Calculation of Slow Speed Wear of Lubricated Gears

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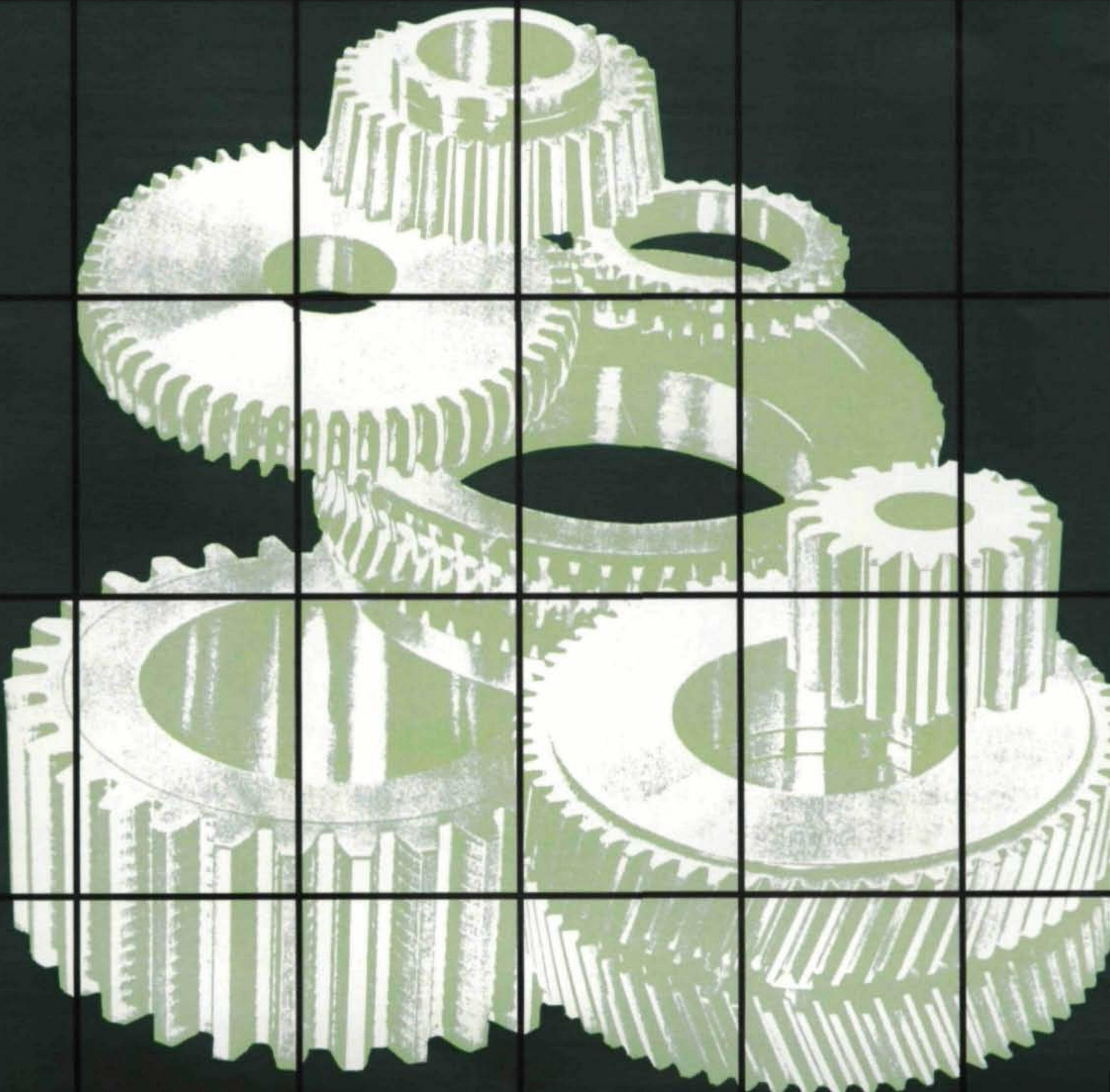

Summary

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.

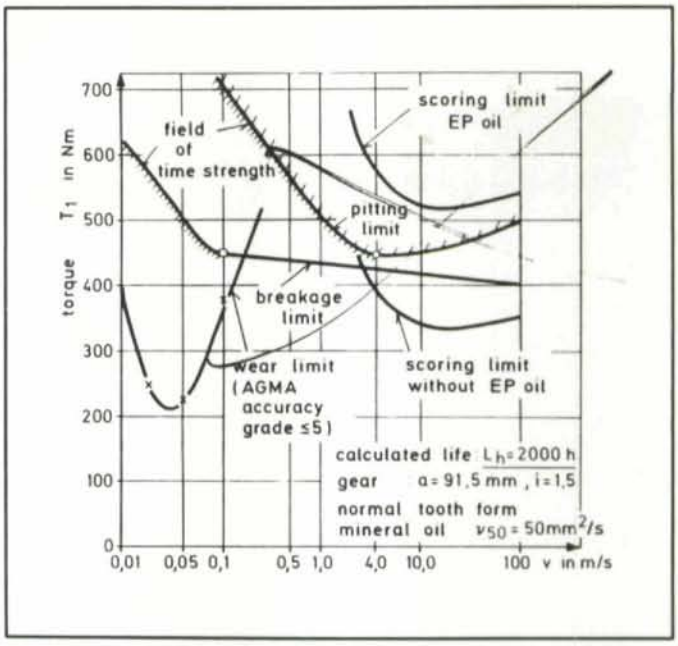

Fig. 1 – Failure limits of a casehardened gear pair. (example)

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.[1] Attempts to relate this failure to load – lubrication conditions were made during the last twenty years.

From industrial gear case studies, Dudley[2] derived an approximate film thickness at the pitch line below 0.2 μm 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,[3] 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[4] found that gear tooth surface distress is related to the specific film thickness λ. Here λ is the ratio of calculated oil film thickness to the magnitude of the composite surface texture. As another observation they divert:

When λ 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[4] published a wear formula which is based on a relationship between a wear coefficient k and the specific film thickness λ, see also Seireg.[5]

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 ± 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.

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- Influence of the pitch line velocity on quantitative wear behavior for two material combinations (casehardened/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 demonstated 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-

Fig. 2 – Wear for constant transmitted power (above) and constant running time (below) dependent on the pitch line velocity for oils of different viscosity.

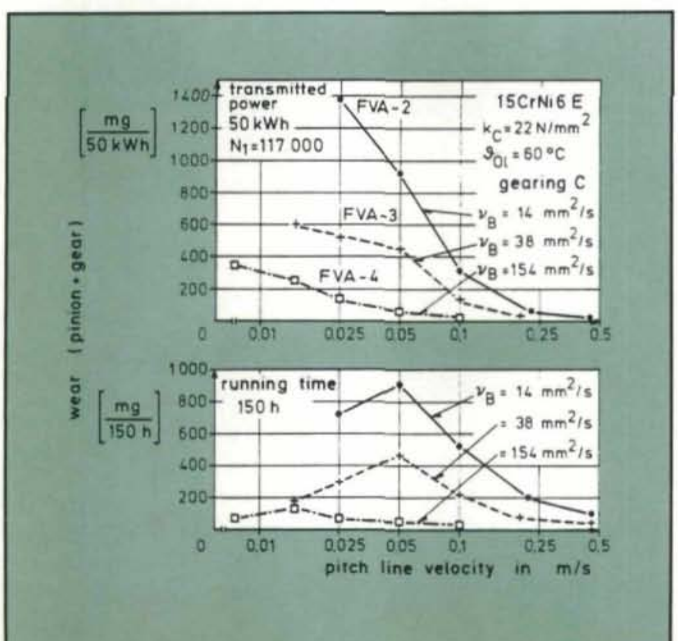


Table 1: Data of test gears

	tooth form C(x)	range of test data
center distance	91,5 mm	constant
gear ratio z_1/z_2	16/24	32/48 ··· 12/18
module	4,5 mm	2,25 ··· 6 mm
addendum modification	$x_1=x_2=0,181$	$x_1=+0,86$; $x_2=-0,5$
active facewidth	20 mm	14 ··· 30 mm
working pressure angle	20°	constant
wear-relevant specific sliding ζ_w	0,74	0,46 ··· 0,79
surface finishing	ground CLA<0.5μm	hobbed CLA<2μm shaved CLA<0.5μm

(x) Mainly used for the tests

Table 2: Data of material and heat treatment

15 CrNi 6 E 16 MnCr 5 20 MnCr 5	case carburized hardness: 700-750 HV 10
42 CrMo 4	flame hardened hardness: 650-700 HV 10
42 CrMo 4 GS 42 CrMo 4	through hardened hardness: 300-330 HB
31 CrMo V9 (gas) Ck 45 (tufftrided)	nitrided hardness: 650-750 HV 10 hardness: 350 HV 2
42 CrMo 4 GGG-70 GGG-90	borided hardness circ. 1700 HV 0,015 sphäroidal graphite iron pearlitic 280-290 HB austenitic-bainitic, 360-375 HB

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

Fluids: Straight mineral oils	kinematic viscosity ν_{40} in mm²/s	dyn. viscosity η_{60} in mPas
Diesel fuel	2,5	1,3
FVA-1	16,7	7,1
FVA-2	30	11,9
FVA-3	98,5	33,0
FVA-4	500	135,5
Synthetic fluids:		
PIV – Varifluid	28	11
Santotrac 70	90	38
BP – Energol GRS 450 EP	235	107
Flow grease: Different base oil viscosities: $\nu_{40}=68$, 120 and 340 mm²/s NLGI 00, Na – soap.		

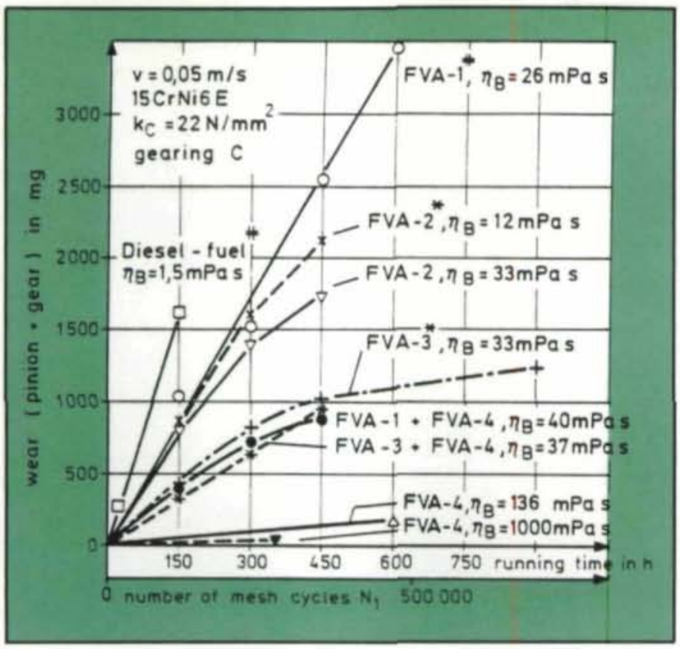


Fig. 3—Slow speed wear. Influence of mineral oil viscocity.

tion of the running time for constant pitch line velocity. Lubrication with mineral oils of increasing viscosity (for decreasing temperature as well as for increasing base oil viscosity) 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 changings of 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:[6]

$$h_{min} = 2.65 \cdot (\alpha \cdot E')^{0.54} \cdot \left(\frac{\eta_O \cdot v \cdot \sin \alpha_w}{E' \cdot \varrho_c}\right)^{0.7} \cdot \left(\frac{2 \cdot T \cdot 10^9}{d_b \cdot b \cdot E' \cdot \varrho_c}\right)^{-0.13} \tag{1}$$

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 diction 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.[7]

Influence of Load for Different Materials

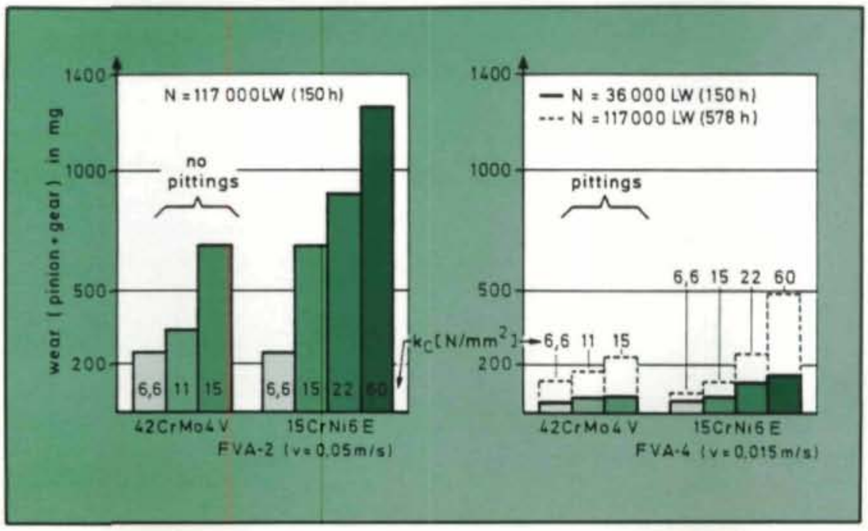
For the most usual materials in gear practice, casehardened and through hardened gears, the influence of load was tested, applying 2 mineral oils of different viscosity at their critical speeds. Fig. 4 shows the results for different test parameters. The left side graph shows the effect of load – variation for lubrication with a mineral oil of low viscosity at a speed of $v = 0.05$ m/s. Up to mean loads of $k_C = 15$ N/mm², both materials show about equal amounts of wear. Therefore, at low Hertzian pressures there is no real advantage in wear reduction for casehardened gears in relation to through hardened gears of 42 CrMo 4 steel.

Application of lubricants of higher viscosity at the critical speed of $v = 0.015$ 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 casehardened gears instead of through hardened gears. Under such conditions, casehardened 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.

Fig. 4—Influence of load on slow speed wear for two gear materials and lubricants.



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.

Therefore, 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.[8]

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.[4] They based their

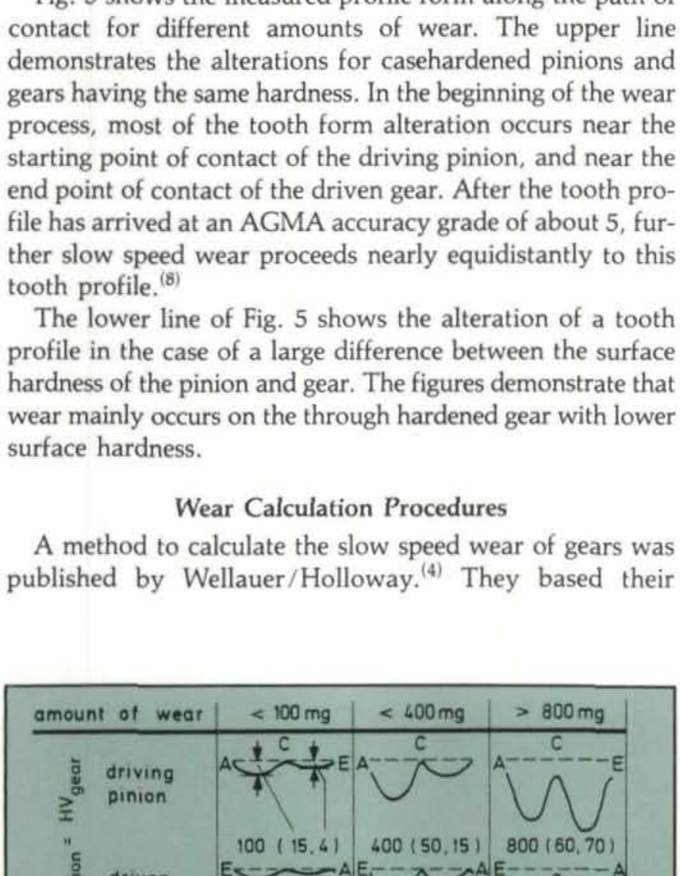


Fig. 5 – Alteration of tooth profile along the path of contact for different amounts of wear and two material combinations (casehardened/casehardened and casehardened/through hardened gears).

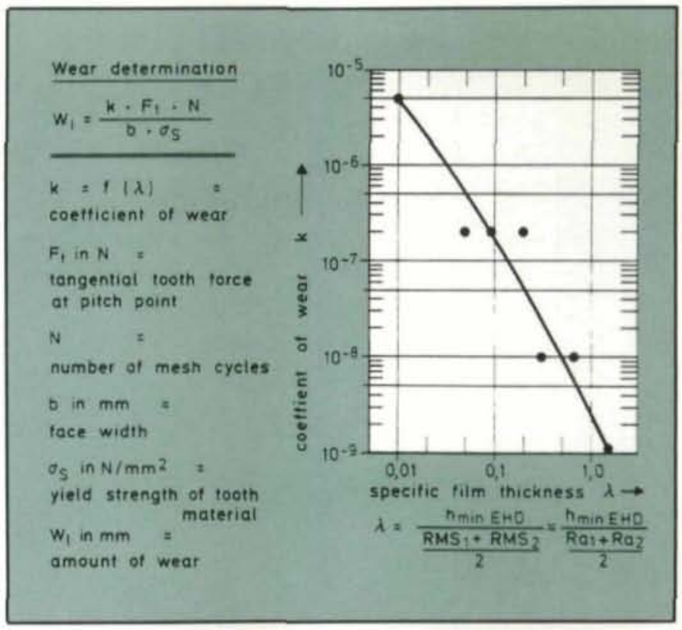


Fig. 6 – Calculation of wear according to Wellauer/Holloway /4/ and Seireg /5/ (The medium line was added by the authors).

calculation method on a very common wear equation:

$$W_v = K \cdot 1 \cdot A_a \tag{2}$$

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

Fig. 6 shows the wear equation according to Wellauer/Holloway and Seireg.[5] They published this gear wear formula for the region $\lambda < 0.7$. The dependence of the wear coefficient k from the specific film thickness λ 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:

$$W_m = a' \cdot N^{b'} \tag{3}$$

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 ≥ 0.98.

As wear coefficient, we defined the gradient of the best

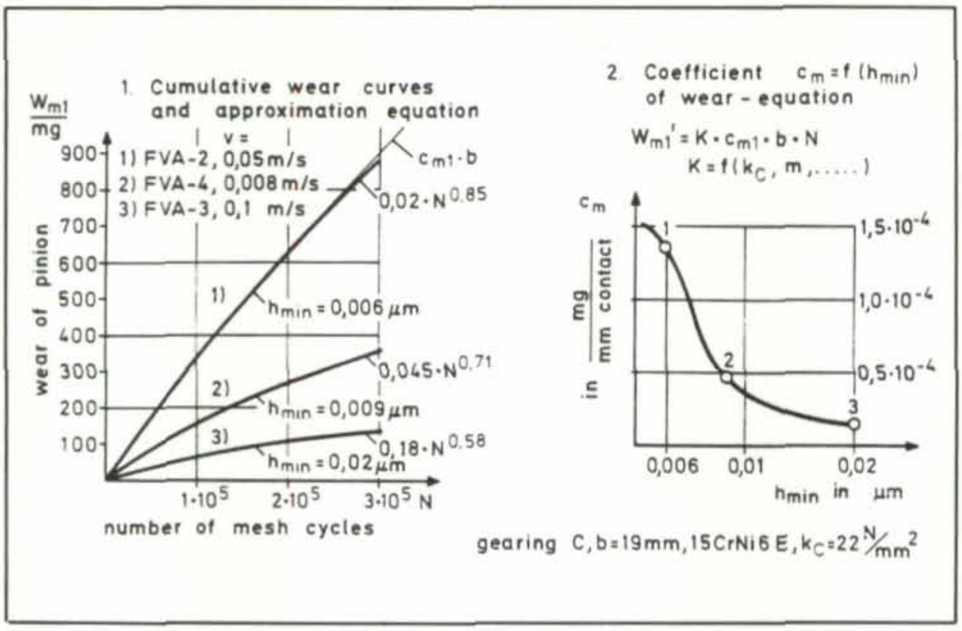


Fig. 7 – Evaluation of wear coefficient c_m from wear curve. (Example for tooth form C according table 1).

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.

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

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_{lt} 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):

$$I_{P1} = \frac{2 \cdot b_H}{v_1} \cdot (v_1 - v_2) \cdot N_1$$

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

Looking at the tooth contact, there are variable slip and

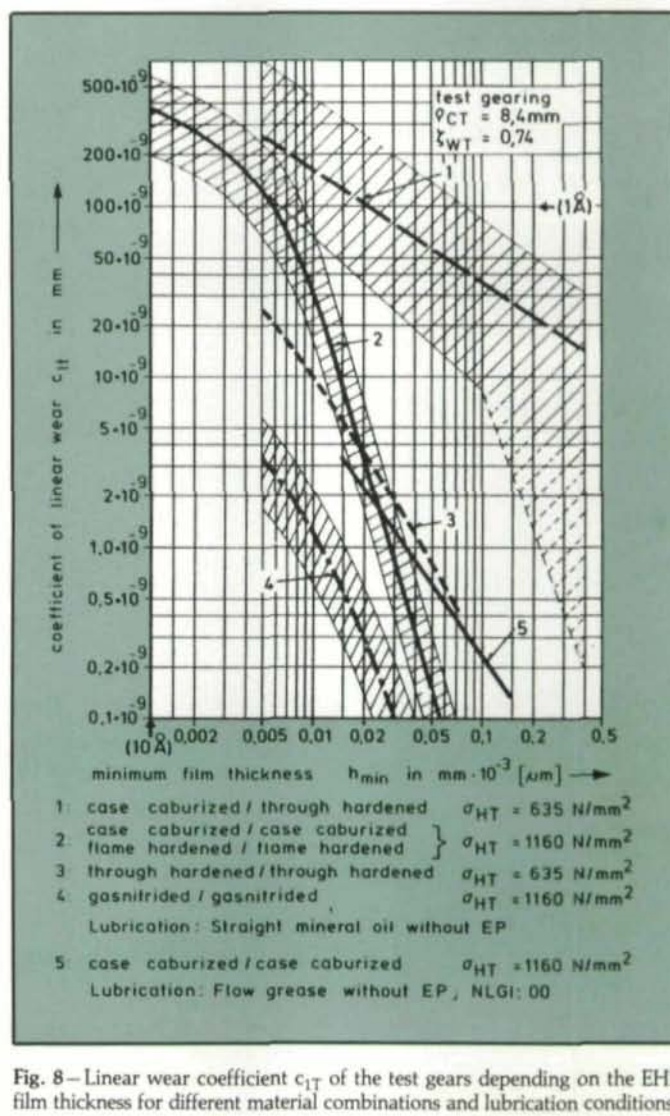


Fig. 8 – Linear wear coefficient c_{lT} of the test gears depending on the EHL film thickness for different material combinations and lubrication conditions.

contact conditions. As an equivalent to the slip, there is the variable specific sliding ζ 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:

$$\zeta w = \frac{\zeta E1 \cdot e_1 + \zeta A2 \cdot e_2}{e_1 + e_2} \qquad (6)$$

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 ($2b_{HC}$). Relating to the test gears (Index T), the following relationship for gears of different size at the same pressure can be given:

$$b_H/b_{HT} \sim \varrho_c/\varrho_{CT} \qquad (7)$$

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:

$$\frac{c_I}{c_{IT}} = \left(\frac{k_C}{k_{CT}}\right)^{0.7} = \left(\frac{\varphi_H}{\varphi_{HT}}\right)^{1.4} \qquad (8)$$

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.

$$h_{min} = 2.65 \cdot (\alpha \cdot E')^{0.54} \cdot \left(\frac{\eta_O \cdot v \cdot \sin \alpha_w}{E' \cdot \varrho_c}\right)^{0.7} \cdot \left(\frac{2 \cdot T \cdot 10^9}{d_b \cdot b \cdot E' \cdot \varrho_c}\right)^{-0.13} \qquad (9)$$

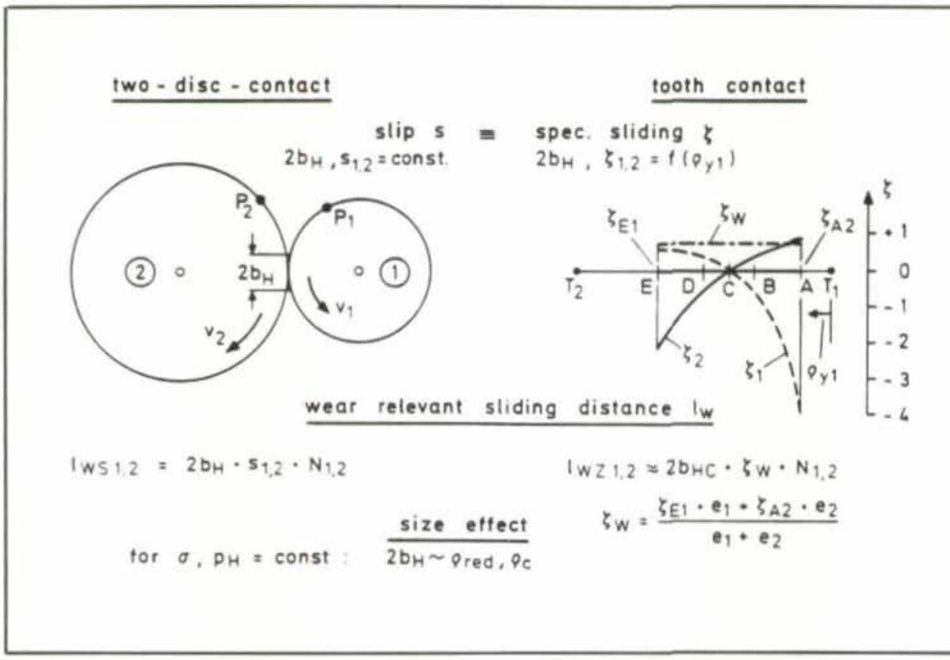


Fig. 9 – Calculation of sliding distance for disc-contact and tooth contact.

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 $h_{min} \leqslant 0.4\ \mu m$ 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_{1T}, 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:

$$W_e = c_{IT} \cdot \left(\frac{\varphi_H}{\varphi_{HT}}\right)^{1.4} \cdot \left(\frac{\varrho_C}{\varrho_{CT}}\right) \cdot \left(\frac{\zeta_W}{\zeta_{WT}}\right) \cdot N \qquad (10)$$

Herein φ_{HT}, ϱ_{CT} and ζ_{WT} are the test gear data on which c_{1T} is based.

4. Calculated life according wear failure:

$$L_{hW} = \frac{W_{lzul}}{c_{IT}\left(\frac{\varphi_H}{\varphi_{HT}}\right)^{1.4} \cdot \left(\frac{\varrho_C}{\varrho_{CT}}\right) \cdot \left(\frac{\zeta_W}{\zeta_{WT}}\right) \cdot n \cdot 60} \qquad (11)$$

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

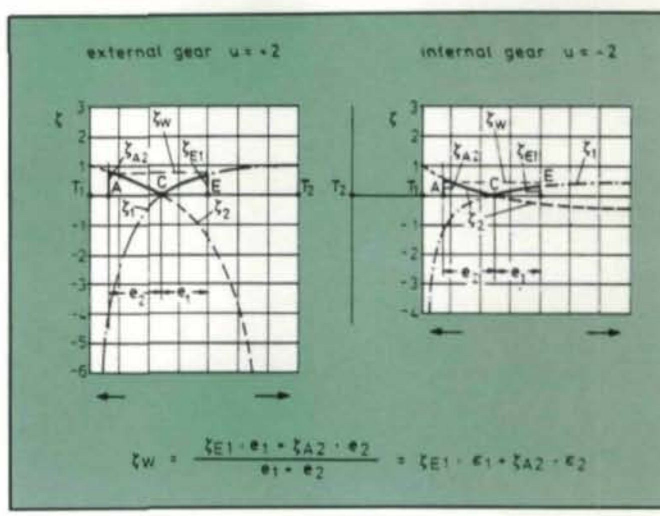


Fig. 10 – Wear relevant specific sliding ζ_W.

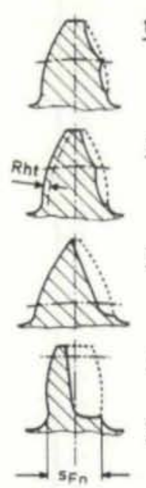
Modes of failure and allowable amounts of linear wear: Fig. 11 shows the failure tooth profile forms and the allowable depth of wear for different tooth geometry and heat treatment.

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.



1. Worn out tooth flanks

$$W_{l\,f\,zul} = \frac{f_{f\,zul}}{3}$$

$f_{f\,zul}$ = profile form error

2. Wear of surface layer

$W_{l\,Rht\,zul}$ = 1 × hardness penetration depth

Rht = f (module , heat treatment)

3. Minimum tooth thickness

$$W_{l\,Sp\,zul} = \left[d_a \cdot \left(\frac{\pi + 4 \cdot x \cdot \tan\alpha_n}{2z} + \operatorname{inv}\alpha_t - \operatorname{inv}\alpha_{at}\right) - 0{,}1 \cdot m_n\right] \cdot \cos\alpha_{at}$$

4. Safety factor for tooth root stress

$$W_{l\,F\,zul} = s_{Fn}\left[1 - 0{,}85\sqrt{S_{FW}/S_F}\right]$$

5. Wear particles in the lubricant

$$\frac{W_m \text{ unfiltrated}}{\text{mass of lubricant } m_s} \approx 0{,}1 - 0{,}5\,‰ \approx 100 - 500\,\frac{mg}{kg}$$

Fig. 11 – Wear failure modes and allowable linear amount of wear. (S_F, S_{FW} = safety factors; new, worn out).

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 μm. The material combination casehardened/through hardened gears showed critical wear conditions for a film thickness up to 0.4 μm.

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.[7]

Increasing loads below the endurance limit for Hertzian stress results in degressive 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:

Number of teeth

Sun pinion	$z_S = 17$	module = 6mm (circ. 4 DP)
Planet gear	$z_P = 49$	pressure angle $\alpha = \alpha_w = 20°$
Internal gear	$z_I = 115$	active facewidth = 60 mm
Transmitted power	72 kW,	driving speed $n_S = 100$ min.$^{-1}$

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

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/mm^2	stress acc. Stribeck 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/mm^2	stress acc. Hertz at pitch point
s_{Fn}	mm	tooth thickness in the critical section at the tooth root
v	m/s	pitch line velocity
x	—	addendum modification coefficient
z	—	number of teeth
A	mm^2	active tooth contact area
E	N/m^2	effective elastic modulus
L_{hW}	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	m^2/N	lubricant pressure – viscosity coefficient
α	°	pressure angle
α_a	°	pressure angle at tip diameter
α_w	°	working pressure angle
ζ	—	specific sliding
ζ_W	—	wear-relevant specific sliding
ϵ_a	—	transverse contact ratio
η_0	mPas	lubricant viscosity at amibient conditions
ϱ	mg/mm^3	density
ϱ_C	mm	relative radius of curvature at pitch point
ϱ_H	N/mm^2	surface stress at the pitch point acc. ISO

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

From this data we derive the following results:

	sun gear/plant	planet/internal gear
Rolling speed	0.465 m/s	0.465 m/s
Surface stress acc. Hertz φ_H	1504 N/mm²	581 N/mm²
Relative radius of curvature at pitch point ϱ_C	12.95 mm	87.5 mm
EHL-minimum film thickness h_{min}	0.053 μm	0.12 μm
Coefficient of linear wear of test gears (from Fig. 8) c_{IT}	$0.1 \cdot 10^{-9}$mm/ contact	$35 \cdot 10^{-9}$mm/ contact
Wear relevent specific sliding ζ_W	0.762	0.183
Real number of mesh cycles /min (3 planets)	$n_S = 261.36\ \text{min}^{-1}$	$N_I = 38.64\ \text{min}^{-1}$
Linear wear after 1000 h of service life W_I	0.004 mm	0.185 mm
Allowable amount of wear	~1.2 mm (case depth)	~4 mm(1) (minimum thick-nness at tip cylinder)
Service life due to wear L_{hW}	335 000 h	22 000 h

(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 $W_m \approx 120$ g after 1000 h.

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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 Leverton 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

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