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Running-in of an artificial rough rolling-sliding contact using Finite Element Analysis

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ABSTRACT – The aim of the present paper is simulating a rolling-sliding contact using finite element analysis (FEA) to analyze the change in surface topography and contact. Finite element simulations are done for the two-disc configuration with an artificial rough surface of a sphere and a smooth cylinder. The results show that the contact pressure reaches a stabilized value after a certain number of rotations, indicating the end of the running-in phase and the beginning of the steady-state phase.

1. INTRODUCTION

In the study of rolling–sliding contacts of mechanical components such as roller-bearings [1], cam and followers [2], gears [3] and micro gears [4], wear is regarded as a surface phenomenon that has been identified as a critical factor for controlling the lifetime of these components. However, the investigations on wear of rolling-sliding contacts were often conducted when operating in the steady-state phase without considering the running-in phase. During running-in, friction and wear between two contacting bodies may vary considerably over time.

In the investigations conducted in the past decades in developing running-in models and performing running-in experiments, attention was paid to observing changes in the coefficient of friction [5-7] and the surface topography [8-10] during running-in. However, there are other parameters whose contribution cannot be neglected for a successful running-in phase. Hsu et al. [11] observed that besides surface roughness, contact pressure and interface layer, the establishment of an effective lubricating film is also adjusted in the transient period of running-in to the steady-state condition. These parameter adjustments induce surface conformity, oxide film formation, material transfer, lubricant reaction products, martensitic phase transformations and subsurface microstructure reorientation.

Recently, Argatov and Fadin [12] indicated that based on mathematical modelling, using the theory of elasticity in conjunction with Archard's law of wear, the contact pressure is very important in determining the end of the running-in phase and the start of the steady-state phase. This paper aims to conduct a rolling-sliding contact using finite element analysis (FEA) to analyse wear, change in surface topography and contact pressure as a function of sliding distance/time.

2. METHOD

Finite element simulations were done using ABAQUS for the two-disc configuration with an artificial rough surface of the upper sphere (R_h = 4 mm) and a smooth lower cylinder (R_c = 4 mm). An arrangement of asperities covers the upper surface as depicted in Fig. 1. A contact load F_N of 100 N and the radius of the artificial asperities, R_{as}, of 50 μm is used in the simulations. The circumferential distance between the asperities is 120 μm. The rolling and sliding contact situation is performed by determining the velocity of the upper ball, and the velocity of the lower cylinder are 800 mm/s and 880 mm/s whereas the specific wear rate is 1 x 10-5 mm^3/Nm and the coeffictien of friction is 0.6.

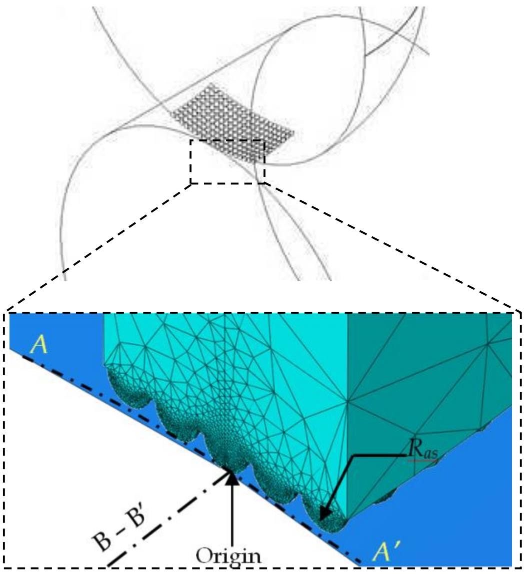

Figure 1 Two-"disc" contact where an artificial rough surface is modelled on the upper surface. The zoom-in view depicts the contact between the rough surface and the smooth lower surface and the mesh refinement at the asperities.

The model contains three stages in the simulation procedure: determination of the contact pressure, calculation of the wear based on Archard's wear equation and updating of the geometry. The wear simulation lasts until the sliding distance (S_{max}) is obtained.

3. RESULTS

The initial contact between the rough sphere and the smooth cylinder when loading the system initially contact occurs on the axi-symmetrical line A-A' (see Fig. 1). Then, as the contact load increases and reaches, in this case, 100 N, other asperities in perpendicular direction of line A-A' come in contact, marked with number *4* and *5* as illustrated in Fig. 2. It shows that the nominal contact area becomes larger and the number of contacting asperities increases as the number of rotation increases.

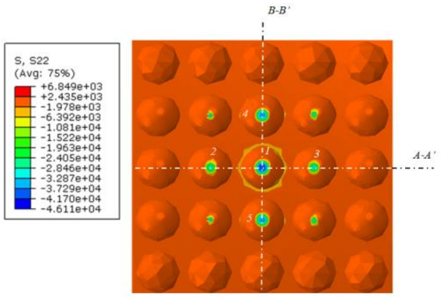

Figure 2 Contact area between the rough sphere and smooth cylinder. The contact pressures are given in MPa.

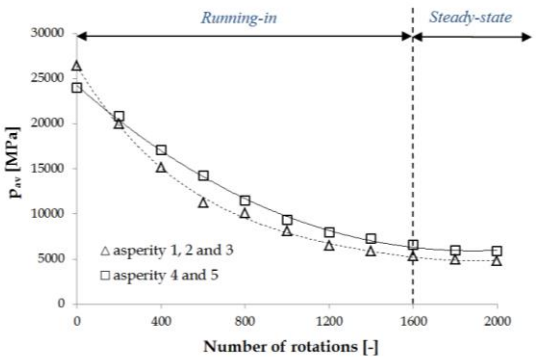

Figure 3 Calculated contact pressure evolution of asperity 1-5 as a function of the number of rotations.

The evolution of the average contact pressure of the asperities with increasing number of rotations is shown in Fig. 3. The average contact pressure of asperity 1, 2 and 3, located along A-A' line, show a similar behavior. Asperity 4 and asperity 5 have the same distance to line A-A' and are assumed to have similar average contact pressure. Initially the contact pressures of asperity 1, 2 and 3 are higher than asperity 4 and 5. As the number of rotations increases, the contact pressures of asperity 1, 2 and 3 decrease and are finally lower than the average contact pressure in asperity 4 and 5. Then, the contact pressures of the asperities are "stabilized" after 1600 rotations indicating the end of the running-in phase and the beginning of the steady-state phase.

Using the similar method in FEA, the previous research [4,13] employed the smooth surfaces in contact and the asperities were not taking into account. The present model will be developed further using a real rough surface model in predicting the running-in wear.

4. CONCLUSION

The present FEA based deformation and wear model is used to perform simulations in which an artificial rough hemisphere is in rolling-sliding contact with a smooth cylinder. The results show that the contact area becomes larger and the number of contacting asperities increases as the contact load increases. The asperity contact pressures "stabilize" after a certain number of overrollings indicating the end of the running-in phase and the beginning of the steady-state phase.

5. REFERENCES

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