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A two scale mixed lubrication wearing-in model, applied to hydraulic motors

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ABSTRACT

Wearing-in of a machine component can increase the conformity between contacting pairs and smoothen the surface topography. A two scale model, combining the wearing-in effects, resulting in changes in the surface topography, with the wear that occurs on the component, is presented. The geometry of the components is represented with measured coordinates. Wear leads to changes of the geometry, which has an effect on several tribological conditions, such as contact forces, relative velocities and conformity. Due to the wear on the topography scale, the load sharing is also affected. The model is applied to orbital hydraulic motors. The wear depth predicted with the model is qualitatively in good agreement with the wear depth recorded in experiments.

1. Introduction

The early life of a machine component is characterized by high wear rates. Under desirable circumstances, this wear rate will drop gradually and a steady state of low wear rate will occur. The process is called running-in. Running-in is caused by a number of factors. Changes in microstructure, activation of lubricant additives and changes in surface topography are examples of changes that can help us to protect the surface from further damage. Wearing-in is when wear during the early life of a machine component affects the wear rate through geometrical changes [6]. Wearing-in can be seen as a two-scale process, which from a global scale perspective increases the conformity between the surfaces and on the local scale smoothens the topography. Smoothening of the topography, in turn, influences the load sharing between asperities and hydrodynamic lift, caused by pressure build up in the lubricant. Wear which leads to a smoothening of the surfaces, often with a low wear rate, is called mild wear. Wear causing an increased surface roughness, usually under high wear rates, is called severe wear. Favourable running-in is commonly initiated by severe wear, which is followed by a state of mild wear [7].

Akbarzadeh and Khonsari [1] used a model to vary loads, initial roughness and sliding speed to investigate the influence of these parameters on wearing-in. The model illustrates mild wear wearing-in. The model and experimental investigations have shown that running-in under mild wear conditions can be favoured by high loads. This is only the case up to a limit, where a transition to severe wear occurs, as noted by Jamari [21]. Experimental investigations of surface roughness by Jahanmihir and Suh [20] indicate that high loads will quickly cause a roughening of a surface, and that wear under low loads can be reduced by low initial surface roughness. Other investigations show both increases [13] and decreases [4] in surface roughness due to wear. The transition to severe wear can be explained by *e.g.* breakdown of oxide films, and depend on the conditions of the tribological system [25]. A model investigating the details of the transition to severe wear has been published by Bosman and Schipper [8]. The study included a transition from mild to severe wear due to local surface temperatures. The running-in is further complicated, and improved, by including lubricant additives in the tribological system. Changes in surface roughness due to the influence of lubricant additives have been investigated experimentally [16,35] and numerically [3].

Full scale component wear modelling can be used to estimate the wear behaviour of machines and machine components. Such models allow new designs to be tested numerically. Contact conditions have been numerically evaluated using the finite element method [27,29], the boundary element method [33,37,2] and the elastic foundation material model [15,34]. Numerical procedures to efficiently evaluate the wear problem have been elaborated on by *e.g.* Põdra and Andersson [28] and Lengiewicz and Suptkiewicz [24]. Mattei et al. [26] compared a number of running-in models which assume changes in wear rate with time for the case of hip joints. Zhu et al. [41] treated the lubrication as a separate model component, solving the lubrication problem

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including a wear effect. Although wear models keep improving, many of them do not consider roughness, running-in and lubrication in a satisfying way. Treating the roughness deterministically is not always possible as time and geometry scales for a component grow large. Therefore, simplified running-in models are needed for full scale component wear modelling.

Orbital hydraulic motors are suitable for delivering high torque at a low speed [19]. In the centre of the orbital hydraulic motor there is a gear set, illustrated in Fig. 1. The gear set consists of an inner gear which is moving in an epicyclic motion inside an outer gear. The inner gear consists of $m-1$ teeth. As illustrated in Fig. 1, geometric positions on the teeth can be related with an angle θ. The inner gear orbits around the centre of the outer gear in a circle, which is called the excentre. The inner gear also rotates around its own centre. The time it takes for the inner gear to complete a rotation around its own centre is $m-1$ times the time it takes for the inner gear to orbit one lap on the excentre.

Contact between the inner and outer gear causes wear, which in the worst case, can lead to seizure. Mixed lubrication effects and wear in orbital hydraulic motors were investigated by the present authors in a previous publication [18]. Results from the study indicated that improvements of the mixed lubrication model would be needed in order to explain the deviance between simulation and experiments.

In this study a two scale wearing-in model is presented. Global wear is modelled through material removal from the geometry of the components. Topography scale wear is modelled by changes in the mixed lubrication behaviour of the individual contact points between the surfaces. Measured surface topographies are used to tune the topography scale wear model. The two scales are illustrated in Fig. 2. The model is applied to carry out numerical

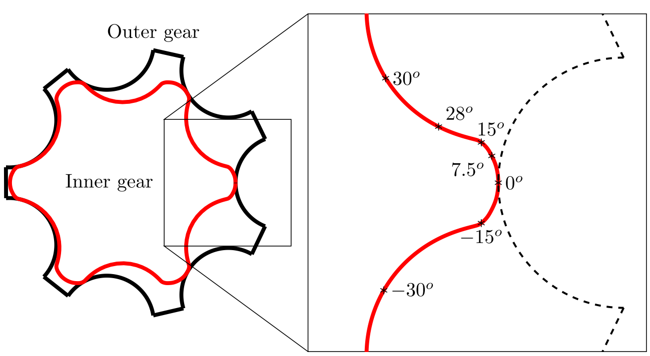

Fig. 1. The gear set of an orbital hydraulic motor. The inner gear has $m-1$ teeth and the outer gear has m teeth. In this work, $m=7$. The geometry is designed in such a way that the inner gear moves in an orbit inside the outer gear, due to hydraulic expansion of the chambers, which are isolated by contact between the inner and the outer gear. One inner gear in contact with an outer gear is illustrated to the right. The angle θ is a coordinate that describes the inner gear profile.

simulations of wearing-in behaviour of gear sets in an orbital type hydraulic motor. The model is applied to unique, measured gear set geometries. The numerical simulation results include changes in the forces between inner and outer gear, as well as the changes in the load sharing between lubricant and asperities. Finally, the component scale wear recorded in the experiments is compared with the wear depths predicted by numerical simulations.

2. Method

A numerical model, developed by the authors of this work, which was presented in [17], is used to estimate tribological conditions in a number of orbital type hydraulic motor gear sets. The hydraulic pressure in the motor is balanced by contact forces F_{tot}, at each contact point. The contact forces together with speeds, geometry and material data constitute known tribological conditions. The conditions, in combination with measured surface roughness, are used to estimate the extent of mixed lubrication. The load which is carried by direct asperity contact, F_a, is then used to calculate a pressure distribution. The process is repeated for a number of time increments, as the inner gear moves all the way around the excentre cycle. Once the whole rotation is complete, a theoretical model is used to estimate the wear on two scales. On both scales material is removed based on an Archard type of wear model. The numerical simulation results are compared with experimental results, in terms of wear, for the gear set geometry presented in Fig. 1.

2.1. Numerical model

The numerical model includes calculations of forces, contact points and velocities, based on what is observed on the global scale. These *tribological conditions* are calculated from measured surface profiles, as presented in [18]. On the local scale, a relation between applied force and resulting average surface separation between two rough surfaces is calculated. This local scale is defined here as the topography scale. The curve depicting the relation between contact force and separation is referred to as the contact stiffness or contact stiffness curve. The two scales of the model, the topography scale and the tribological conditions from the global scale, are considered in the mixed lubrication model. Wear is included in the model both on the topography scale and on the global scale.

The gears of the gear set, shown in Fig. 1, are symmetric around each inner and outer gear tooth. This means that the tribological conditions are cyclic and by using the symmetry, the calculation of tribological conditions can be done $(m-1)m$ times more efficiently. For a more detailed explanation, the reader is referred to [18]. The model presented in this paper does, however, not rely on

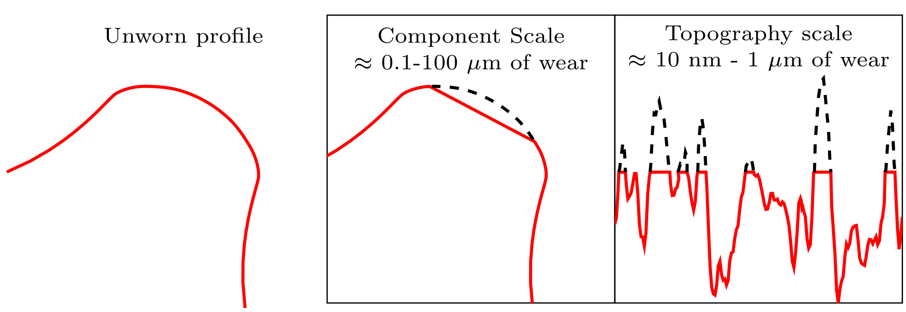

Fig. 2. A schematic image of two scales of wear which occur during wearing-in. Component scale wear changes the conformity between the machine pair, and topography scale wear affects the roughness. The image of component scale wear is not in scale, in terms of wear depth. Furthermore, also in the image of component scale wear, the location of wear is not related to the real application.