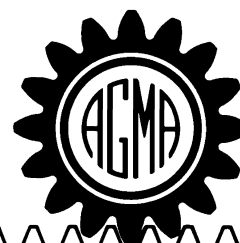


Hypoid Gear Lapping Wear Coefficient and Simulation

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American Gear Manufacturers Association



TECHNICAL PAPER

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Claude Gosselin and Qimi Jiang, Laval University, Kevin Jensi and Jack Masseth, American Axle and Manufacturing

[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

Because of the large volume of manufactured sets, hypoid gears are usually hard finished after heat treatment using the lapping process. In the lapping process, a gear set is run at varying operating positions and under a light load in order to lap the tooth surface. An abrasive lapping compound is used as a metal removal media.

Because of the rolling and sliding motion inherent to hypoid gears, the lapping compound abrades and refines the tooth surface to achieve smoothness in rolling action and produce high quality gear sets. The pinions and gears are lapped in pairs and must therefore remain as coordinated pairs for the rest of their lives.

However, heat treatment distortion can vary significantly from batch to batch, and even within one batch if the temperature is not consistent throughout the heat treatment furnace. Thus, developing a lapping sequence for manufacturing requires both time and experienced technicians who can establish lapping operating positions and sequence times to produce quality gear sets both in terms of performance and cost. This development is generally trial and error as past operator experiences factor heavily into the process.

In this paper, the lapping process is simulated using advanced modelling tools such as gear vectorial simulation for the tooth surfaces and path of contact and reverse engineering to analyze the tooth contact pattern of existing gear sets under load (static LTCA). Test gear sets are measured using a CMM prior to a special lapping cycle where the position of the gear sets on the lapper does not change, and then re-measured after lapping in order to establish how much, and where, material was removed. A wear constant named "wear coefficient" specific to the lapping compound composition is then calculated.

Based on the obtained wear coefficient value, an algorithm for simulating the lapping process is presented. Gear sets lapped on the production line at American Axle and Manufacturing are used for simulation case studies. Results show that it is possible to predict how much and where material will be removed on the tooth surface, thereby opening the door to better understanding of the lapping process.

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HYPOID GEAR LAPPING WEAR COEFFICIENT AND SIMULATION

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1. Introduction

Hypoid gears are widely used in vehicles. Two basic cutting processes can be used, either face milling or face hobbing [1-3], the latter now often being preferred for its cost effectiveness. Whichever cutting process is used, tooth surface errors appear, in part because of tooling wear and machine distortion, in part because the gears are heat treated after cutting which releases internal strain and causes tooth flank distortion. Therefore, a finishing operation is necessary.

Finishing can be done by grinding the gear teeth, but it is expensive in terms of tooling and time, and is therefore usually limited to small productions such as aerospace gears. Finishing can also be done by lapping, whereby the gear set is operated for a short time, under limited speed and torque, at varying positions such that the abrasive lapping compound improves the contact surfaces. Lapping is normally applied to hypoid gear finishing because it is economical for large production volumes. As a result, excellent smoothness and quietness of operation can be obtained.

To evaluate tooth surface errors, gears are measured with a Coordinate Measurement Machine (CMM) to obtain the actual tooth flank topography. The CMM coordinates are compared to reference coordinates obtained from a 3 dimensional modelling software where the actual machine settings and cutter dimensions allow precise calculation of the theoretical tooth flank.

The CMM coordinates may be used to reverse engineer the actual tooth flank as developed by Gosselin [4]. Reverse engineering provides the machine settings of a theoretical reference

matching the actual tooth such that the kinematics of an actual gear set may be calculated the same way the theoretical gear set is calculated.

In practice, lapping is an abrasive wear process. The pinions and gears are lapped in pairs and must therefore remain as coordinated pairs in operation. Because of the variability in tooth surface errors caused by different machine production lines, varying tool wear, and different positions in the heat treatment furnace, tooth surface errors vary significantly from batch to batch, piece to piece, and even tooth to tooth. As a result, the contact pattern is different from gear set to gear set, while the lapping cycle must be same.

There are many factors affecting the lapping wear rate (weight removal per unit of time), including grain size, shape, material and hardness of the lapping grit, applied torque, pinion RPM, hardness of the gear set, etc. Thus, developing a lapping cycle for production requires both time and experienced technicians to establish lapping operating positions and cycle time to produce quality gear sets.

So far, theoretical knowledge on the relationship between material removal, lapping time and applied torque is limited. To help address this problem, one key parameter dubbed "wear coefficient" can be determined.

Since the wear coefficient can be affected by factors such as grain size, shape, material and hardness of the lapping grit, this work is based on lapping tests made using the same lapping compound and conditions such that the above factors can be neglected. In other words, this work focuses on presenting an algorithm working with data measured from hypoid gear sets lapped

under the same conditions (same production lines).

The lapping process is reproduced using advanced modelling tools such as gear tooth simulation for the tooth surfaces and reverse engineering to analyze the tooth contact of the test gear sets. The test gear sets are measured using a CMM prior to a special lapping cycle in which the position of the pinion and gear does not change; the gear sets are measured after lapping in order to establish the distribution of material removal. A wear coefficient is thus calculated and used to simulate the lapping process of production gear sets.

Results show significant scattering of tooth heat treatment distortion, from tooth to tooth and from gear set to gear set, which makes the simulation process difficult. However, results also provide a rather constant wear coefficient, which is used to predict how much and where material will be removed on the teeth.

2. Main Nomenclature

A	contact area on the surface (m^2)
H	Knoop hardness of the soft material (N/m^2).
L	sliding distance, (m)
V_m	volume of material removed by wear, (m^3)
W	load normal to the tooth surface, (N)
h	thickness of removed material. (m)
k	abrasive wear coefficient,
r_j	ratio of the radius at point P_j to the radius at the mean point.
dS_j	difference in residual surface errors after and before lapping at corner point P_j ($j = TB, TT, HT$ or HB) (m);
σ_i	average contact pressure at point i , (Mpa)
v_i	average relative rolling and sliding speed between the tooth surfaces of pinion and gear at point i (m/s)
ω	rotation speed of the pinion or gear (RPM).

3. Wear Coefficient Calculation

3.1 Abrasive Wear

Basically, there are two abrasive wear mechanisms in hypoid gear lapping, generally referred to as two-body and three-body abrasive wear respectively [8].

In two-body wear, a hard rough surface is in contact with another softer surface; in three-body wear, large hard abrasive grains are in contact with two softer surfaces.

Literature results show that the wear equations of these two basic types of abrasive wear have the same form [9-10]:

$$V_m = k \frac{W L}{H} \quad (1)$$

Eqn.(1) shows that the observed wear rate is proportional to the load and the sliding distance, but inversely proportional to hardness. Kruschov [11], Rabinowicz et al. [12], Aleinikov [13], Mulhearn and Samuels [14], Avient et al. [15] verified experimentally the validity of Eqn. (1). So, the details of the two-body and three-body abrasive wear mechanisms will be neglected.

Avient et al. [15], Spurr et al. [16], Lopa [17], Kruschov and Babichev [18], Samuels [19], Toporov [20], Rabinowicz et al. [12, 21] experimented with different materials, and obtained typical values for k .

Wear rate, and therefore the wear coefficient, can be affected by factors such as hardness [22-25], the size of abrasive materials [12, 14, 15], the atmospheric moisture content [26], etc. Kallas investigated the wear energy in the abrasive wear process [27]. Kato discussed the abrasive wear mode and abrasive wear rate from the viewpoints of effective work for plastic deformation and fracture [28]. Bradley et al. analyzed the relative abrasive wear rate using wall friction instead of hardness [29]. Mason and Rooij investigated the abrasive wear between rough surfaces in deep drawing [30].

From the above, one concludes that it is impossible to obtain a constant value for k which can be applied to all materials and situations. Thus, it is not possible to select from the literature

a suitable value for the abrasive wear coefficient k that would be directly applicable to lapping as lapping is a different abrasive wear process from the above literature.

3.2 Formulation and Procedure

To address wear in hypoid gear lapping, one may rewrite Equation (1) as:

$$h = k \frac{W L}{H A} \quad (2)$$

At any contact point, under load a contact region is created between the hypoid pinion and gear tooth surfaces where contact pressure is distributed in an elliptical fashion.

If one divides the contact area into many small elements, as shown in Figure1, the worn thickness of element e_i during a period of time Δt can be expressed as:

$$\Delta h_i = k \frac{W_i v_i \Delta t}{H A_i} = k \frac{\sigma_i v_i \Delta t}{H} \quad (3)$$

So the wear rate (thickness of removed material) of element e_i will be:

$$\delta_i = \frac{\Delta h_i}{\Delta t} = k \frac{\sigma_i v_i}{H} \quad (4)$$

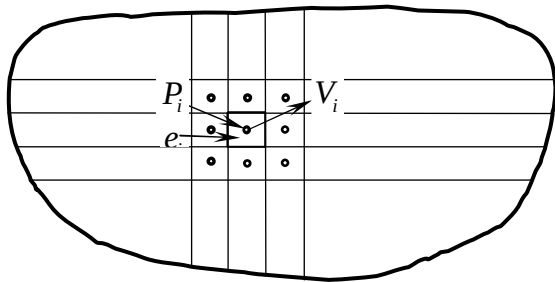


Figure1: Discretized contact area

When the size of element e_i becomes very small, δ_i will be the wear rate of point P_i . During one mesh, the sliding speed of point P_i can be regarded as unchangeable with time if the

operating position of the gear set remains constant: $v_i(t) = v_i$.

However, the contact pressure of point P_i is a function of time, $\sigma_i = \sigma_i(t)$, as shown in Figure

2. Thus, the wear rate of point P_i is a function of time over one mesh.

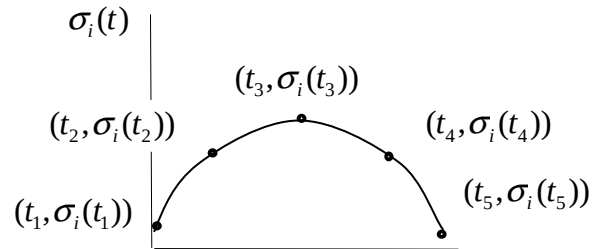


Figure 2: Contact pressure in one mesh

Suppose the contacting time of point P_i , that is the time over which point P_i is submitted to contact pressure, is t_i . The worn thickness at P_i for one mesh will be:

$$\begin{aligned} h_i^1 &= \int_0^{t_i} \delta_i(t) dt \\ &= \int_0^{t_i} k \frac{\sigma_i(t) v_i}{H} dt \\ &= \frac{k v_i}{H} \int_0^{t_i} \sigma_i(t) dt \end{aligned} \quad (5)$$

Contact pressure is obtained through a Loaded Tooth Contact Analysis software module [4, 5]. Five values for the contact pressure, at every contact point, are used for calculation. As shown in Figure 2, the five contact pressure values occur at:

- 1) time t_1 when contact point P_i comes into mesh where the contact pressure can be regarded as 0, i.e. $\sigma_i(t_1) = 0$;
- 2) time t_2 between t_1 (above) and the moment when the contact pressure becomes maximum;
- 3) time t_3 when the contact pressure becomes maximum;

4) time t_4 between t_3 and when meshing at point P_i ends;

5) time t_5 when meshing at point P_i ends, i.e. $\sigma_i(t_5) = 0$.

In Figure 2, as a general rule, Hertzian pressure is not symmetric about point t_3 .

Building up a coordinate frame as in Figure 2, the contact pressure $\sigma_i(t)$ can be interpolated with two parabolas as in Equation (6) such that the integral in Equation (5) can be calculated using numerical methods.

$$\sigma_i(t) = \begin{cases} \text{for } 0 \leq t \leq t_3 \\ \frac{t_2 \sigma_i(t_3) - t_3 \sigma_i(t_2)}{t_2 t_3 (t_3 - t_2)} t^2 + \frac{t_3^2 \sigma_i(t_2) - t_2^2 \sigma_i(t_3)}{t_2 t_3 (t_3 - t_2)} t \\ \text{for } t_3 \leq t \leq t_5 \\ \frac{(t_4 - t_5) \sigma_i(t_3) - (t_3 - t_5) \sigma_i(t_4)}{(t_3 - t_4)(t_3 - t_5)(t_4 - t_5)} t^2 + \frac{(t_3^2 - t_5^2) \sigma_i(t_4) - (t_4^2 - t_5^2) \sigma_i(t_3)}{(t_3 - t_4)(t_3 - t_5)(t_4 - t_5)} t + \frac{t_4 t_5 (t_4 - t_5) \sigma_i(t_3) - t_3 t_5 (t_3 - t_5) \sigma_i(t_4)}{(t_3 - t_4)(t_3 - t_5)(t_4 - t_5)} \end{cases} \quad (6)$$

To calculate the wear coefficient, two test gear sets underwent special lapping where the values of the gear set positioning axes do not change, i.e. the values of the three lapping parameters (i) hypoid offset E , (ii) pinion mounting distance P and (iii) gear mounting distance G are kept constant.

Suppose wear thickness of point P_i after the gear set has been lapped for T seconds is h_i . Then the wear coefficient value calculated at this point will be:

$$k_i = \frac{60 h_i H}{\omega T v_i \int_0^{t_i} \sigma_i(t) dt} \quad (7)$$

However, the shape of the contact pattern changes as wear progresses during special lapping. Figures 3 and 4 respectively show the calculated contact patterns on the tooth surface of the gear member before and after special lapping (both have been reverse-engineered).

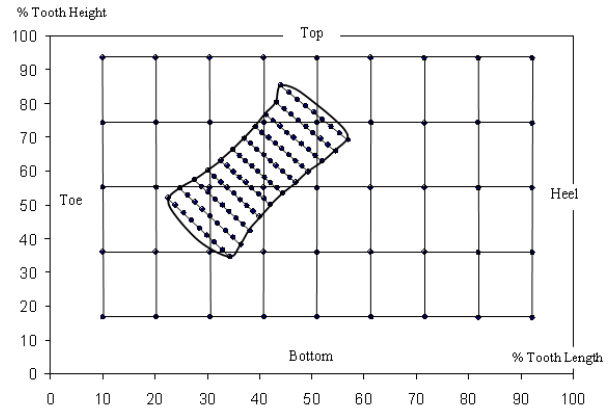


Figure 3 : Calculated contact pattern before lapping – gear IB tooth flank (using the error surface)

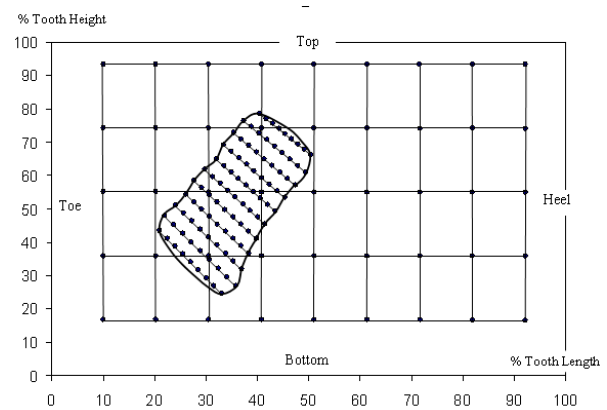


Figure 4 : Calculated contact pattern after lapping – gear IB tooth flank (using the error surface)

The origin is set at toe-bottom. The grid shows the location of the CMM measured points (CMM grid).

Because of the variation in contact pressure caused by material removal as lapping proceeds, it is reasonable to average the calculated contact