Prediction of wear in rolling and sliding contacts

by

Sören Andersson
Machine Elements, Department of Machine Design
Royal Institute of Technology (KTH),
S-100 44 Stockholm, Sweden
soren@damek.kth.se

Mild wear of rolling and sliding contact surfaces can in some cases cause an increased risk for surface fatigue. Prediction of mild wear in roller bearings, gears and cam-follower contacts have therefore been studied at our laboratory. The procedure used is based on wear prediction as an initial value problem. The simulation is normally rather time consumable but some simplifications can be use to reduce the simulation time.

The main uncertainties when predicting wear in rolling and sliding contacts are, however, the relevance or accuracy of the wear model and of the parameters used. It is thus important that much more research efforts are put on interdisciplinary research between basic wear phenomena studies and developments of wear models for quantitative determination of wear.

Introduction

The possibility to predict behaviour and performance of technical systems is of vital importance for successful development and use of products. In many applications the interacting contact surfaces have a strong influence on the behaviour and performance. Many of these surfaces will wear, depending on the running conditions and the properties of the interacting surfaces and the interfacial environment. In most cases a product with its elements are well designed. The interacting surfaces will show only some mild wear mainly during the running-in phase followed by a nearly no wear continuing running. This is what we strive for and also is the case in many high performance products. In some cases a severe situation with unacceptable high friction and wear will occur and thus must be avoided. Prediction of that risk for transition from a mild running condition to a severe unacceptable condition is thus very important. However, even mild conditions can cause some secondary effects that can lead to failure. Surface form changes due to mild wear, has been observed to have a strong influence on the risk for surface fatigue in some high performance rolling and sliding contacts. That phenomenon has thus been the subject for a number of studies at our laboratory.

I will in this paper present our approach to predict mild wear in rolling and sliding contacts and some of the results obtained so far. I will also indicate a research field that have to be more focused on in order to improve the possibility to predict wear in general and in rolling and sliding contacts in particular. The results presented in this paper are limited to mixed and boundary lubricated rolling and sliding contacts.

Modelling of wear

The problem of predicting wear of surfaces can generally be looked at as an initial value problem. If we know the properties and the state of the contact surfaces at an arbitrary point of time $t = t_0$ and we can determine the relative motion and the transmitted loads of the surfaces, we can predict the wear depth h at any point on a contact surface for any time $t > t_0$ provided that a relevant wear model is available. A relevant wear model should then have the general form:

$$\frac{dh}{dt} = f\left(p, v, h, T, t, H_v, \mu, \eta, \ldots\ldots\ldots\ldots\ldots\ldots\right) \tag{1}$$

where p is contact pressure at the point, v is the sliding speed, T is temperature at the point, H_v is the hardness of the surface, μ is the coefficient of friction and η is the viscosity of the oil. Many other factors that might influence the wear could have been added to the list as well. By integrating equation (1) for different time, we get the wear depth distribution as a function of time.

The most used wear model so far is:

$$\frac{dh}{dt} = kpv \tag{2}$$

which sometimes is named "generalised Archard's equation" and can be rewritten in the following form since $ds = vdt$, namely:

$$\frac{dh}{ds} = kp \tag{3}$$

where s is the sliding distance the point has slid against the opposite interacting surface, k is the dimensional wear coefficient and v is the sliding speed at the point studied. k can be a function of many parameters and state values and may vary with time. Here we assume that k is a constant, since experimental results from mild wear tests often show that, the wear per sliding distance is nearly constant after a short running-in period and that the wear is approximately proportional to the applied pressure as long as the wear mechanism is the same. Transitions from one type of wear mechanism to another need, however, a much more complex description.

Sliding wear in rolling and sliding contacts

In rolling and sliding contacts a point on a surface is in contact a very short time. For an ideal stiff point contact, a point on the surface will theoretically not slide any distance against the opposite surface during a mesh, although there is a sliding speed between the surfaces. However, with real materials the contact surfaces are deformed and the contact width is not zero. A point passing through the contact will thus slide against the interacting opposite surface. It has been found that this sliding and particularly the total sliding distance is one of the most important parameters that determines the wear in rolling and sliding contacts.

An approach we have used to determine the sliding distance in rolling and sliding contacts was first introduces by the author in 1975 [1] for gears. The approach has been named "single point observation" and later used for prediction of wear in both roller bearings and gears [2,7]. Each point on a contact surface is followed when it passes through the contact and the distance the points have slid against the opposite interacting surface during the mesh is determined. This may be exemplified by two rolling and sliding wheels in contact. At $t=0$ we observe a point P_1 on surface 1 just entering the contact. We also observe a point on the other surface P_2 that is opposite point P_1 at $t=0$. The sliding speed of surface 1 and surface 2 are v_1 and v_2 respectively. Surface 1 moves faster than surface 2. After a short time interval $2a/v_1$ point P_1 leaves the contact and has then moved a distance 2a. Point P_2 ,however, has only moved $2av_2/v_1$. The sliding distance point P_1 has slid against the opposite surface 2 when it leaves the contact is thus:

$$s_1 = 2a - \frac{2a}{v_1}v_2 = 2a\left(1-\frac{v_2}{v_1}\right) = 2a\xi \qquad (4)$$

where $2a$ is the contact width in the rolling direction and ξ is the relative slip.

For gears one can derive equations in the same way for the sliding distances points on gear tooth flanks slide against the interacting surfaces during a mesh, see figure 1 [1,2,7]. However, the equations derived are much more complex than equations (4) for rollers.

The prediction of wear can be made by integrating equation (3) for all points on a contact surface according to:

$$h = \int_0^s kp\,ds \qquad (5)$$

If we assume that k and p are approximately constant during a mesh, the integral equation 5 can be simplified to the following numerical equation:

$$h_{P,n} = h_{P,(n-1)} + kp_{P,(n-1)}s_P \qquad (6)$$

where $h_{P,n}$ is the wear depth at point P after n iteration steps. $h_{P,(n-1)}$ is the wear depth after n-1 iterations, $p_{P,(n-1)}$ is the pressure at the point after n-1 iterations and s_P is the sliding distance the point has slid against the opposite interacting surface during the time interval considered. The pressures are determined by the Hertz formulas and the sliding distances are determined with the derived equations for sliding distances respectively. This simplified way to predict the wear of rolling and sliding contacts has been used in some works at

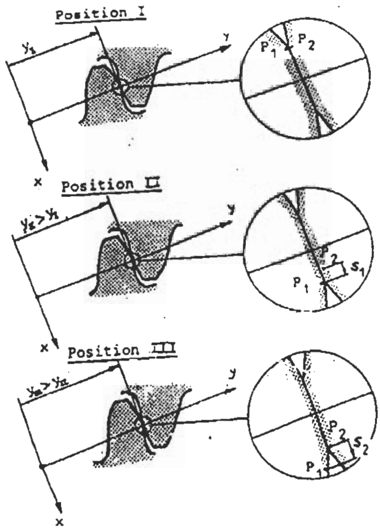
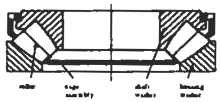


Figure 1. Gear tooth flank motion relative each other during one mesh.



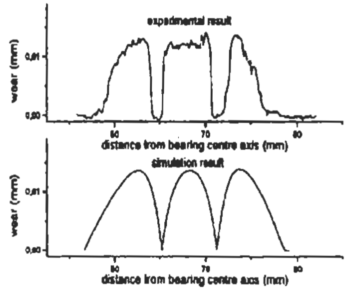


Figure 2. Experimental and simulated wear distribution of a washer surface of a spherical roller thrust bearing

the laboratory. Figure 2 show the experimental measured and predicted wear of a housing washer of a slow running spherical roller thrust bearing [5]. In this case slicing of the contact surfaces is also used to simplify the determination of pressure distribution in the contacts. Figure 3 show the results of wear simulation with this simplified approach on spur gears [2,7].

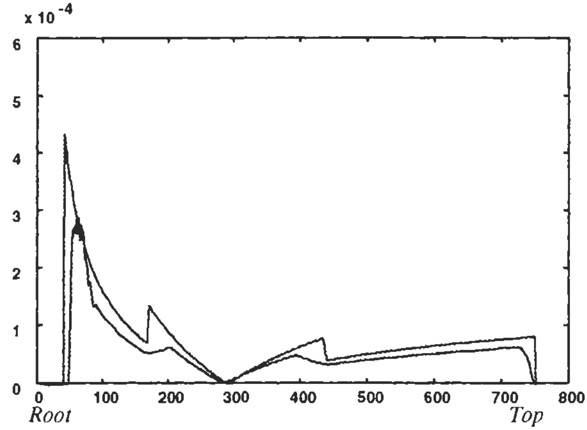


Figure 3. Simulated wear distribution of a spur gear pinion tooth flank. The upper curve is obtained with the assumption of constant pressure during the mesh [2,7]. The lower curve is obtained with a Winkler surface model [7].

Wear of the surfaces will change the surface form and the contact pressure can not in some cases be determined accurately enough with Hertz formulas. Other methods to determine the pressure distribution in the contacts must therefore be used. Modelling of surfaces as Winkler models have been made. Equation (2) has then been numerically integrated with small time steps. Results from simulations of the wear in helical gears are shown in figure 4 [6].

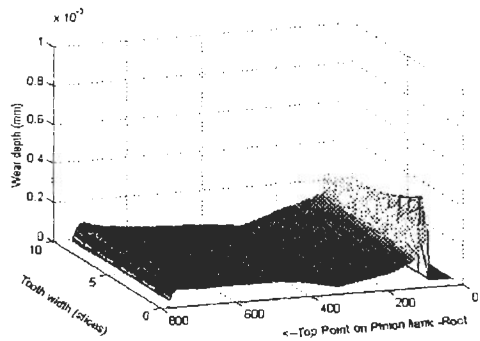


Figure 4. Simulated wear distribution of a helical gear [6].

Another case studied, where rolling and sliding occurs and where wear and surface fatigue sometimes is a problem, is a cam follower contact in a valve mechanism to a commercial Diesel engine. Results from simulations of the cam follower contact is shown in figure 5 [8]. The contact pressure distribution in this case was determined with an accurate numerical method developed by Björklund and Andersson [4].

Accurate determination of a pressure distribution is time consuming and slow down the simulation. However, computer power and numerical code developments have improved the possibility to use numerical contact mechanics programs and commercial codes for the determination of contact pressure distribution. Integration of wear simulation procedures in such codes have also been done and will be more common in the future.

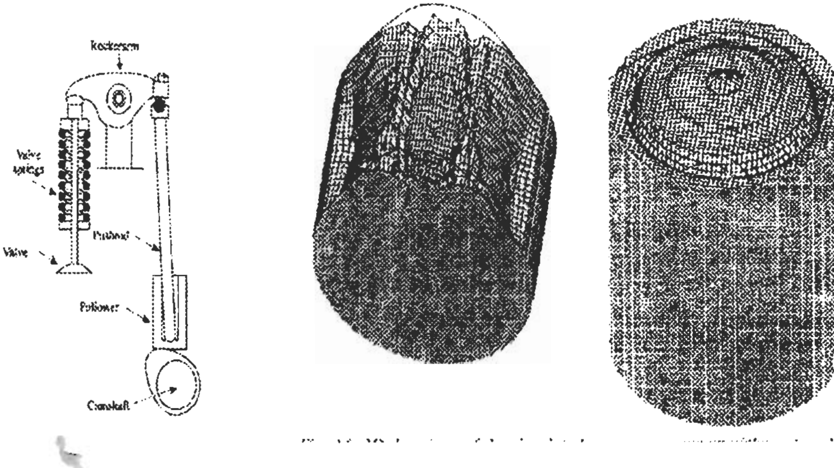


Figure 5. Simulated wear distribution of a cam and a follower of a valve mechanism [8].

Disussion

The interacting surfaces of the three cases presented are rolling and sliding against each other. In the gear case the motion of the contact surfaces is *motion controlled.* The rolling and sliding between the interacting gear tooth flanks are determined by the geometry and the rotation of the gears and the position on the surfaces. The main problem with gears is the determination of the transient loading and pressure distribution rather than the determination of the rolling and sliding between the surfaces. For roller bearings the motion of the rollers are *force controlled.* The motion of the rollers are then determined by the friction forces between the washers and the rollers. However, for the bearing studied the changes of the rollers rotation relative the washers have been very small and it has been possible to accurately determine zero sliding points that in its turn determines the motion at all other contact points in the contacts. In the third case, the cam follower, the motion of the contact surface of the cam is well defined by the geometry of the cam and the cam-shaft rotation. The motion of the follower, however, is determined by friction forces between the interacting surfaces. Since the load is varying a lot due to the motion and the dynamics of the valve mechanisms, the wear distribution and the contact pressure distribution are rather complex and change with time. The motion of the cam-follower contact and thus the wear over the surfaces is dependent of the frictional shear stress distribution in the contacts which in its turn is dependent of the pressure distribution in the contacts. The results of these interdependencies for one case with the same wear resistance on both surfaces are the wear distributions shown in figure 5 [8].

The changes of surface form has been found to have a strong influence on the risk for surface fatigue. For the spherical roller thrust bearings, the mild wear of the washers cause a gradually less favourable pressure distribution that will increase the risk for surface fatigue and thus reduce the life of the bearing [3]. A similar situation is observed for cam-follower contacts [8]. It is probable that this effect also has some relevance in gears [6].

The wear coefficient used in the cases studied have been $5 \cdot 10^{-10}$, $1 \cdot 10^{-10}$ and $1 \cdot 10^{-12}$ mm^2/N for the gear, roller bearing and cam follower case respectively. These values are typical for mixed and boundary lubricated sliding contacts running under mild conditions. If you estimate how much is worn each time a point pass through the contact, you will find that the value will be extremely small. It is thus not possible that the process is continuous but instead must be discontinues or local and distributed over the surfaces but for a macro observer will appear as a smooth continuous process. The base or connection between micro phenomena and macro appearance is therefore of great interest. You may find a big number of wear models in the literature. Most of them are not general applicable and others are difficult to use due to the difficulty to find relevant parameters for the particular case studied. The different models are however based on some general interpretation that can roughly be summarised by the following interpretation of the wear coefficient k. The most common interpretation of the wear coefficient k is as the probability that an asperity interaction results in a formation of a wear particle. The wear coefficient has also been related to a fatigue process i.e. the number of load cycles required to form wear particles. The coefficient has also been proposed to reflect the complex of surface oxide films and other boundary surface films that normally are formed and removed by for example a delamination process or a desorption process. Although, many fundamental wear mechanism studies have been made, the quantification of the different micro processes and how they interact and together give the overall wear is not satisfactory treated yet. Much more effort must be put on that part before we can predict mild wear accurate enough.

Conclusions

Sliding wear in all type of rolling and sliding contacts can be predicted by the principles developed at our laboratory. The prediction of wear is looked at as an initial value problem and the wear at all points on contact surfaces can be predicted by numerical integration. Since the wear prediction is formulated as an ordinary dynamic field problem, the interaction between the surrounded system and the contact studied can easily be considered. However, an accurate simulation is normally time consuming but some simplifications can be used to reduce simulation time considerable.

The main uncertainty when predicting wear in rolling and sliding contacts is, however, the relevance or accuracy of the wear model itself and parameters used. How well will it imitate the process? Do we pass any transformation that can not be considered by the model? These are typical questions that one never can get any good answer to. Therefore, it is very important that much more works are focused on the interdisciplinary area between basic wear phenomena and wear models for quantitative determination of wear.

Referenses

1. Andersson S. Partial EHD Theory and Initial Wear of Gears. Doctoral Thesis. Royal Institute of Technology, Stockholm. 1975.
2. Andersson S. and Eriksson B.. Prediction of the Sliding Wear of Spur geras. Proceeding of NORDTRIB'90, ISBN 87 98353802.
3. Olofsson U. Fatigue life reduction due to wear in boundary lubricated spherical roller thrust bearings. Wear 207 (1997) 74-78.
4. Björklund S. And Andersson S. A numerical method for real elastic contacts subjected to normal and tangential loading. Wear 179 (1994) 117-122.
5. Olofsson U., Andersson S. and Björklund S.. Simulation of Mild Wear in Boundary Lubricated Spherical Roller Beraings. AUSTRIB'98, Proceeding of the 5th International Tribology Conference in Australia, Brisbane 6-9 Dec. 1998.
6. Flodin A. and Andersson S.. Simulation of Mild Wear in helical Gears. AUSTRIB'98, Proceeding of the 5th International Tribology Conference in Australia, Brisbane 6-9 Dec. 1998.
7. Flodin A. and Andersson S.. Simulation of Mild Wear in Spur Gears. Wear 207 (1997) 16-23.
8. Hugnell A. B. J. , Björklund S. and Andersson S.. Simulation of mild wear in a cam-follower contact with follower rotation. Wear 199 (1996) 202-210.