

Investigations on the Scuffing Resistance of High-Speed Gears

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ABSTRACT:

Test results with a high speed four square gear test rig ($n_{\max} = 26\,000$ rpm) show the influence of speed, viscosity, base oil, type and amount of additive on the scuffing load. In some cases at high speed the scuffing load can be more than twice the value calculated according to ISO/DIS 6336/4. The reason for the speed-dependency of the scuffing load is explained by the kinetics of the chemical reaction between the metal and the additive. It is described, how a calculation method could take account of the high scuffing load at high speed.

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Investigations on the Scuffing Resistance of High-Speed Gears

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Test results with a high speed four square gear test rig ($n_{\max}=26\ 000$ rpm) show the influence of speed, viscosity, base oil, type and amount of additive on the scuffing load. In some cases at high speed the scuffing load can be more than twice the value calculated according to ISO/DIS 6336/4. The reason for the speed-dependency of the scuffing load is explained by the kinetics of the chemical reaction between the metal and the additive. It is described, how a calculation method could take account of the high scuffing load at high speed.

1 Introduction

Besides tooth breakage and pitting, scuffing is one of the main factors, limiting the power transmitting capacity of gears. Of course, scuffing damage can nearly always be avoided by an adequate amount of Extreme Pressure (EP) additives. But especially in cases, when the lubrication of gears is not the only task of the lubricant, it is often not possible to add enough EP-additive to give sufficient scuffing protection:

In applications where the lubricant is also used as a hydraulic fluid (e.g. for the control of steam- or gas turbines or in hydraulic torque converters in locomotive drives) the air release property of the oil may be reduced because of EP-additives.

Marine gears are often lubricated with diesel engine oils.

The oxidation properties of the oil as well as seal materials are affected by EP-additives.

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It is known, that the scuffing load of medium speed gears is lower than that of low speed gears. For high speed applications, however, increasing scuffing load was first reported in the 1950's by Borsoff [1]. Ku and Baber [2], Lechner [3], Drozdov and Gavrikov [5] and Seitzinger [4] also found a minimum in scuffing load at a certain speed, while other investigators, e.g. Schauerhammer [6], did not. None of these authors could give a complete explanation for this effect, and no commonly used calculation method considers the increasing scuffing load at high speed.

It was the objective of the research work, the results of which are presented here, to investigate the main factors which influence the increasing scuffing load with increasing speed, to contribute to possible explanations and calculation methods.

2 Test Rig, Test Gears

A four-square gear test rig with a mechanically closed power circuit having 91.5 mm center distance was used (Fig. 1). The pinion speed is continuously variable between 500 and 26 000 rpm, the maximum pitch line velocity is 100 m/s ($\approx 20\ 000$ ft/min). The torque is applied while the gears are running using a hydrostatic load device, measured by strain gages and controlled by an electrical con-

trol circuit. The maximum pinion torque is 350 Nm, which corresponds to a Hertzian stress of $p_C = 1500 \text{ N/mm}^2$ ($\approx 217\,500 \text{ psi}$) at the pitch point of the test gears (type A and C).

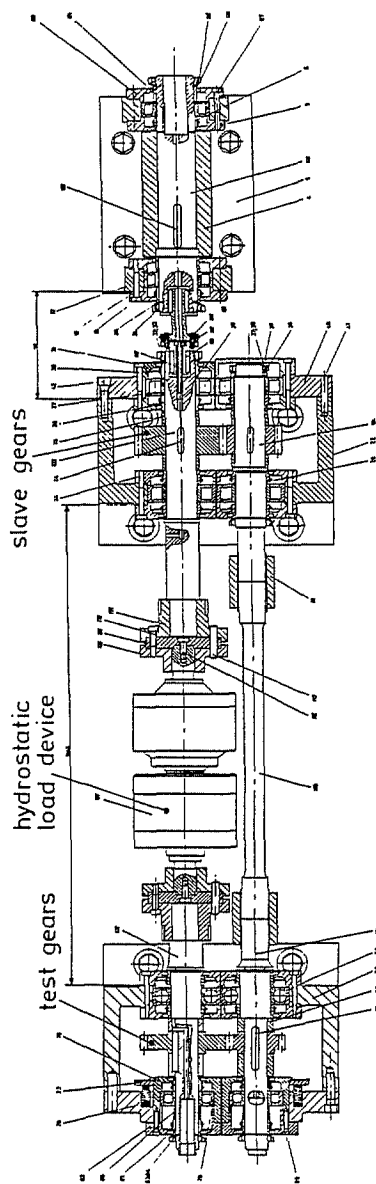


Fig. 1 Four square high speed gear test rig with hydrostatic load device. Center distance 91.5 mm. Maximum transmittable power 1000 kW

The temperature at the center of one pinion tooth, (1·module below the addendum circle) is measured by an K-type-thermocouple (chromel-alumel). One rotating amplifier each modulates the temperature and the torque signal and transduces them by an infrared LED / phototransistor (PCM-system) to the stationary display and registration unit.

The distribution of load, speed and contact time is shown in Fig. 2 and Fig. 3, the main data of the standard test gears type A and C are shown in Fig. 4. The material of the test gears is 20 Mn Cr 5, case carburized (case depth 0.6 - 0.8 mm), surface hardness 60 - 62 HRC. Retained austenite at the surface is less than 20%.

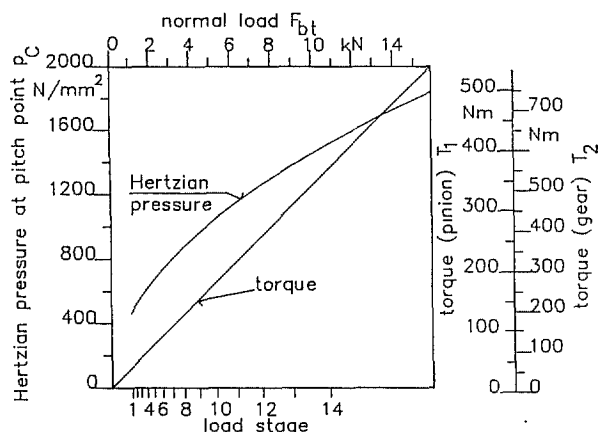


Fig. 2 : Relation between load-stages, torque, normal load F_{bt} and Hertzian pressure p_C at the pitch line of the test gears

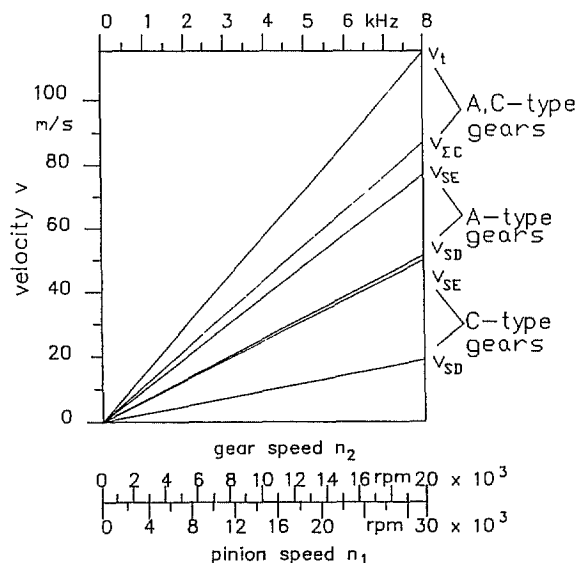


Fig. 3 Pitch line velocity v_v , sum velocity v_{xc} at pitch point C, sliding velocity v_{se} , v_{sd} at the engagement points D and E of the test gears

All test gears are 15°-MAAG ground. The deviations of pitch, profile and involute were measured

		type A	type C
center distance	a (mm)	91.5	
number of teeth (pinion / gear)	z_1 / z_2	16 / 24	
module	m (mm)	4.5	
tooth width	b (mm)	20	
working pitch diameter (pinion / gear)	d_{w1} / d_{w2} (mm)	73.2 / 109.8	
addendum diameter (pinion / gear)	d_{a1} / d_{a2} (mm)	88.77 / 112.50	82.64 / 118.64
addendum modification factor	pinion x_1	0.8635	0.1817
	gear x_2	-0.5000	0.1715
pressure angle	α_n (deg)	20.0	
helix angle	β (deg)	0	

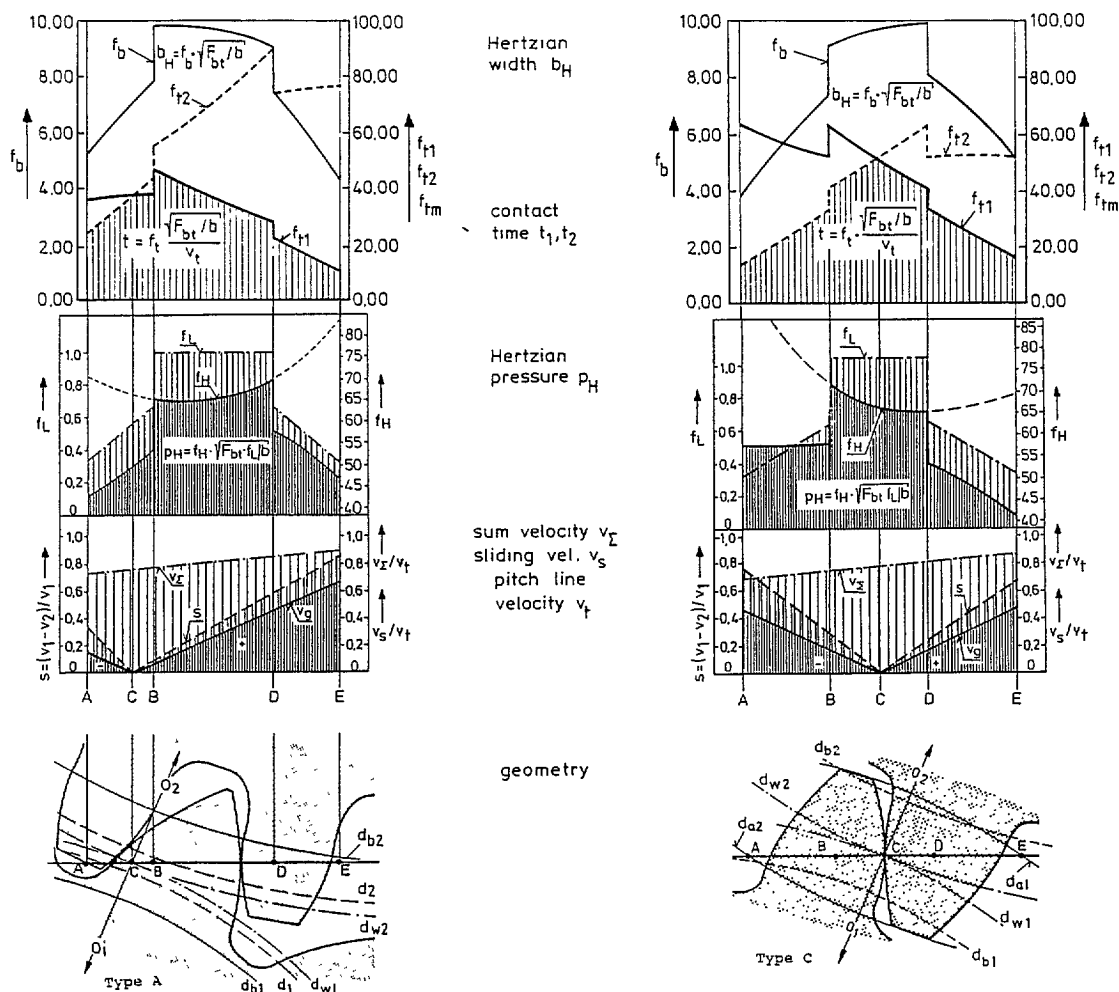


Fig. 4 Geometry, load and speed distribution along the line of contact of test gears type A and C

before the tests. The gear accuracy corresponds to DIN-quality 5 and better (AGMA-quality 12). The surface roughness of three teeth of each gear was also measured. The composite arithmetic (CLA) roughness value, measured in direction of the involute, is $R_a = 0.56 \pm 0.12 \mu\text{m}$ at the new pinions (measurement length $l_t=4.8 \text{ mm}$, cut-off $\lambda=0.8 \text{ mm}$) and $R_a = 0.39 \pm 0.11 \mu\text{m}$ at new gears ($l_t=2.5 \text{ mm}$, $\lambda=0.25$).

3 Test Procedure, Interpretation of the Results

In the standard tests the pinion drives the gear. The oil is sprayed into the mesh with an oil flow rate of 5.5 l/min, the oil temperature is $90 \pm 3 \text{ }^\circ\text{C}$.

The test starts at the lowest load stage No.1 ($F_{bt}/b = 50 \text{ N/mm}$). After 15 minutes running time the gears are inspected visually with a stereo microscope. If there are no scuffing marks visible, the load is increased by 20 % (load stage No.2) and the same process starts again, etc..

If there are scuffing marks visible, the patterns of the scuffed areas of pinion and gear are noted. If the damaged area is more than 20 % of the active flank of pinion and gear the failure limit is reached and the test is terminated. If the failure limit is not yet reached, the load is increased by only 10 %. The load corresponding to 20% of scuffed area of pinion and gears is interpolated and reported as scuffing load.

Because scuffing as an instantaneous failure is not influenced by the load cycles, but mainly by the temperature of the gear, the same running time is applied for the whole speed range. The bulk-temperature of the gears has to reach steady state conditions. This is the case after 8-12 minutes running time, depending on the load and only very little on the speed. Therefore a running time of 15 minutes in each load stage was chosen.

It is known, that the scuffing load of gears is strongly affected by smoothing of the surfaces due to running-in [7]. For the test it is required, that the surface roughness does not change during the test procedure. Therefore all gears are run-in before the test at medium load (7th load stage) and low speed ($v_t = 8.3 \text{ m/s}$). So the roughness of the run-in flanks before scuffing occurs is always the same ($R_a = 0.39 \pm 0.04 \mu\text{m}$ (pinion); $R_a = 0.30 \pm 0.07 \mu\text{m}$ (gear)).

4 Appearance of the Scuffing Failure

At all speeds the scuffing patterns of the mating teeth of pinion and gear correspond each other exactly, but the appearance of the scuffing damage differs remarkably depending on the speed. Fig. 5 shows some typical examples.

At low speed ($v_t < 20 \text{ m/s}$) the damages of pinion and gear look very similar. The flanks are rough and the profile of the involute of pinion and gear changes because of material loss (Fig. 5, 1a,b).

At high speed ($v_t > 50 \text{ m/s}$), on the addendum of pinion and gear there is nearly no material loss. The flanks look grey. In some cases, the grinding marks are still visible in the scuffed areas. The corresponding dedendum shows a large amount of material loss, the flanks are very smooth. This effect is independent of whether the pinion or the gear are driving (Fig. 5, 6a,b; 7a,b).

In the medium speed zone ($v_t = 20\text{...}50 \text{ m/s}$) smooth parts with and rough parts without material loss are visible in addendum and dedendum of pinion and gear. The higher the speed, the lower is the material loss at the addendum.

The described effect of speed on the profile of the scuffed area is nearly independent of the load or of the oil used. The scuffing appearance changes at that speed, where the scuffing load has its minimum.

The grey areas look macroscopically very similar to areas of micropitting, but the damage is different. Scanning electron micrographs (Fig. 6) show that the damage mechanism at low speed and at high speed is the same: parts of material of the gear flanks weld with the corresponding pinion flanks. Material is torn out of the dedendum flank and adhered to the addendum surface. At high speed, these parts are very small and thin, so the grinding marks are sometimes still visible under the welded parts. At low speed the parts are larger and grinding marks can no longer be detected.

The reason, why at the beginning of the scuffing process always parts of the dedendum adhere to the addendum of the mating gear is, according to Michaelis [7], that the shearing stresses below the surface, resulting from frictional force and flash