Influences on the lubricant supply of grease lubricated gears

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Abstract. Greases have a variety of advantages when special operating conditions apply. Mainly related to large, slow-running gear drives such as used in heavy industry applications, grease lubrication can also be the preferred solution for small, fast running gear drives. Consequently, the calculation of the wear service life within the gear design process is essential. Due to their flowing properties, there is a danger of losing the lubrication supply depending on the operating conditions, boundary conditions and grease properties. While a circulating lubricant ensures a continuous lubricant supply to the gear mesh, channeling includes the risk of starved lubrication and consequently, a discontinuation of the lubricating film that can lead to heavy damages of the gearbox. The experimental investigations on a modified FZG back-to-back test rig show a strong effect of operating conditions and grease properties on the lubricant supply: A higher amount of grease in the gearbox and a higher lubricant temperature support circulating, whereas, a higher consistency of the grease supports channeling. Based on the results, a first calculation approach is developed that approximates the lubricant supply of grease lubricated gears for the gear design process.

1. Introduction

Special operating conditions of a gearbox can require a grease lubrication. Especially in large, slow-running and open gear drives such as used in heavy industry applications (i.e. in the primary or mining sector), but also for small, fast running gear drives i.e. in power tools, greases are the preferred solution compared to oils. They have a variety of advantages that allow low-maintenance lifetime lubrication. In robot gearboxes they ensure a lubrication of the gear flanks, regardless of the spatial position the gearbox is in.

Continuous (or slow speed) wear (Figure 1a) is often the lifetime limiting factor when gears are running under grease lubrication. Wear appears continuously during the whole runtime and can be seen as scratches on the whole tooth flank. An exception is the pitch diameter, where almost no wear occurs, because the sliding speed is close to zero [1,2]. Consequently, the lubricating film thickness is higher than in other flank areas. For the gear applications mentioned prior, the calculation of the wear service life within the gear design process is essential. One common way to predict the wear lifetime of gears is the calculation method acc. to Plewe [3]. The calculation has originally been developed for oil lubrication and thus, assumes a continuous lubricant supply of the gear mesh. Greases can result in more critical lubrication conditions due to their flowing properties [4,5]. Different influences like operating conditions, boundary conditions and grease properties have strong effects on the lubrication supply and hence, on the lifetime of the gear pairing. While a circulating lubricant ensures a continuous lubricant supply to the gear mesh, channeling includes the risk of starved lubrication and accordingly, a discontinuation of the lubricating film. This discontinuation can lead to heavy damages or an immediate failure of the gearbox. As a consequence, continuous wear is no longer the critical gear...
failure mode, but scuffing (Figure 1b) or macro pitting (Figure 1c). Scuffing occurs typically in flank areas with high sliding speed and high contact temperatures. If a critical temperature is exceeded, the meshing tooth flank surfaces are welded instantaneous. The visual appearance of scuffing are deep scratches in sliding direction in corresponding flank areas of pinion and wheel. Macro pitting, in contrast, is a damage due to fatigue, but influenced by the lubricating conditions. It appears as larger, typically shell shaped, material breakout from the gear flank surface. Pittings are typically located on the tooth flank below the pitch diameter in areas of negative specific sliding [4,5].

![Image](image_url)

**Figure 1.** Typical gear failure modes

In order to prevent channeling by the gears, the knowledge of the expected lubricant supply, during the gear design process is essential. A first approach for the estimation of the lubrication supply mechanism of grease-lubricated gear pairings was proposed by Schultheiß [6] as a size-velocity-factor. Based on current experimental investigations, that first approach is extended within this course to include further influences into the model.

### 2. Lubricant supply mechanism

If a gear pairing operates under oil lubrication at a sufficient fill level, a lubricant supply of the tooth contact can be generally assumed. The oil is circulating with the rotation of the gears. With increasing circumferential speed, the lubricant film thickness increases and thus, causes better hydrodynamic conditions in the tooth contact (acc. to [7]). For grease lubrication, certain restrictions have to be considered. Although the wear behaviour can be related to the lubricant film thickness of the base oil [3], several more influences have an effect on the service life. The complex interactions between the grease components result in a different behaviour between a grease and its base oil in a sliding/rolling contact. The investigations of Poon [8] on a disc machine even showed a decrease of the lubricant film depending on the run time. Later, Fukunaga [9] could prove that the consistency and therefore, the thickener to have a greater effect on the grease characteristics than any base oil properties. The thickener determines the flowing properties that are essential to the lubrication of the tooth contact. If a grease is too viscous, there is a risk of starved lubrication. Consequently, the tackiness of a grease plays an important role for the ranking of greases acc. to Georgiou [10]. A lot of other investigations are dealing with grease lubrication, but the majority is based on investigations with the help of a ball-on-disc apparatus or a disc machine [8,11,12,13] and cannot directly be transferred to the conditions of a gear stage. Other experiments have been run with grease lubricated gears, but under non-critical conditions concerning a starved lubrication in the gear mesh [9,14,15]. First, Stemplinger and Schultheiß [4,5] observed the consequences of insufficient conditions on the gear service lifetime. They referred the results to two possible supply mechanisms grease lubricated gears can have. On the one hand, there is the so called “circulating” supply mechanism (Figure 2a) that mainly occurs at lower circumferential speeds and with low consistency greases; on the other hand so called “channeling” (Figure 2b) that mainly occurs at higher circumferential speeds and with high consistency greases. During circulation the grease builds a lubricating ring surrounding the gear and is transported into...
the tooth contact. The good lubricant supply enables a good mixing of the grease in the gearbox and thus, a good heat dissipation [4,5].

During the supply mechanism channeling, no or only very little grease can flow back into the tooth gaps behind the gear mesh. Moreover, at higher speeds the lubricating grease is radially thrown off. Only a minimal amount of bleed oil can prevent the gear pairing from losing the separating lubricant film. Otherwise, the starved lubrication can result in an immediate failure of the gearbox. The insufficient mixing of the grease sump causes a poor heat dissipation and thus higher temperatures around the gears [4,5].

Based on the investigations of Schultheiß [6] a first calculation approach was developed in order to predict, whether a gear stage has a tendency towards circulating or channeling. The so called “size-velocity-factor” considers the rotational speed of a gear pairing and its size and therefore, the geometry of the gear teeth and their gaps in-between. Due to the large number of other influences on the lubrication mechanism and the complex interactions between them, this calculation approach needs to be validated and further developed. Especially, the transition area between the two supply mechanisms circulation and channeling is of particular interest.

3. Experimental investigation
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” [17]. Additionally, within the herein described investigations an acrylic glass cover in the test gearbox allows to observe the lubricant supply by visual inspection as well as by a high-speed camera.

The test runs start at a slow rotational speed of the pinion of $n_1 = 9 \text{ min}^{-1}$ (acc. to the slow speed wear test [18]). Because the observation is difficult, if the speed increases constantly, 22 speed steps are defined up to a pinion speed of $n_1 = 3000 \text{ min}^{-1}$. At each step, the lubricant supply mechanism is observed as soon as constant conditions can be seen. If not otherwise described, both gears are completely surrounded by the grease and the sump temperature $\vartheta_s = 90^\circ \text{C}$. Only a minimal torque of $T_1 = 35.3 \text{ Nm}$ is applied to ensure, the gears are in contact. Since the lubricant supply is in the centre of the investigation, there is no further documentation of failure mechanisms within this course.

The test gears are case-hardened standard gears of the type C-PT [18]. The main data of the gears are described in Table 1. To evaluate the influence of the grease consistency, three lubricants have been investigated within this course, which correspond to common greases for these applications (Table 2). They are based on a paraffin mineral oil with a nominal base oil viscosity $\nu_{40} = 110 \text{ mm}^2/\text{s}$. The thickener is a lithium complex soap. The three lubricants only differ in their thickener concentrations in order to avoid further influences.

![Figure 3. FZG back-to-back test rig [17,18].](image-url)

Table 1. Data of the C-PT gears.

| Gear parameter         | Symbol | Unit | Numerical 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            | $\beta$ | ° | 0              |
| Normal pressure angle  | $\alpha$ | ° | 20             |
| Profile modification factor | $x_1 / x_2$ | – | 0.1817 / 0.1715 |
| Tip diameter           | $d_{a1} / d_{a2}$ | mm | 82.46 / 118.36 |
| Pitch diameter         | $d_{w1} / d_{w2}$ | mm | 73.2 / 109.8   |
| Flank roughness        | $R_{a12}$ | $\mu$m | 0.2 – 0.4 |

Table 2. Data of the investigated greases.

| Code               | M110LiX00 | M110LiX1 | M110LiX2 |
|--------------------|-----------|----------|----------|
| Description        | –         | Grease   | Grease   | Grease   |
| NLGI grade         | –         | 00       | 1        | 2        |
| Base Oil           | –         | Mineral oil | Mineral oil | Mineral oil |
| base oil viscosity $\nu_{40}$ | mm$^2$/s | 113.1 | 113.1 | 113.1 |
| base oil viscosity $\nu_{100}$ | mm$^2$/s | 12.1 | 12.1 | 12.1 |
| Thickener type     | –         | Lithium-complex | Lithium-complex | Lithium-complex |
| Percentage Thickener | wt%   | 4.3      | 9.2      | 11.5     |
| Worked penetration (25 °C) | 0.1 mm | 412 | 318 | 285 |
| Oil separation (90 °C) | % | 5.2 | 5.3 | 3.1 |
4. Observations of the test runs

The results show a strong effect of the investigated operating conditions and grease properties: A higher amount of grease in the gearbox and a higher lubricant temperature support the supply mechanism circulating. In contrast, a higher torque and a higher consistency of the grease support channeling. Figure 4 shows under which conditions and at which rotational speeds circulating or channeling of the grease sump is more probable. Between these two supply mechanisms, there is a range, which cannot be clearly assigned to any of the mechanisms. It is called “transition zone”. In a first approximation the mean value of this zone has been selected as the critical rotational speed for the further analysis.

In the beginning of the first test “Ref” under reference conditions, both the pinion and the wheel rotate within the lubrication sump of the grease M110LiX1. A ring of grease is rotating with the gears. Large parts of the grease sump remain stationary. The tooth spaces on the wheel are completely filled with the lubricant (comparable to Figure 5a). At a rotational speed of \( n_1 = 845 \text{ min}^{-1} \), the grease starts to lift off from the gears and intermittently splashes against the gearbox housing. With increasing speed at \( n_1 = 1250 \text{ min}^{-1} \), the thrown off grease jet becomes holey and inconsistent. Its flow is rather turbulent. This condition of the lubricant supply can no longer be described as pure circulating. The lubricant supply is in a transition zone. However, up to \( n_1 = 3000 \text{ min}^{-1} \), there is still a certain amount of grease in the tooth gaps. Only at this rotational speed, the sump behaviour starts to match the criteria of channeling (comparable to Figure 5b).

4.1. Influence of the filling level

The tests “Fill mid” and “Fill low” are investigating the influence of the level of grease in the gear box. For “Fill mid”, the grease level is up to the middle of the shaft centre (acc. to [18]). In relation to the immersion depth of the gear, “high” corresponds to 100 %, “medium” corresponds to 50 % and “low” corresponds to 25 %. It is suspected that the fill level of the gearbox has an effect on the lubricant supply, because the dipping time, the tooth gaps can be refilled with the lubricant, decreases with a smaller amount of grease inside the housing.

The comparison of the three tests shows a very clear influence of the fill level. In the beginning of the test “Fill mid”, the grease forms a circulating ring around the gears. As the circumferential speed increases, a decrease of the lubricant in the tooth gaps can be observed immediately. Beyond a rotational speed of \( n_1 = 386 \text{ min}^{-1} \), the first grease splashes are found on the gearbox housing. At a speed of \( n_1 = 620 \text{ min}^{-1} \), large parts of the lubricating sump are thrown off. Because the tooth gaps are only a
short time inside the lubricating sump, the grease can hardly flow back. The lubrication supply mechanism is clearly channeling. During the test “Fill low”, the grease circulates at the first speed stages. However, even at those speeds, the tooth gaps are not completely filled. Already at a rotational speed of \( n_1 = 105 \text{ min}^{-1} \), the first grease splashes are visible on the housing. From a speed of \( n_1 = 269 \text{ min}^{-1} \), the supply mechanism can be described as channeling.

4.2. Influence of the temperature

The runs “60 °C” and “25 °C” are testing the influence of the sump temperature on the lubricant supply. Since the greases become more viscous at lower temperature, it is hypothesized that a reduction of the flowing properties (compared to the reference conditions) promotes a channeling of the test gears.

The test “60 °C” is operated with a sump temperature of \( \theta_S = 60 \text{ °C} \). In the beginning, at low speeds, the gears rotate within the lubrication sump in the area of circulating (Figure 5a). The tooth gaps are completely filled. The flowing behaviour is worse than under reference conditions. A large part of the lubricant sump is stationary. At a circumferential speed of \( n_1 = 620 \text{ min}^{-1} \), the parts of the grease are thrown off the gears. The fill level inside the tooth gaps is very low and the supply mechanism can no longer be described as circulating. At a rotational speed of \( n_1 = 1677 \text{ min}^{-1} \), the lubricant cannot flow back into the tooth gaps and the gear mesh is insufficiently supplied by the lubricant. At this point, the supply mechanism is clearly channeling (Figure 5b).

![Image](image_url)

a) “60 °C” at \( n_1 = 386 \text{ min}^{-1} \)  
b) “60 °C” after \( n_1 = 2286 \text{ min}^{-1} \)

**Figure 5.** Visual inspection of the lubricant supply [16]

In the test “25 °C”, the sump temperature is at the room temperature of \( \theta_S = 25 \text{ °C} \). At low rotational speeds, the gears are rotating within the lubrication sump in the area of circulating. The gaps between the teeth are completely filled, a circulating grease ring is visible. The surrounding lubrication sump does not take part in the lubrication, but rather gives the impression of a shrouding. The first splashes on the housing can be seen at \( n_1 = 503 \text{ min}^{-1} \). Due to the poor flowing behaviour at this low temperature, only a small part of the tooth spaces is filled with the grease. From the rotational speed of \( n_1 = 1250 \text{ min}^{-1} \), the lubricant supply can clearly be described as channeling.

4.3. Influence of the consistency

Two tests consider the effects of different grease consistencies. The low-consistency grease M110LiX00 (“NLGI00”) and the high-consistency grease M110LiX2 (“NLGI2”). They cover a concentration range of the thickener soap from 4.3 % up to 11.5 %. Comparable to the prior tests runs at different temperatures, an influence on the lubricant supply due to the different flowing properties of the greases is expected.

The comparison of the runs shows a clear trend that a high-consistency grease promotes channeling. The grease M110LiX00 is gradually thrown off the gears against the gearbox cover at \( n_1 = 620 \text{ min}^{-1} \). It seems almost liquid during the whole test run and thus, the lubrication supply can be described as circulating up to the maximum speed \( n_1 = 3000 \text{ min}^{-1} \). Under identical test conditions, the grease M110LiX2 shows similar behaviour at low speeds and is initially comparable with the previous tests. Due to the low rotational speeds, the gears are circulating within the lubrication sump. As a re-
result of the poor flowing properties, only a small part of the tooth gaps is filled. At a rotational speed of \( n_1 = 386 \text{ min}^{-1} \), the amount of grease in the gear mesh is reduced and the lubrication supply is in the transition zone. At \( n_1 = 1058 \text{ min}^{-1} \), parts of the grease are thrown off and spray onto the housing. The supply mechanism can be referred to as channeling.

5. Calculation approach

Based on the observations of the experimental investigations, a calculation approach is developed to determine the rotational speed \( n_{\text{LSM}} \), when a grease lubricated gear pairing presumably changes its lubricant supply mechanism and starts channeling (1). The grease M110LiX1 is used as a reference, because its grease components are widely used in practical applications. The boundary rotational speed of 2135 \( \text{min}^{-1} \) is the mean value of the transit zone in this first approximation of the lubricant supply. The different factors \( F_K \), \( F_F \) and \( F_\theta \) consider the results of the experimental investigations and can change the limiting speed \( n_{\text{LSM}} \) depending on the running conditions, boundary conditions or grease properties. They result from different regression analyses of the test results.

\[
n_{\text{LSM}} = 2135 \text{ min}^{-1} \cdot F_K \cdot F_F \cdot F_\theta \quad (1)
\]

The grease consistency and therefore, the thickener concentration have a strong influence on the supply mechanism. A higher thickener concentration promotes the tendency of channeling (2). Due to the poorer flowing behaviour, the amount of lubricant in the tooth contact decreases significantly even at lower speeds (Figure 6).

The tests with different gear filling levels show a clear influence of the gear immersion depth on the supply mechanism. With a higher filling level, circulation of the lubricant is promoted (Figure 7). The longer a gear is inside the sump, the better is the possibility for the grease to flow back into the tooth gaps. Consequently, it enables better lubrication conditions and has a positive effect on the wear life of a gear pair (3).

\[
F_K = 76.64 \cdot 0.44 \frac{V_{hl}}{W} \quad (2)
\]

\[
F_F = X_{\theta h} \quad (3)
\]

\[
F_\theta = 4.6 \cdot 10^{2.6 \cdot 1.002 \left(\frac{\theta_\xi}{\pi}\right) - 4} \quad (4)
\]

The temperature of a lubricating grease has an influence on the supply mechanism as well. At higher temperatures the grease becomes more fluid and have a better ability to flow back into the tooth gaps (Figure 8). This promotes circulation of the lubricant. When the sump temperature decreases, the grease is not able to refill the tooth gaps behind the gear mesh. The missing refill results in a tendency of channeling (4).
6. Conclusion
The results of the investigations show a strong effect of the investigated operating conditions and grease properties: A higher amount of grease in the gearbox and a higher lubricant temperature support the supply mechanism circulating. In contrast, a higher consistency of the grease supports channeling. Based on these results, a calculation approach was developed that offers the possibility to estimate in a first approximation the lubricant supply of grease lubricated gears for the gear design process. It calculates the rotational speed of the pinion $n_{LSM}$, at which a channeling of the gear pairing becomes probable. Therefore, it accounts the influences of the fill level, the temperature and the consistency. In order to reach a most possible validity for this approach, it is based on test results with standard test gears and a plain grease with a mineral base oil and a lithium complex thickener. These are common components in a lot of gearbox greases and have been used in a variety of research projects before. However, for a deeper understanding of the lubricant supply mechanism of greases and to consider other influences, it is necessary to further develop the investigation. Besides the running conditions and grease properties investigated in this course, several other aspects can have an effect on the lubrication supply of the gears. Thus, the geometry of the gear pairing (module resp. diametral pitch and tooth width) as well as grease components like the base oil, its viscosity and the type of thickener should be in the focus of additional investigations. It can be assumed that they have an influence of the flowing properties of a grease and its ability to get into the gear mesh. Consequently, these aspects can promote either circulating or channeling. Further tests can investigate these influences quantitatively and extend the validity of the calculation approach. This way, the relevant parameters as well as the interactions between the influences can be observed, analysed and taken into account in the calculation approach.

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References
[1] Siewerin B J, Dobler A, Tobie T and Stahl K 2019 Applicability of an Oil Based Calculation Approach for Wear Risk and Wear Lifetime to Grease Lubricated Gear Pairings Proceedings of the ASME IDETC/CIE 2019
[2] Linke H 2010 Stirnradverzahnung: Berechnung - Werkstoffe - Fertigung Carl Hanser Verlag Munich
[3] Winter H and Plewe H-J 1985 Calculation of Slow Speed Wear of Lubricated Gears Gear Technology 2(6) pp. 8–16
[4] Stemplinger J-P, Stahl K, Höhn B-R, Tobie T and Michaelis K 2014 Analysis of lubrication supply of gears lubricated with greases NLGI 1 and 2 and the effects on load-carrying capacity and efficiency NLGI Spokesman 78(1) pp. 18–22
[5] Schultheiss H, Stemplinger J-P, Tobie T and Stahl K 2016 Influences on failure modes and load carrying capacity of grease lubricated gears Gear Technology 33(1) pp. 42–47
[6] Schultheiss H, Tobie T and Stahl K 2017 The Effect of Selected Operating Parameters, Material Pairing, and Gear Size on the Wear Behavior of Grease Lubricated, Small Module Gears Journal of Tribology 139(6)
[7] Dowson D and Higginson G R 1966 Elastohydrodynamic Lubrication Pergamon Press Oxford [u.a.]
[8] Poon S 1972 An experimental study of grease in elastohydrodynamic lubrication Journal of Lubrication Technology 94(1) pp. 24–37
[9] Fukunaga K 1990 Grease for gear lubrication Lubrication Engineering 46(9) pp. 557–564
[10] Georgiou E P, Drees D, De Bilde M and Anderson M 2018 Can We Put a Value on the Adhesion and Tackiness of Greases? *Tribology Letters* **66**(60)

[11] Kaneta M, Ogata T and Naka M 2000 Effects of a thickener structure on grease elastohydrodynamic lubrication films *Proc. Instrn. Mech. Engrs*

[12] Cousséau T, Björling M, Graça B, Campos A, Seabra J and Larsson R 2012 Film thickness in a ball-on-disc contact lubricated with greases, bleed oils and base oils *Tribology International* **53** pp. 53–60

[13] DeVaal P, Möller V and Langenhoven J 2013 Investigating Seizure Load and Wear Characteristics *World Tribology Congress*

[14] Krantz T L and Handschuh R F 2004 A Study of Spur Gears Lubricated With Grease-Observations From Seven Experiments *58th Meeting of the Society for Machinery Failure Prevention Technology (MFPT)*

[15] Krantz T L, Oswald F and Handschuh R F 2007 Wear of Spur Gears Having a Dithering Motion and Lubricated With a Perfluorinated Polyether Grease *Proceedings of the ASME IDETC/CIE 2007*

[16] Siewerin B J, Tobie T and Stahl K 2021 Calculation Method and Boundary Criteria for the Continuous Wear Behaviour of Grease Lubricated Gears with Regard to Different Material Combinations *German Society of Petroleum and Coal Science and Technology (DGMK e.V.) Hamburg*

[17] DIN ISO 14635-1:2006-05 2006 Gears – FZG test method – part 1 FZG test method A/8.3/90 for relative scuffing load carrying-capacity of oils

[18] Bayerdörfer I, Michaelis K and Höhn B-R 1997 Slow Speed Wear Test in the FZG Gear Test Rig *5th CEC International Symposium on the Performance Evaluation of Automotive Fuels and Lubricants*