Calculation of Slow Speed Wear of Lubricated Gears

by
H. Winter and H. J. Plewe
Forschungsstelle für Zahnrader und Getriebebau – FZG
(Gear Research Laboratory) Technical University Munich
West Germany
Summary

On gear drives running with pitch line velocities below 0.5 m/s so called slow speed wear is often observed. To solve some problems, extensive laboratory test work was started 10 years ago. A total of circ. 300,000 h running time on FZG back-to-back test rigs have been run in this speed range. The test results showed a correlation between calculated EHL film thickness and a wear coefficient. Based on these experimental results and some wear data of different size industrial gear drives, a method for calculating slow speed wear was derived and is presented in this paper.

To calculate the service life of slow speed gears, the limits of wear for different modes of failure are discussed. Design guides to increase slow speed wear load capacity are presented.

Introduction

As an example, Fig. 1 shows the limits of the load capacity of case hardened gears to resist pitting, scoring (with and without EP-additives) and breakage. At pitch line velocities below 0.5 m/s, the oil film may become so thin that it is disrupted by asperities or approaches a condition in which boundary lubrication occurs. Slow speed wear results, and probably limits the service life. Gears working in this speed range can be found in open running gear drives of tube mills, in final gearings of infinite variable drives, antenna drives and, as a very new application, in gear drive mechanisms of solar energy reflectors (Heliostats). Slow speed wear can cause gear failure according to change in involute geometry or breakage of the worn off tooth.

Early observations of this kind of failure are described in 19th century papers. Attempts to relate this failure to load - lubrication conditions were made during the last twenty years.

From industrial gear case studies, Dudley derived an approximate film thickness at the pitch line below 0.2 \( \mu m \) as an inadequate oil film with probable wear. He found that slow speed gears with thin lubricants may wear as much as 1/4 in. in 10 million cycles of operation.

Experimental studies of Landen, using a roller test machine, showed relations between wear rate and calculated oil film thickness. At specific conditions, he produced rippled surfaces on the rollers and gave some explanations for this phenomenon.

By application of EHL Oil Film Theory to Industrial Gear Drives Wellauer and Holloway found that gear tooth surface distress is related to the specific film thickness \( \lambda \). Here \( \lambda \) is the ratio of calculated oil film thickness to the magnitude of the composite surface texture. As another observation they diver:\

When \( \lambda \) is less than 0.7 boundary lubrication prevails and lubricant, surface physical and chemical interactions, loads and temperatures are said to have a strong effect on distress modes and rates. To predict wear performance, the authors published a wear formula which is based on a relationship between a wear coefficient \( k \) and the specific film thickness \( \lambda \), see also Seireg.

Experimental

It was the main object of the experiments, to evaluate transferable wear coefficients from steady state region of slow speed wear. Therefore, long time tests (normally more than 500 h) were run on the FZG - gear rig test machine. (For data of the gears see Table 1).

The range of the pitch line velocities investigated was from \( v = 0.007 \) up to \( v = 1 \) m/s. To find correlations of gear failure and experience at higher velocities, tests at \( v = 2.76 \) m/s were also performed. The gears were lubricated by an oil jet with about 1 1/min into the tooth contact. Special tests (for example with grease) were run with splash lubrication. After a first running, in periods of 10,000 revolutions of the pinion and later every 150 h running time, the gears were dismounted and the weight loss was determined by a balance (accuracy \( \pm 1 \) mg). The change of the involute geometry was measured with an involute gear test machine.

To solve the complex tribologic problem the following factors of influence were investigated.

AUTHORS:

PROF. DR.-ING. HANS WINTER has studied as an associate of Prof. Dr.-Ing. H.c. Gustav Niemann. He received his Doctoral degree at the Technical University of Munich. In 1956, he began his work in the German gear industry: Zahnradfabrik, Friedrichshafen (Calculation, research, manufacturing). Demag, Duisburg (Research, development, design, selling). Since 1969 he has been the head of the Laboratory of Gear Research and Gear Design (FZG) at the Technical University of Munich.

DR.-ING. HANS-JUERGEN PLEWE has studied as an associate of Prof. Dr.-Ing. H. Winter. He received his Doctoral degree at the Technical University of Munich. In 1980, he began his work with MAN Munich as development engineer.
• Influence of the pitch line velocity on quantitative wear behavior for two material combinations (case-hardened/casehardened and through hardened/through hardened) with three mineral lubricants of different viscosity. (Table 2 and 3)
• Influence of the kind of lubricant (mineral, synthetic, grease . . . ) and material combination for a critical pitch line velocity which results in a relatively large amount of wear within a time period.
• For two material combinations and some lubricants, we investigated the influence of - load - surface finishing - profile correction and - tooth geometry and module

Influence of Pitch Line Velocity on Weight Loss

Fig. 2 shows the weight loss of the gears (pinion + gear) for 3 mineral oils of different viscosity. The upper graph shows the cumulative wear of the gears after a transmitted power of 50 kWh. For decreasing speed, the lubrication regime goes bad and the cumulative amount of wear increases. Increasing viscosity results in decreasing the amount of wear.

The cumulative wear after 150 h running time is demonstrated in the lower graph. For each viscosity, a speed with maximum wear exists. This speed we call "critical speed" because this speed will give the minimum working life. As a result of the decreasing number of rotations, further decreasing speed leads to less weight loss. This behavior is important for designing a slow speed gear for a defined working service.

Influence of Lubricant — Viscosity

Fig. 3 shows typical curves of cumulative wear as a func-

Table 1: Data of test gears

<table>
<thead>
<tr>
<th>Tooth form C(x)</th>
<th>Range of test data</th>
</tr>
</thead>
<tbody>
<tr>
<td>center distance</td>
<td>91.5 mm</td>
</tr>
<tr>
<td>gear ratio</td>
<td>16/24</td>
</tr>
<tr>
<td>module</td>
<td>4.5 mm</td>
</tr>
<tr>
<td>addendum modif.</td>
<td>x₁ = x₂ = 0, 181</td>
</tr>
<tr>
<td>active face width</td>
<td>20 mm</td>
</tr>
<tr>
<td>working pressure angle</td>
<td>0,74</td>
</tr>
<tr>
<td>wear-relevant specific</td>
<td>ground CLA&lt;0.5µm</td>
</tr>
<tr>
<td>sliding r_w</td>
<td></td>
</tr>
<tr>
<td>surface finishing</td>
<td></td>
</tr>
</tbody>
</table>

(x) Mainly used for the tests

Table 2: Data of material and heat treatment

<table>
<thead>
<tr>
<th>Material</th>
<th>Heat Treatment</th>
<th>Hardness</th>
</tr>
</thead>
<tbody>
<tr>
<td>15 CrNi 6 E</td>
<td>case carburized</td>
<td>700-750 HV 10</td>
</tr>
<tr>
<td>16 MnCr 5</td>
<td>hard</td>
<td>650-700 HV 10</td>
</tr>
<tr>
<td>20 MnCr 5</td>
<td>hard</td>
<td>300-330 HB</td>
</tr>
<tr>
<td>42 CrMo 4</td>
<td>flame hardened</td>
<td>650-750 HV 10</td>
</tr>
<tr>
<td>GS 42 CrMo 4</td>
<td>through hardened</td>
<td>350-500 HB</td>
</tr>
<tr>
<td>31 CrMo V9 (gas)</td>
<td>nitrided</td>
<td>1700 HV 0,015</td>
</tr>
<tr>
<td>Ck 45 (tufftrided)</td>
<td>hard</td>
<td>350 HV 2</td>
</tr>
<tr>
<td>42 CrMo 4</td>
<td>borided</td>
<td>280-300 HB</td>
</tr>
<tr>
<td>GGG-70</td>
<td>hard</td>
<td>360-375 HB</td>
</tr>
<tr>
<td>GGG-90</td>
<td>hard</td>
<td>360-375 HB</td>
</tr>
</tbody>
</table>

Table 3: Data of lubricants (oil sump temperature 60°)

<table>
<thead>
<tr>
<th>Fluids: Straight mineral oils</th>
<th>Kinematic viscosity ν₄₀ in mm²/s</th>
<th>Dynamic viscosity η₄₀ in mPas</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel fuel</td>
<td>2,5</td>
<td>1,3</td>
</tr>
<tr>
<td>FVA-1</td>
<td>16,7</td>
<td>7,1</td>
</tr>
<tr>
<td>FVA-2</td>
<td>30</td>
<td>11,9</td>
</tr>
<tr>
<td>FVA-3</td>
<td>98,5</td>
<td>33,0</td>
</tr>
<tr>
<td>FVA-4</td>
<td>500</td>
<td>135,5</td>
</tr>
<tr>
<td>Synthetic fluids:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>PIV - Varifluid</td>
<td>28</td>
<td>11</td>
</tr>
<tr>
<td>Santotrac 70</td>
<td>90</td>
<td>38</td>
</tr>
<tr>
<td>BP—Energol GRS 450 EP</td>
<td>235</td>
<td>107</td>
</tr>
<tr>
<td>Flow grease:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Different base oil viscosities:</td>
<td>ν₄₀=68, 120 and 340 mm²/s</td>
<td></td>
</tr>
<tr>
<td>NLGI 00, Na — soap.</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Fig. 3 - Slow speed wear. Influence of mineral oil viscosity.

The running time for constant pitch line velocity results in decreasing the weight loss of the gears. Tests with very thin oils (Diesel Fuel, FVA 1) at very slow speeds showed fretting corrosion due to changes in the wear mechanisms.

It can be seen that qualitatively there is the same influence of the two parameters pitch line velocity and oil viscosity on the weight loss. A quantitative analysis shows nearly the same amounts of wear for a constant result of speed and working viscosity. This also corresponds to a constant calculated EHL-film thickness acc. Dowson:

\[ h_{\text{min}} = 2.65 \cdot (\alpha \cdot E')^{0.54} \cdot \left( \frac{\eta \cdot v \cdot \sin \alpha_w}{E' \cdot \rho_c} \right)^{0.7} \cdot \left( \frac{2 \cdot T \cdot 10^9}{d_b \cdot b \cdot E' \cdot \rho_c} \right)^{-0.13} \]  

Although there are some further important factors in gear tooth surface performance (surface texture, coefficient of friction), we found only the film thickness as a constant characteristic quantity to calculate slow speed wear. Moreover, we tested the effectiveness of EP-additives to reduce wear. The results were divergent, it seems that no general solution is possible. For example, 4% of a phosphor sulfur EP package in FVA - 3 oil showed increasing wear rates. Additional tests were performed with grease lubrication. The wear behavior cannot be described only by the base oil viscosity. For slow speeds, the additional soap grid results in an increase of "effective viscosity" or decreasing wear rates.

Influence of Load for Different Materials

For the most usual materials in gear practice, case hardened and through hardened gears, the influence of load was tested, applying 2 mineral oils of different viscosity at their critical speeds. Fig. 4 shows the results for different test parameters. The left side graph shows the effect of load - variation for lubrication with a mineral oil of low viscosity at a speed of \( v = 0.05 \text{ m/s} \). Up to mean loads of \( k_C = 15 \text{ N/mm}^2 \), both materials show about equal amounts of wear. Therefore, at low Hertzian pressures there is no real advantage in wear reduction for case hardened gears in relation to through hardened gears of 42 CrMo 4 steel.

Application of lubricants of higher viscosity at the critical speed of \( v = 0.015 \text{ m/s} \) results in much smaller amounts of wear (right side graph). Here, at all loads, it is an advantage in wear reduction to take case hardened gears instead of through hardened gears. Under such conditions, case hardened gears are superior (in wear load capacity) compared to through hardened gears.

Effect of Surface Finish

A common way to evaluate the danger of wear failure is to compare the calculated film thickness with a characteristic roughness height. As indicated in the introduction, experts use different methods for choosing an appropriate roughness height measure.
To investigate these surface effects, gears of different surface finish were run. After a minimum running time of circ. 500 h, hardly any difference in wear behavior was observed between hobbed and ground through hardened gears. The worn in surface texture was dependent on the working conditions.

Thereupon, we decided not to consider the surface conditions of manufacturing as an important influence factor for a coefficient of wear.

Change of Tooth Profile According to Slow Speed Wear

Slow speed wear alters the involute tooth profile. Therefore, we decided to take the weight loss to measure the amount of wear in a test run. The alteration of tooth profile is dependent on the amount of wear and the material hardness — combination of the gear pair.

Fig. 5 shows the measured profile form along the path of contact for different amounts of wear. The upper line demonstrates the alterations for casehardened pinions and gears having the same hardness. In the beginning of the wear process, most of the tooth form alteration occurs near the starting point of contact of the driving pinion, and near the end point of contact of the driven gear. After the tooth profile has arrived at an AGMA accuracy grade of about 5, further slow speed wear proceeds nearly equidistantly to this tooth profile. 

The lower line of Fig. 5 shows the alteration of a tooth profile in the case of a large difference between the surface hardness of the pinion and gear. The figures demonstrate that wear mainly occurs on the through hardened gear with lower surface hardness.

Wear Calculation Procedures

A method to calculate the slow speed wear of gears was published by Wellauer/Holloway. They based their calculation method on a very common wear equation:

$$W_v = K \cdot 1 \cdot A_s$$  \hspace{1cm} (2)

Where K is the coefficient of wear, 1 the sliding distance and $A_s$ the real Hertzian contact area.

Fig. 6 shows the wear equation according to Wellauer/Holloway and Seireg. They published this gear wear formula for the region $\lambda < 0.7$. The dependence of the wear coefficient $k$ from the specific film thickness $\lambda$ is based on a theoretical model of Seireg. To compare these results with the test results reported here, the medium line in Fig. 6 was added by the authors.

Evaluation of the Wear Coefficient

Equation 1 describes a linear wear process. Examples of typical experimental slow speed wear curves are given in Fig. 2. The linear running in wear, is followed by the regime of steady state wear. The wear coefficient describes the wear behavior in the steady state regime. The test showed that the running in process lasts, approximately, up to 500 h running time, independent of the working conditions.

Fig. 7 demonstrates the way we evaluated the wear coefficient from experimental wear curves. The left graph shows wear curves at three different working conditions. They represent the influence of the film thickness.

By searching for the best fitting approximation formula, we found an exponential equation of the form:

$$W_m = a' \cdot N^{b'}$$  \hspace{1cm} (3)

where N is the total number of tooth contacts. The coefficient $a'$ and the exponent $b'$ are dependent on the material and working conditions. Linear regression of the test results and equation (3) showed correlation factors $\geq 0.98$.

As wear coefficient, we defined the gradient of the best
1. Cumulative wear curves and approximation equation

2. Coefficient $c_m = f(h_{\text{min}})$ of wear-equation

$W_m = K \cdot c_m \cdot b \cdot N$

$K = f(k_c, m, \ldots)$

---

Fig. 7 – Evaluation of wear coefficient $c_m$ from wear curve. (Example for tooth form C according table 1).

Fitting exponential equation at 500 h running time.

The right graph in Fig. 7 demonstrates this mass - wear coefficient $c_m$ over the EHL-film thickness.

$\frac{dW_m}{dN} = \frac{b}{c_m}$

where $b$ is the active facewidth.

Tooth form alterations and wear failure modes are best described by linear amounts of wear. The mean coefficient of linear wear $c_l$ can be derived from the measured coefficient of mass wear $c_m$. In Fig. 8, this linear wear coefficient of the test gears $c_l$ is plotted against the EHL minimum film thickness. Moreover, the influence of different material combinations, loads and lubrication conditions is shown.

Proposed Wear Calculation Method

Sliding Distance and Influence of Size.

Several different wear formulae have been published by now. One common characteristic of all these methods is the use of the sliding distance to calculate the amount of wear.

Fig. 9 demonstrates how to calculate the sliding distance for disc and gear contacts. At the two-disc-contact, the sliding distance for one point on a disc is given by the contact time in the contact area and the sliding speed. (For example for point 1 on disc 1):

$I_{11} = 2 \cdot b_H \cdot (v_1 - v_2) \cdot N_1$

Herein $(v_1 - v_2)/v_1$ is the (constant) slip related to disc 1. The Hertzian contact length $2b_H$ is also constant for the whole circumference.

Looking at the tooth contact, there are variable slip and...
contact conditions. As an equivalent to the slip, there is the variable specific sliding \( \xi \) along the line of action. Because of the alteration of the contact-radius, there is also a small variation of the Hertzian contact length.

As described in Fig. 5, only during the running in process is the alteration of tooth profile nearly corresponding to the slope of the specific sliding. During a steady state process, however, no additional alteration of the tooth profile is observed. Therefore, from tests with gears of different module, profile correction and gear ratio, we derived a wear relevant specific sliding, which considers the sliding conditions in the dedendum and addendum path of contact according to equation 6:

\[
\xi = \xi_1 \cdot e_1 + \xi_2 \cdot e_2
\]

The relations for internal and external gears are shown in Fig. 10. As a representative Hertzian contact length, we determined the values at the pitch point \( (2b_H) \). Relating to the test gears (Index T), the following relationship for gears of different size at the same pressure can be given:

\[
b_H/b_{HT} \sim \frac{\varphi_c}{\varphi_{CT}}
\]

This relation represents a tribological size effect. To check this influence by experiment with large test gears would be too expensive. Considering information about slow speed wear behavior of large industrial gears, at somewhat different working conditions, showed results which are in good agreement with equation 7.

**Quantitative Load Influence**

Some experimental results of load influence were shown in Fig. 4. Considering more tests with different material combinations, we found the following relationship:

\[
\frac{c_1}{c_{IT}} = \left( \frac{k_C}{k_{CT}} \right)^{0.7} = \left( \frac{\varphi_H}{\varphi_{HT}} \right)^{1.4} \tag{8}
\]

When exceeding the endurance limit for Hertzian stress, the exponent in equation 8 seems to increase.

**Rule for Calculation of Slow Speed Wear for Mineral Oil or Flow Grease Without EP Additives.**

1. To evaluate the danger of slow speed wear distress, one first calculates the EHL-film thickness according to equation 9.

\[
h_{\text{min}} = 2.65 \cdot (\alpha \cdot E)^{0.54} \cdot \left( \frac{\eta_0 \cdot v \cdot \sin \alpha}{E \cdot \varphi_c} \right)^{0.7} \left( \frac{2 \cdot T \cdot 10^6}{(E \cdot b \cdot E' \cdot \varphi_c)} \right)^{-0.13}
\]

**HOGlund**

**FORMED CBN GEAR GRINDING WHEELS**

**GEAR FORMS FROM THE PEOPLE WHO KNOW WHAT AN INVOLUTE IS!**

**FOR YOUR:**

**DETOIT & RED RING GRINDERS**

**HELICALS & SPURS**

**SPECIALISTS IN GEAR GRINDING TECHNOLOGIES**

**HOGlund TRI-ORDINATE CORP. 141 NYDER AVE. BERKELEY HTS., NJ 07012 (201) 464-0200 TWX 710-964-7965 CIRCLE A-2 ON READER REPLY CARD**

14 Gear Technology
Normally only the oil jet or sump temperature is to be considered for viscosity because only small friction heating occurs at slow speeds.

Danger of distress must be expected in $h_{\text{min}} \leq 0.4 \, \mu m$ for gearings with large hardness differences (for example casehardened mating with through hardened gears). In the case of the pinion and the gear having the same surface hardness, the critical figure is in the order of $0.05 \, \mu m$.

2. Determination of wear coefficient $c_{IT}$

If no experience from actual gears is available, Fig. 5 can be used to evaluate a wear coefficient $c_{IT}$, which depends on the calculated EHL-film thickness. This refers to mineral oils without EP-Additives.

For flow grease lubrication, the viscosity of the base oil at working temperature can be taken for a first approach.

3. Calculation of the linear amount of wear $W_e$:

$$W_e = c_{IT} \cdot \left( \frac{\varphi_H}{\varphi_{HT}} \right)^{1.4} \cdot \left( \frac{\varrho_C}{\varrho_{CT}} \right) \cdot \frac{\xi_{W} W_T}{\xi_{WT}} \cdot N \quad (10)$$

Herein $\varphi_{HT}$, $\varrho_{CT}$ and $\xi_{WT}$ are the test gear data on which $c_{IT}$ is based.

4. Calculated life according wear failure:

$$L_{W} = \frac{W_{\text{lin}}}{c_{IT} \left( \frac{\varphi_H}{\varphi_{HT}} \right)^{1.4} \cdot \left( \frac{\varrho_C}{\varrho_{CT}} \right) \cdot \frac{\xi_{W} W_T}{\xi_{WT}} \cdot n \cdot 60} \quad (11)$$

Guides to estimate the allowable amount of linear wear $W_{1_{zul}}$ can be taken from Fig. 11.

Modes of failure and allowable amounts of linear wear:

Fig. 11 shows the failure tooth profile forms and the allowable depth of wear for different tooth geometry and heat treatment.

1. Worn out tooth flanks:

For precision gears, the profile form error, resulting from wear, causes reduced accuracy in torsional transmission. Consequently, increased dynamic loads must be expected at high speeds in the case of gears running at different speeds.

2. Wear of surface hardness layer:

If the surface hardness layer is worn-out, danger of pitting and plastic deformation increases.

November/December 1985 15
Wear failure modes and allowable linear amount of wear. (SF, SFW = safety factors; new, worn out).

1. Worn out tooth flanks
   \[ W_{1,\text{zul}} = \frac{f_{1,\text{zul}}}{3} \]
   \[ f_{1,\text{zul}} = \text{profile form error} \]

2. Wear of surface layer
   \[ W_{1,\text{RhZul}} = 1 \times \text{hardness penetration depth} \]
   \[ R_{\text{th}} = f \text{ (module, heat treatment)} \]

3. Minimum tooth thickness at the tip cylinder:
   A figure of \( s_a = 0.1 \ m_n \) perhaps can be taken as a limit.

4. Reduction of the safety factor for tooth root stress:
   The factor 0.85 (instead of 1.0) was found from the test results of the authors.

5. Wear particles in the lubricant:
   Abrasive particles in the lubricant result in increasing rates of wear. So, only a certain percentage is acceptable.

Conclusions
Extensive long time tests on FZG back to back test rigs in the speed range below 1 m/s showed the influence of the main parameters on slow speed wear.

For straight mineral oils the influence of pitch line velocity and viscosity at working conditions can be described by the EHL-film thickness. We found the film thickness as a parameter to evaluate the danger of wear distress. It can also be used to determine a wear coefficient to calculate the amount of wear. A danger of wear distress is given at film thicknesses below 0.05 \( \mu \)m. The material combination case-hardened/through hardened gears showed critical wear conditions for a film thickness up to 0.4 \( \mu \)m.

Tests with different additives in the mineral oil showed divergent results. Their effect cannot generally be predicted. A calculation based on the EHL-film thickness is not possible. The test with grease lubricated gears showed that the additional soap grid results in an increase of “effective viscosity” or decreasing wear rates. (?)

Increasing loads below the endurance limit for Hertzian stress results in degreasing increase of wear. Loads above this limit can also result in pitting.

Calculation Example
Planetary gears are often used in high loaded, slow speed drives. The typical material combination (casehardened planetary gears/through hardened or normalized internal gear) of planetary gears gives a high probability of slow speed wear failure. As an example, the danger of wear failure and the expected life due to wear in a planetary gear drive is calculated by the described method.

**Description of the Gear:**
The power of the driving sun pinion (casehardened) is branched into three planetary gears (casehardened) which are in contact with the fixed internal gear (through hardened).

**Gear Data:**

<table>
<thead>
<tr>
<th>Number of teeth</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sun pinion ( z_5 = 17 ) module=6mm (circ. 4 DP)</td>
</tr>
<tr>
<td>Planet gear ( z_5 = 49 ) pressure angle ( \alpha = \alpha_n = 20^\circ )</td>
</tr>
<tr>
<td>Internal gear ( z_5 = 115 ) active facewidth=60 mm</td>
</tr>
</tbody>
</table>

Transmitted power 72 kW, driving speed \( n_5 = 100 \text{ min}^{-1} \)

Lubricant: Mineral oil without EP additives, viscosity at working temperature = 33 m Pa s, pressure-viscosity coefficient \( \alpha = 17.6 \times 10^{-9} \text{ m}^2/\text{N} \).

---

**Fig. 11 – Wear failure modes and allowable linear amount of wear.** (SF, SFW = safety factors; new, worn out).
Nomenclature

- **a**: mm, center distance
- **b**: mm, active facewidth
- **b_H**: mm, half of Hertzian contact length
- **c_1**: mm/U, coefficient of linear-wear
- **c_m**: mg/mmU, coefficient of mass-wear
- **d_a**: mm, tip diameter
- **d_b**: mm, base diameter
- **d_w**: mm, pitch diameter
- **e**: mm, length of addendum path of contact
- **h_min**: mm, minimum film thickness (acc EHL)
- **k_C**: N/mm², stress acc. Striebeck at pitch point
- **l**: mm, sliding distance
- **m**: mm, module
- **m_S**: kg, total mass of lubricant
- **n**: min⁻¹, pinion speed
- **p_C**: N/mm², stress acc. Hertz at pitch point
- **S_F**: mm, tooth thickness in the critical section
- **v**: m/s, pitch line velocity
- **x**: - , addendum modification coefficient
- **z**: - , number of teeth
- **A**: mm², active tooth contact area
- **E**: N/m², effective elastic modulus
- **L_W**: h, working life acc. wear
- **N**: - , number of mesh cycles
- **S_F**: - , safety factor for tooth root stress
- **S_FW**: - , safety factor for tooth root stress (of worn out gear)
- **T**: Nm, torque
- **W_1**: mm, amount of linear-wear
- **W_m**: mg, amount of mass-wear
- **α**: m²/N, lubricant pressure – viscosity coefficient
- **α_α**: °, pressure angle
- **α_w**: °, pressure angle at tip diameter
- **δ**: °, working pressure angle
- **δ_W**: - , specific sliding
- **ε_a**: - , wear-relevant specific sliding
- **η_0**: mPas, lubricant viscosity at amibient conditions
- **q**: mg/mm³, density
- **q_C**: mm, relative radius of curvature at pitch point
- **q_H**: N/mm², surface stress at the pitch point acc. ISO

The index T characterises the actual working parameters at test conditions.

---

**CNC & CAM CONTROLLED DRESSING SYSTEMS**

**FOR YOUR GLEASON GEAR GRINDERS**

**ADVATAGES:**

1. Perfect blends of roots & flanks
2. Better control of critical geometry
3. Reduced setup times
4. Same dresser for all pressure angles

---

**SPECIALISTS IN GEAR GRINDING TECHNOLOGIES**

Gear grinding dressers - CBN formed gear grinding wheels
Spur & Helical gear grinder remanufacturing, & CNC upgrading

HOGlund TRI-ORDINATE CORP. 343 SnyDER AVE, BERKELEY HTS, N.J. 07922 (201) 464-0200 : Twx 710-984-7965

CIRCLE A-4 ON READER REPLY CARD

November/December 1985 17
From this data we derive the following results:

<table>
<thead>
<tr>
<th></th>
<th>Sun gear/plant</th>
<th>Planet/internal gear</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Rolling speed</strong></td>
<td>0.465 m/s</td>
<td>0.465 m/s</td>
</tr>
<tr>
<td><strong>Surface stress acc. Hertz</strong></td>
<td>1504 N/mm²</td>
<td>581 N/mm²</td>
</tr>
<tr>
<td><strong>Relative radius of curvature at pitch point</strong></td>
<td>12.95 mm</td>
<td>87.5 mm</td>
</tr>
<tr>
<td><strong>EHL-minimum film thickness</strong></td>
<td>0.053 µm</td>
<td>0.12 µm</td>
</tr>
<tr>
<td><strong>Coefficient of linear wear of test gears</strong></td>
<td>0.1 ( \times 10^{-9} ) mm/ contact</td>
<td>35 ( \times 10^{-9} ) mm/ contact</td>
</tr>
<tr>
<td><strong>Wear relevant specific sliding</strong></td>
<td>0.762</td>
<td>0.183</td>
</tr>
<tr>
<td><strong>Real number of mesh cycles/ min (3 planets)</strong></td>
<td>( n_s = 261.36 \text{ min}^{-1} )</td>
<td>( n_t = 38.64 \text{ min}^{-1} )</td>
</tr>
<tr>
<td><strong>Linear wear after 1000 h of service life</strong></td>
<td>0.004 mm</td>
<td>0.185 mm</td>
</tr>
<tr>
<td><strong>Allowable amount of wear</strong></td>
<td>( \sim 1.2 \text{ mm} ) (case depth)</td>
<td>( \sim 4 \text{ mm}^{(1)} ) (minimum thickness at tip cylinder)</td>
</tr>
<tr>
<td><strong>Service life due to wear</strong></td>
<td>335 000 h</td>
<td>22 000 h</td>
</tr>
</tbody>
</table>

(1) Another allowable amount of wear should be considered: The percentage of wear particles in the lubricant. For the above example the mass wear of the internal gear results in an amount of \( W_m = 120 \text{ g} \) after 1000 h.

References

This paper was previously presented at the Fall Technical Conference of AGMA, P. 219.16, 1982.

E-1 ON READER REPLY CARD

TECHNICAL CALENDAR

SME’s annual “Gear Processing and Manufacturing” clinic and tabletop exhibits will be held at The Dearborn Inn in Dearborn, Michigan November 19-21, 1985. Current technologies in the gear industry will be covered. J. Richard Newman, formerly of National Broach and Machine Division of Lear Siegler, and Carl S. Eckberg of Bourn & Koch Machine Tool Co., are co-chairing this three-day clinic. An evening of vendor tabletop exhibits will accompany the daily technical presentations. For further information on the clinic or the exhibits, contact Dianne Levertont at SME, 313/271-1500, extension 394.

**March 3-5**
2nd World Conference on Gearing
Institute de l’Engrenage et des Transmissions
Paris, France

The Paris Congress has brought together various groups of technicians, practitioners, buyers, production, lubrication and control experts (150 speakers from all over the world) to present the results of their research on gearing techniques. To obtain further information contact: Maurice Allard, Director, Institute de l’Engrenage et des Transmissions, 162 Boulevard Malesherbes, 75017, Paris, France. Telephone: 43.80.04.09

**March 17-19**
International Conference on Austempered Ductile Iron, Ann Arbor, Michigan

For further information contact: Dale Breen, Gear Research Institute, P.O. Box 353, Naperville, IL 60566. Telephone: (312) 355-4200.