Increased Tooth Bending Strength and Pitting Load Capacity of Fine-Module Gears

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Miniaturization is one of the major trends for future drivetrain design. Hence more and more small motors and gearboxes are available at the market. Also components like gears becoming smaller. Modules of 1 mm and less are not unusual. But the common calculation methods according to ISO 6336 and DIN 3990 are mainly verified for gears with module 3 mm — 10 mm. These investigations showed a decreasing load-carrying capacity for tooth bending strength and pitting resistance with increasing gear size. But for gears with module less than 5 mm, a size effect is not thus far considered in the calculation methods.

Therefore theoretical analysis and experimental investigations were done to verify the load-carrying capacity of small-sized gears. The results prove an increased tooth bending strength and pitting resistance of approx. 30% for case-carburized gears with module 0.6 mm, compared to gears with module 5 mm. Hence a proposal for an extended size factor for the calculation method according to DIN 3990 and ISO 6336 was derived.

Introduction

Miniaturization is one of the major trends in drivetrain design. Therefore, gearboxes with gears with a module of 1 mm or less are increasingly used (Refs. 10 and 18). For industrial robots, waste-heat-recovery units, and rapidly accelerating pick-and-place applications, high power densities are required; hence these gears are often made of case-carburized steels. The design of the gears is typically based on the calculation methods according to DIN 3990 (Refs. 3–4) and ISO 6336 (Refs. 6–7) for tooth root bending strength and pitting resistance. These calculations are based on manifold tests on gears with $m_n = 5$ mm. Special research projects (Refs. 21 and 23) prove decreasing load-carrying capacity with increasing gear size. But only a thin data basis is available for smaller gears in the size range of approx. $m_n = 1.5\ldots$ 5 mm. For smaller sizes, the calculation methods are not verified at all. Therefore, no gear size influence is considered in the calculation for gears with a module of less than 5 mm. Hence small-sized gears have wasted load-carrying capacity. For this reason the load-carrying capacity of small-sized gears was theoretically and experimentally investigated.

Theoretical Influence of Gear Size on Load-Carrying Capacity

The common calculation methods according to DIN 3990 and ISO 6336 are based on a comparison of occurring stress and allowable stress. The influence of gear size on the load-carrying capacity is considered with the size factors Y_X (tooth root bending) and Z_X (pitting), but there are further influences, which should be considered.

In the following, major influences of gear size on the load factors as well as on the permissible tooth root bending and contact stress will be discussed.

2.1 Influence of gear size on the load factors K_v, $K_{H\alpha}$, $K_{H\beta}$, $K_{F\alpha}$, $K_{F\beta}$. The influence of gear size on the occurring stress is limited to the load factors. The common calculation methods for tooth root stress and contact stress are presented in Equations 1 and 2.

$$\sigma_F = K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha} \cdot \frac{F_t}{b \cdot m_n} \cdot Y_F \cdot Y_S \cdot Y_\beta \tag{1}$$

$$\sigma_{H½} = \sqrt{K_A \cdot K_V \cdot K_{H\beta} \cdot K_{H\alpha}} \cdot Z_{B/D} \cdot Z_\varepsilon \cdot Z_\beta \cdot Z_H \cdot Z_E \cdot \sqrt{\frac{F_t}{d_1 \cdot b} \cdot \frac{u+1}{u}} \tag{2}$$

The application factor K_A considers externally induced overload. The dynamic factor K_v takes into account internal dynamic loads, while the transverse load factor $K_{F\alpha}/K_{H\alpha}$ and the face load factor $K_{F\beta}/K_{H\beta}$ consider the influence of uneven load distribution at the meshing teeth respectively along the face width; detailed descriptions of all factors are summarized in ISO 6336.

The dynamic factor K_v mainly depends on the operating conditions (here: resonance ratio N), tooth deviations (from manufacturing and profile modification, here: factor K) and the variation of the meshing stiffness (Eq. 3).

$$K_V = N \cdot K + 1 \tag{3}$$

For a smaller gear with constant main geometry, the resonance ratio N is proportional to the module m_n and the revolution speed n_1 (Eq. 4). Hence the subcritical operating range is becoming wider with decreasing gear size. Thus for small-sized gears, higher speeds are acceptable.

$$N \sim m_n \cdot n_1 \tag{4}$$

The factor K in Equation 3 is a function of the ratio of manufacturing tolerances to load-induced deviations. While the load-caused deviations are proportional to the module, the manufacturing tolerances remain the same due to technological limitations. This may lead to a worse dynamic behavior and increased stresses.

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The manufacturing tolerances also influence the load distribution, considered with the factors $K_{F\alpha}$, $K_{H\alpha}$, $K_{F\beta}$ and $K_{H\beta}$. Furthermore, shaft deviations, bearing displacements and housing deformations, have an increasing effect on the load distribution with decreasing gear size. Hence uneven load distribution is a major problem of small-sized gears. Therefore extraordinary high manufacturing quality, adequate flank modifications and the application of bearings with reduced clearance are recommended for small-sized gears. For evaluating gear quality, the used standardization has to be considered. According to DIN 3961 (Ref. 1) and DIN 3962 (Ref. 2), the limiting values for gear quality are the same for gears equal or less than $m_n = 1$ mm. In most cases no extrapolation is used. It is recommended to use ISO 1328 (Ref. 8) to determine the quality of small-sized gears. This standard gives different limiting values down to $m_n = 0.5$ mm. However, there is no linear relation between the limiting values and gear size, so small-sized gears with quality 5 may have more deviations compared to the gear size than larger gears of the same quality.

2.2 Influence of gear size on the permissible tooth root bending stress. The permissible tooth root bending stress σ_{FP} is calculated as:

$$\sigma_{FP} = \frac{\sigma_{Flim} \cdot Y_{ST} \cdot Y_{NT}}{S_{Fmin}} \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X \tag{5}$$

Thereby σ_{Flim} is the allowable bending stress number of the reference test gears with a module of $m_n = 5$ mm for 1% failure probability. $Y_{\delta relT}$ is the relative notch sensitivity factor that compares the notch sensitivity of the actual gear with that of the reference gear. The size factor Y_X considers the influence of gear size on the tooth root strength. It depends on the material, heat treatment and module. For module sizes $m_n \leq 5$ mm, the size factor $Y_X = 1$, according to DIN 3990 and ISO 6336.

However, according to the general mechanics of materials local notches like the tooth root fillet lead to locally increased stresses. But the stress peaks are reduced by the support of the surrounded material through plastic micro-deformations. This leads to increased gear strength. The range of support n_χ can be calculated:

$$n_\chi = 1 + \sqrt{\rho' \cdot \chi^*} \tag{6}$$

It is depending on the relative stress gradient χ^*:

$$\chi^* = \frac{d\sigma}{dy} \sim \frac{1}{m_n} \tag{7}$$

The slip-layer thickness ρ' is a function of the material (Ref. 7), but there are further results of research projects that determine higher values (Ref. 19).

The supporting properties of the material are also depending on the notch parameter q_S. For $q_S = 2.5$ the calculated values according to the different calculation methods are shown (Fig. 1).

While DIN 3990 shows no effect for decreasing the gear size from module 5 mm to 0.5 mm, the other calculation methods promise increased gear strength of 20% – 55%, depending on the assumed slip-layer thickness.

The relative surface factor Y_{RrelT} is taking into account the influence of surface roughness at the tooth root fillet on the local stresses. Depending on whether the tooth fillet is ground, the roughness is comparable to that of the gear flank. However, the resulting roughness of common grinding processes is limited to values of approx. $Ra = 0.2$ µm. Hence the influence of roughness is increasing with decreasing gear size. This may lead to an additional stress increase, caused by roughness notches in the tooth fillet.

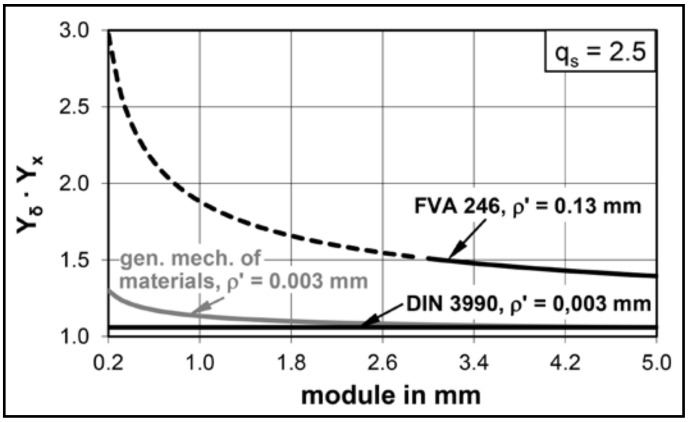

Figure 1 **Supporting properties of the material depending on gear size according to DIN 3990 (Ref. 4), general mechanics of materials (Ref. 17), and FVA 246 (Ref.19).**

2.3 Influence of gear size on the permissible contact stress. The permissible contact stress is calculated:

$$\sigma_{HP} = \frac{\sigma_{Hlim} \cdot Z_{NT}}{S_{Hmin}} \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_W \cdot Z_X \tag{8}$$

The allowable contact stress σ_{Hlim} of the reference test gears is explained in DIN 3990 and ISO 6336. It is dependent upon the material and heat treatment. The size factor Z_X considers the influence of gear size on the permissible contact strength. The major parameters on this factor are the material quality and heat treatment (statistical influence), as well as the radius of flank curvature and the case depth (supporting effect). The size factor $Z_X = 1$ for $m_n \leq 10$ mm (Ref. 3) respectively, $Z_X = 1$ for all modules (Ref. 6). But with decreasing gear size the stress gradient at the gear flank is increasing as well. On one hand, the depth of maximum shear stress beneath the surface is proportional to the relative radius of curvature. Hence with decreasing gear size the depth is decreasing. On the other hand, theoretical studies show that with decreasing gear size the friction coefficient increases. This leads to higher shear and thermal loads at the flank surface. Therefore the shear stresses at the surface, as well as near to the surface, are increased as well. The main influence on the pitting resistance is the first one because of the depth of pitting cracks; it leads to increased permissible contact stresses for small-sized gears.

The complex loading conditions at the tooth flank can only be calculated with adequate software, e.g. — *ROSLCORHR* (Ref. 13). This was done for the reference test gears as well as for gears with the same geometry — but with a module of 0.5 mm (Ref. 12). Thus the relative radius of curvature at the pitch point was 10 mm compared to 1 mm. The operating conditions were assumed as constant. The results show that the supporting properties depend on the local point at the gear flank. For the calculated gears with module 0.5 mm, an increase of 10% – 20% can be derived.

The factors Z_L, Z_v and Z_R consider the influences of lubricating conditions (oil viscosity, circumferential velocity, surface roughness) on the permissible contact stress. The roughness factor Z_R depends on the flank roughness and the center distance (Ref. 3), respectively the relative radius of curvature (Ref. 6). Because flank roughness is almost independent of gear size, this factor worsens with decreasing gear size. Furthermore, the circumferential velocity is decreased for small-sized gears, even if they are operated at higher rotational speeds. Hence the velocity factor Z_v is also decreased. These effects reduce the pitting resistance of small gears.

The worse lubricating conditions at small-sized gears lead to a higher risk for micropitting and wear, which should be considered.

Table 1 Geometry of the test gears

Test Aim		Tooth Bending Strength				Pitting Load Capacity			
Test Rig		Back-To-Back		Pulsator Test Rig		Back-To-Back			
Designation		BR06K	BR06U	FL06P	FL045(U)P	FL06K	FL06U	FL045K	FL045U
Module	m_n / mm	0.60		0.60	0.45	0.60		0.45	
Centre Distance	a / mm	35.28		-		15.00		11.25	
Tip Diameter	d_a / mm	36.38 / 36.45		19.38	14.54	13.08 / 19.38		9.81 / 14.54	
Face Width	b / mm	9.00		9.00	6.75	10.00 / 9.00		7.75 / 6.75	
Number of Teeth	z_1 / z_2	57 / 58		29		19 / 29			
Profile Shift Coefficient	x_1 / x_2	0.903 / 0.500		0.500		0.450 / 0.689			
Helix Angle	β / °	0							
Profile Crowning	C_h / µm	5 / 5	-	-		5 / 5	-	3.5 / 3.5	-
Lengthwise Crowning	C_b / µm	1 / 0	-	-		3 / 0	-	3 / 0	-
Helix Angle Correction	C_p / µm	-	-	-		-13 / 0	-	1 / 0	-

Table 2 Operating conditions

Test Aim	Tooth Bending Strength	Pitting Load Capacity	
Test Gears	BR06	FL06	FL045
Module	0.6	0.6	0.45
Rotational Speed Of Pinion	2000 rpm	7000 rpm	9000 rpm
Tangential Speed At Pitch Point	3.66 m/s	4.18 m/s	4.03 m/s
Lubricant	mineral oil ISO VG 100 +EP-additive		
Lubricating Conditions	regulated circulationg dip lubrication @60°C (140°F)		
Limit Load Cycles	10 million	50 million	
Failure Criteria	tooth root breakage	pitting area > 4 % of tooth flank	

Experimental Investigations of Gear Size Influence on Load-Carrying Capacity

In a first research project (Refs. 9, 14 – 15) an increased load-carrying capacity of involute gears within a module range of 0.3 – 1.0 mm had already been basically confirmed, although there were some difficulties with the heat treatment of the test gears. And so a second project (Ref. 11) was started to determine reliable values for this potential increase.

Test gears and operating conditions. For the determination of the load-carrying capacity two special gear geometries were designed. The gear ratio for tooth root breakage testing was 57/58, for pitting testing 19/29. The tests were performed with modified and unmodified test gears of module $m_n = 0.45$ mm and $m_n = 0.6$ mm on a FZG small-gear, back-to-back test rig. The detailed gear geometry is presented in Table 1. In addition, pulsator tests were performed with the wheels of the pitting gear design, partly with unground tooth fillet.

All test gears were made of 16MnCr5. After gear hobbing they were case-carburized to 700–750 HV surface hardness and 0.1 – 0.2 mm case hardening depth (limit hardness 550 HV). After heat treatment the gears were ground. The gear quality acc. to DIN 3962 was Q ≤ 5. The tooth root fillets were ground due to lack of small enough protuberance hobbing tools. Only the gears of FL045UP had unground fillets. They were only hobbed and heat treated. The test conditions for the different tests at the FZG small gear test rig are shown (Table 2). Prior to the tests all gearsets were run a two-stage running-in procedure.

The pulsator tests were run at an electromagnetic resonance pulsator, as it is described, e.g., in (Ref. 21). Each gear was clamped over 6 teeth. Hence the force was applied near the outer point of single tooth contact. The test frequency was in a range of 50 to 60 Hz. The pulsator tests were run until tooth root breakage occurred or a maximum of 6 million load cycles.

Evaluation of Test Results

Tooth root bending strength. For the analysis of running tests the equations of DIN 3990 (Ref. 4) or ISO 6336 (Ref. 7) are typically used to determine the tooth root stress, depending on the applied torque. But the special test gear geometry leads to transverse contact ratios of two or even higher for the loaded contact. Hence the tooth root stresses have been calculated using the FZG-FVA software *RIKOR* I (Ref. 22). Furthermore, the actual tooth fillet geometry was considered. The tooth root stress for the endurable torque was multiplied with the factor of 0.86 (Ref. 20) to convert the stress for 50% failure probability to 1% failure probability $\sigma_{Flim,exp}$. This stress was compared to the allowable bending stress σ_{Flim} of the reference test gears with 5 mm module.

For evaluation of the pulsator tests, pulsator stress was calculated depending on the pulsator force as:

$$\sigma_{F,pulsator} = K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha} \cdot \frac{F_{PN} \cdot \cos(\alpha_n)}{b \cdot m_n} \cdot Y_F \cdot Y_S \qquad (9)$$

Of course the real tooth root fillet geometry was considered for the calculation of the form factor Y_F and the stress correction factor Y_S. The pulsator stress for the endurable force was multiplied with the factor 0.90 to convert the pulsator result to one of a running test (Ref. 20). This stress was additionally converted to a failure probability of 1%.

Evaluation of the pitting load capacity. For the evaluation of the pitting load capacity, the endurable contact stress was calculated according to Equations 2 and 8. This stress was multiplied with the factor 0.92 (Ref. 20) to convert the result for 50% failure probability to 1% failure probability $\sigma_{Hlim,exp}$. Subsequently the stress was compared to the allowable contact stress of the standard reference test gears.

Test results. Figures 2 and 3 show typical examples of tooth root breakage at the test gears after running tests. The damage is comparable to those at gears with a module of 5 mm. The cracks start at the surface near the 30° tangent at the tooth root fillet. As expected for a gearset without flank modifications, the crack initiation at BR06U is near the face side of the pinion. For the gearsets with adequate flank modifications the crack starts near the middle of the face width. The tooth root breakages at the pulsator test rig are comparable to those of BR06K.

The pitting damages at the pinions with module 0.45 mm and 0.6 mm are also comparable to those of gears with module 5 mm. The pitting occurred preferentially in the flank area with negative specific sliding. The test gears with adequate flank modifications show uniform pitting damage along the whole face width (Fig. 4). In contrast, the pitting at the gears without modifications occurred near one face side. Therefore, the adjustable bearing plates of the test rig were insufficient to compensate for the load-caused deformations and deflections. Furthermore, micropitting was observed on all test gears (Fig. 5).

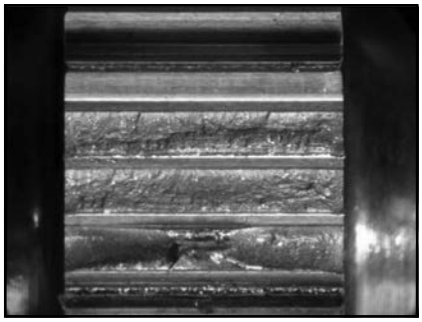

Figure 2 Tooth root breakage at the BR06K pinion (T_1 = 53 Nm, 5.01·10^6 LC_{Pi}).

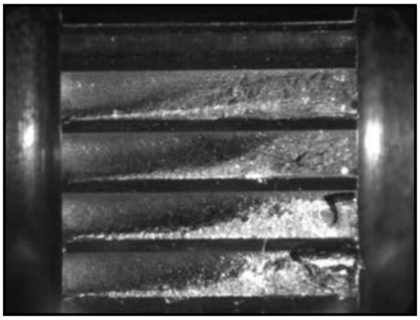

Figure 3 Tooth root breakage at the BR06U pinion (T_1 = 60 Nm, 474849 LC_{Pi}).

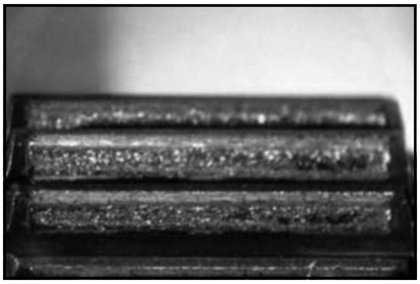

Figure 4 Pitting at the FL06K pinion (T_1= 8.4 Nm, 26.0·10^6 LC_{Pi}).

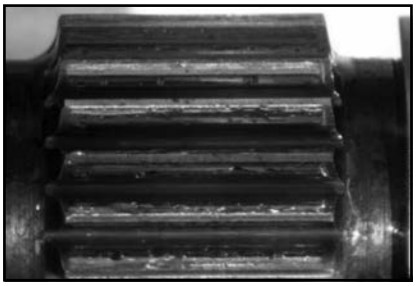

Figure 5 Micropitting at the FL06U pinion (T_1 = 10 Nm, 27.0·10^6 LC_{Pi}).

Figure 6 shows major results for the tooth bending strength. The presented S-N curves of the pulsator tests are in good accordance with the results of the additional running tests with modified and unmodified test gears (Fig. 8) when considering the load-caused, real transverse ratio.

For the pitting load capacity, exemplary S-N curves of the test gears with module 0.6 mm are shown (Fig. 7). The results are comparable and correspond well with those for module 0.45 mm. The allowable stress numbers σ_{Hlim} are approximately equal for all pitting test variants and on a high level. The determined allowable stress numbers for tooth root breakage σ_{Flim} and for pitting load capacity σ_{Hlim} of all test variants are shown (Fig. 8). Additionally, the stress numbers acc. to DIN 3990 (Ref. 5) (material quality MQ) are presented for comparison.

Proposals for Extended Size Factors

As seen in Figure 8, the experimentally determined allowable stress numbers of the gears with modules of m_n = 0.45 and 0.6 mm are significantly higher than those given for standard reference gears. The values are comparable to those which were expected from the theoretical investigations (see again sections *Influence of gear size on the permissible tooth root bending stress,* and *Influence of gear size on the permissible contact stress*). Hence, there is a need for new, extended size factors. These factors should consider the higher strength of small sized gears.

For the derivation of the new size factors, additional experimental data of further research projects were analyzed (Refs. 9, 16, 21 and 23). In these projects case-carburized spur gears with different module sizes were tested. The proposals for the

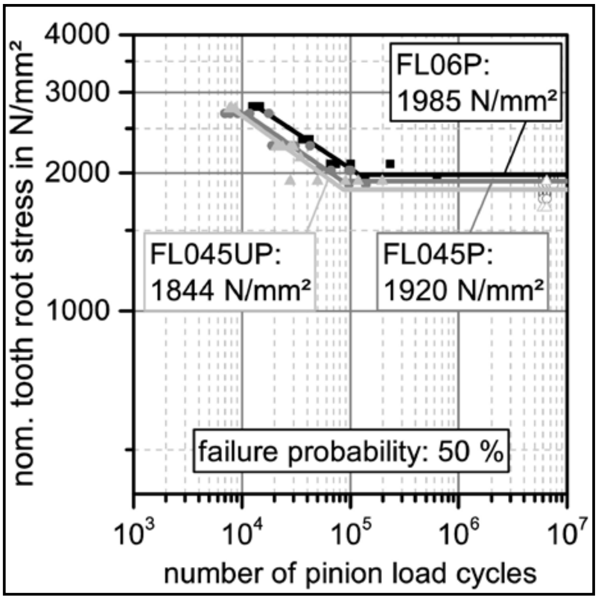

Figure 6 S-N curves for tooth bending strength pulsator tests ***only***.

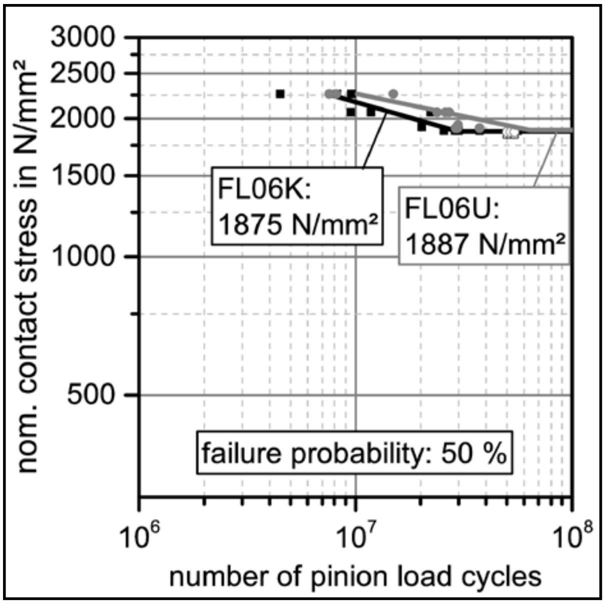

Figure 7 S-N curves for pitting load capacity of FL06K and FL06U.

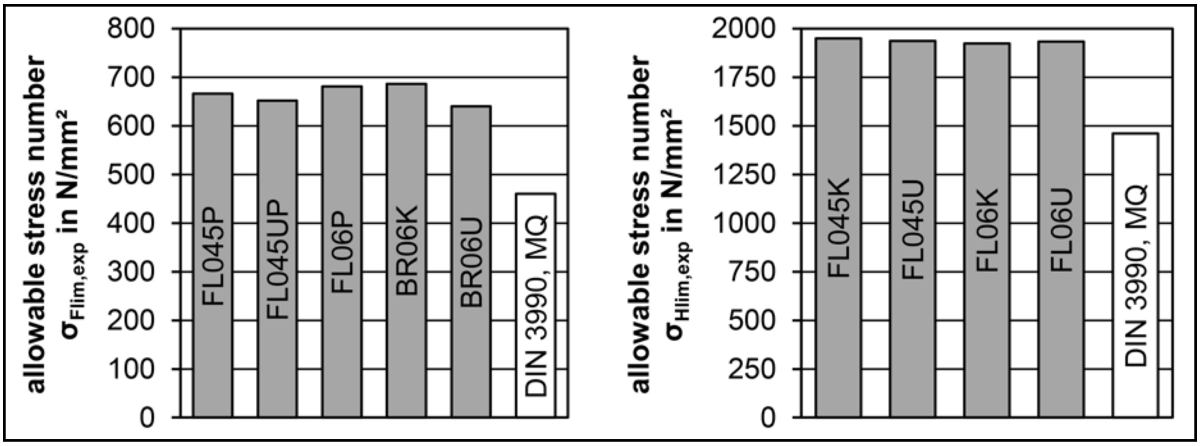

Figure 8 **Experimentally determined allowable stress numbers for tooth root breakage (left) and pitting (right) in comparison to reference gears (Ref. 5).**

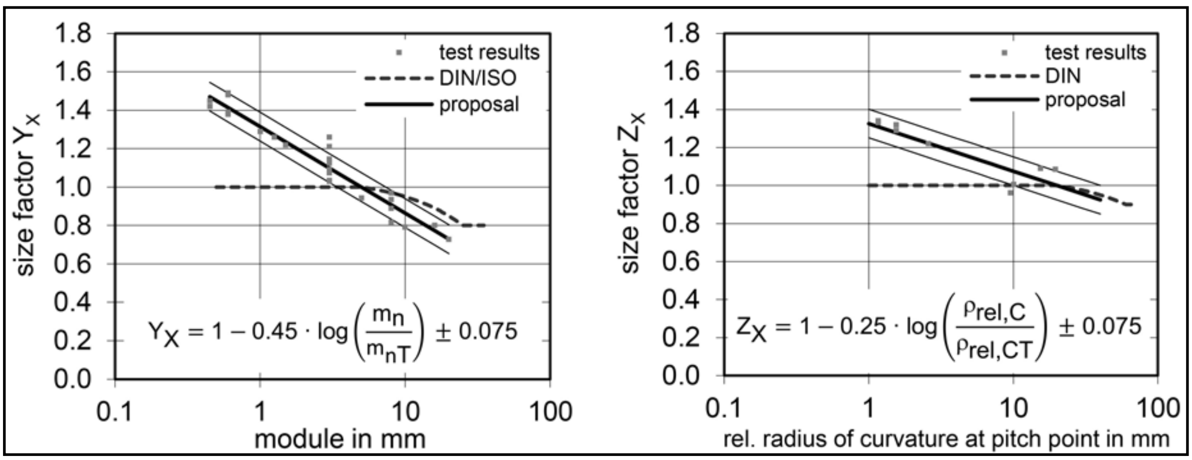

Figure 9 **Proposed size factors for tooth bending strength (left) and pitting load capacity (right) for case-hardened steel 16MnCr5 based on test results.**

new extended size factors Y_X and Z_X, as well as the existing ones of DIN 3990/ISO 6336, are presented (Fig. 9). The test results of different research work are included, too.

For tooth root breakage, the relevant size parameter is the module (see again *Influence of gear size on the permissible tooth root bending stress*). Hence the calculation method for Y_X is depending on the module of the actual gear m_n, compared to the module of the reference test gears $m_{nT} = 5$ mm. For pitting resistance, the relevant size parameter is the relative radius of curvature at the pitch point C. Therefore the calculation of the size factor Z_X considers the relative radius of curvature of the actual gear $\rho_{rel,C}$ as well as the one of the standard reference gears $\rho_{rel,CT} = 10$ mm.

Since reliable manufacturing and heat treatment of high-quality gears with module sizes $m_n < 0.45$ mm is extremely demanding, the validity range of the proposals should be limited to module sizes $m_n \geq 0.45$ mm. For high gear quality and proper heat treatment, the upper range of tolerance can be used. If there are uncertainties, it is recommended to use the size factors according to the lower range of tolerance.

Summary

Tooth bending strength and pitting load capacity increase with decreasing gear size. Since no comprehensive verification of the carrying capacity of fine module gears has been available thus far, common calculation methods do not state a positive size effect for gears with module sizes smaller than 5 mm. Increased tooth bending strength and pitting load capacity are proven theoretically and experimentally in this work. On this basis proposals for extended size factors for calculation acc. to DIN 3990 / ISO 6336 are given.

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Maria Hergesell studied mechanical engineering at Technical University of Dresden, receiving in 2004 her Diplom-Ingenieur. From 2004-2011 she worked as a research associate at the Gear Research Centre (FZG), Technical University of Munich. She received her PhD in 2013 for her work, "Formation of Micropitting and Pitting on Case-Carburized Gears of Medium and Small size. For one year (2011-2012) she worked for Siemens AG and was responsible for the optimization of business processes. Hergesell in 2012 was named head of technology management for Wittenstein GmbH.

Dr.-Ing. Thomas Tobie studied mechanical engineering at the Technical University of Munich (TUM), Germany. Today he is head of the Load Carrying Capacity of Cylindrical Gears department at the Gear Research Centre (FZG), where he specializes in gear materials, heat treatment, gear lubricants and gear load carrying capacity research. Concurrently, Tobie brings to that work a particular focus on all relevant gear failure modes such as tooth root breakage, pitting, micropitting and wear, as well as sub-surface-initiated fatigue failures.

Prof. Dr.-Ing. K. Stahl studied mechanical engineering at the Technische Universitat Muenchen and also served as a research associate at the University's Gear Research Centre (FZG). In 2001 he received his PhD degree (Dr.-Ing.) in mechanical engineering, and that same year started as gear development engineer at the BMW group in Dingolfing, subsequently becoming head of the Prototyping, Gear Technology & Methods group in 2003. In 2006 he moved to the BMW/MINI plant in Oxford, UK, and the next year (2007) became department leader — Validation Driving Dynamics and Powertrain. In 2009 Stahl returned to Munich as manager for Pre-Development and Innovation Management within BMW Driving Dynamics and Powertrain in Munich.In 2011, he became both a full professor at the Institute for Machine Elements and head of the Gear Research Centre. The FZG employs about 80 associates — 50 of them PhD candidates and more than 200 students. Organized in 5 departments, Prof. Stahl's research focuses on experimental and analytical investigations of endurance, tribology, NVH, materials and fatigue life analysis. Components in the focus are cylindrical, bevel, hypoid and worm gears; clutches; synchronizers; rolling-element bearings; and drive systems. Professor Stahl is editor of 6 scientific journals, a member of scientific committees of 7 national and international conferences, holds the VDI ring of honor and is a board member of the board of 2 scientific associations and a member of 4 ISO working groups. He has published more than 100 scientific papers and presentations.