Extended Wear Lifetime Calculation for Grease Lubricated Gears in Consideration of Surface Hardness and Roughness

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Abstract. Greases have a variety of advantages when special operating conditions apply for gear lubrication, e. g. improved properties against continuous (or slow speed) wear. Often, gears with different surface hardness are combined, in order to reduce costs during manufacturing and heat treatment. However, this has detrimental effect on the wear lifetime. Consequently, the calculation of the wear service life within the gear design process is essential. One common way of wear lifetime prediction is the calculation method acc. to Plewe that is typically combined with a standardized wear test with case carburized gears. It allows to calculate the wear carrying capacity for different gear-lubricant-systems, but a conversion between gear stages of different gear materials is not possible so far. Based on theoretical studies, influences from different surface hardness and surface roughness values as well as lubricating conditions were investigated on FZG back-to-back test rig. The results prove that hard-soft gear pairings have a higher risk for strong wear compared to case hardened gear stages, but basically follow the same trends and mechanisms. Furthermore, surface roughness of the harder gear has an effect on the wear life time of the softer gear. Based on these results the existing calculation method for the wear lifetime of gears was extended.

1 Introduction

In large gear drives in heavy industry applications like the primary industry or the mining sector, case-hardened pinions can be paired with softer, through-hardened ring gears or internal gears under certain circumstances. If a gear stage has got a high gear ratio, a single tooth of the larger wheel is less frequently in gear mesh. Therefore, it can be designed from a softer material with a lower allowable contact stress compared to the case-hardened pinion. That offers economic advantages to the gear designer, but involves an increased risk of wear during operation, especially on the wheel. The knowledge of the wear lifetime is essential for the engineer already during the design process to guarantee a safe operation of the machine. Furthermore, grease lubrication is often the best solution for gearboxes running under these conditions. It has several advantages compared to oil lubrication that can reduce the effort of sealing or allow low-maintenance lifetime lubrication.

Various test methods (e. g. [1-3]) and calculation approaches (e. g. [4,5]) exist that enable the gear designer to evaluate the expected amount of wear in field during the operation based on characterization of the gear-lubricant-system during the test. However, all standard tests methods are regularly conducted with case-hardened gears only. The wear characteristics between a gear stage of two case-hardened gears of similar surface hardness and a gear stage of a case-hardened pinion and a softer, through-hardened wheel differ a lot. For this reason, a direct transfer of the tests results to the operating conditions in field is not possible so far.

2 Wear in Gear Applications

Continuous or slow speed wear as gear damage describes the material that is continuously worn off of the active gear flank. It usually appears during the whole runtime

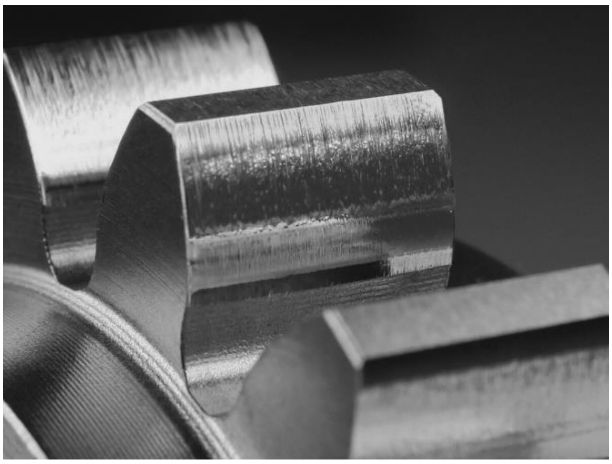

Fig. 1. Typical scratches on the addendum and dedendum of a tooth flank due to slow speed wear [6].

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and is visible as scratches over the whole tooth flank. At the pitch diameter, a wear minimum can be observed (Figure 1), because the sliding speed is close to zero. Almost pure rolling prevails in the contact zone. Accordingly, wear shows characteristic profile form deviations at the addenda and dedenda of the tooth flanks [6].

Different calculation approaches exist to estimate the expected amount of wear in a gear stage in field applications. However, none of these is covered by a standardization (international nor national) so far. This paper focuses on the wear calculation developed by Plewe [4]. It is a common way to predict the wear lifetime for a gear pairing based on test results with a comparable gear-lubricant-system (e. g. [1,2]). The center of the calculation method is the so-called “Plewe diagram” (Figure 2). It relates a wear coefficient c_{lT} with the lubricant film thickness in the tooth contact (calculated acc. to [7]).

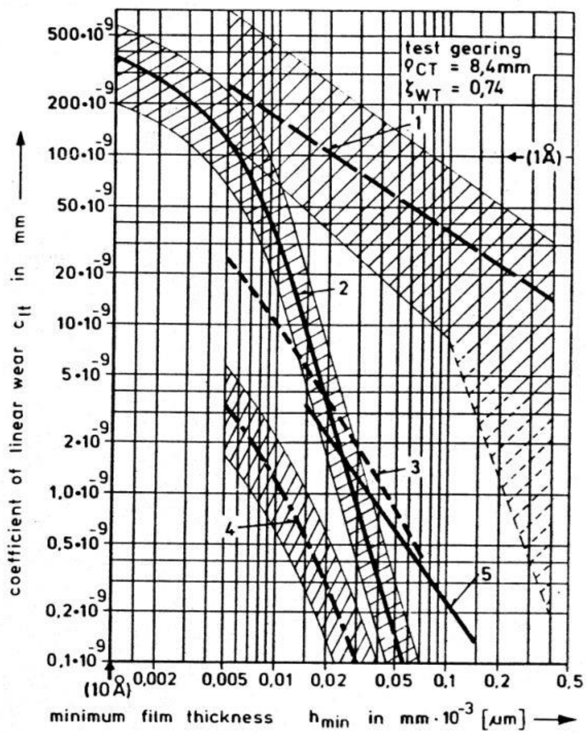

Fig. 2. Wear diagram acc. to Plewe [4] (displayed wear curves are assigned to gear pairings of following heat treatment: For oil lubrication: 1 – case-hardened/through-hardened, 2 – case-hardened/case-hardened, 3 – through-har-dened/through-hardened, 4 – gas-nitrided/gas-nitrided and for grease lubrication: 5 – case-hardened/case-hardened).

The wear coefficient is calculated from the measured loss in mass of the gear during a test run. It accounts several influences like the gear geometry and the density of the gear material to indicate the average linear loss of material, normal to tooth surface, during one revolution of the gear. Assuming continuous wear over time, the calculation allows to compare the results of different test run times and speeds. Plewe discovered the trends of these wear coefficients in dependence of lubricant film thickness as shown in Figure 2. Thereby, he could assign different wear curves to gear pairings of different heat treatments. They have been determined in extensive, experimental investigations with unalloyed mineral oils and fluid greases [4].

2.1 Wear calculation for grease lubricated gears

The suitability of the wear calculation acc. to Plewe [4] has been proven for grease lubricated gear applications in several investigations. Therefore, the calculation of the minimum lubricant film thickness is (in this case) based on the assumption that the grease behaves like its base oil. That keeps the calculation approach easy in terms of handling. The investigations could show sufficiently accurate results compared to Plewe’s wear curve [3,6,8]. An exact calculation of the film thickness is not possible by analytic methods that are appropriate for a standardized calculation approach. Due to the components and properties of a grease, more complex simulations that consider the elastohydrodynamic contact are necessary. Thus, the rheological behavior of the thickener has to be considered, because it takes part in the lubricant film thickness and increases it. Consequently, the measurements have shown differences between the film thickness of a grease, its base oil and the bled oil [9-12]. Furthermore, gear lubricated by greases can enter a state of channeling under certain operating conditions (e. g. at high speeds, with higher thickener concentrations or at low ambient temperatures). In this case, a starved lubrication is predominant. Only a reduced amount of grease is able to refill the tooth gaps of the rotating gears in the grease sump and makes a reduced amount of grease available to the gear mesh [8,13]. This promotes other failure modes that may replace wear as the lifetime limiting factor.

In conclusion, it is possible to calculate the wear carrying capacity of a grease lubricated gear stage respecting the given limitations. If the prevailing lubricant supply mechanism in the gearbox is known, the wear calculation acc. to Plewe [4] is a good compromise between a sufficiently accurate result and good handling.

2.2 Wear calculation for gear pairings of different heat treatment

Moreover, it is to be expected acc. to Plewe [4] that the wear characteristics of a gear pairing containing of a pinion and a wheel of different materials and heat treatment differ from the characteristics of a pairing with gears of same hardness. Whereas the amount of wear is similar between the pinion and wheel of a gear pairing with similar surface hardness (following curve 2, Figure 2), a gear pairing with differences in surface hardness ΔHV over 60 HV leads to a significant increase of wear on the softer gear (following curve 1, Figure 2) [4]. The wear of the harder gear is still within a certain range around curve 2. Nevertheless, the amount of wear increases in general. A transfer of the wear coefficients between these two gear stage types is not possible at the moment.

The investigations of this study will have a closer look on the behavior of grease lubricated gear stages with a combination of case-hardened (hard) and through-hardened (soft) gears. Based on the results, it aims to describe the wear behavior mathematically and thus, enables a transfer of the wear calculations between the two types of gear sets with respect to the influences of surface hardness and grease properties.

3 Experimental Investigations

The experimental investigations have been executed on a modified FZG back-to-back test rig. "It utilizes a recirculating power loop principle, also known as a four-square configuration, to provide a fixed torque (load) to a pair of precision test gears […]. The slave gearbox and the test gearbox are connected through two torsional shafts [, one of which] […] contains a load coupling used to apply the torque through the use of known weights […] hung on the loading arm. The test gearbox contains heating elements to maintain and control the minimum temperature of the lubricant. A temperature sensor located in the side of the test gearbox is used to control the heating system as required by the test operating conditions" [14].

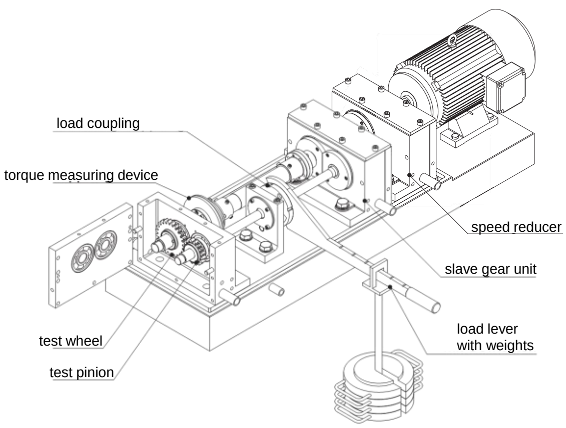

Fig. 3. FZG back-to-back test rig [14].

A standard gear geometry, called "type C-PT" [15], is chosen as test gear. The main data of the gears are described in Table 1. The pinions are case-hardened with a surface hardness of 719 HV. They are in the comparative variant in contact with case-hardened wheels of the same surface hardness. This hard-hard gear pairing represents the state of the art from the wear test acc. to DGMK 377-01 [2]. In the hard-soft variant, the case-hardened pinion is in contact with a through-hardened wheel. It has a surface hardness of 280 HV. Furthermore, the influence of the surface roughness of the harder pinion is investigated. It is varied between a smooth surface of $Ra_s = 0.14$ µm and a rough surface of $Ra_r = 0.76$ µm. The third condition of $Ra_m = 0.26$ µm matches the standard requirements of the type C-PT test gears. The surface roughness of all wheel flanks is Ra = 0.26 µm as well.

Table 1. Data of the test gears.

Gear parameter	Symbol	Unit	Value
Shaft center distance	a	mm	91.5
Module	m	mm	4.5
Number of teeth	z_1/z_2	–	16/24
Face width	b	mm	14
Helix angle	β	°	0
Normal pressure angle	α	°	20
Profile mod. factor	x_1/x_2	–	0.18/0.17
Tip diameter	d_{a1}/d_{a2}	mm	82.46/118.36
Pitch diameter	d_{w1}/d_{w2}	mm	73.20/109.80

The test variants are examined in up to three different speed stages with three different lubricant film thicknesses (see Table 2). It allows to observe the evaluation of the wear coefficients acc. to Plewe [4] in relation to the lubricant film thickness. Wear is measured as loss in mass during a test run by weighting the test gears before and after a test run with an accuracy of 1 mg. The slower speed stages 1 and 2 with circumferential speeds of $v_{t1} = 0.05$ m/s and $v_{t2} = 0.57$ m/s are based on the test conditions of the slow-speed wear test [2]. Additionally, a third speed stage is set to $v_{t3} = 2.78$ m/s. The selected speeds prevent channeling of the test gears and therefore ensures that wear is the predominant failure mode. In all test runs, a torque of $T_1 = 149.5$ Nm is applied.

Table 2. Test variants.

Influence	Grease	Rough-ness (pinion)	Speed Stage in m/s		
			0.05	0.57	2.78
Hard-Hard	M110LiX1	0.26 µm	x	x	x
	M110Ph1	0.26 µm	x	x	x
Hard-Soft	M110LiX1	0.26 µm	x	x	x
	M110Ph1	0.26 µm	x		
Roughness	M110LiX1	0.14 µm	x		
		0.76 µm	x		

The influence of the surface roughness of the harder pinion is investigates in test runs at speed stage 1 for the softer and the rougher pinion variant. Moreover, another grease M110Ph1 with polyurea thickener is tested. It has a closer meshed thickener net [16] and therefore, should show a higher wear resistance even under more wear critical conditions. Each test run is repeated to validate the result. The test greases are based on the same paraffin mineral oil with a nominal base oil viscosity $\nu_{40} = 113.1$ mm²/s. The thickener is a Lithium complex soap for the grease M110LiX1 and for M110Ph1 a polyurea soap. Both test greases have a NLGI grade 1 (see Table 3). The lubricant temperature is set to 90 °C in all test runs. The test gearbox is filled with the greases to a level that both gears, pinion and wheel, are operating completely immersed in the lubricant sump.

Table 3. Test greases.

Code	M110LiX1	M110Ph1
Base oil	Mineral oil	Mineral oil
Base oil viscosity	113.1 mm^2/s	113.1 mm^2/s
Thickener type	Lithium complex	Polyurea
Thickener (wt%)	9.2 %	19.3 %
Additive	–	–
NLGI grade	1	1

4 Test Results

In all test runs, a significant higher amount of wear can be observed on the wheel, when it is designed softer than the pinion. The amount of wear on the harder pinion is minimal in this case and therefore negligible. However, the amount of wear in general is higher than with gears of the same hardness under comparable running conditions. In contrast, the loss in mass is distributed evenly, when pinion and wheel have a similar surface hardness. This behaviour is known from observations of the standard wear test. Based on these findings, only the wear coefficients acc. to Plewe [4] of the through-hardened wheels are necessary to be considered in the evaluation of the test results. They are lifetime limiting. For the comparative variants of a case-hardened gear pairing, pinion and wheel are taken into the wear evaluation. The linear wear coefficients acc. to Plewe [4] from the experimental investigations can be seen for the grease M110LiX1 in Figure 4 and for M110Ph1 in Figure 5. Firstly, the wear coefficients for both greases and the hard-hard gear variants follow the (acc. to [4] translated) wear curve for grease lubricated gears within a narrow range. It should be noted that in the lower field of the diagram, even small deviations can cause an apparently large shift of the entered points due to the logarithmic axis division. However, the results of this study confirm the course of the wear curve for grease lubricated gears acc. to Plewe [4], as long as they are ini sufficient lubricating conditions (see [8,17]). The wear coefficients of the through-hardened wheels show for the first speed stage v_{t1} clearly higher values compared to the case-hardened pairing. The values are even above Plewe's wear curve for the corresponding gear material pairing with oil lubrication [4] (dotted line in Figure 4). With an increase of the lubricant film thickness (at higher circumferential speeds), the wear coefficients decrease for the through-hardened wheels. This is due to the better hydrodynamic conditions that prevail at higher speeds. It is noticeable that the wear curve of hard-soft gear pairing under grease lubrication does not follow Plewe's reference curve of a hard-soft gear pairing (dotted curve), but it is in the double logarithmic diagram almost parallel to Plewe's reference curve for grease lubrication (solid curve). Therefore, the reference curve 5 (Figure 2) of grease lubricated gears seems not only to be valid for a case-hardened gear pairing of similar surface roughness but is also the fitting curve for a hard-soft pairing that is lubricated by a grease.

Moreover, further tests investigated the influence of the surface roughness of the harder pinion as well as the

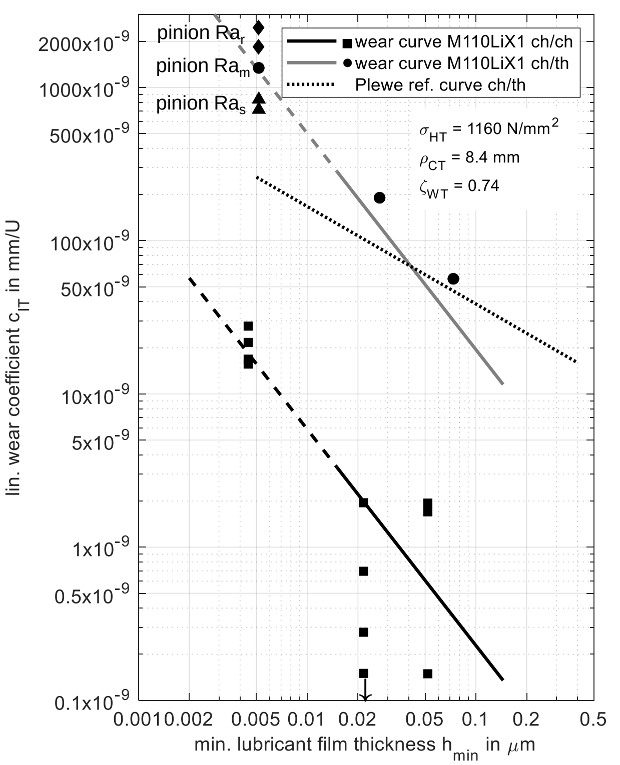

Fig. 4. Wear coefficients acc. to [4] for grease M110LiX1.

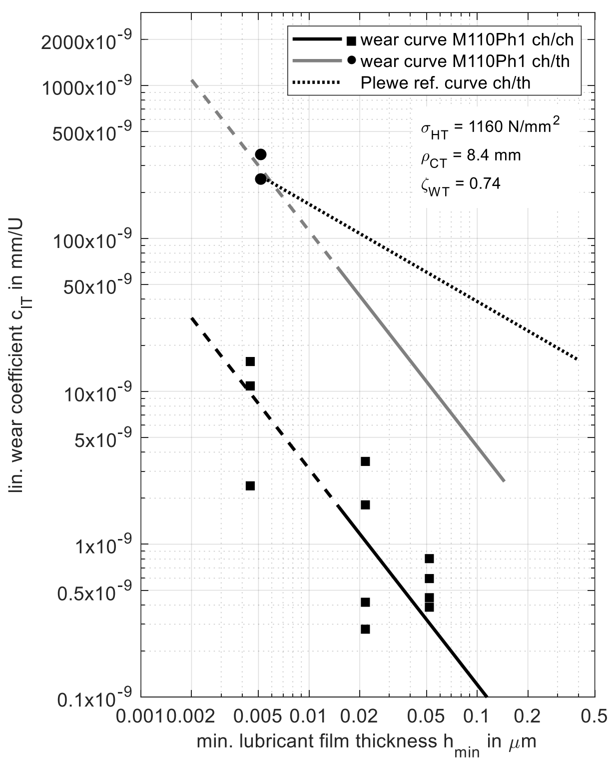

Fig. 5. Wear coefficients acc. to [4] for grease M110Ph1.

thickener type. The test runs are conducted in the first speed stage v_{t1}. It can be seen in Figure 4 that the wear coefficients with a rougher flank surface Ra_r are above the values with the medium roughness Ra_m. Consequently, the values for the hard-soft variant with the smoothest pinion surface Ra_s are below the medium variant. Finally, the test grease M110Ph1 is investigated in two test runs. The different thickener type shows an improved wear protection and has smaller wear coefficients than the Lithium complex grease. This trend is visible for the case-hardened gear pairing as well as the hard-soft pairing.

5 Modified Calculation Approach

The results from the speed stage investigations are taken to derive a conversion of the wear coefficients between gear pairings of the same surface hardness and a different hardness. Since the curse of the wear coefficients in Figure 4 is almost parallel for both variants, a first approximation can convert the wear coefficients by moving Plewe's reference curve vertically, in direction of the y-axis. A similar procedure is known from the influence of the surface hardness on the tooth flank carrying capacity that is expressed by the Work Hardening Factor Z_X in ISO 6336-2 [18]. Analogous, all possible influences from this study are taken as factors into the Equation (1) to calculate the linear wear coefficient $c_{lT2,X}$ for a hard soft gear pairing.

$$c_{lT2,X} = c_{lT} \cdot F_X \cdot F_{Ra} \cdot F_{HV} \cdot F_V \tag{1}$$

The discussed influences are presented in the factors for conversion F_X, surface roughness of the pinion F_{Ra}, surface hardness F_{HV} and for the lubricant F_V. Thus, a transfer from results of the standard slow speed wear test with hard-hard test gears to a hard-soft gear pairing in a field application is possible by multiplication with factors. A comparison of the fitted wear curves equations for M110LiX1 prove the validity of this approach. Both Equations (2) and (3) have almost the same exponent and therefore, almost the same slope.

$$c_{lT,chch} = 0.029 \cdot h_{min}^{-1,11} \tag{2}$$

$$c_{lT,chth} = 2.57 \cdot h_{min}^{-1,19} \tag{3}$$

By dividing the two equations, it is possible to determine the factor F_X that sets the vertical translation between the curves. Therefore, it is postulated that they are parallel in the Plewe diagram and have got the same exponents. In consequence, the translation factor F_X for conversion of the wear coefficients between a hard-hard and a hard-soft gear pairing is presented in Equation (4).

$$F_X = 88{,}6 \tag{4}$$

The influence of the surface roughness of the harder, case-hardened pinion can be developed by a linear regression of the plotted wear coefficients from Figure 4 related to the surface roughness at beginning of the test run. The resulting factor F_{Ra} can be calculated acc. to Equation (5). It moves the wear curve for grease lubricated gears in vertical direction as well and is normalized to the standard surface roughness Ra_1 = 0.3 µm of a test gear type C-PT. Therefore, F_{Ra} is set to a value of one for Ra_1 = 0.3 µm.

$$F_{Ra} = Ra_1 / 0.565\ \mu m + 0.469 \tag{5}$$

Furthermore, theoretical studies and test results with the test grease M110Ph1 have shown further influences on the wear amount by using a grease with polyurea thickener. Thus, an additional influence of the thickener can be assumed and is included in the calculation with an influence factor F_V (6,7). It fits the vertical translation between the wear curve of a hard-hard gear paring and hard-soft gear pairing for a certain grease.

$$F_V \text{ (lithium complex thickener)} = 1 \tag{6}$$

$$F_V \text{ (polyurea thickener)} = 0.41 \tag{7}$$

$$F_{HV} = 1 \tag{8}$$

Finally, it can be assumed based on the theoretical studies that the difference of the surface hardness between pinion and wheel also has an influence on the amount of wear. This is known from the calculation of the flank load carrying capacity acc. to ISO 6336-2 [18] and could have been observed for wear carrying capacity, too [19]. A smaller difference in surface hardness reduces the amount of wear and may increase the wear lifetime of the gears. To capture this influence, a factor for the difference of the surface hardness F_{HV} is introduced (8). Initially, the value is set to one, since a reliable statement is not possible based on the results of this study at the moment. By further experimental investigations under variation of this parameter it will be possible to extend the conversion approach in the future.

6 Conclusion

In this study, the wear behaviour of grease lubricated gear pairings in the combination case-hardened/case-hardened as well as case-hardened/through-hardened has been tested on an FZG back-to-back test rig and compared. The test runs have been conducted at three speed stages. It is noticeable in the hard-soft tests that almost the complete loss in mass occurs on the softer wheel, whereas it is distributed evenly between pinion and wheel, if they have a similar surface roughness. Furthermore, the difference in surface hardness between pinion and wheel as well as the surface roughness of the harder contact partner (pinion) have been systematically varied. The results show that the wear coefficients from all test runs follow the wear curve acc. to Plewe [4] within a small range. The wear behaviour of grease lubricated gears with a hard-hard pairing as well as a hard-soft pairing both follow the trend of Plewe's reference wear curve for grease lubricated gears (Figure 2, curve 5). The influences in dependence of the surface roughness of the harder rolling contact partner and in dependence of the grease thickener have been quantified: With a rougher surface roughness of the harder gear, the wear carrying capacity decreases. In contrast, finer surfaces or a polyurea thickener (compared to a Lithium complex soap) have a positive effect on the wear resistance. In order to include these findings in the wear calculation, influence factors were introduced that are able to mathematically translate the wear curve acc. to Plewe [4] for the prevailing operating conditions. This enables the gear designer to make sufficient statements about the service life of a grease lubricated, hard-soft gear stage under operating conditions from the results of a standard wear test for gears [3]. The difference in the surface hardness of the two gears in contact may also have further effects. In a newly developed calculation approach based on wear tests with oil lubricated gears, a conversion from a hard-hard gear pairing to a hard-soft gear pairing is realized by a horizontal translation of the wear curve [19]. However, a direct transfer to grease lubricated gears is not possible. Therefore, further investigations are necessary to combine these two approaches.

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