Micropitting of Large Gearboxes: Influence of Geometry and Operating Conditions

Dr.-Ing. Khashayar Nazifi and Prof. I.R. Dr.-Ing. Wolfgang Predki

The focus of the following presentation is two-fold: 1) on tests of new geometric variants; and 2) on to-date, non-investigated operating (environmental) conditions. By variation of non-investigated geometric parameters and operation conditions the understanding of micropitting formation is improved. Thereby it is essential to ensure existent calculation methods and match them to results of the comparison between large gearbox tests and standard gearbox test runs to allow a safe forecast of wear due to micropitting in the future.

Introduction

Micropitting is the word used to describe the modified and dull-grey appearance of case-hardened gear flanks. In particular, the areas of negative specific sliding from engagement, beginning with (Fig. 1, left picture, line A), up to the pitch point (Fig. 1, left picture, line C) are susceptible to micropitting.

Numerous, small material chunks fallen from the surface are detectable by microscope in these grey areas (Fig. 1, middle). A further zooming of the area of interest (Fig. 1, right) shows directed edges of the pitting due to sliding and friction forces over these damaged surfaces. Because of this geometry the edges create shades when the light impacts in one certain direction — from pitch circle to dedendum — and the dedendum of the flank appears dull-grey.

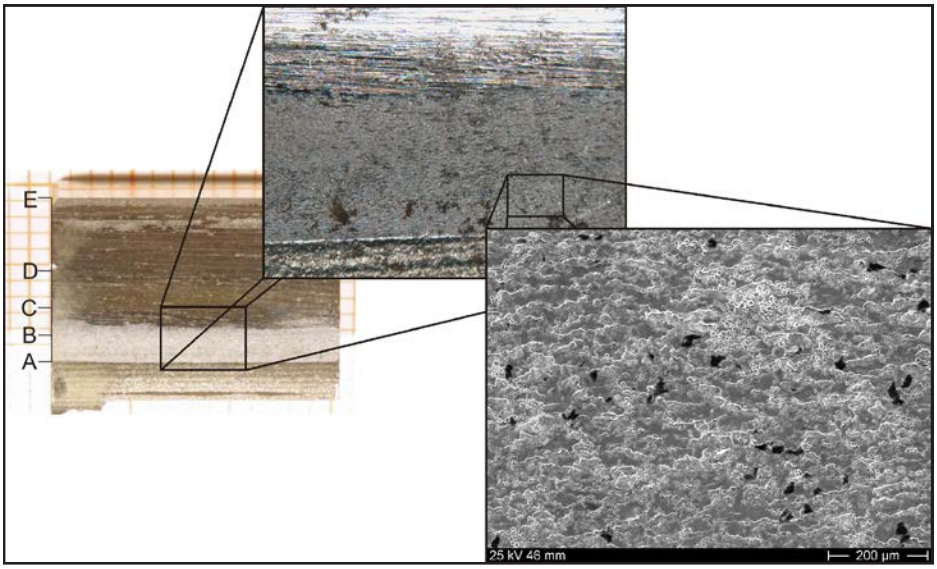

Figure 1 Micropitting on gear flank with different zoom factors.

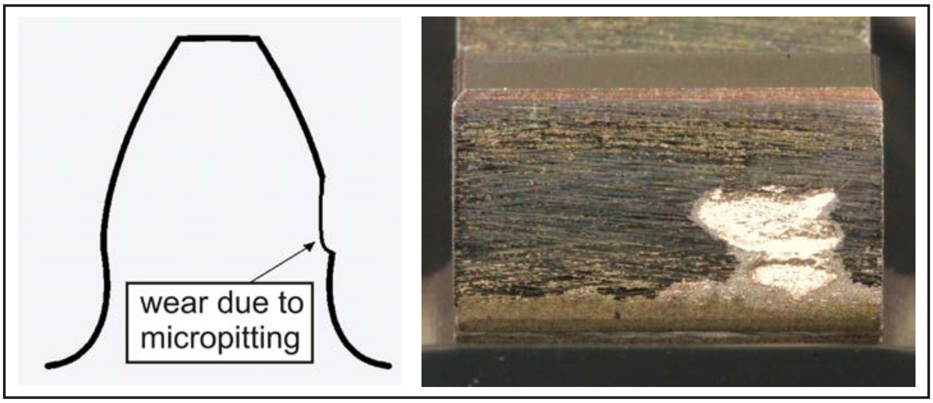

Figure 2 Micropitting (left) deviations on gear flank; pitting (right) caused by micropitting.

With an increasing number of revolutions these material breakouts can develop along the path of contact and into the material depth — detectable as profile form deviations — especially in the dedendum area of gear flanks. Figure 2 (left) shows these deviations caused by wear in an exaggerated illustration.

Observations often show that excessive pitting initiates in or at the edge of micropitting areas (Fig. 2, right). There are at least as many cases with no pitting appearance and stopping micropitting growth.

Since the early 1980s, investigations of the micropitting carrying capacity of gears have been carried out. These investigations were mainly centered on gears with small modules up to 10 mm. Starting with Haske (Ref. 1) and Lützig (Refs. 2-3), the Ruhr-University Bochum began test runs with gears module 22 mm. The investigations of the small gears lead to a calculation formula for local prediction of profile form deviations due to micropitting wear described in FVA Project 259 I (Ref. 4). First comparisons between calculation and test bench results (Refs, 2-3) show deviations when applied on large modules of 22 mm. Further investigations (Ref. 5) of large gears as described in this paper shall lead to a better understanding of micropitting wear and profile form deviations. To achieve this goal, new geo-

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metrical and operational parameters are considered. In order to determine size effects, test runs are carried out on test benches of two different sizes.

Test Benches

The largest spur gear test bench in the world at a research facility is the one at the Ruhr-Universität Bochum. It is specially made for tests on large gearbox teething. The key parameters of the test bench are shown in Table 1.

The test bench is devised as a back-to-back arrangement, making it only necessary to compensate the losses deriving from the bearings, seals and gears as driving power (Fig. 3). The wheel shaft of a spur gearbox is driven over a one-staged, adjustment gearbox by a variable speed electrical motor. This so-called power return gearbox is connected to a second spur gearbox that contains the testing gearset. The torque measurement system is applied to the connecting shaft of the big wheels. The back-to-back connection is closed by the shaft linking the pinions of the spur gearboxes using a hydraulic bracing unit. The bracing unit consists of a hydraulic turning motor that receives hydraulic lubricant through a rotary feedthrough. The change of load level and direction is controlled by a proportional and way valve. The valves are controlled via computer, allowing an exact adjustment of the applied torque in operation. Therefore, load collectives and change of torque direction are possible.

The lubrication systems of the test and of the power return gearbox work separately and have their own lubrication tank. The lubricant injection systems are equipped with temperature-regulated cooling water valves and heat exchangers. Furthermore, extra cooling devices keep the room temperature at a constant level. The lubrication system of the test gearbox has additional immersion heaters that heat the lubricant to the desired test temperature before test start. The tank is filled with 600 l of the test lubricant; from there the lubricant is pumped with a flow rate of 180 l/min to the injection nozzles within the gearbox housing. The lubricant is injected into starting and ending mesh via flat fan nozzles. More injection nozzles are directed to the bearings for lubrication.

Table 1 Test bench parameters

Centre distance	a	447 mm
Maximum cycling power	P_{cyc}	6 MW
Maximum brace torque	T_2	114 kNm
Power of electrical drive	P_{Drive}	240 kW
Load appliance	back-to-back with controlled hydraulic bracing unit	

Several measurement systems with torque, temperature and vibration sensors serve to document the operating condition for further test evaluation. Furthermore, a facility of this size needs intensive monitoring to be able to shut down the test bench automatically in case of damaged bearings or gears.

Test Scheme

Overall there are eight test runs with module 22 mm, and two with module 16 mm gears carried out on the large test bench. Due to the much lower running costs of the small test bench and production costs of small gearsets, a higher number of test runs have been carried out on the small test bench. Therefore 17 test runs with module 4.5 mm and four with module 3.27 mm are made to investigate the size influence on the one hand, and explore non-investigated parameters on the other. Besides the module ratio of nearly five, both gear sizes have almost similar geometry. This leads to a comparable Hertzian contact stress distribution along the path of contact. All other operating conditions have been kept constant to all test runs, as there are lubricant, lubrication temperature $\vartheta = 90°C$, circumferential speed $v_t = 8.3$ m/s and minimal lubricant film thickness. In order to achieve distinguishable results, the damage load stage against micropitting of the lubricant is chosen low — = 8. The test gears of the large test bench have an arithmetical mean roughness of the flanks that ranges between Ra = 0.1 µm up to 0.7 µm, and long linear tip reliefs beginning at the points of single engagement that are between $C_a = 50$ µm and 100 µm. For the test runs on the small test bench, the aritmetical mean roughness of the flanks is adjusted in order to achieve the same minimal lubricant film thickness as on the large test bench; the tip reliefs of the small gears are chosen for the same Hertzian contact stress on the pitch circle as for the large gears. The low costs for small gears enable variants of the profile shift and tip relief investigations, in addition to the size-effect test runs.

Table 2 shows the investigated parameters on the large gearbox. Earlier investigations have shown the major influence of the arithmetical mean roughness on the appearance of micropitting. New production technologies such as super finishing enable a reduction in the

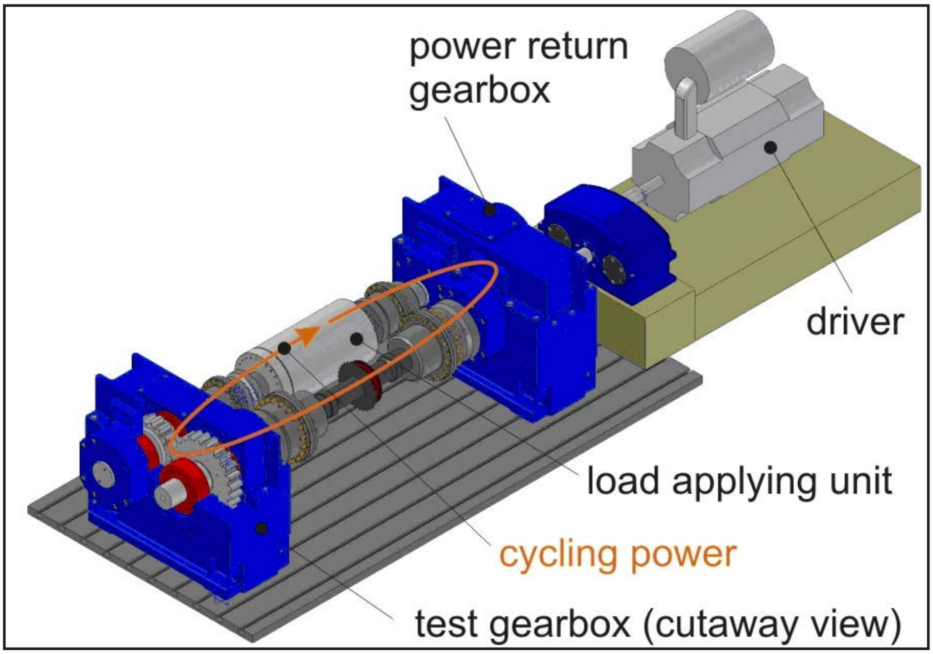

Figure 3 Big gearbox test bench.

surface roughness of gear teeth down to an arithmetic mean roughness of Ra = 0.1 μm — and even lower. For the large gearbox two gearsets have undergone this treatment and started the test runs with Ra = 0.1 μm and Ra = 0.3 μm.

Investigations to date have focused on the gear engagement within the regular contact zone; pre-engagement due to mesh interference has not been taken into account. The idea that the lubricant is wiped off the tooth surface by mesh interference must be verified by one experiment with extremely tip-edge-rounded gears on the last 2-5% of the path-of-contact.

All existing test results on gear micropitting have been carried out on gears with modules up to m_n = 10 mm (Ref. 4) and on the described large gearbox with module m_n = 22 mm (Refs. 1, 2). To close this module gap in the investigations, one gearset with module m_n = 16 mm is used in the test runs. The geometry is designed to achieve similar sliding velocities along the path-of- contact.

The profile shift of the regular C gears is designed for equal specific sliding velocities in the dedendum of the test gears. Typical gears in most applications have increased profile shifts around x_1 = 0.4 and higher. One gearset is designed with a profile shift of x_1 = 0.5 to account for typical applications.

For generating energy with wind or water turbines, high torques with low speeds are transformed to low torques with high speeds. Within the used gearboxes the big wheel drives the pinion. This operating condition is not common to all other gearbox applications and is not yet investigated. One test run has thus far been carried out to ascertain the influences on micropitting.

Gearboxes are open breathing systems. During operation air within the gearbox heats up and its volume increases and excess air is pressed out. On production stops the air cools down again, the air volume decreases, and wet air from outside is sucked into the gearbox. Water condenses on the interior of the housing, runs down the walls, and attains the lubricant. Contamination with 500 ppm water within the lubricant of one test run has to show the effect on micropitting appearance.

Test Results

The size comparison tests show that the profile form deviations deriving from mesh interference have a more significant impact on larger module than on smaller module gears. This can be explained by the fact that in order to receive equal Hertzian contact stresses at the pitch circle for geometrically similar gears, the line load increases in proportion to the module. Therefore the deformation of the gears increases proportionately to the module and creates higher profile form deviations. Conversely, observations show that the profile form deviations caused by micropitting do not increase by the module. Even the morphology of micropitting does not change significantly between different module sizes like crack width, length, angle to surface and/or distance between cracks.

Moreover, it can be stated that gear flank pitting does not grow out of the micropitting area. In fact, pitting develops at the transition of the micropitting area to the undamaged flank as a result of the unworkable geometry. The reasons for this disadvantageous geometry can be, apart from micropitting wear, shaving marks due to mesh interference or unfavorably chosen profile modifications. If wear on the gear flanks is unavoidable, the profile form deviations caused by the wear have to be directed by profile modifications to have the least effect on pitting occurrence.

Figure 4 shows the flank of a superfinished gearset with arithmetical mean roughness of Ra = 0.12 μm within an endurance test run. At left, in the figure, the flank can be seen shortly before the pitting failure. Except for the shaving line in the dedendum nothing else can be observed on the flank. After the next run of the endurance test the whole flank is failed due to massive pitting, as shown on the right side.

Figure 5 shows that the minimum radius of curvature r_{min} of a rougher gear flank moves with increasing load cycles towards the flank and therefore the region with highest contact stresses moves along the flank as well; a low surface roughness avoids this movement along the path of contact. Hereby the material is highly loaded at the same position and pitting occurs. Test runs with gears possessing flanks with arithmetical surface roughness between Ra = 0.1 μm and 1.0 μm show that, with increasing micropitting areas, the pitting-free lifespan can be increased. In contradiction to this effect, stagnating micropitting areas that only grow into the material create higher-profile form deformations, and therefore reduce the minimum radius of curvature r_{min} and decrease the pitting-free lifespan.

Tip relief and tip-edge-rounding have proven to be a proper method to reduce mesh interference. Tip-edge-rounding can be applied purposefully by grinding or as a side effect of other finishing technology. A relation between the duration of the superfinishing procedure and the

Table 2 Overview of investigated parameters on large gearbox

Investigated parameters	Standard as defined in [1], [2]	Variation
Arithmetic mean roughness	0.8 μm	0.1 μm – 0.3 μm
Tip edge roundness	20 μm	250 μm
Module	22 mm	16 mm
Profile shift	x_1 = 0.18	x_1 = 0.5
Driving gear	Pinion	Large wheel
Water contamination	0 ppm	500 ppm

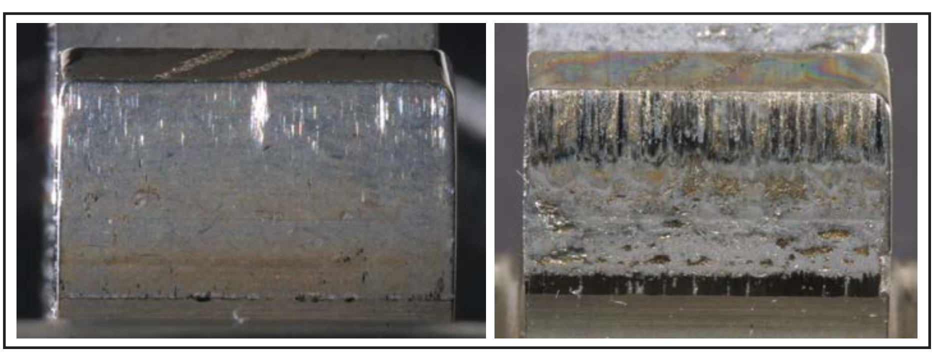

Figure 4 Gear flank m_n = 4.5 mm, Ra = 0.12 μm. (Left) N = 33.3·10^6; (right) N = 43.7·10^6.

tip-edge-roundness has been observed within the investigations. The superfinishing process is continuously optimized towards higher contour accuracy of the flanks so that very fine surfaces can be achieved with shorter process times. These optimizations lead to less roundness of the tip edges and the positive side effect declines.

Aside from the reduction of shaving due to mesh interference, the tip-edge-roundness leads to better lubrication of the flank at start-of-contact. This can be observed with gears of module $m_n = 4.5$ mm and grinded tip-edge-roundness. Although the mesh interference effect of these gears is not so pronounced on the large gearsets, the test results show a higher resistance to micropitting and fewer profile form deviations.

Without considering the mesh interference, the correlation between arithmetical mean roughness and profile form deviations had been lost for gears with low tip relief in the former project (Ref. 2). By the newly gained knowledge about mesh interference and the resulting damages, the old results can be re-evaluated. With the re-evaluation the correlation between arithmetical mean roughness and profile form deviations can be reconstructed again; it shows that with increasing flank roughness the profile form deviations caused by micropitting grow for small as well as for large gears.

The influence of arithmetical mean roughness on micropitting creation is demonstrated for large gears featuring flanks with very low roughness of Ra = 0.1 µm. Chemically accelerated vibratory-finished gears show very small micropitting areas due to the very low surface roughness. Test runs with gears module $m_n = 22$ mm have shown that an arithmetical mean roughness of Ra = 0.3 µm is completely sufficient for suppressing micropitting. The results of gearsets with practical profile shifts on large and standard test benches show congruently the influence of specific sliding velocities on micropitting development. By increasing the positive profile shift on the pinion, the amount of specific sliding velocities in the dedendum of the pinion is reduced. This area of the flanks shows — compared to the regular C gears — very small micropitting. As a consequence of the fix center distance of the test bench gearboxes, the profile shift sum must be kept constant. As a result of this boundary condition extreme, negative profile shifted big wheels occur with high amounts of negative specific gliding velocities in the dedendum. This is now the place where large micropitting areas can be observed after the test runs.

Test runs with a driving wheel have illustrated that the driving direction has an influence on the resulting profile form deviations. The micropitting-driven profile form deviations are a function of sliding velocities and surface roughness of the flanks, and both values are not affected by changing from driving pinion to driving wheel.

Profile form deviations do not have different causes, yet these deviations appear not separately on the flank and lay on each other. This is what is meant by superposition. Essential for mesh interference is the stiffness in the outer point of single engagement. Is the driving direction changed the outer point of single engagement swaps from point D on the path of engagement to point B. The regular C gear is much stiffer in point B than in point D. Therefore the resulting profile form deviation is lower for driving wheels than for driving pinions. As expected, the place of maximum profile deviation is not the dedendum of the pinion but rather the dedendum of the wheel. The tip-edge shows by entering the engagement with the sharp corner towards the mating flank (Fig. 6, left) and shaves over it. Leaving engagement, the obtuse corner is pulled over the mating flank (Fig. 6, right). Therefore the shaving at pre-engagement is rated as more damaging than at post-engagement.

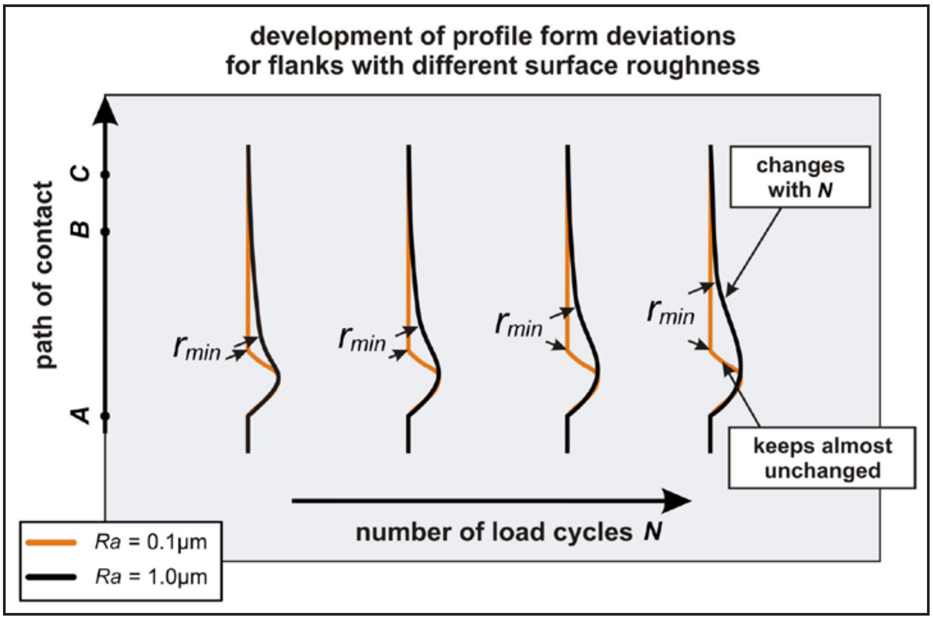

Figure 5 Development of profile form deviations for flanks with different surface roughness.

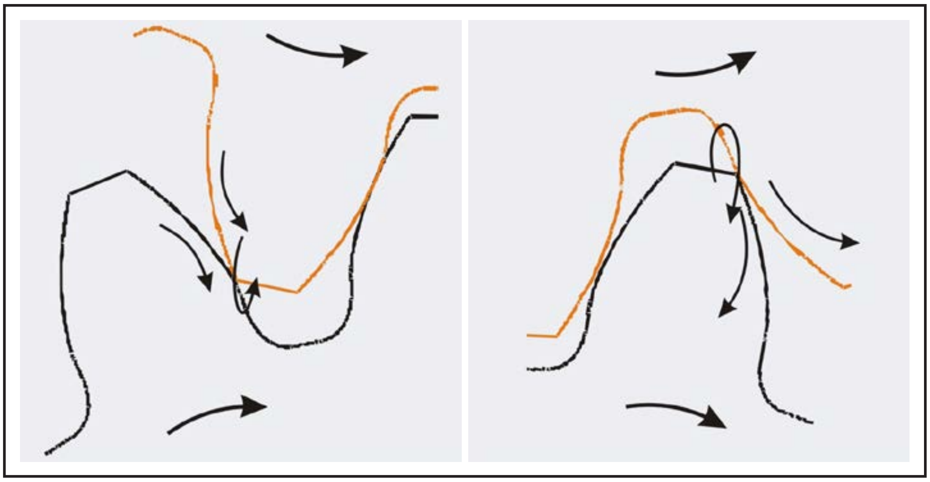

Figure 6 Kinematics of gear teeth: (left) at pre-engagement; (right) at post-engagement.

The lubricant used for test runs appears to be relatively impervious to water contamination, and the micropitting carrying capacity is unchanged. The water quantity used in the test runs is W = 500 ppm, although there is a dependency between contamination duration and wear amount.

On the first flank the reaction duration of additive and water is probably too short, and no negative influence can be observed. On the second flank the wear increases. This flank has been exposed to the contaminated lubricant as working as well as the non-working flank. Pretests with lubricants used in wind turbine gearboxes show vastly differing reactions to water contamination. With dependency on the used additives, the wear on the flank increases. Two different cases have to be considered in the water contamination of lubricants:

1. Water is completely insolvent in lubricant.
2. An oversaturated lubricant with free water molecules is given.

In the case of solved water in lubricant, the reaction between water and additives is critical. The pre-tests have shown that oxidation on flanks is not critical respective to wear.

Calculation Results

Within project FVA 259 I (Ref. 4) a new approach for the calculation of micropitting-caused wear is shown. This method considers different local lubrication

$$f_{fm}(Y)=f_{fm,GFT}\cdot\left(\frac{\zeta_{GF}(Y)}{\zeta_{GFT}}\right)\cdot\left(\frac{b_{H,GF}(Y)}{b_{H,GFT}}\right)\cdot\left(\frac{p_{H,GF}(Y)}{p_{H,GFT}}\right)^{0.25}\cdot\left(\frac{\lambda_{GF}(Y)}{T_{\lambda}\cdot\lambda_{GFT}}\right)^{-1.25}\cdot\left(\frac{N_{GF}}{N_{GFT}}\right)^{0.25}\cdot\left(\frac{m_{GF}}{m_{GFT}}\right)^{-1}\cdot C$$

Figure 7 New formula for calculating micropitting-caused wear.

conditions, loads along the tooth profile, micropitting carrying capacity of the lubricant, and profile modifications. In order to calculate wear for a gearbox, the operating conditions of the regarded gearbox (index „GF") are compared to the operating conditions of the standard gearbox (index „GFT"). The calculation formula is adjusted with the knowledge that micropitting wear does not grow with module size. The old calculation method is enhanced by the ratio of investigated module to standard module as shown in Figure 7. Figure 8 shows a comparison between test results, i.e., previous calculation method vs. enhanced calculation. As can be seen, the result is a very good fit.

Summary

- By virtue of detailed analysis, profile form deviations detected in test runs are cause-determined.
- Having this knowledge enables study of each deviation and cause on its own.
- It can thus be shown that micropitting morphology is very similar between different module sizes.
- Deviations caused by micropitting are not relative to module, and deviations caused by mesh interference are relative to module.
- Moreover, the influence of micropitting and, in particular, the influence of flank geometry on pitting occurrence is shown to be related to high local Hertzian stresses.

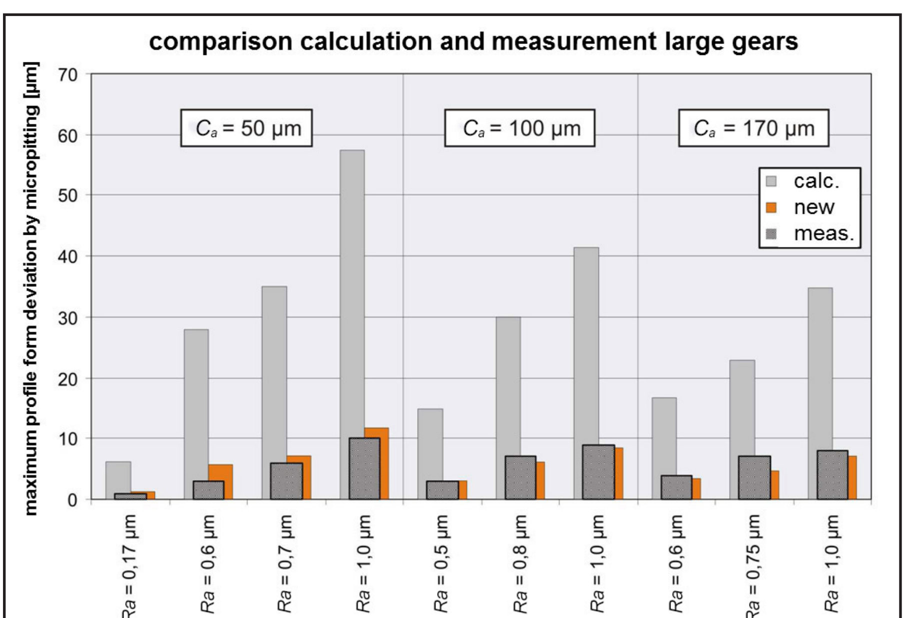

Figure 8 Comparison between test results; old calculation and new method.

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Dr.-Ing. Khashayar Nazifi holds the position of Dorsten plant manager for the Zollern Company since March 2013. Concurrently, he was additionally named head of the research and development department of Zollern's Drive Technology unit since April 2014.Upon receipt of his engineering degree he began his career as research assistant at the Chair for Mechanical Components, Industrial and Automotive Power Transmission (LMGK). Nazifi's industrial profession first started at Vestas, until his move in 2011 to Zollern to work initially as team leader/research and later as plant R&D manager.

Until his retirement in 2012, **Prof. Dr.-Ing. Wolfgang Predki** served with distinction as Head of Chair for Mechanical Components, Industrial- and Automotive Power Transmission (LMGK) at Ruhr-University Bochum since 1992.Upon receiving his doctorate in 1982, Predki began work at the former Flender AG. His last position was as Head of Engineering, where he was responsible for worm, standard-helical, and planetary gears. In his 20 years as a professor Predki supervised many dissertations, published countless reports, and headed the DIN and FVA working groups for worm gears.