Analysis of Wear Phenomena in Sliding Contact Surfaces

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ABSTRACT

A general approach to numerically simulating wear in rolling and sliding contacts is presented in this thesis. A simulation scheme is developed that calculates the wear at a detailed level. The removal of material follows Archard's wear law, which states that the reduction of volume is linearly proportional to the sliding distance, the normal load and the wear coefficient

Careful attention is paid to stress properties in the normal direction of the contact. A Winkler method is used to calculate the normal pressure. The model is calibrated either with results from Finite Element simulations (which can include a plastic material model) or a linear-elastic contact model. The tangential tractions and the sliding distances are calculated using a method that incorporates the effect of rigid body motion and tangential deformations in the contact zone. Results experiments (full-scale, pin-on-disc and disc-on-disc) were used to establish the wear\and friction coefficients under different operating conditions. For the disc-on-disc simulation, there was good agreement between experimental results and the simulation in terms of wear and rolling friction under different operating conditions.

Keywords: disc on disc, disc-on-disc, pin-on-disc, Archard, wear simulation, Winkler, rolling, sliding

1. INTRODUCTION

One of the basic tasks in the study of machine elements has traditionally been the characterization of wear. Wear is defined as the material loss or change in surface texture occurring when two or three surfaces of mechanical components contact each other. There are many different types of wear and a widely varying range of working conditions,
between the wear analyses of different machine elements such as roller-bearings [2], cam followers [3] and gears [4]. While the dynamics and geometry are different, the material is more or less the same. What is known as Archard's linear wear law has traditionally been used in the study of both sliding and rolling-sliding contacts. This law assumes that wear is proportional to normal load, the sliding distance and a wear coefficient, divided by thesurface hardness.

Since the 1980s, wear modelers have begun to use relevant theories from other fields of engineering to explain such wear phenomena as plastic deformation, fatigue, heat generation, oxidation, and crack formation and propagation. Many of these phenomena have been studied in detail in other fields and validated theories have been developed. The adopted theories have also been used to describe variations in working conditions and some single phenomena during the wear process.

1.1 CLASSIFICATION IN RELATION TO SEVERITY OF WEAR

Wearing systems have been classified in terms of the severity of wear on the wearing surfaces. Archard and Hirst [6] proposed two broad types of wear phenomena: severe wear and mild wear. Severe wear is characterized by high wear rates, extensive plastic deformation, transfer of material to the harder counter face, and flake-like metallic wear debris. Mild wear, by contrast, is characterized by low wear rates, minimal plastic deformation, formation of a surface film protecting against metal-to-metal contact, and oxide wear debris.

A wearing system consists of a number of mechanisms that need to be precisely defined in order to avoid overlaps in wear analysis. For example, the severity of wear needs to be defined in terms of precise, well-accepted definitions of such features as the amount of mass loss, the coefficient of friction and the surface roughness in order to accurately distinguish mild and severe wear. Without such precise definitions, models of wear may produce different results on the basis of different classifications

1.2 DISC ON DISC SLIDING CONTACT ANALYSIS

Continuum rolling contact theory started with a publication by Carter [18], in which he approximated the disc by a cylinder and the disc by an infinite half-space. The analysis was two-dimensional and an exact solution was found. Carter showed that the difference between the circumferential velocity of a driven disc and the translational velocity of the disc has a non-zero value as soon as an accelerating or a braking

couple is applied to the disc. This difference increases as the couple increases until
the maximum value according to Coulomb's law is reached. Carter formulated a creep-force law relating the driving–braking couple and the velocity difference. Carter's theory is adequate for describing the action of driven discs (for example, it
is capable of predicting the frictional losses in a locomotive driving disc). However, it is not sufficient for vehicle motion simulations that involve lateral forces as well as the motion in rolling direction [19].

Johnson [20] generalized Carter's results to circular contacts and longitudinal and lateral creep. Vermilion and Johnson [21] generalized this theory to elliptical contact areas. Shen et al. [22] improved the results by replacing the approximate values for
the creep coefficients given by Vermeulen and Johnson with more accurate values. All of this work is Hertzian-based, giving contact solutions for a class of geometrical objects satisfying the half-space restriction [23]. In the development of wear modeling in the discway context, an Archard-like wear model developed by Li and Kalker [24] has also been used in which the normal load is replaced by the frictional load or, in other words, the normal load is multiplied by the friction coefficient

There is a lack of work connecting different phenomena, and too many oversimplifications that attempt to deal with the whole issue in terms of stresses only or, at the opposite extreme, attempt to apply various wear coefficients in applications
without having much of a theoretical structure and an understanding of the sources and circumstances of wear. It is important to remember that shearing and stress related failures happen around the sticking region of the contact because of the static hooking of asperities that can move back and forth. Adhesive wear occurs primarily under sliding conditions, where asperities are beating each other under a transient load and stress-related effects may also be present

1.3 THE GOAL

Accurate wear modeling requires detailed description of many different wear phenomena that occur simultaneously on wearing surfaces if analytical models to explain wear phenomena in wear system.

The goal of this paper is to standardize the mathematical expression of different wear phenomena. The long term goal must br to devise a wear classification scheme based primarily on mechanisms by which the asperities deform and particles are detached. The first steps will be take towards constructing a flexible simulations models in which the nature of the wear mechanism can change depending on various geometric, kinematics and structural parameters rather produce a new wear equation

The study in paper showed that there are three methods of disc-disc wear analysis: - shakedown and plasticity effects, which operate continuously in real contacts; the non-elliptical shape of the contact zone, especially for worn profiles; - the velocity difference between the disc head and disc edge, which can be more than 1m/s, and which changes direction rotationally, causing spin. In order to model these observations correctly, it is necessary to investigate them

2. RESEARCH METHODOLOGY

The simulation tool for wear analysis represents an attempt to achieve an integrated understanding of wear and other degradation mechanism. Analysis of metal to metal contact is a common element in machine design, as it is clear from large no. of wear tests need that attempt to systematize and responds flexibly to working conditions that produces wear.

For the purpose specialized software's has been used throughout the world like Vampire,Nucars wet . All of these are highly specialise4d.There are potential function based mathematical models such as simplified theoryfastism bt J.J Kalker.In thsis theory the surface displacement at one unique point depends only on the surface traction at that point(Winkler Model what is often referred as complete theory. was implemented in a computer program called Contact[32] which is based on boundary element method .Kalketr extended his theory of rolling contact between arbitrary bodies tro the case where shape of contact area is non-elliptical.in order to get the approximate solution contact area is divided in the rectangular elements.

The experimental form measurements showed that there was a significant change in the disc profile due to both wear and plastic deformation and that both processes influence the form of a disc that has been in use for more than 5 years. The surface hardness measurement showed that the hardness of the new disc increased, but that after 2 years' use it had not yet reached the hardness of the old disc. These experimental results show that plastic deformation is a necessary element in disc contact analysis.

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2.1 FE ANALYSIS

First to overcome the limitations inherent in traditional approaches and their lack of ability to analysis plasticity, a tool for FE based. The routines are written in ANSYS[34].the meshing can be based on disc on disc profiles. The kinematics constraints are enforced with kinematic hardening. The quasi-static loads were obtained from disc to disc dynamic calculations. With special purpose MBS software.

In finite element method, plasticity is modeled according to establish plasticity theories but the time taken to do this is impractical for wear analysis

2.2 CONTACT LOCALITY ROLLING-SLIDING ANALYSIS

The FE method has undergone significant improvement, but parallel progress of FE **method and** faster numerical methods are also interesting.

The contact locality is rectangular area that is large enough to envelope the true contact region. The benefits of solutions only on contact locality are as follows

a) it's not necessary to model the bodies as only the surface is discretised

b) the tangential solution can be achieved by one computation.

The transformation of contact locality are performed in following steps

a) To create 3 dimensional geometries of both discs.

b) To rotate disc geometries for normal solution. Both profiles are rotated to resultant normal load direction and contact localities of the both discs together.

c) To move both discs back to their positions in space after the normal solutions. The penetrations calculated in normal solution are now included and velocities are calculated.

d) To transform the 3 dimensional geometries to a fat surface. This velocity and creep system is rotated so that Euler angle becomes zero. this simplifies the calculations of tangential of tangential forces since only the components in x and y directions are used.

e) To rotate the disc profiles again to the resultant normal load directions where the original 2 dimensional data is updated with form change due to wear.

f) To transform back to original position in order to create a new contact localities of disc.

The application of rigid body velocities is particular important when updating the geometries and developing a sort of "rough surfaces" during the wear .The time step is associated and length and is imp-ortant when using time dependent equation:

$$t = \frac{\arctan(x/R)}{\Omega}$$

where x is length in rolling direct ion R is radius of disc and Ω is constant of rotation velocity[rad/sec].At different velocities and load conditions different wear coefficients apply. The removal of material follows Archard's law.

$$W_{i,j} = \frac{k.P_{i,j}\ x\ [U^2 x_{,i,j}\ \mathrm{SLIDING} + U^2 y_{,i,J}\mathrm{sliding}]^{½}}{H}$$

Where H is hardness of material. Using the formula W=ΣW following parameters are calculated

-wear volume per meter determined form volume knowing that the steady state solution gives wear volume per discretization unit ▲x using W/▲x

-mass loss per meter

-wear volume per disc revolution

-Wear depth

2.3 NORMAL SOLUTION

In the earlier studies presented in paper, a finite element method was employed to predict the changes in contact properties when subjected to high loads. A particular elastic-plastic material model was simulated. There is a number of different material
Models, and therefore the results are qualitative, predicting a rise in the contact area and a decrease in the maximum contact pressure. A comparable method as regards computation time versus accuracy of the structural properties is the Winkler mattress method used in papers C and D. The normal displacement is related to the normal contact pressure by

$$\mathbf{Uz = p / KN}$$

Where KN is the linear modulus of the foundation. According to equation 3, the influence function becomes a constant in equation 6, i.e., Cz = 1/KN. KN can be determined by experimental work, by FEM analysis or by comparison with another
calculation theory, such as Hertzian theory [36].
The method outlined in paper is valuable when focusing on linear-elastic contacts. The Winkler brush model (used in papers C and D) leaves some parameters as unknowns. The method adopted in paper establishes the link between a pure linear elastic solution and the Winkler brush method that has to be adjusted for different Solutions. The method is based on several assumptions that challenge the assumptions in previous implementations of influence functions [32, 37]. These assumptions are:
15- that the pressure distribution on each rectangular cell is approximately constant. Love's [38] solutions cover the area using rectangular elements. This solution meets the boundary conditions and the superposition principle is valid. Although there are still restrictions on the curvature, Hertzian geometries are certainly valid; - that the solution is to treat the initial overlapping of two bodies as a purely geometric problem. The overlap is gradually eliminated by proceeding in discrete steps, each a predetermined fraction (1/1000 or the like) of the maximum overlap; that every discrete step may consist of several equal lengths in different places (the corresponding pressure is automatically found) and that the order of succession within this step is not important; that every discrete subtraction length also successively subtracts the influence lengths at neighboring cells so that the total subtraction is made for the entire overlap (bodies);
- that those discrete lengths (without the neighbor effects) are accumulated because they are directly proportional to pressure (force). The addition of
those discrete lengths can be stopped if any total load restriction is met, enabling computation to be either load-based or approach-based. The normal component in equation 3 is expressed in this case as

$$Cz(x,y)=(1-v)(gAxx+gBxx)/2\pi G$$

where gAxx and gBxx are given in equation 5 and are the functions that scale the displacements for the neighboring cell. The problem is solved using only geometric parameters. Exact Hertzian solutions are obtained. Moreover, the bodies may be described as general polynomials (with variable curvatures) in any order and several concurrent contacts can be solved solely on the basis of geometric overlapping.

2.4 TANGENTIAL SOLUTION
This section contains a review of selected aspects of wear analysis focusing on the Formulation of tangential contact problems for deformable discrete surfaces. In contact problems, frictional effects are generally accounted for by the introduction of
a friction law that relates the sliding velocity to the contact forces. The tangential component of the contact tractions, or frictional traction, can be exerted without sliding, i.e., under stick conditions, until a certain threshold is overcome to allow sliding. According to Coulomb's law, the threshold is proportional to the magnitude of the normal pressure. When sliding occurs, the frictional tractions always oppose the sliding velocity and are, therefore, dissipative

We shall be concerned with the motions of a deformable body, but first the rigid body motions are determined. Rigid body kinematic expressions provide many commonly used functions for dealing with rigid body attitude coordinates. The rotation matrix includes the time dependent Euler angles ϕ, θ, ψ are roll, pitch and yaw, respectively.
In the present case $\phi \rightarrow \gamma$ will be the inclination matrix of the perpendicular of roller
curvature and $\theta = \Omega t$ is the angle of curvature of the roller radius.

The basic steps for tangential solution as described in Paper are as follows:

- Calculate relative velocities and creep for the rigid body;

- Use an artificial displacement field created by creep ratios and enlarged by the influence of neighboring cells. The influence of a neighboring element is determined logarithmically by the solutions of potential theory for constant traction on a rectangular area.

-Creep times the discretisation unit, Δx in square is linearly (in every index separately) divided by the artificial displacements;

- Cumulatively sum the result from the beginning of the contact (in the rolling direction). The results are directly proportional to the tangential surface tractions. Check surface tractions by the frictional bounds and, if applicable, reduce them to level $\mu \cdot \mathbf{P}$;

- Modify the elastic displacements in line based on the previous restriction.

The part that was cut is the sliding component used in Archard's wear
equation;

- Calculate the wear volume using Archard's wear law.

3. CONCLUSION AND RESULTS

The analysis of disc-disc interaction has been presented by using FEM technique. However this work give rise to thoughts about possible applications in various fields The main results of FE modeling are as follows:

1. A FE tool of discon disc analysis is developed. This tool allows easy changing of the geometry of contact. Measured disc profiles are used.

2.The results of 2 sets cases presented show that difference in max contact pressure between contact and hertzian method and FE method are negligible where radii of curvature of 2 contacting bodies at contact point where large compared with significant dimensions of contact area.

4. FUTURE WORK

The focus so far has been on disc-. analysis. However, the work done in this field suggests future possibilities in a number of engineering fields using the methods introduced here.

4.1 RANDOMNESS, TIME-DEPENDENCE AND ROUGH SURFACE

The use of variation in discretisation introduces the possibility of making greater use of what is known as
the Monte Carlo technique if the variables change randomly based on their probabilistic distribution. In the present case, only the lengths of Δx and Δy are variable. MBS data such as attitude angles between the disc and the . and the global creep ratio νr GLOBAL, can be varied, as can the pin-on-disc data coefficient of friction μ, and the wear coefficient *k*. Many other parameters may be varied within their probabilistic bounds between the different cells in contact or between the computation steps, because the computation steps progress with reasonable frequency. For instance, Beckmann and Dierich [40] proposed that wear prognoses must take account of the statistical nature of hardness. The methods robustness can be analyzed by comparing input and output variation and more general relationships can also be found. The introduced transformation by the time-dependent Euler angles and the corresponding velocities (accelerations) permits the study of transient motions.It will also be possible to study the effects of a rough surface on a rectangular area.
Such a study could be statistical, in the form of what is known as the Abbott curve implementation, or could involve precisely measured asperities mechanically attached (although this approach would be rather time-consuming).

4.2 WEAR

Future work in regard to wear can be divided into long-term and short-term plans. The short-term plans involve disc-. analysis to study how lubricated and wet conditions affect the degradation mechanisms in disc-. wear. With lubrication, the elastic tension in the tangential direction is shortened due to the decrease in the friction coefficient and the plastic flow effect is reduced. In the long-term perspective, the aim will be to study range of materials to determine
how the plastic limit indicated as equivalent stress on the surface affects the wear coefficient. A related problem is the 'softening' of the so far optionally linear-elastic tangential solution in the proposed method in paper of roughness on friction and the wear coefficient is not only interesting in general but also affects the validity of the method through discretisation and the half space assumption. An important part of curved geometries is determining what metric length constituting a valid discretisation length

4.3 PERSPECTIVES ON PLASTIC FLOW AND FATIGUE ANALYSIS

In the approach adopted in this thesis, sliding displacements and elastic displacements are separated. The loading of a train disc on . is usually such that plastic flow occurs in the . with every disc passage, imparting a small increment of plastic strain in the opposite direction to traction. This strain accumulates until it reaches the ductility of the material, at which point rupture occurs. The incremental rise in surface sliding is obvious at the disc flange. The relative transversal velocities for the t.ing disc in a bogie have motions in the flange contact in the opposite direction. However, the leading disc has a greater effect on the flange contact because it is steering the bogie.

5. REFERENCES

[1] Nilsson, R., 'Disc and . wear – measured profile and hardness changes during 2.5 years for Stockholm commuter traffic, .way Engineering – 2000, London, 5-6 July, 2000

[2] Olofsson, U., Andersson S. and Björklund S. 'Simulation of mild wear in boundary lubricated spherical roller thrust bearings' *Wear* 241 pp**.** 180**–**185, 2000

[3] Hugnell, A. B**.**-J. **'***Simulation of the Dynamics and Wear in a Cam-Follower Contact'* Ph.D. thesis, Stockholm**,** 1995

[4] Flodin, A. *'Wear of spur and helical gears'* Ph.D. thesis, Stockholm**,** 2000

[5] Archard, J. F. 'Contact and rubbing of flat surfaces' *Journal of Applied Physics*, Vol. 24 pp. 981–988, 1953

[6] Archard, J. F. and Hirst, W. 'Wear of metals under unlubricated conditions' *Proc. R. Soc., London*. A, 1956, 236, 3–55

[7] Peterson, M. B. and Winer, W. O., *Wear Control Handbook* ASME, New York, 1980

[8] Finnie, I. 'Some observations on the erosion of ductile metals' *Wear* 19 pp. 81, 1972

[9] Jennings, W. H., Head, W. J. and Manning, C. R., Jr. 'A mechanistic model for the prediction of ductile erosion' *Wear* 40 pp. 93, 1976

[10] Evans, A. G. 'Impact damage Mechanics: Solid Projectiles' Treatise on material science and technology', Vol. 16, *Erosion*, C. M. Preece (ed.) , Academic, New York, p. 1, 1979

[11] Hutchings, I. M. 'A model for the erosion of metals by spherical particles at normal incidence' *Wear* 70 pp**.** 269–281, 1981

[12] Ringsberg, J. W., Loo-Morrey, M., Josefson, B. L. and Beynon, J. H. 'Prediction of fatigue crack initiation for rolling contact fatigue', Int. J. of Fatigue, 22, pp. 205–215, 1999

[13] Waara, P. 'Wear prediction performance of . flange lubrication' Licentiate thesis**,** Lulea, 2001

[14] Knothe, K. and Grassie, S. L. 'Modelling of .way track and vehicle/track interaction at high frequencies' *Vehicle System Dynamics*, 22(3–4): pp**.** 209–262, 1993

[15] Igeland, A. and Ilias, H. '. head corrugation growth predictions based on non-linear high frequency vehicle/track interaction' *Wear* 213 pp**.** 90–97, 1997

[16] Jendel, T. 'Prediction of disc profile wear: Methodology and verification' Licentiate thesis, Royal Institute of Technology, Sweden, TRITA-FKT 2000:49

[17] Jendel, T. 'Prediction of disc profile wear: Comparison with field measurements', *Proceedings of Contact Mechanics and Wear of ./Disc Systems, Tokyo, 25–27 July, 2000*

[18] Carter, F. W. On the action of a locomotive driving disc, *Proc. Roy. Soc., London*, Ser. A, pp**.** 151–157, 1926

[19] Kalker, J. J. 'Disc-. rolling contact theory', *Wear* 144 pp. 243–261, 1991

[20] Johnson, K. L. 'The effect of a tangential contact force upon the rolling motion of an elastic sphere on a plane', *J. Appl. Mech.*, 25, pp. 339–346, 1958

[21] Vermeulen, P. J. and Johnson, K. L. 'Contact of non-spherical bodies transmitting tangential forces', *J. Appl. Mech.*, 31, pp**.** 338–340, 1964 27

[22] Shen, Z. Y., Hedrick, J. K., and Elkins, J. A. 'A comparison of alternative creep-force models for . vehicle dynamic analysis', *The Dynamics of Vehicles, Proc. 8th IAVSD Symp.*, Hedrick, J. K. (ed.), Cambridge, MA, Swets and Zeitlinger, Lisse pp**.** 591–605, 1984

[23] Johnson, K. L. *Contact Mechanics*, Cambridge University Press, 1985

[24] Li, Z. L. and Kalker, J. J. **'**Simulation of Severe Disc-. Wear'**,** *Proc. of the 6thInternational Conference on Computer Aided Design, Manufacture and Operation in the.way and Other Advanced Mass Transit Systems:* Computational Mechanics Publications,Southampton, pp**.** 393–402, 1998

[25] Li, Z. L., Kalker, J. J., Wiersma, P. K. and Snijders, E. R. 'Non-Hertzian disc-. wear simulation in vehicle dynamical systems' *Proc. of the 4th Int. Conf. on .way Bogies and Running Gears, Budapest, 1998*

[26] Linder, C. and Brauschli, H. 'Prediction of disc wear' *Proc. of 2nd Mini Conf. on Contact Mechanics and Wear of ./Disc Systems, Budapest, July 29–30, 1996*

[27] Fries, R. H. and Davila, C. G. 'Analytical methods for disc and . wear prediction' *Proceedings 9th IAVSD Symposium, Linköping, 1985*

[28] Andersson, E., Berg, M. and Stichel, S. *Dynamics of . Vehicles* (in Swedish), .way Technology, Dept. of Vehicle Engineering, Royal Institute of Technology, KTH, Stockholm, Sweden, 2000

[29] Gensys, *Gensys User's Manual*, Release 9910, Desolver, 1999

[30] Knothe, K. L., Strzyakowski, Z., and Willner, K. '. vibrations in the high frequency range', *Journal of Sound and Vibration*, 169(1), pp. 111–123, 1994

[31] Kalker, J. J. 'A fast algorithm for the simplified theory of rolling contact', *VehicleSystem Dynamics*, 11, pp. 1–13, 1982

[32] Kalker, J. J. 'Disc-. wear calculations with the program CONTACT', in Gladwell, G. M. L., Ghonen, H. and Kalousek, J. (eds.), *Proc. Int. Symp. on Contact Mechanics and Wear of .-Disc Systems II, Kingston, RI, July 1986,* University of Waterloo Press, Waterloo, Ontario, pp. 3–26, 1987

[33] Telliskivi, T., Olofsson, U., Sellgren, U. and Kruse, P. 'A tool and a method for FE analysis of disc and . interaction' *Proceedings ANSYS Conference in Pittsburgh, Pennsylvania, 2000*

[34] Ansys, *ANSYS Theory Reference*, release 5.4, ANSYS, 1997

[35] Lim, S. C. and Ashby, M. F. 'Wear-mechanism maps' Acta Metal. vol. 35, no. 1, pp. 1-24, 1987

[36] Põdra, P. and Andersson, S. 'Wear simulation with the Winkler surface model', *Wear* 207 pp. 79–85, 1997

[37] Johnson, K. L. 'The application of shakedown principles in rolling and sliding contact' *Eur. J. Mech*., A/Solids, 11, pp. 155–172, 1992

[38] Love, A. E. H. 'The stress produced in a semi-infinite solid by pressure on part of the boundary', *Phil. Trans. Royal Society*, A228, 377, pp. 54–55, 1929

[39] Olofsson, U. 'Characterisation of wear in boundary lubricated spherical roller thrust bearings', *Wear* 208 pp. 194–203, 1997

[40] Beckmann, G. and Dierich, P. 'A.