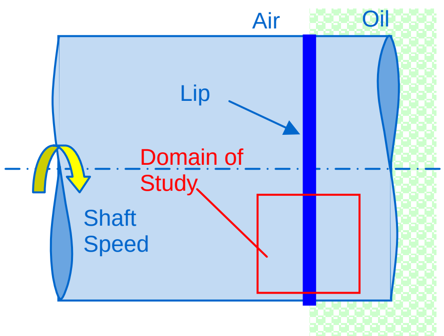

Figure 6. Pressure distribution as a function of rotational speed

4.1 Pressure Distribution

Figure 7 shows the distribution in pressure at speed ranging from 1m/s to 10 m/s. It can be seen that pressure varies in both the circumferential and axial direction. In the circumferential direction, pressure builds up to flatten the asperities. In the axial direction, pressure follows the dry pressure condition and intensifies at the point where the lip fist touches the shaft. Maximum pressures are the most important for lower velocities. In this situation, almost no pressure builds up between the asperities and the pressure formation remains localized to the top of the asperities. As the velocity increases, pressures develop within the valleys of the rubber surface, which entrain a reduction in maximum pressure.

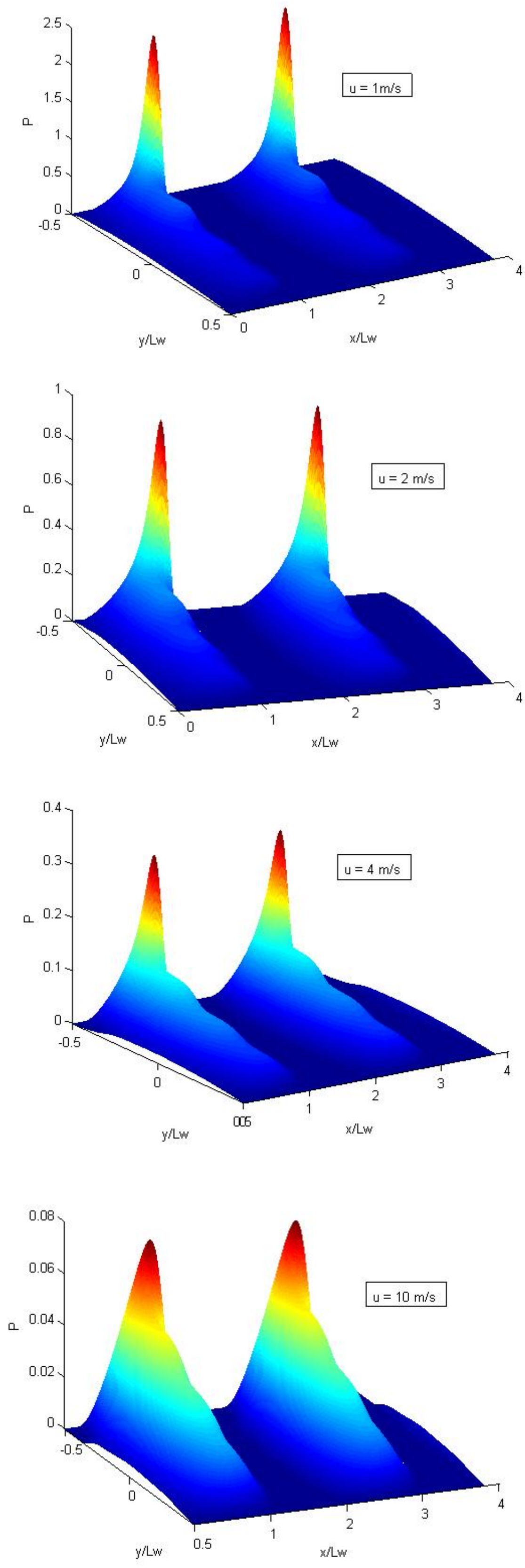

Figure 7. Pressure distribution as a function of rotational speed

4.2 Film Thickness

The variation in the film thickness with speed at the top of the asperities is indicated in Figure 8. As expected the film thickness increases with the rotational speed of the shaft. For speed up to 4m/s, the film thickness remains vary low at the point where the lip touches first the shaft. The lubrication conditions at this point are prone to be in the mixed lubrication regime with development of friction and wear.

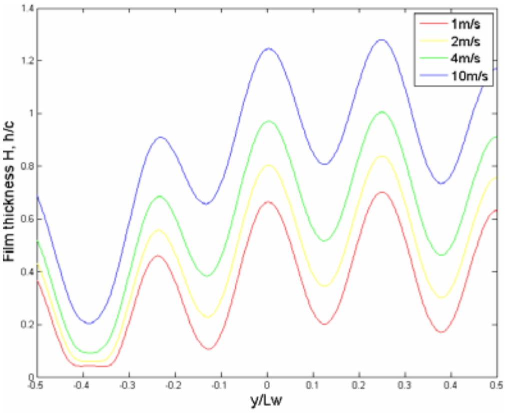

Figure 8. Film thickness as a function of the shaft rotational speed

5. CONCLUSIONS

The conditions where the surface contains a roughness represented by an array of asperities have been analyzed. The results show that the micro-asperities provide sufficient hydrodynamic pressures to separate the surfaces and to develop a full lubricated film in some operating conditions. As the shaft speed is increased, the peak pressure decreases and a thicker lubricated film is developed. These conclusions are consistent with the results obtained in previous studies (Salant et al., 1995-2004).

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