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A method for characterizing running-in of sliding contacts

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Abstract. Although in terms of conservation wear is undesirable, however, running-in wear is encouraged rather than avoided. Running-in is rather complex and most of the studies related to the change in micro-geometry have been conducted statistically. The purpose of this study was to characterize the running-in of sliding contacts using finite element analysis based on measured micro-geometries. The developed model combines the finite element simulation, Archard's wear equation and updated geometry to calculate the contact pressure distribution and wear depth. Results show that the proposed model is able to predict the running-in phase of sliding contact system.

Introduction

Changes which occur between start-up and steady state of contacting surfaces during rolling/sliding motion are associated with running-in (also called breaking-in or wearing-in). Although in terms of conservation wear is always undesirable, running-in wear is encouraged rather than avoided. GOST (former USSR) Standard defines running-in as: "The change in the geometry of the sliding surfaces and in the physicomechanical properties of the surface layers of the material during the initial sliding period, which generally manifests itself, assuming constant external conditions, in a decrease in the frictional work, the temperature, and the wear rate" [1].

Running-in is very complex problem. There are many tribological characteristics, such as friction, wear quantity and surface topography during the running-in process. Changes to the surface micro-geometry during the running-in phase of a sliding contact are usually related to a mild wear processes, as described by the Archard's wear concept [2] and by later researchers in more details [3], that includes wear particle removal and abrasive wear [4]. On a macro scale, the sliding contact between two contacting bodies is often referring to an elastic contact situation. The macroscopic wear volume has been studied extensively. Most of the researches are essentially experimental. The change in surface topography and the transition from the running-in phase to the steady-state phase is expressed using statistical surface roughness parameters. The local change of the surface topography of simple repeated contact has been studied by Jamari and Schipper [5]. However, the local changes of the surface topography during the running-in process did not get much attention. When sliding occurs, it is known that the elastic-plastic contact situation on asperity level plays an important role in the change of the asperity shape. The coefficient of friction and the wear rate of the contacting materials are the main parameters to distinguish the running-in and steady-state phase.

In attempting to predict the local changes of the surface topography during running-in of sliding contacts, a developed elastic-plastic finite element analysis based model is presented in this paper. The model combines the finite element method, the Archard's wear equation and the updated geometry.

Method

The simulation scheme for predicting wear in a sliding contact is schematically depicted in Fig. 1. Basically, 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 reached.

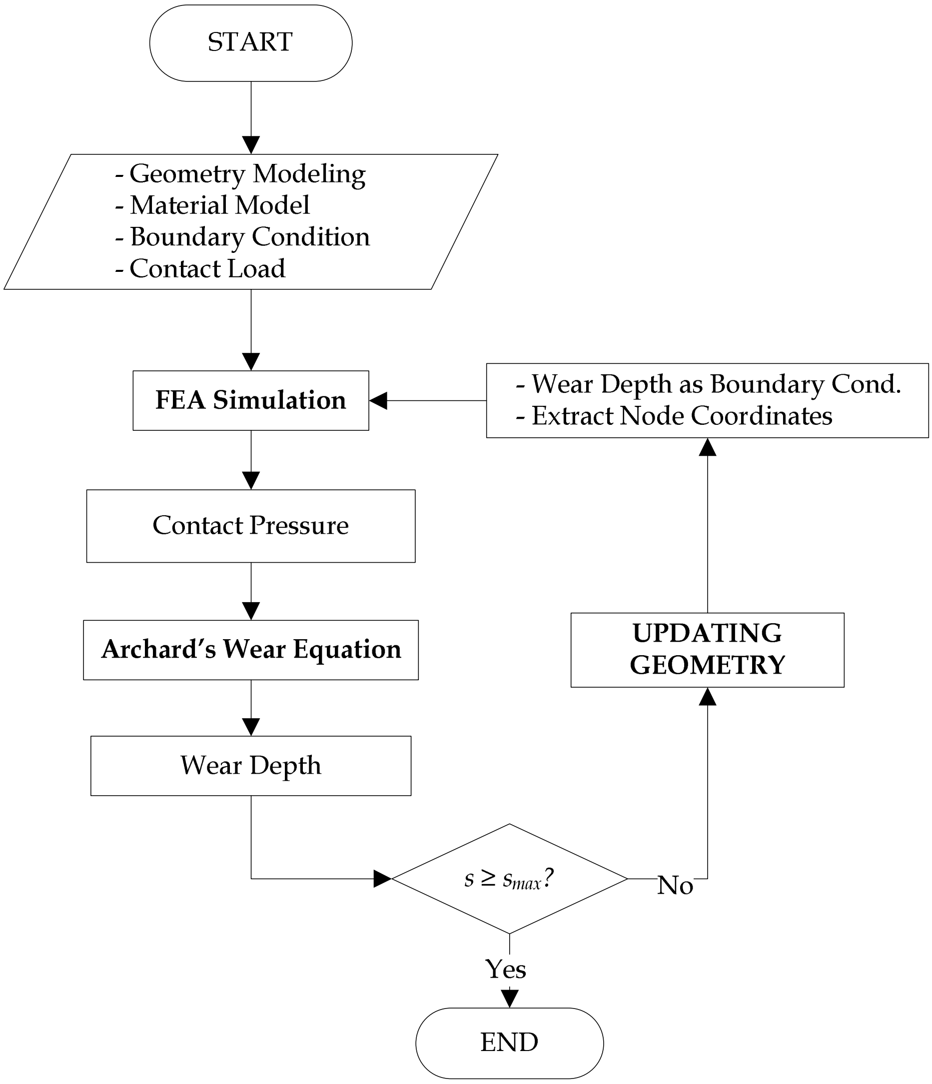

Fig. 1. Flow chart for predicting wear in sliding contacts.

In the first stage, the inputs are the geometry, material model, boundary conditions, and contact load. The simulation starts with a finite element analysis to obtain the contact pressure for each node on the contacting surface. Then, the contact pressure p is used as input in the second stage for calculating the local wear by employing the Archard's wear equation [2]. Here, the wear depth, h^w, of the contacting system was determined using the incremental sliding distances Δs as well as the wear rate, k_D, of the system. $h^w = k_D p \Delta s$. Third, the geometry of the contact system was updated with the amount of wear, h^w, calculated in the previous stage. In this stage, the nodes and the boundary conditions are updated. The routine is repeated until a certain defined sliding distance (S_{max}) was obtained.

A pin-on-disc contact system as is schematically shown in Fig. 2a was used in this study. In the elastic-plastic finite element simulation, such a contact is simplified to a contact between an axis-symmetric hemisphere and a flat (Fig. 2b). The mesh was refined in regions near the contact area of the hemisphere and the flat body, as is depicted in Fig 2c, to increase the accuracy of the calculation. The simulation does not aim to simulate the entire sliding process of the contact system but instead treats the problem of sliding wear as 'quasi-static' to save the computational expense. In the simulations, the hardness of the material and the wear rate, k_D, were assumed to be constant during sliding. The analysis on wear focuses on the pin geometry while the wear of the disc is not discussed.

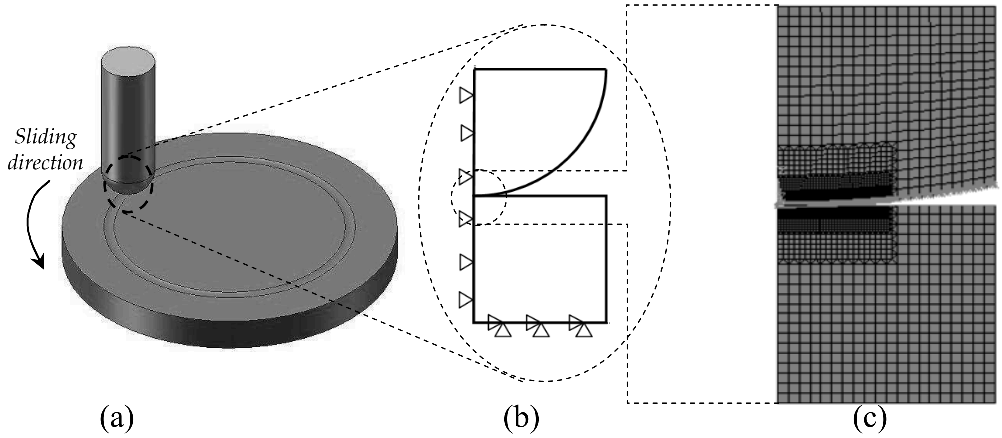

Fig. 2. (a) Pin-on-disc contact system, (b) the model and its boundary conditions and (c) the finite element mesh and its refined location.

Results and Discussion

For the pin-on-disc configuration, a pin radius R_P of 0.794 mm was in sliding contact with a disc of R_D of 4 mm on a wear track with radius R_{WT} of 3 mm. The thickness of the disc, t_D, was 1 mm and a contact load F_N of 200 mN was applied. The material used for pin and disc in the present study was ceramic Si_3N_4 with a modulus of elasticity E = 304 GPa and Poisson's ratio υ = 0.24. A coefficient of friction μ = 0.45 and wear rate k_D = 13.5 x 10^{-9} mm^3/Nm was used. The calculated contact pressure distributions are compared with the work of Hegadekatte et al. [6]. Figure 3a shows the Hegadekatte contact pressure distribution [6] while Fig. 3b shows the contact pressure distribution of the present model as a function of the sliding distance. Differences in pressure distribution are found; (i) the contact pressure distribution of the present model coincides with the contact pressure of Hertz for s = 0 and (ii) for the following cycles the present model show pressure distributions as discussed in [7, 8]. When the hemisphere starts to wear and flattening occurs, the maximum in the contact pressure moves from the centre to the edge of the contact area. There is no pressure increase at the edges in Hegadekate's graphs. The discrepancies could be due to the fact that in the present model the elastic-plastic behaviour is considered and the wear occurs on the pin surface only.

The present model is also compared with the work of Hegadekatte et al. [6]. They compared a finite element simulation with their experimental data and found a good agreement. Figure 4 depicts the wear depth evolution for the present results and those of [6] as a function of the sliding distance. The present simulation is limited to a sliding distance of approximately 70 mm. The wear depth predicted with the proposed FEM model is in good agreement with the results of Hegadekatte [6].

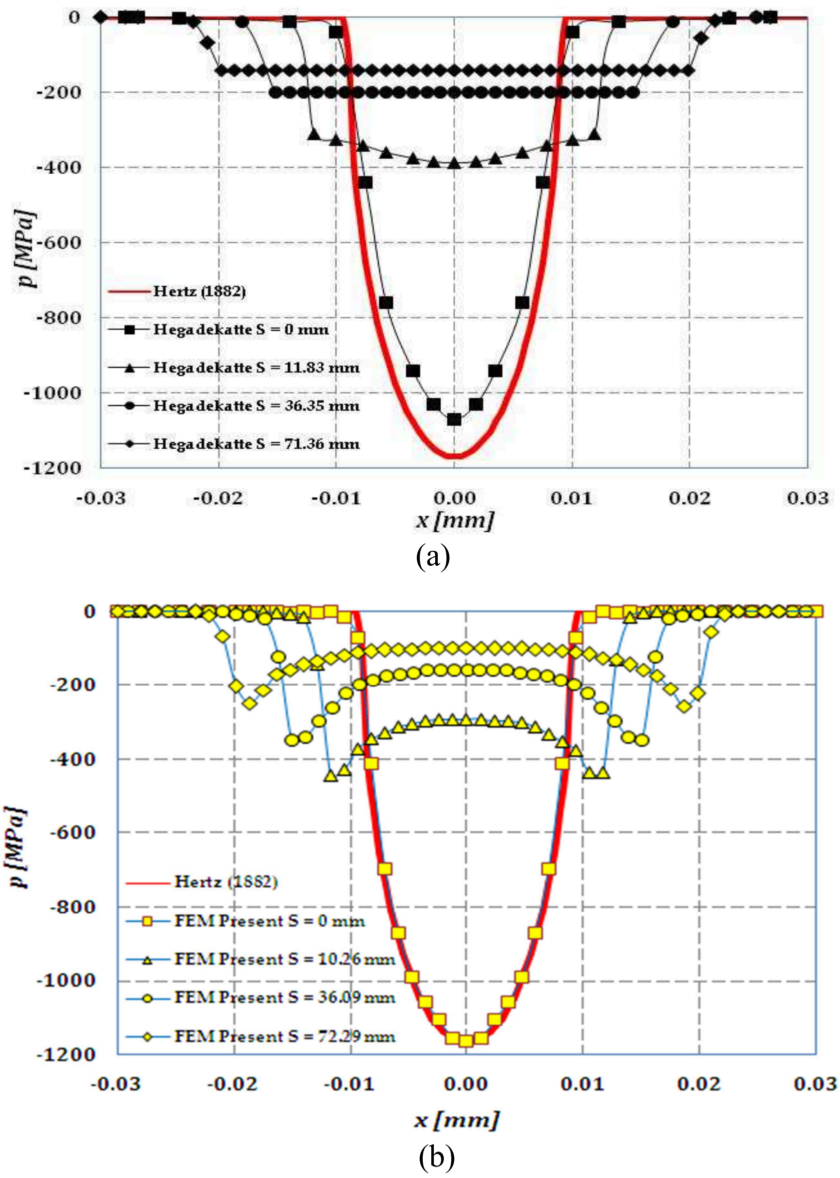

Fig. 3. The comparison of the contact pressure between: (a) Hegadekatte et al. [6] and (b) present FEM model.

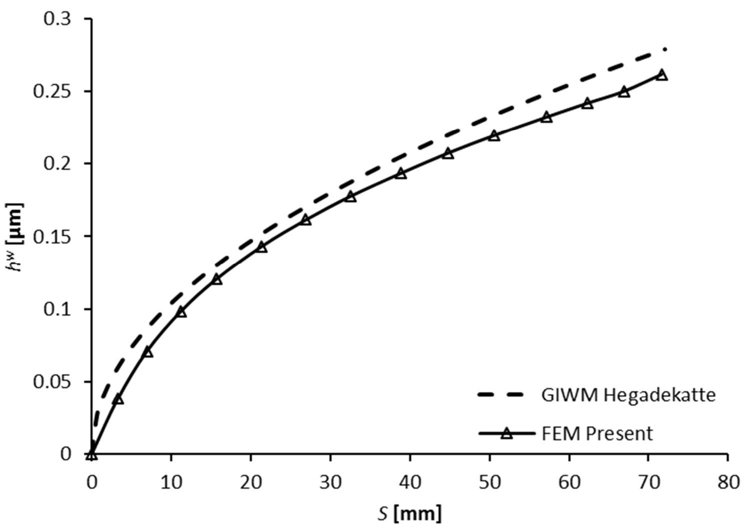

Fig. 4. Comparison of the wear depth of a pin of Hegadekatte et al. [6] and present model simulation.

From the aforementioned results and discussion, the predicted wear depth of the pin does not consider the different phases in the wear process, i.e. running-in and steady-state. Therefore, a new method is proposed for characterising the running-in parameter. Here, two contact pressures are introduced, namely the contact pressure at the center of the pressure distribution (p_{center}) and the average contact pressure (p_a). At the start of the sliding contact, where the running-in phase occurs, the p_{center} and p_a are initially high and decrease gradually, until a more or less "steady-state" is reached. This method has been applied using the present model to determine the transition of the running-in to the steady-state phase of the work of Hegadekatte et al. [6] as is depicted in Fig. 5 and Fig. 6 for different load. It is found that the average contact pressure reaches its "steady-state" after approximately 34 m sliding of $F_N = 21$ N and 26 mm sliding of $F_N = 200$ mN.

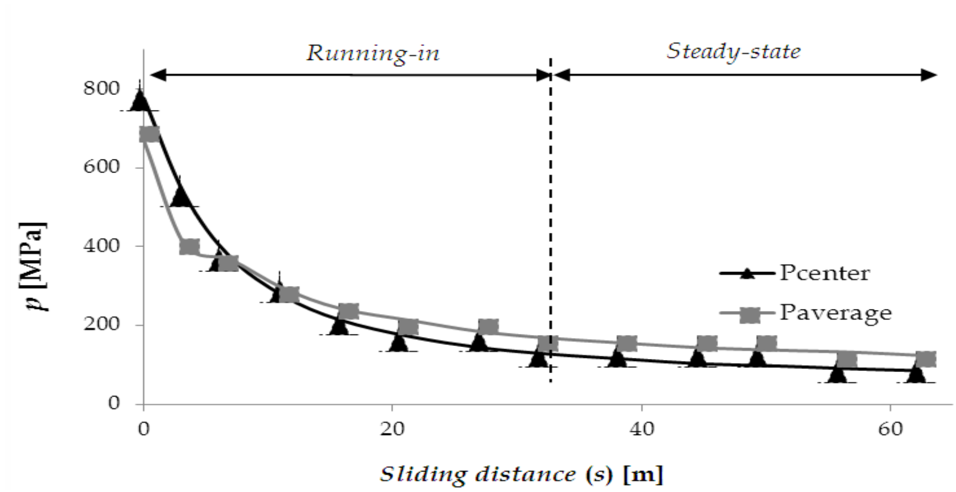

Fig. 5. Determining the transition of the running-in to the steady-state phase, Hegadekatte et al. [6], using the present model. $F_N = 21$ N, $R_{ball} = 5$ mm, $E = 210$ GPa, $\upsilon = 0.3$ and $K_D = 1.33 \times 10^{-10}$ mm^3/Nm.

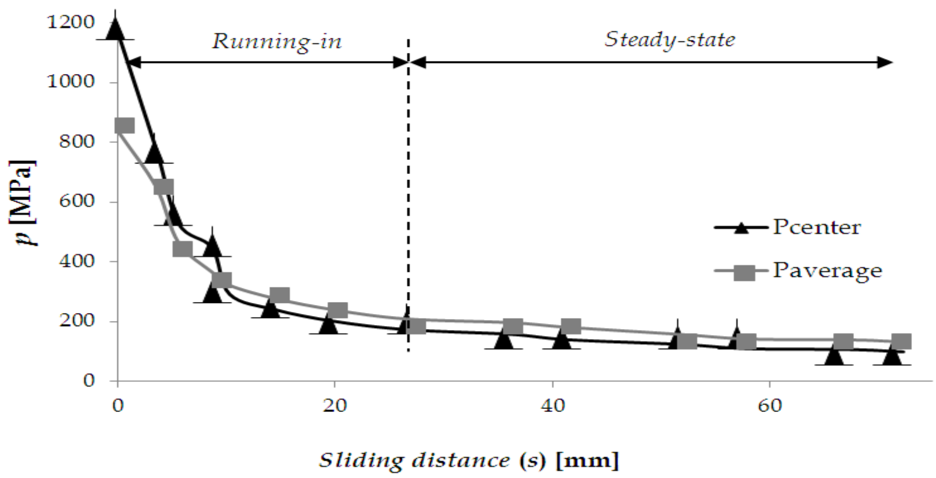

Fig. 6. Determining the transition of the running-in to the steady-state phase, Hegadekatte et al. [6], using the present model. $F_N = 200$ mN, $R_{ball} = 0.794$ mm and $K_D = 13.5 \times 10^{-9}$ mm^3/Nm.

Conclusion

Study for characterizing the running-in of sliding contacts based on finite element analysis has been conducted. Contact pressure evolution, change in topography, and wear depth of sliding contacts have been calculated by a method of incorporating finite element analysis, Archard's wear equation and updated geometry. A new method to characterize the running-in of sliding contact was introduced. It was found that the developed model is able to predict the transition between the running-in phase and the steady-state phase.

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