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ORIGINALARBEITEN/ORIGINALS

Investigation on the flank surface durability of gears with increased pressure angle

Christian Weber[1] · Thomas Tobie[1] · Karsten Stahl[1]

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Abstract Within a funded research project (reference number 0325244C, BMWi – Federal Ministry for Economic Affairs and Energy of Germany), the load-carrying capacity of alternative involute gears was investigated. To find qualified variations for use in gear drives, increased pressure angles ($\alpha_n > 20°$) and/or asymmetric tooth shapes ($\alpha_n < 20°$ and $\alpha_n > 20°$) have been examined experimentally. Optimization goals were the power to weight ratio of the transmission and the power density in the drive train.

In this report gears with reference tooth shape ($\alpha_n = 20°$) as well as gears with modified tooth shape ($\alpha_n = 28°$) are discussed with focus on the load carrying capacity of the tooth flank with special regard to the damage mechanisms pitting, micro-pitting and scuffing. The results of experimental and test accompanying studies are shown. They allow a direct comparison between standard and special gears as well as a classification in context of the actual state of knowledge.

1 Introduction and objective

Mechanical engineering uses almost exclusively symmetric involute gears with pressure angles $\alpha_n \approx 20°$ according to the basic rack as specified in ISO 53 [7] in today's power transmitting gear drives. Besides the economic advantages of the manufacturing process using straight-flank hobbing tools the broad application of this toothing is due to its uniform transmission of motion. This even applies to shaft center distance deviations from the setpoint.

For the design of symmetric involute toothings with $\alpha_n \approx 20°$ comprehensive calculation tools and methods such as ISO 6336 [6] are available. These calculation methods are based on extensive theoretical and experimental studies on appropriate standard gears and are well-proven for years. Nevertheless, these calculation methods do not claim to universal validity. The calculation method according to ISO 6336 [6] for example is limited to a pressure angle of $\alpha_n \leq 25°$ according to the basic rack specified in ISO 53 [7].

However, for a reliable application of special involute gears with an increased pressure angle in industrially running transmissions a comprehensive and confirmed identification and verification of the load-carrying capacities is needed for such toothings. This also requires an experimental verification of existing tooth flank load-carrying capacity calculation methods.

2 Investigated failure mechanisms

The herein described experimental investigations were carried out in order to determine the flank load carrying capacity against the most common gear failures, i. e. pitting, micro-pitting, and scuffing. For all tests regarding the listed types of flank failures, selective tests have been performed. The operating conditions were adjusted to cause the desired type of flank failures, if necessary.

Furthermore, investigations concerning the tooth root bending strength of gears with increased pressure angles as well as of gears with asymmetric tooth shape were done [14]. The aim of these investigations was amongst others the validation of the advanced calculation method for tooth root bending strength according to FZG/Fröh [13]. The results of these investigations are no object of this paper.

✉ Christian Weber
c.weber@fzg.mw.tum.de

[1] Gear Research Centre (FZG), Technical University of Munich, Garching, Germany

In the following the investigated flank failure mechanisms are described recapitulatory according to DIN 3979 [2], ISO 10825 [4], [8], [9] and [12].

2.1 Pitting

Pitting is a fatigue damage occurring on the tooth flank caused by the periodic rolling/sliding load in the tooth contact. Usually, pittings are the result of surface or subsurface fatigue cracks caused by local metal-to-metal contact at the roughness asperities. In addition to Hertzian stress, oil viscosity and temperature, specific sliding, surface roughness and circumferential speed have a significant effect on the pitting resistance. Pitting damages on case carburized gears are characterized by a shell-shaped material fracture in the area of the active flank, typically located in the area of negative specific sliding. Usually, this leads to increased noise and additional dynamic loads.

2.2 Micro-pitting

Micro-pitting is a tooth flank fatigue phenomenom that can be observed often on case-carburized gears with a high surface hardness. It is strongly influenced by the conditions of the tribological system consisting of the tooth flank surface and the lubricant and usually starts in flank areas with high negative specific sliding [4]. Being a surface damage resulted from numerous surface cracks generation, it causes profile deviations of wear type on the active flanks as well as dynamical additional forces and gear noise increase. The micro-pits can coalesce to produce a continuous fractured surface which appears as a dull, matte surface during unmagnified visual inspection [12].

2.3 Scuffing

This damage is based on a breakdown of the lubricating film due to high contact stresses and temperatures [11]. It is characterized by a local welding of the corresponding tooth flanks with subsequent damage/scoring in this flank region. This damage results in a roughening of the flank surface and profile form changes leading to increased dynamics. Scuffing is a spontaneous damage mainly influenced by the lubricant, the circumferential speed, the contact stress, the mass temperature and the lubricant temperature.

3 Test program and test method

3.1 Test rig

The experimental investigations were carried out on FZG back-to-back gear test rigs [1] with a center distance of a = 91.5 mm (gear size m_n = 5 mm) as well as on a FZG back-to-back gear test rig with a center distance of a = 200 mm (gear size m_n = 8 mm), as shown schematically in Fig. 1.

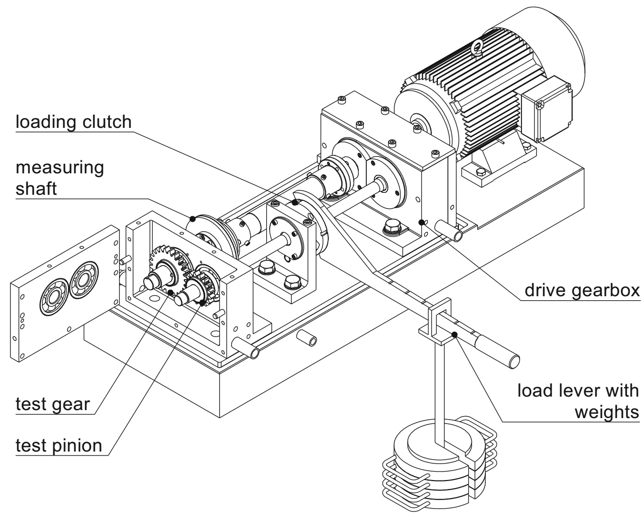

Fig. 1 FZG back-to-back gear test rig with center distance a = 91.5 mm [1]

The FZG back-to-back gear test rig utilizes a recirculating power loop principle, also known as a foursquare configuration, in order to provide a fixed torque to a pair of test gears. The test rigs are driven by a three-phase asynchronous engine with a constant speed. Test pinion and test gear are mounted on two parallel shafts which are connected to a drive gear stage with the same gear ratio. The shaft of the test pinion consists of two separate parts which are connected by a load clutch. A defined static torque is applied by twisting the load clutch using defined weights on the load lever or by twisting the load clutch with a bracing device. The torque can be controlled indirectly at the torque measuring clutch as a twist of the torsion shaft. The gears were loaded in such way that the pinion drove the gear.

3.2 Test gears and test conditions

The main nominal geometry of the test gears for investigations on the pitting load-carrying capacity can be seen in Table 1.

In addition to the investigations on the reference test variant with a pressure angle α_n = 20° a modified reference test variant with the same pressure angle α_n = 20° has been examined. This modified reference variant has been designed with the same transverse contact ratio than the test variant with increased pressure angle α_n = 28°.

The main nominal geometry of the test gears for investigations on micro-pitting and scuffing are shown in Table 2.

All gears of one design size were made of the same batch of material 18CrNiMo7-6 and were case-hardened and conventionally ground (only flank area) after heat treatment.

Table 1 Main nominal test gear geometry (pitting)

Denomination	Symbol	Unit	Reference	Modified reference	Increased pressure angle
Center distance	a	mm	200		
Module	m_n	mm	8		
Number of teeth	z_1/z_2	–	24/25		
Face width	b	mm	18		
Pressure angle	α_n	°	20/20	20/20	28/28
Helix angle	ß	°	0		
Transverse contact ratio	ε_α	–	1.45	1.39	1.39
Tip diameter	d_{a1}/d_{a2}	mm	211.8/219.4	211.0/218.7	212.2/220.7
Tip relief	C_{a1}/C_{a2}	µm	50/50	50/50	45/45
Case hardening depth	CHD_{550}	mm	1.6–1.7		
Single pair tooth contact factor	Z_B/Z_D	–	1.017/1.012	1.021/1.015	1.011/1.007
Conversion factor $T \rightarrow \sigma_{H0}$	$X_{T\rightarrow\sigma H0}$	–	31.44	31.82	28.93

Table 2 Main nominal test gear geometry (micro-pitting, scuffing)

Denomination	Symbol	Unit	Reference	Increased pressure angle
Center distance	a	mm	91.5	
Module	m_n	mm	5	
Number of teeth	z_1/z_2	–	17/18	
Face width	b	mm	14	
Pressure angle	α_n	°	20/20	28/28
Helix angle	ß	°	0	
Transverse contact ratio	ε_α	–	1.21	1.21
Tip diameter	d_{a1}/d_{a2}	mm	98.7/102.6	99.6/103.5
Tip relief	C_{a1}/C_{a2}	µm	25/25	25/25
Case hardening depth	CHD_{550}	mm	0.75–0.85	
Single pair tooth contact factor	Z_B/Z_D	–	1.027/1.027	1.027/1.023
Conversion factor $T \rightarrow \sigma_{H0}$	$X_{T\rightarrow\sigma H0}$	–	75.70	72.82

Table 3 Running-in conditions

Failure mechanism	Number of load cycles	Lubricant temperature	Nominal contact stress σ_{H0}	Rotational speed at pinion
Pitting	20,000	40 °C	1100 N/mm^2	1500
	180,000	90 °C	1200 N/mm^2	1500
Micro-pitting	130,000	90 °C	510 N/mm^2	1500

Table 4 Test conditions

Failure mechanism	Lubricant	Type of lubrication	Lubricant temperature
Pitting	FVA3 + 4% Anglamol 99	Oil injection	60 °C
Micro-pitting	FVA3 + 4% Anglamol 99	Oil injection	90 °C
Scuffing	FVA2	Splash lubrication	90 °C

Concerning the investigations on the pitting load-carrying capacity as well as the micro-pitting load-carrying capacity a running-in is performed for each set of test gears before starting the test run on the FZG back-to-back gear test rig. The conditions of this running-in depending on the investigated failure mechanism are shown in Table 3.

All tests were performed with lubricants from the FVA reference oil catalogue [10]. For most of the tests, the reference oil FVA 3 (ISO VG 100) with an additive content of 4% Anglamol 99 by volume has been used. The lubricant, the type of lubrication as well as the lubricant temperature of the test runs of the three different investigations on the flank surface durability are shown in Table 4.

For all of the test variants an average number of between two and four test runs were performed. The test results of the gears with increased pressure angle $\alpha_n = 28°$ were evaluated in direct comparison with the results of the reference gears with pressure angle $\alpha_n = 20°$.

4 Results

4.1 Pitting load-carrying capacity

The investigations on the pitting load-carrying capacity were performed by a constant speed of 2200 rpm at the driving test pinion. The maximum running time of each test run was $50 \cdot 10^6$ load cycles on pinion. The flanks of the pinion and of the wheel were inspected visually after certain test intervals for damage. The test runs were stopped when more than 4% of the working tooth flank area of one single tooth or when 0.5% of the total working area of all teeth were damaged by pitting. If no pitting damage occurred within the maximum running time or if the failure criterion mentioned above was not met within this load cycle number, the test runs were stopped and rated as a pass. All tests were performed in the region of endurance life. The determined transmitted torques, relating to the load level on which no pitting damage occurred on the reference variant are shown in Fig. 2.The load level on which no pitting damage occurred on the reference variant corresponds with the state of the art for case-hardened gears. The conversion factors $X_{T \rightarrow \sigma H0}$ and the single pair tooth contact factors Z_B and Z_D are given in Table 1. The two test variants with pressure angle $\alpha_n = 20°$ show a comparable transmittable torque. The pitting load-carrying capacity of the test variant with pressure angle $\alpha_n = 28°$ is about 20% higher than for the two variants with standard pressure angle. This confirms a higher transmittable torque for gears with increased pressure angle.

For evaluating the test results, the pitting load-carrying capacity is calculated according to ISO 6336 [6]. Therefore, the nominal contact stress at the pitch point σ_{H0} is calculated. The determined nominal contact stresses, relating to the load level on which no pitting damage occurred on the reference variant are shown in Fig. 3. All three test variants show comparable transmittable nominal contact stresses. These results correspond with the expectations as all test gears were made of the same material in the same production batch. Therefore all tested sets of gears should fail at the same nominal contact stress. Thus, it can be assumed that the increased pressure angle of $\alpha_n = 28°$ is considered correctly by the calculation method according to ISO 6336 [6]. It should be noticed that the test variant with increased pressure angle shows a lower scattering regarding the load cycles than the test variants with pressure angle $\alpha_n = 28°$.

Exemplary photographs of pinion tooth flanks after end of the tests are shown in Fig. 4. The photographs show an exemplary tooth flank without any pitting damage (α_n = 20°), with a typically shell-shaped pitting damage (α_n = 28°) and an advanced pitting damage which grew fast between two inspection cycles (α_n = 20°). The pitting damages shown in Fig. 4 are comparable for all test variants and are all located in the tooth flank area of negative specific sliding. All testes gears show a comparable case hardening depth.

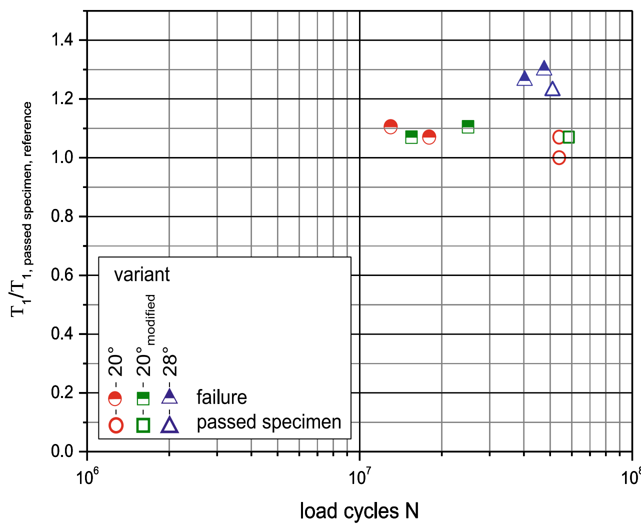

Fig. 2 Transmitted torques, relating to the load level of the reference variant on which no pitting damage occurred

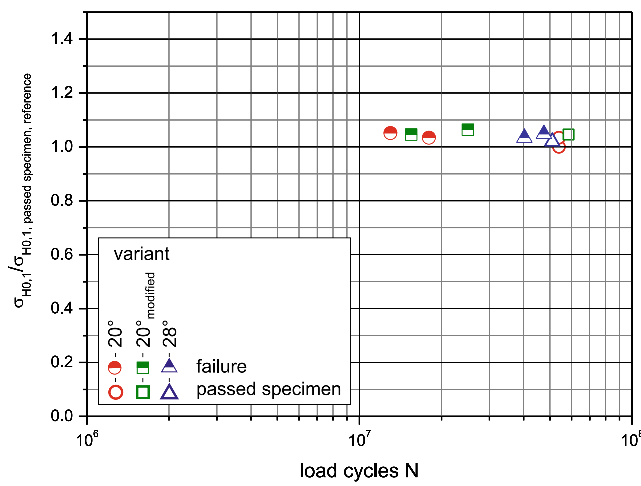

Fig. 3 Calculated nominal contact stresses at the pitch point σH0, relating to the load level of the reference variant on which no pitting damage occurred

4.2 Micro-pitting load capacity

The investigations regarding the micro-pitting load capacity were carried out based on the standard micro-pitting test C-GF/8.3/90 described in the FVA information sheet 54/7 [3]. These tests were performed with the reference oil FVA 3 (ISO VG 100) with an additive content of 4% Anglamol 99 by volume. Two test runs were performed for each test variant.

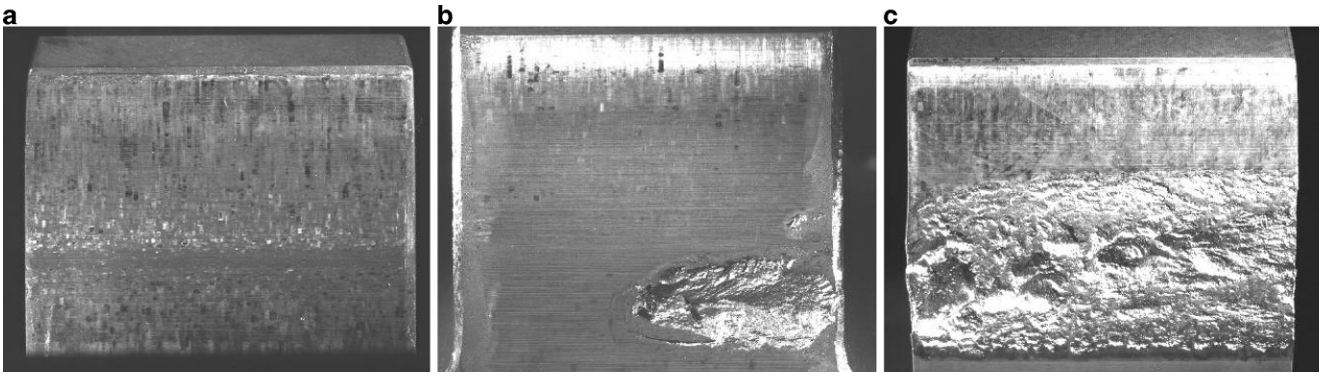

Fig. 4 Exemplary tooth flanks of pinion without any pitting damage (**a** $\alpha_n = 20°$), with a shell-shaped pitting damage (**b** $\alpha_n = 28°$) and an advanced pitting damage (**c** $\alpha_n = 20°$) after test end

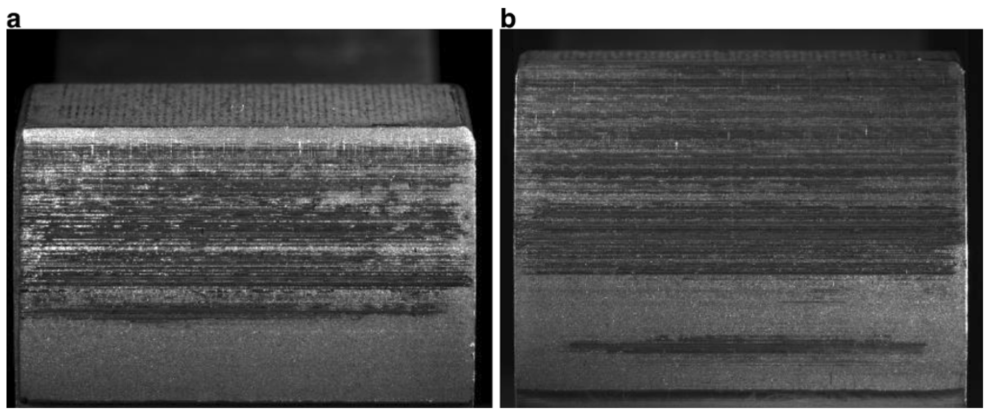

Fig. 5 Examples of tooth flanks of pinion damaged by micro-pitting of the reference test variant with pressure angle $\alpha_n = 20°$ (**a**) and the test variant with increased pressure angle $\alpha_n = 28°$ (**b**) after test end

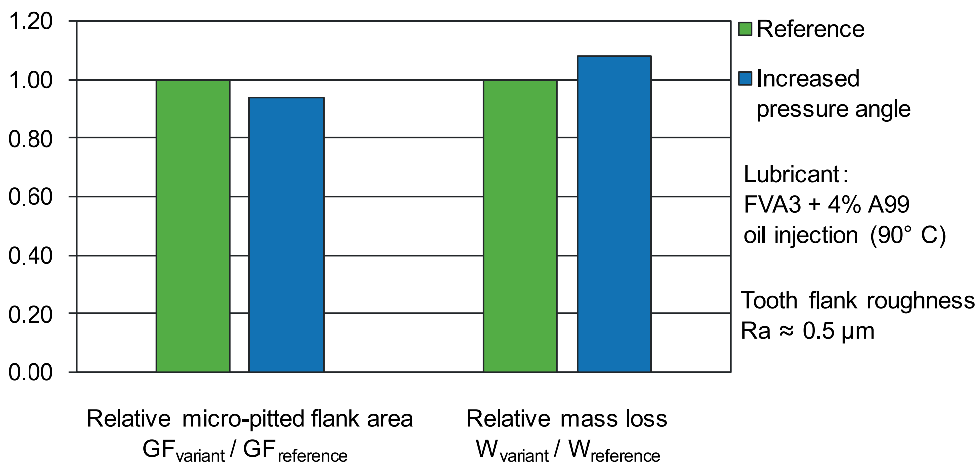

Fig. 6 Relative micro-pitted flank area and mass loss of the reference test variant with pressure angle $\alpha_n = 20°$ and the test variant with increased pressure angle $\alpha_n = 28°$

The torque was adjusted in such a way that both variants have the same nominal contact stresses at the pitch point on each load stage. This means that the test variant with increased pressure angle is loaded with a higher torque than the reference test variant on the same load stage. It should be noted that the wheels of the used gear sets were evaluated because of the lower micro-pitting resistance.

Fig. 5 shows the tooth flanks of the reference variant with pressure angle $\alpha_n = 20°$ and the variant with increased pressure angle $\alpha_n = 28°$ exemplarily. Both photographs show tooth flanks after load stage 10.

Both test runs of the two test variants were evaluated together each. The results of the determined micro-pitted flank area and mass loss of the test gears are shown in Fig. 6. For a better comparison between the reference variant with a pressure angle $\alpha_n = 20°$ and the variant with increased pressure angle $\alpha_n = 28°$, the results are shown relating to the reference variant.

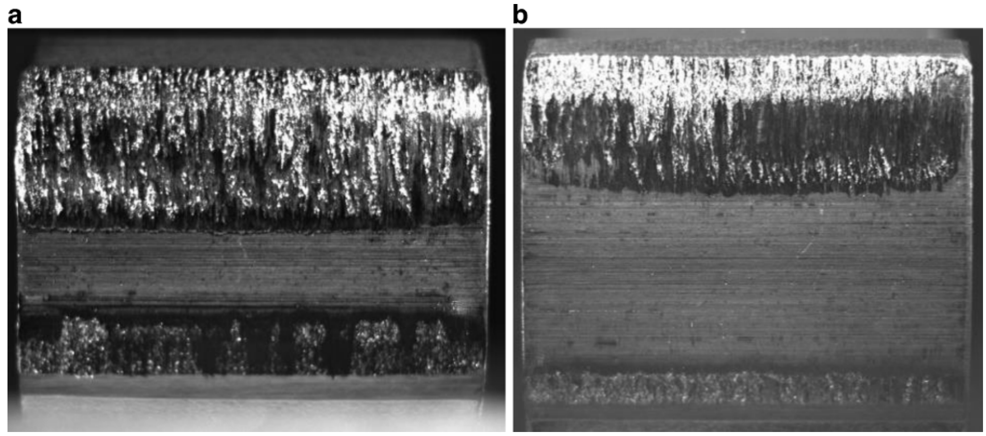

Fig. 7 Examples of tooth flanks of pinion damaged by scuffing of the reference test variant with pressure angle $\alpha_n = 20°$ (**a**) and the test variant with increased pressure angle $\alpha_n = 28°$ (**b**) after test end

The investigations on the micro-pitting load capacity show comparable results for the relative micro-pitted flank area as well as for the relative mass loss of the considered gears. On that basis it may be concluded that both test variants have a similar micro-pitting load capacity when loaded with the same nominal contact stresses at the pitch point. This also means that the transmittable torque until micro-pitting occurs on the tooth flank is higher for the test variant with increased pressure angle. Furthermore it can be assumed that the standard micro-pitting calculation method described in ISO/TR 15411 [5] could also be used for gears with increased pressure angle.

4.3 Scuffing resistance

The scuffing load capacity of the two test variants was tested in the FZG scuffing test acc. to DIN ISO 14635 [1] using the reference oil FVA 2 without additivation. In this standardized test, the load is increased gradually under defined test conditions until a damage of the tooth flank due to scuffing is visible, as shown in Fig. 7. The load stage on which the scuffing occurs is called the scuffing load stage. The torque was adjusted in such a way that both variants have the same nominal contact stresses at the pitch point on each load stage. This means that the test variant with increased pressure angle is loaded with a higher torque than the reference test variant on the same load stage.

For the reference variant with pressure angle $\alpha_n = 20°$ both test runs failed on load stage 6. For the test variant with increased pressure angle $\alpha_n = 28°$ one test run failed on load stage 5 and one test run failed on load stage 6. Based on the failure torque load, for both test variants a comparable load safety factor against scuffing could be calculated. Therefore the test results show that the calculating method for scuffing resistance according to DIN ISO 14635 [1] could also be used for gears with increased pressure angles. In addition, the test results confirm a higher transmittable torque until scuffing occurs for the test variant with increased pressure angle.

5 Conclusion

In this paper the influence of an increased pressure angle on the tooth flank load carrying capacity was discussed. Therefore, the most common gear failures, i. e. pitting, micro-pitting, and scuffing were investigated on gears with a reference tooth shape ($\alpha_n = 20°$) as well as on gears with an increased pressure angle ($\alpha_n = 28°$). The results of these investigations were evaluated based on standardized test procedures and calculation methods. It could be shown that the transmittable torque until flank damages occur can be increased when using gears with an enlarged pressure angle. Besides, the calculation and test methods according to ISO 6336 [6] (pitting), FVA information sheet 54/7 [3]/ISO/TR 15144 [5] (micro-pitting) and DIN ISO 14635 [1] (scuffing) were applied on gears with increased pressure angles. The results show, that these calculation methods could be used for those gears. It should be noticed that all results are based on a small number of tests. For a higher reliability systematically investigations on the influence of an increased pressure angle should be done.

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