

Micropitting of Big Gearboxes: Influence of Flank Modification and Surface Roughness

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Management Summary

Micropitting—or grey staining—is a fatigue failure on tooth flanks that is mainly influenced by lubrication conditions that can compromise the dynamics and noise behavior of gearboxes. Furthermore, it can lead to larger cracks and, finally, to pitting. Present experiments on micropitting have been conducted, primarily on small-module gears. Test runs on spur gears with a module of 22 mm on a large-gear test bench (center distance 447 mm) and, for comparison with previous investigations, test runs on a standard gear test machine were performed.

The experiments concentrated on lubrication behavior impacted by changing additives, temperatures and circumference velocities on the one hand, and surface roughness and profile form modification on the other. A comparison of calculations according to FVA 259 (Ref. 1) and actual test results help to validate the method for large modules.

Introduction

Gears have long been integral parts of machinery, and vast knowledge of gearboxes and gears has been gained through research. The results of these projects have been used as a basis for a high number of design instructions and calculation methods. Yet a lot of these research projects are carried out on relatively small gears. The positive points of this approach are pretty obvious: The whole test bench gets smaller, easier to handle and, most importantly, more economical. Often, high revolution speeds can be achieved that lead to a high number of load cycles up to the range of infinite life in a substantially short time. But not all results achieved on small gearboxes can be scaled up to big applications. Therefore, tests on big gearboxes with teeth close to the size that are actually used in the application are necessary. Several producers of industrial gearboxes have test benches for big gearboxes that are used for regular, functional and noise tests on gearboxes out of the production line.

Nevertheless, these test benches cannot be used for comprehensive investigations due to cost and capacity reasons.

Moreover, the utilization of gearboxes in daily use for systematic investigations with changing parameters like speed and torque is not possible, because they are tied to an economical operation management. As a result, research facilities at universities are challenged.

Big-Gearbox Test Bench at the Ruhr-Universität Bochum

The big-gearbox test bench at the Ruhr-Universität Bochum is designed for tests on big gearbox teeth. It is the biggest spur gear test bench worldwide at a research facility. The key parameters of the test bench are shown in Table 1.

The test bench is built up similarly to the standard back-to-back test bench according to DIN 51354 (Ref. 2), while the center distance is five times bigger. Figure 1 shows a principal sketch of the test bench arrangement. The wheel shaft of a spur gearbox is driven over a one-staged adjustment gearbox by an electrical motor with a variable speed. This spur gearbox is known as a “power return gearbox.” A second spur gearbox contains the testing gear set. The tooth geometry of the power return gear set is equal to the test gear set, yet the

face width of the test gear set is reduced. The shafts of both spur gearboxes are connected to each other via curved tooth couplings. The torque measurement system is applied to the connecting shaft of the big wheels. The back-to-back connection is closed through the shaft linking the pinions of the spur gearboxes by use of a hydraulic bracing unit. The bracing unit consists of a hydraulic turning motor that is applied hydraulic lubricant through a rotary feed-through. 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 the change of torque direction are possible.

The lubrication systems of both the test and power return gearbox work separately—each with its own lubrication tank—so that both gearboxes can be operated with different lubricants. 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 liters 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 the starting and ending mesh via flat fan nozzles. More injection nozzles are directed to the bearings for lubrication.

The gearbox housings are prepared to be equipped with axial bearings, so the ability exists to perform tests with spur as well as helical gears. Several measurement systems with torque, temperature and vibration sensors allow for documentation of the operating condition for further test evaluation. Furthermore, a facility of this size needs intensive monitoring and the ability to shut down the test bench automatically in case of damaged bearings or gears.

Aside from the described back-to-back test bench arrangement (Fig. 2, right), often a driver-break arrangement (Fig. 2, left) is used. Therefore, a second inverse connected test gearbox is used to increase the speed and decrease the torque at the break. This second arrangement is closer to the practical application of a common system with driver, gearbox and working machine. Yet this arrangement demands that driver and break are always working at the full power that is applied on the test gearbox. These high powers and the long durations of tests, though necessary for research studies, result in disproportionate effort in energy and heat removal. However, in the back-to-back arrangement the driver just needs to compensate the losses deriving from the bearings, seals and gears. Only the heat resulting from these losses needs to be removed by the cooling facilities.

Micropitting on Gear Teeth

As stated, micropitting is gear wear that is mainly influenced by the lubricant and by the lubrication conditions. The matte-grey appearance of the teeth is due to numerous small flakes and cavities. Micropitting often appears with flank loadings below the calculated pitting endurance strength according to ISO 6336 (Ref. 3). Figure 3, left shows micropit-

continued

Table 1—Test bench parameters		
Parameter	Abbreviation	Value
Center 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
Module	m_n	22 mm
Tip circle diameter	d_{a1}/d_{a2}	403 mm/570 mm
Number of teeth	z_1/z_2	16 / 24
Face width	b	100 mm
Profile modification	x_1/x_2	0.18 / 0.17
Load appliance	back-to-back with controlled hydraulic bracing unit	

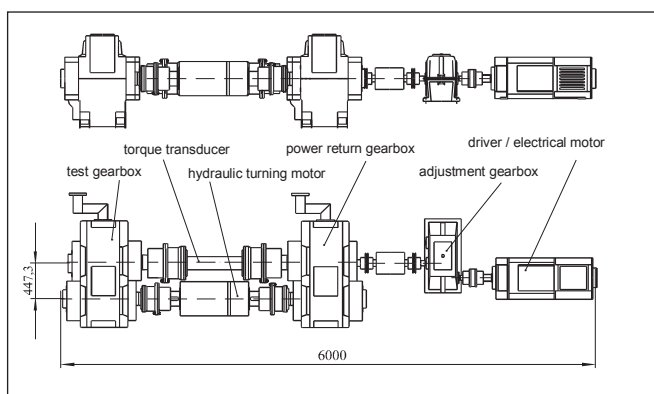


Figure 1—Big-gearbox test bench.

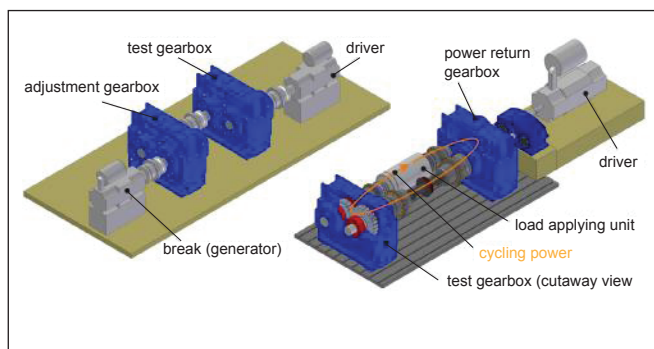


Figure 2—Test bench arrangement concepts: driver-break (left); and back-to-back (right, with a cut section through the test gearbox).

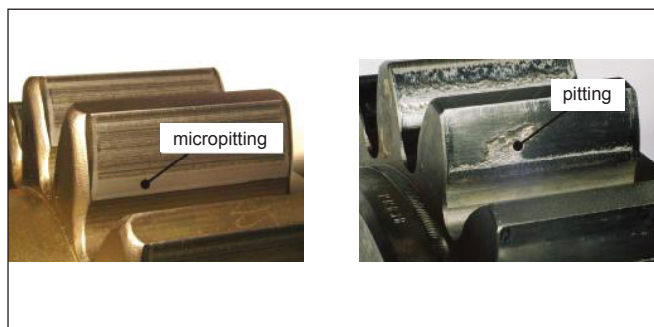


Figure 3—Gear flanks with micropitting (left) and pitting (right).

ting on a gear flank.

The occurrence of micropitting is accompanied by material removal—which leads to profile form deviations. These deviations, with the grey-stained area and weight loss, are conducive to an assessment of micropitting development. Micropitting can lead to deep cracks and eventually to larger spalling (Fig. 3, right). These profile form deviations can change gear dynamics and acoustic behavior.

A popular model of the development mechanisms of micropitting identifies the surface roughness and the sliding

speed as the most important influences on appearance. Via the sliding between the meshing teeth, roughness peaks on the surfaces are pushed aside and folded over, causing a flaked structure. Figure 4, right, shows a flaked surface through a scanning electron microscope. While these flakes are built, the material is plastic-deformed and weakened so that cracks can occur. The loading due to the Hertzian contact—as well as the shear forces caused by friction—make these cracks propagate in a relatively flat manner beneath the surface. A chunk of material drops off when two cracks meet; new flakes are built at the edges of the spalling on every further revolution. This progression can be continued, and the cracks can propagate and cause large pitting.

Often it can be observed that micropitting stops propagating by itself. The most influencing parameters for the micropitting propagation are the lubricant and additives used, the lubrication temperature, the geometry of the teeth, the surface roughness and the applied load.

Standard Micropitting Test

The investigations on micropitting are done in a standardized test (Ref. 4). Since the lubricant has a significant influence on micropitting occurrence, this standard test is mainly used to prove a certain micropitting load-carrying capacity of lubricants. This test consists of a step and then an endurance component. A defined test gear set is increasingly loaded in six load stages. Each load stage consists of 2.1 million revolutions of the pinion. After each load stage, the pinion is dismantled and weighed, the profile form deviation is measured and the surface is visually inspected. The step test is followed by the endurance test with another six intervals of 10.5 million revolutions of the pinion on a high-load level. For the characterization of the load, the Hertzian pressure at the working diameter is determined. The test-achieved Hertzian pressure of 1,550 N/mm² is close to the material strength limit of the surface according to pitting. The standard test is performed with relatively small gears, with a module of 4.5 mm, a center distance of 91.5 mm and a gear ratio of 24/16. The damage-load stage is defined as the load stage where the limiting profile form deviation of the pinion is exceeded.

Micropitting Test Runs on a Big-Gearbox Test Bench

The test runs on the big-gearbox test bench follow the same scheme as the standard test (Fig. 4, left). The other test conditions and the gear set geometry are chosen according to the standard test, but this time scaled by the factor of five. Therefore, the comparability between results of the standard test bench and the big-gearbox test bench is shown. The Chair of Mechanical Components and Power Transmission employs several standard test benches in addition to the big-gearbox test bench for these kinds of comparisons. The key parameters of the used gears are listed in Table 1, and Figure 5 displays the size difference of the gears. The difference of size has its biggest impact on the testing time—i.e., due to the significant influence of the circumferential velocity on the micropitting behavior of gears, the velocity must remain constant over the different gear sizes. For that reason, the revolutions-per-minute are decreased by increasing gear diameters. A complete stage test on the big-gearbox test bench takes about 25 days, not including assembly time. Compared to the big gearbox, the same test takes only six days on the standard test bench.

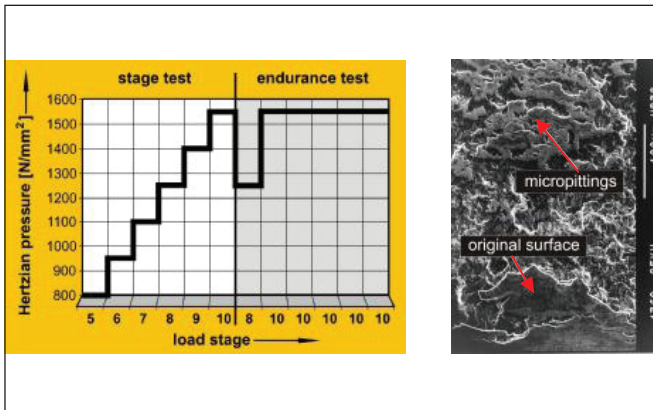


Figure 4—Standard micropitting test procedure (Ref. 4) (left); tooth surface through a scanning electron microscope (right).

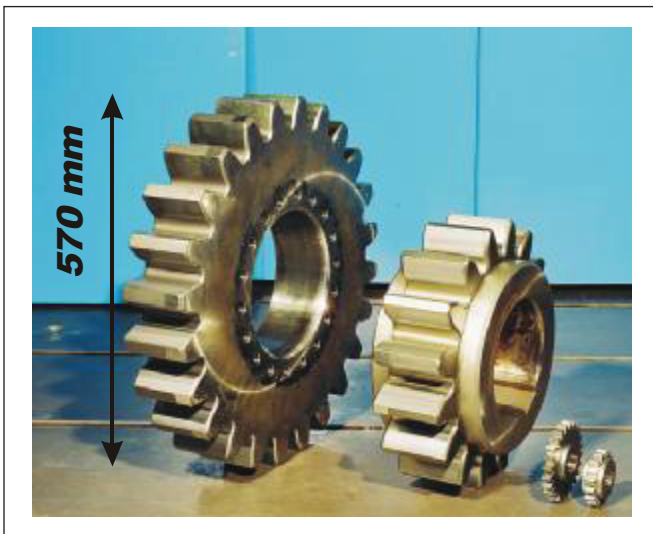


Figure 5—Test gear sets: big (left); and standard (right).

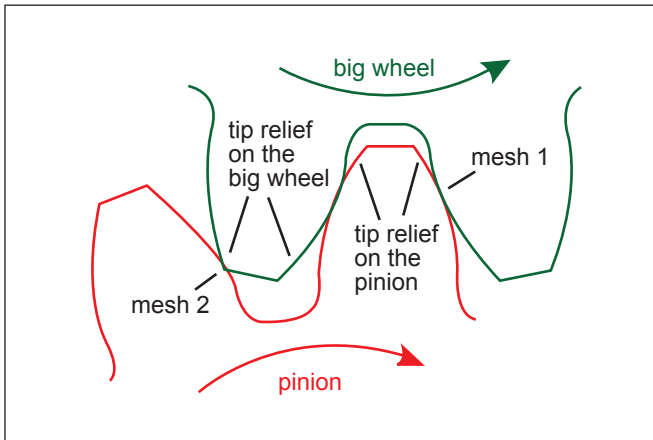


Figure 6—Engagement of gears, with/without tip relief.

Because of the amount of labor expended for assembling and disassembling, the measuring of the big gearbox's pinion after each load stage is not possible. The solution to this problem is found in a replica of the gear tooth flank. With a special-made device, the replica can be produced with good repeatability and can later be arranged on a gear measuring center for measure.

Test Scheme

As stated, the tests on the big gearbox are very time-consuming. Moreover, the costs for gear sets in the desired high quality are very high. Therefore, testing is necessarily limited. The main aim of the tests on the big gearbox is the spot-check comparison of gained knowledge on the standard gearbox with that of the tests carried out on big gear sets. Through this approach the ability exists to see which effects can be transferred linearly to big gear sets and where the size influences the micropitting occurrence.

Thus far, two micropitting research projects have been carried out on the big-gearbox test bench.

The first project (Ref. 5) focused on the following parameters:

- **Lubricant:** Two mineral oils with different additive packages and one synthetic lubricant (polyglycol)
- **Operation temperature:** Oil injection temperatures of 60° C and 90° C
- **Circumferential velocity:** Standard velocity of 8.3 m/s according FVA 54 I-IV [4] and a reduced velocity of 4.2 m/s

The second project (Ref. 6) focused on two parameters:

- **Surface roughness:** Arithmetic mean roughness between $R_a = 0.5 \mu\text{m}$ and $1.0 \mu\text{m}$
- **Profile modifications:** Linear and parabolic tip relief between $C_a = 50 \mu\text{m}$ and $170 \mu\text{m}$

Profile modifications are directed deviations from the ideal involute of the teeth. The purpose of the modifications is to compensate the load-applied deformations of housings, shafts and gear teeth. But these modifications also influence load distribution between the teeth. The meshing of two spur gears is divided into phases by which two teeth pairs are in contact and phases where just one pair is meshing. The phases—with two pairs in the mesh of the addendum and the dedendum of the meshing flanks—are in contact.

Figure 6 shows a pinion driving the big wheel—two teeth form the Mesh 1 (right). In the shown position an additional tooth pair is entering the mesh (Mesh 2, left). The teeth addenda have a profile form modification—i.e., there is material loss at the tip of the teeth. The teeth in Mesh 1 now have to bend more until the teeth in Mesh two come into contact; thus Mesh 1 is carrying a heavier part of the load than Mesh 2. This effect can be seen in the run of the Hertzian pressure along the pinion flank (Fig. 7) for the tooth completely on the left side in Figure 6. The unmodified gear mesh shows a relatively equal load distribution between the two tooth pairs in the double engagement area. In return there is a relatively harsh step as the second tooth pair leaves the mesh and just one tooth pair takes over the complete load (single engagement area). The picture changes completely for the same gears with a tip relief of 170 μm ; the load distribution changes, and there is a significant lower load at the beginning of the mesh in the

dedendum of the pinion. The load increases continuously and the step between the single- and double-engagement area is much smaller.

Micropitting occurs mostly in the dedendum of the driving wheel. This area shows the most disadvantageous sliding conditions on the tooth flank. This is the area where the tip relief lowers the load, and big impact is to be expected. The same applies for lower surface roughness of the flanks, as the high-roughness peaks are the origin of cracks.

Figure 8 shows the result of a test run. The flank picture **continued**

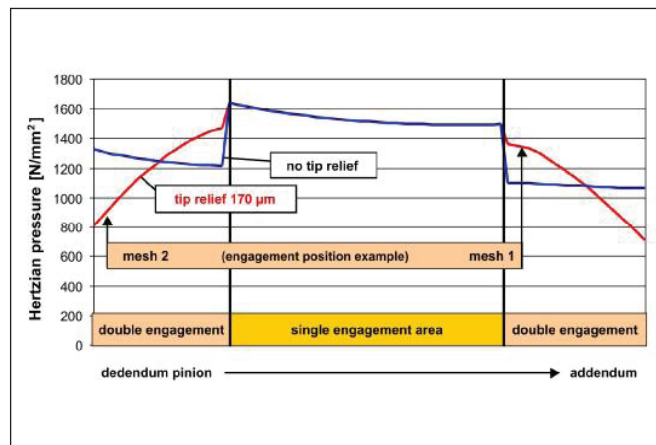


Figure 7—Run of the loading along the pinion flank for modified and unmodified gears.

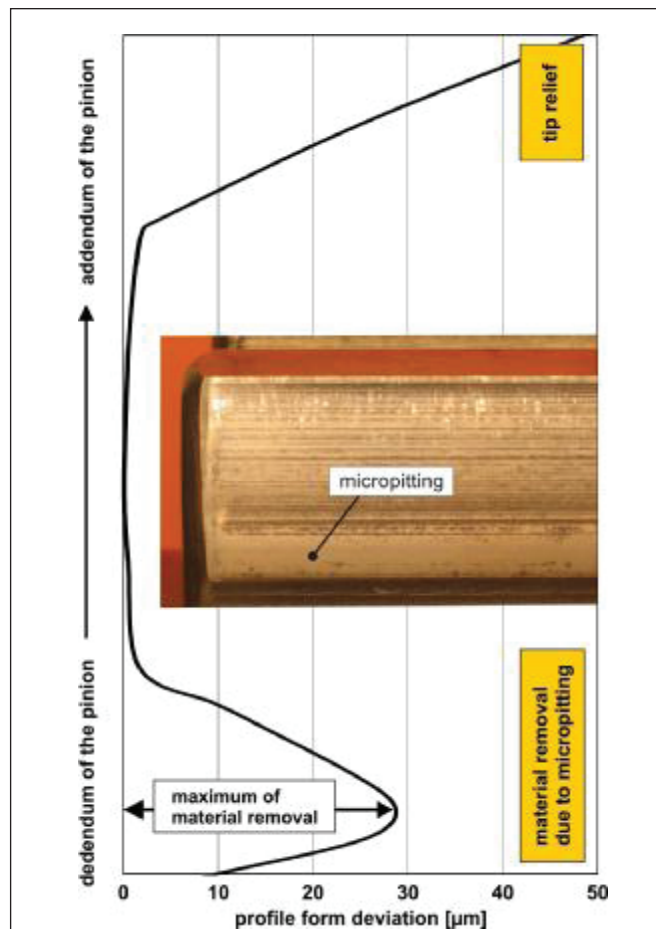


Figure 8—Picture of the flank after test; corresponding profile measurement.

shows micropitting on the dedendum of the pinion flank. The profile measurement diagram shows the deviation of the pinion surface to the ideal involute. On the tip area of the flank, the diagram in Figure 6 shows engagement of gears with/without tip relief, and intentional material removal of the tip relief. Starting at the maximum value of 50 μm , the relief goes linearly back to zero. The middle part of the flank shows almost no deviations. The dedendum area of the flank with micropitting shows a deviation to the unused flank. The microscopic spallings lead to a material removal and a deviation of the flank surface up to a depth of 30 μm . The comparison of these profile form deviations allows an objective relationship between the different test variants. These results are used to validate a calculation method that allows an estimation of the micropitting occurrence on the flanks of a gearbox already in draft stadium.

Test Results

In the first project, lubricants that have a higher micropitting capacity on the standard gearbox show a better micropitting resistance on the big gearbox as well. Yet this improved micropitting behavior does not lead to a lower material removal in the dedendum of the gears. In fact, such lubricants might even lead to a higher possibility of pitting occurrence due to the insufficient flank modifications. The lowered injection temperature from 90°C to 60°C causes higher profile form deviations, thus proving that the additives of the oil must fit to the temperature at and during gear operation. No influence of the circumferential velocity on the profile form deviation depth can be observed. The comparison between the big and the standard gearbox shows that the profile deviations on the big gearbox are deeper, but they don't increase constantly by the proportional factor of 5 between the two gearbox sizes.

In the second project, the systematic variation of the surface roughness proved its influence on the micropitting behavior for big gearboxes. Here high roughness leads to wider micropitting areas. And yet, with low profile modification amounts, the profile form deviations depth becomes independent of the arithmetic mean roughness. The rising of the profile modification amount leads to lower profile form deviations; only in very large modification amounts do we again see higher deviations. Most of the material removal migrates from the dedendum towards the pitch diameter, with rising profile modification amounts. In addition, very small radii were identified at the changing point of the modified flank to the unmodified flank. These radii lead to high pressures and, so it follows, to a higher material removal at these positions.

The comparison calculations with the method described in FVA 259 (Ref. 1) show that for the majority of test runs the calculation leads to larger depth than the experiments; in general, the calculation method is also valid for gears with a large module.

Conclusion

The projects helped to investigate micropitting on big gearboxes. The comparison between the two modules shows that the profile form deviations grow with the module size, yet that there is no constant factor. The lubricant behavior is comparable between the module sizes—i.e., a high micropitting carrying capacity on a small gearbox leads to small micropitting areas on big gearboxes. The influence of the surface roughness on the micropitting behavior is the same for the different module sizes—higher roughness leads to wider micropitting areas. The variation of tip relief forms and amounts shows that this has a huge influence on micropitting. After reaching the minimum of the profile form deviation, a further rising of the tip relief amount leads to deeper profile form deviations again, yet on a different position. The comparative calculations show that in general the calculation method is valid for large modules. Overall, the projects aid in the design of more reliable gearboxes with big modules in respect to micropitting behavior. ⚙

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