Influences on Failure Modes and Load-Carrying Capacity of Grease-Lubricated Gears

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In order to properly select a grease for a particular application, a sound knowledge of the influence of different grease components and operating conditions on the lubrication supply mechanism and on different failure modes is of great benefit.

In this paper the experimental results of a number of research projects with greases ranging from NLGI 00 to NLGI 1 — using the FZG back-to-back gear test rig — are evaluated in context.

Introduction

For many years, greases have been used in the lubrication of ball bearings. For the lubrication of gears, however, oils still play the dominant role. In recent years, the advantages of grease over oil in certain gear applications have led to an increasing significance of grease in gear lubrication. Unlike oil lubrication, for which methods for the calculation of the gear load-carrying capacity have long since been established in standards, such as ISO 6336 or DIN 3990, calculation methods for the load-carrying capacity of grease do not yet exist. This is mainly due to the complex interaction of the influence parameters on the load-carrying capacity of grease-lubricated gears as well as the limited availability of accessible, experimental investigations. A large portion of the conducted experimental work on grease lubrication focuses on the application of grease in the lubrication of bearings. Therefore numerous studies have concentrated on lubricant film thickness investigations with grease. A good summary of these and further, similar work is given by Lugt (Ref. 10). Nonetheless, the number of experimental investigations on the load-carrying capacity of grease-lubricated gears is steadily increasing. For example, Fukunaga (Refs. 2, 1), and in more recent years, Krantz and Handschuh (Ref. 8), and also Krantz et al. (Ref. 9) performed experiments to investigate the effect of different operating conditions and grease parameters on the failure modes of grease-lubricated gears. In the past decade several research projects were conducted at FZG with the aim of expanding general knowledge of grease-lubricated gears. The experimental work therein focuses on the influences of different grease components and operating conditions on different gear failure modes such as scuffing, wear and pitting.

In this paper, selected experimental results from DGMK research projects 591(Ref. 3) 673 (Ref. 4), 670 (Ref. 14), 671 (Ref. 15) and 725 (Ref. 11), with greases ranging from NLGI 00 to NLGI 1, using the FZG back-to-back gear test rig and that have already largely been published on their own in papers and dissertations (see Refs. 12, 16, 5 and 13), are evaluated in context.

Lubrication Supply Mechanism for Grease-Lubricated Gears

The lubrication supply mechanism is of great importance, as it determines the amount of lubricant available in the gear mesh as well as heat dissipation from the gear mesh. These factors, in turn, determine the resulting gear failure mode and lifetime. For oil dip lubrication, under normal operating conditions oil is always available to the gear mesh if the sump fill-level is sufficient. The lubrication supply mechanism is, therefore, always the same and the lubricant film thickness increases with increasing rotational speed. For grease, however, the lubrication supply mechanism is more complex and depends on a number of factors. Fukunaga (Ref. 2) and Stemplinger (Ref. 15) each identify two main lubrication supply mechanisms for grease-lubricated gears, i.e. — "channeling" and "circulating/churning" (Figs. 1 and 2). Circulating refers to the lubrication supply mechanism where the grease in the sump in close proximity to the rotating gears circulates, regularly fills the tooth gaps to a certain degree and is thus available in the gear mesh. According to Stemplinger (Ref. 15), circulating, in comparison to channeling, ensures a better lubricant supply to the gears, better cooling,

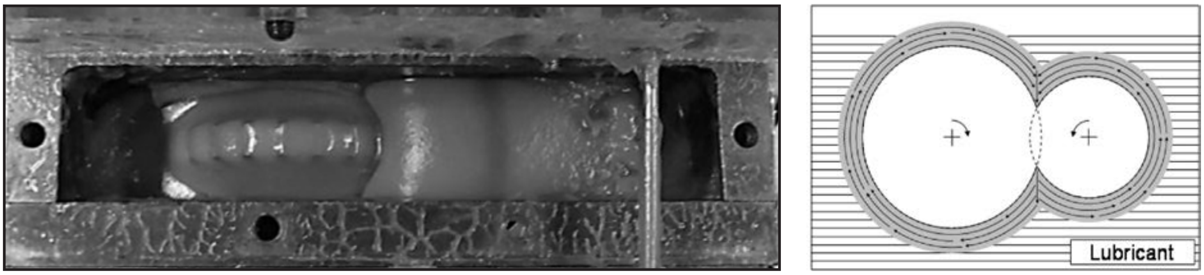

Figure 1 Lubrication supply mechanism "circulating" (Ref. 15).

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and thus lower bulk temperatures in the gears. Furthermore, more homogenous sump temperatures can be observed. Circulating mainly occurs at higher sump fill levels, lower NLGI grades, and lower rotational speeds.

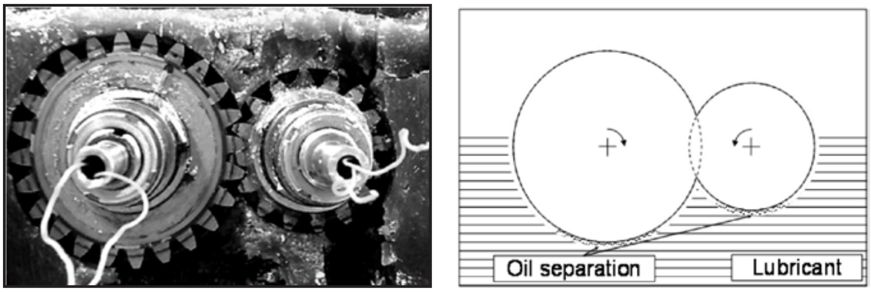


Figure 2 Lubrication supply mechanism "channeling" (Ref. 15).

Channeling, on the other hand, refers to a situation where the grease does not, or at least not to a significant degree, refill the tooth gaps of the rotating gears and thus very little, to no grease at all, is carried into the gear mesh. Often an accumulation of bleed oil can be observed, which may in fact be mainly responsible for the lubrication of the gears (Fig. 2). According to (Ref. 15), a lack of lubrication and cooling can be observed, which can lead to high bulk temperatures in the gears with a heterogeneous temperature distribution in the gearbox. In total, only a small amount of grease participates in gear lubrication and so comparatively low no-load losses can be observed. Channeling mainly occurs for low- to medium-fill levels, high NLGI grades and higher rotational speeds.

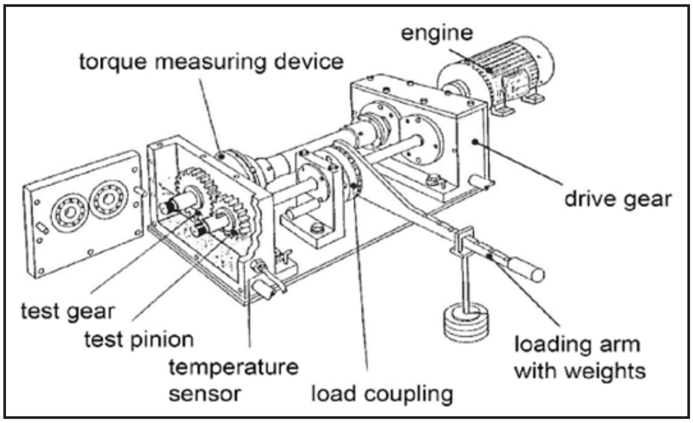


Figure 3 FZG back-to-back gear test rig.

Test Equipment

The experimental investigations of the gear performance under various test conditions were conducted on FZG back-to-back gear test rigs. The schematic setup of the FZG back-to-back gear test rig is shown (Fig. 3). Acc. to ISO 14635-1 (Ref. 6), the test rig utilizes a re-circulating power loop principle — also known as a "four-square configuration" — in order to provide a fixed torque (load) to a pair of test gears.

For the different investigations conducted, the gear types specified in the respective test regulation were used. Table 1 shows an overview of the data for the different test gear types used.

Table 1 Data of the test gears types A, C-PT, C-GF and 21/24

Dimension		Symbol	Type A	Type C-PT	Type C-GF	Type 21/24	Unit
Center distance		*a*	91.5	91.5	91.5	91.5	mm
Module		*m*	4.5	4.5	4.5	4.0	mm
Number of teeth	pinion	z_1	16	16	16	21	–
	wheel	z_2	24	24	24	24	–
Face width		*b*	20	14	14	15	mm
Helix angle		β	0	0	0	0	°
Pressure angle		α	20	20	20	20	°
Working pressure angle		α_w	22.44	22.44	22.44	22.44	°
Profile-shift coefficient	pinion	x_1	0.8532	0.182	0.182	0.22	–
	wheel	x_2	–0.5	0.172	0.172	0.177	–
Tip diameter	pinion	d_{a1}	88.70	82.46	82.46	92.70	mm
	wheel	d_{a2}	112.50	118.36	118.36	103.90	mm
Basic material	pinion	–	20MnCr5	16MnCr5	16MnCr5	16MnCr5	–
	wheel	–	20MnCr5	16MnCr5	16MnCr5	42CrMo4	–
Surface hardness	pinion	–	700–750	700–750	700–750	700–750	HV
	wheel	–	700–750	700–750	700–750	280–290	HV
Flank roughness	pinion	*Ra*	0.35±0.10	0.30±0.10	0.50±0.10	0.30±0.10	µm
	wheel	*Ra*	0.30±0.10	0.30±0.10	0.50±0.10	0.30±0.10	µm

Test Lubricants

Table 2 shows an overview of the lubricants used in the experimental investigations. All the tested lubricants were specially formulated for the investigations and are not commercially available. The EP additive package, however, is a fully formulated, commercially available product.

Influences on the Scuffing Load-Carrying Capacity of Grease-Lubricated Gears

The investigations on the scuffing load-carrying capacity were conducted in the project DGMK 591 (Ref. 3) in load stage tests, acc. to ISO 14635-1 with gears of type A. The pitch line velocity during the tests was $v_t = 8.3$ m/s ($n_1 = 2{,}250$ min-1), the starting sump for graphite as well as molybdenum disulphide (MoS_2) (Fig. 5). Under the given test conditions, MoS_2 performed better than graphite. An increase in the concentration of graphite did not further increase temperature in each load stage 50° C and the fill-level for all tests was at shaft center. For each lubricant, two runs were conducted. The results show that the base oil M680G-o exhibits a much higher scuffing load-carrying capacity than the grease M680Al00-o that is based on this oil (Fig. 4). Furthermore, with increasing base oil viscosity the scuffing load-carrying capacity of grease increases. Tests on the influence of a solid lubricant added to the grease showed an increase in the scuffing load-carrying capacity the scuffing load carrying capacity. Although the added solid lubricants were shown to increase scuffing load-carrying capacity, increased wear could be observed, especially with graphite. Furthermore, the thickener type was found to affect the scuffing load-carrying capacity. Although not shown in the diagrams, grease with a lithium soap thickener was shown to perform slightly better than grease with an aluminum-complex soap thickener.

Table 2 Test lubricants

Lubricant code	Base oil type	Nom. base oil viscosity v40 (cSt.)	NLGI grade	Tickener type	Thickener concentration (wt. %)	Solid lubricant	Additiv package
M680G-o	Mineral	680					
M680G	Mineral	680	–	–	–	–	EP
M70A100-o	Mineral	70	00	Al-X	4.4		
M70A100	Mineral	70	00	Al-X	4.3	–	EP
M680A100-o	Mineral	680	00	Al-X	3.3	–	–
M680A100	Mineral	680	00	Al-X	3.2		EP
M1200A100-o	Mineral	1200	00	Al-X	2.6	–	–
M1200A100	Mineral	1200	00	Al-X	2.5		EP
M70Li00	Mineral	70	00	Li	3.6	–	EP
M680Li00	Mineral	680	00	Li	4.7		EP
M1200Li00	Mineral	1200	00	Li	6.6		EP
P7OLi00	PAO	70	00	Li	approx. 6.0	–	EP
P680Li00	PAO	680	00	Li	approx. 6.0		EP
P1200Li00	PAO	1200	00	Li	approx. 6.0	–	EP
M680ALOO-oC4.2%	Mineral	680	00	Al-X	2.6	4.2% C	
M680A100-C4.2%	Mineral	680	00	Al-X	2.5	4.2% C	EP
M680ALOO-oC11.1%	Mineral	680	00	Al-X	2.1	11.1% C	–
M680A100-C11.1%	Mineral	680	00	Al-X	2.0	11.1% C	EP
M680ALOO-oM	Mineral	680	00	Al-X	2.6	4,2% M	–
M680A100-M	Mineral	680	00	Al-X	2.5	4,2% M	EP
M680Li0	Mineral	680	0	Li	6.65	–	EP
M1000Li0	Mineral	1000	0	Li	6.8	–	EP
M1500Li0	Mineral	1500	0	Li	6.95		EP
M110Li1	Mineral	110	1	Li	8.1	–	EP

C = synthetic graphite solid lubricant
M= MoS_2 solid lubricant
EP = extreme pressure additive package

Al–X= aluminium complex soap thickener
Li = lithium soap thickener
EP = 4 wt. % EP additive package (RC9505)

Influences on the Wear Behavior of Grease-Lubricated Gears

In DGMK 725 (Ref. 11) the influence of various grease parameters on the slow speed wear behavior was investigated with gears of type C-GF. The pitch line velocity during the tests was $v_t = 0.05$ m/s ($n_1 = 13$ min^{-1}), the pinion torque $T_1 = 626.9$ Nm ($p_c = 2{,}385$ N/mm^2) and the fill-level for all tests was at shaft center. The total test duration of the three-stage test used was 120 h. In the first 40 h the sump temperature was kept at 60° C; in the second 40 h at 90° C; and in the third 40 h at 60° C. For each lubricant, two runs were conducted and the average wear sum calculated. Under the given test conditions the lubrication supply mechanism in all tests was observed to be circulating. The results show that for lithium soap grease with either mineral base oil or PAO-based oil, tendentially lower wear occurs with increasing base oil viscosity (Fig. 6). The PAO-based greases show slightly lower wear than their mineral oil-based counterparts. The addition of the solid lubricant graphite to the grease M680Al00 leads to an increase in wear (Fig. 7). The concentration of solid graphite, however, has little effect on the wear behavior. The solid lubricant molybdenum disulphide (MoS_2) also leads to an increase in wear in comparison to the reference grease with no solid lubricant (M680Al00), although the increase in wear is not as large as with the same concentration of graphite.

Figure 8 shows results of wear tests conducted (Refs. 3, 14). After a load stage test acc. to ISO 14635-3 (Ref. 7), a 100 h run at LS 10 ($p_c = 1{,}539$ N/mm^2) followed. Only the wear in the 100 h run was evaluated (Ref. 7).

The NLGI 00 grease and the base oil show comparable wear. Increasing the NLGI-grade to 0 shows a clear increase in wear. This indicates that with NLGI 00 grease the lubrication supply to the mesh is better than with NLGI 0 and thus is likely to be

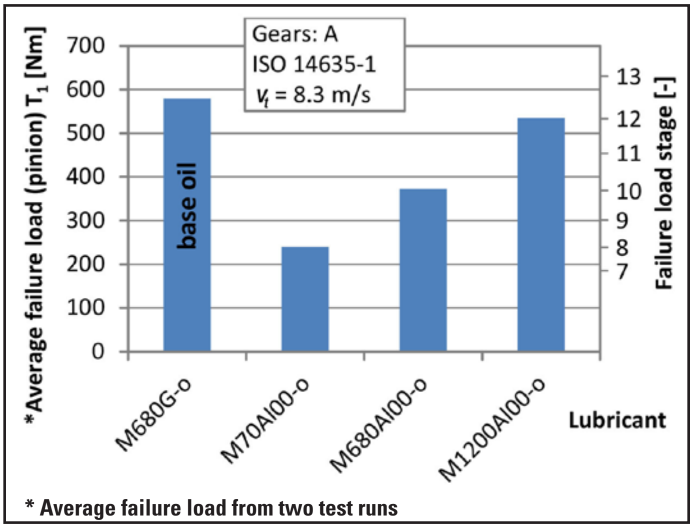


Figure 4 Influence of base oil viscosity on scuffing load carry capacity of NLGI 00 grease without EP additive package (Ref. 3).

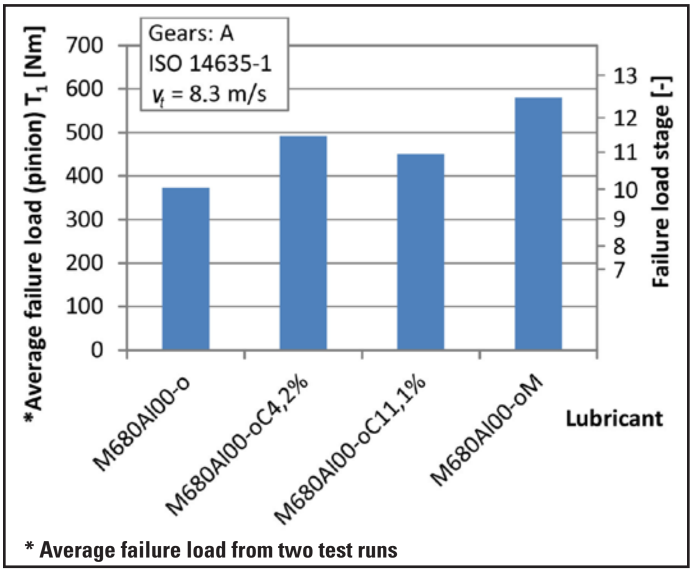
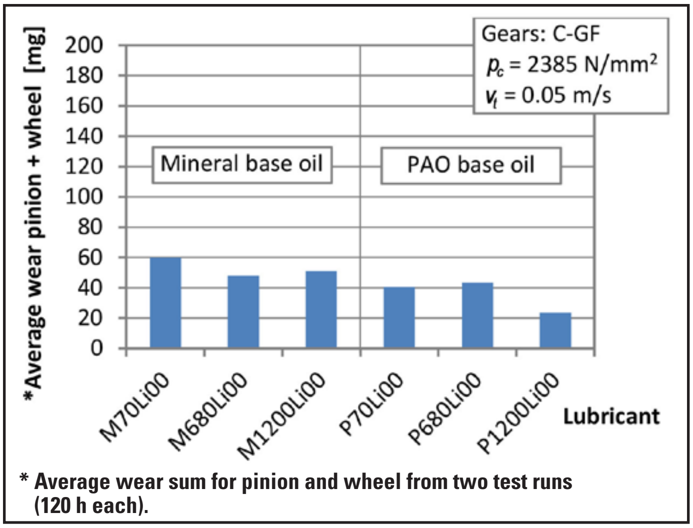


Figure 5 Influence of the solid lubricant type and concentration on scuffing load-carrying capacity of NLGI 00 grease without EP additive package (Ref. 3).



* Average wear sum for pinion and wheel from two test runs (120 h each).

Figure 6 Influence of the base oil-type and viscosity on the slow-speed wear behavior of NLGI 00 grease with EP additive package (Ref. 11).

circulating, while with NLGI 0 channeling effects possibly occur which lead to an increase in wear. An increase in the base oil viscosity of the NLGI 0 grease tends to lead to a further increase in wear.

Influences on the Pitting Lifetime of Grease-Lubricated Gears

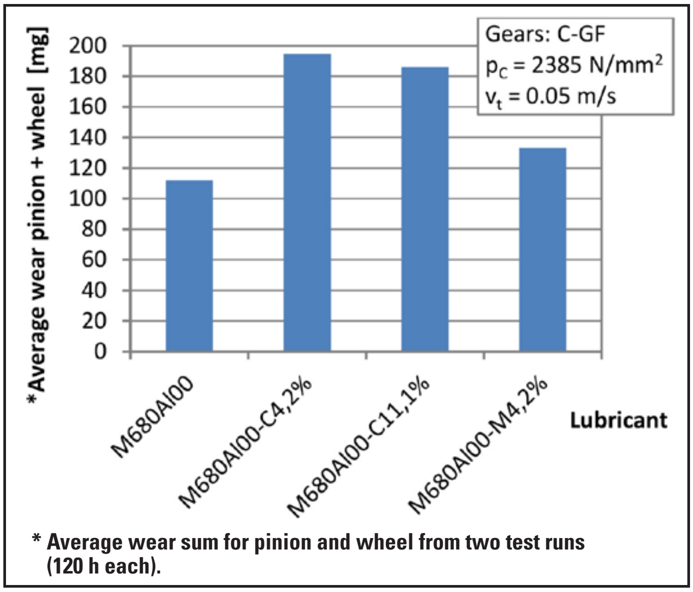
In the projects DGMK 591(Ref. 3), 673 (Ref. 4) and 670 (Ref. 14), tests were run under various conditions and with different gear types to investigate the influence of different grease components and operating conditions on the pitting lifetime of NLGI 00 and NLGI 0 greases. The fill-level for the tests was at shaft center; an overview of the test results is given in Figs. 9 – 11.

The results show that for grease, pitting lifetimes that are almost equal to those of the base oil can be achieved. This can be observed for M680Al00 and base oil M680G at 6.7 m/s (Fig. 9), as well as for M680Li00 and base oil M680G at 5.5 m/s pitch line velocity (Fig. 10). At the higher pitch line velocity (8.3 m/s), however, the pitting lifetime of M680Li00 is much lower than that of its base oil M680G (Fig. 10). This is due to the dominant lubrication supply mechanism. At lower speeds, circulating dominates, at the higher rotational speed dominates, at the higher rotational speed, however, channeling takes the upper hand. Figure 11 shows that for 6.7 m/s the pitting lifetime of the NLGI 0 grease M680Li0 is lower than for its base oil M680G. This shows that the NLGI-grade affects the lubrication supply mechanism — in this case probably leading to dominant channeling effects. It was also observed that increasing the base oil viscosity led to an increase in the pitting lifetime (Figs. 9 – Fig. 11).

Flank Load-Carrying Capacity of NLGI 1 Grease at Different Pitch Line Velocities and Fill-Levels

Figure 12 shows an excerpt of different investigations that were conducted in the research project DGMK 671 (Ref. 15). The diagram shows the different lubrication supply mechanisms for M110Li1 for two different fill-levels of 40 and 80% and the resulting failures. The sump temperature ϑ_S and the material loss of pinion Δm (*pi*) and wheel Δm (*wh*) are also shown.

In the 100h test, the investigated base oil for M110Li1 (not



* Average wear sum for pinion and wheel from two test runs (120 h each).

Figure 7 Influence of solid lubricant type and concentration on the slow speed wear behavior of NLGI 00 grease with EP additive package (Ref. 11).

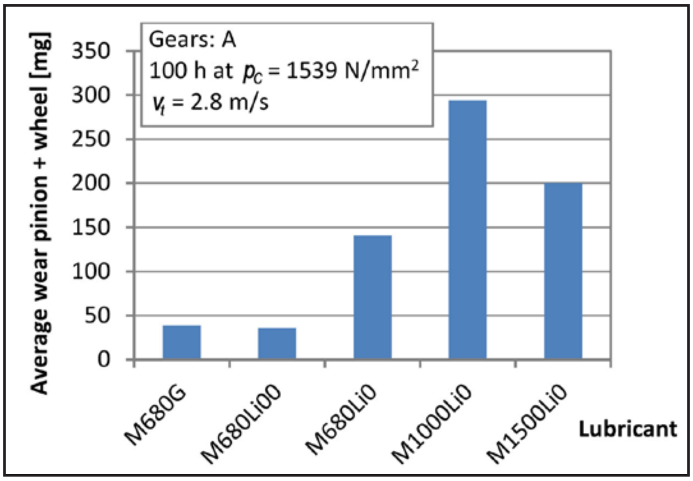


Figure 8 Influence of the base oil viscosity and NLGI grade on the wear behavior of grease with EP additive package (Refs. 3, 14).

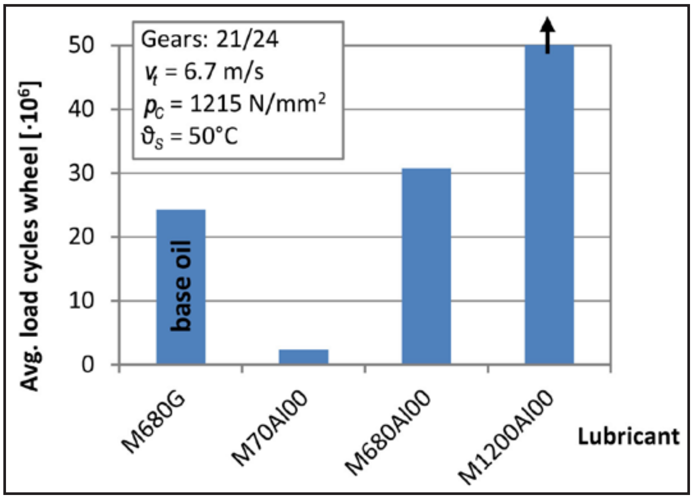


Figure 9 Influence of the base oil viscosity on pitting lifetime of NLGI 00 grease with EP additive package — DGMK 591 (Ref. 3).

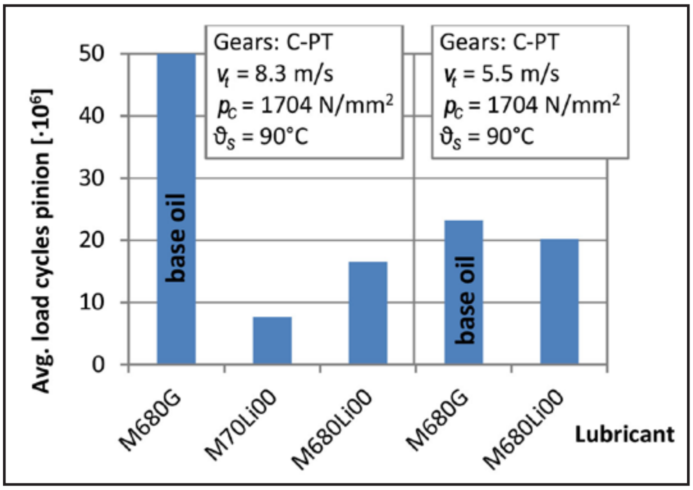


Figure 10 Influence of the base oil viscosity on pitting lifetime of NLGI 00 grease with EP additive package — DGMK 673 (Ref. 4).

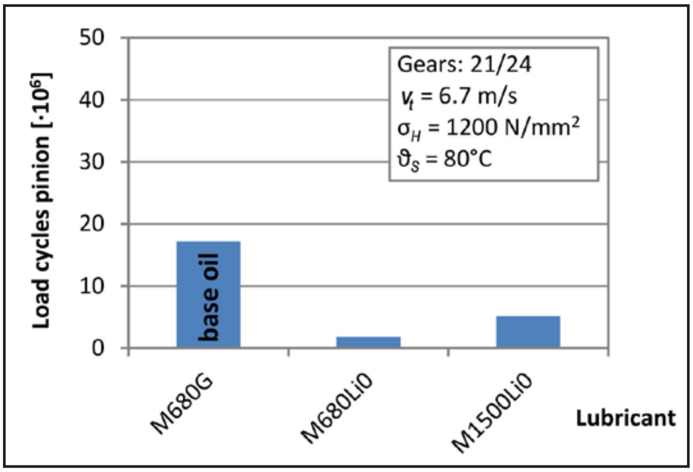


Figure 11 Influence of the base oil viscosity and graphite on pitting lifetime of NLGI 0 grease with EP additive package — DGMK 670 (Ref. 13).

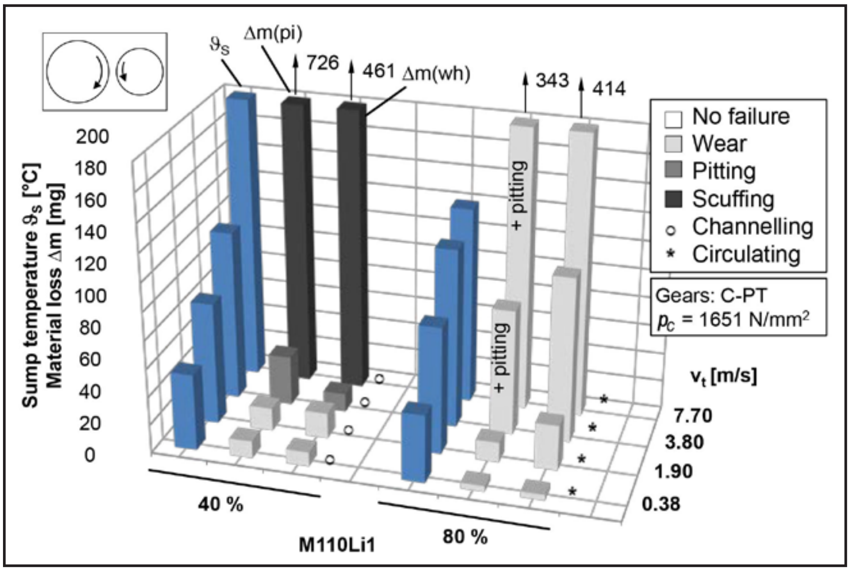


Figure 12 Analysis of sump temperature and material loss for M110Li1 at different speeds and filling levels (Ref. 15).

shown) did not fail and showed the lowest sump temperatures ϑS. With the grease M110Li1, for increasing speed and thus increasing transmitted power and loss torque, increasing sump temperatures are observed. Independent of the fill level, M110Li1 shows no failures at the two lowest speeds, apart from very slight wear. Due to channeling and thus limited heat removal, scuffing occurs at maximum speed for 40% fill-level. At 80% fill-level and circulating, heat removal is better and thus scuffing is avoided but pitting and/or high wear occur instead. At a medium pitch line velocity, $v_t = 3.80$ m/s ($n_1 = 1{,}000\ min^{-1}$), pitting occurred for 40% fill-level. In contrast, for 80% fill-level the gearset shows significant material loss due to wear as well as some pitting. Despite the observed circulating for 80% fill-level, possibly not quite as much grease actually reaches the mesh as initially assumed for this lubrication supply mechanism. The grease in the tests forms a "ring" around the gears (compare Fig. 1), thus evidently "circulating." This leads to better heat dissipation, which averts scuffing at higher speeds; but the question remains on how much grease actually reaches the contact. The high-occurring wear would indicate starved lubrication conditions in the mesh.

Summary

The results of the conducted investigations can be summarized as follows:

- ***Scuffing.*** For test conditions at relatively high speed acc. to ISO 14635-1(Ref. 6), grease shows a lower scuffing load-carrying capacity than its base oil. An increase in the scuffing load-carrying capacity can be achieved by increasing the base oil viscosity of the grease, as well as through the addition of the solid lubricants graphite or MoS$_2$, though this may lead to increased wear. Grease with a lithium soap thickener was found to perform slightly better than grease with an aluminum-complex soap thickener.
- ***Wear.*** At very slow speeds and a circulating lubrication supply mechanism, an increase in the base oil viscosity leads to a reduction in wear. Greases based on PAO-based oil perform slightly better than those based on mineral oil. The addition of the solid lubricants graphite and MoS$_2$ leads to an increase in wear.

  The wear tests acc. to ISO 14635-3 show that an NLGI 0 grease shows higher wear than an NLGI 00 grease, whose wear behavior was shown to be similar to that of the base oil. With the higher NLGI-grade, presumably the lubrication supply mechanism is dominated by channeling effects which lead to insufficient lubricant replenishment of the gear mesh.
- ***Pitting lifetime.*** At the lower rotational speeds (5.5 and 6.7 m/s), the pitting lifetimes of an NLGI 00 grease and its base oil are comparable; at the higher rotational speed (8.3 m/s), however, the pitting lifetime of the grease is lower than that of the base oil. Increasing the NLGI grade also leads to a reduction in pitting lifetime. This can be explained by the lubrication supply mechanism, where circulating dominates at lower speeds, at higher speeds, or higher NLGI grade. However, channeling effects take the upper hand. Furthermore, increasing the

base oil viscosity leads to an increase in the pitting lifetime.

- ***Flank load-carrying capacity of NLGI 1 grease at different pitch line velocities and fill-levels.*** For the lower sump fill-level (40%), channeling occurs. At the highest pitch line velocity this leads to very high temperatures and results in scuffing. For the higher sump fill-level (80%), circulating occurs — resulting in better heat dissipation from the mesh. This leads to significantly lower sump temperatures at the highest pitch line velocity, in comparison to 40% fill-level (channeling) and scuffing does not occur. Higher wear, however, does result. This points to starved lubrication conditions in the gear mesh, despite the evident circulating of the grease occurring in the gearbox.

Conclusions

Overall, grease is suitable for gear lubrication and high load-carrying capacities can be achieved. The lubrication supply mechanism though strongly affects the load-carrying capacity and, therefore, in order to achieve high load-carrying capacities, the choice of grease and a correct analysis of the operating conditions are important. Furthermore, it is necessary to ensure that sufficient grease is available to the gear mesh at all times and that the grease is not destroyed or degraded. All the results shown are from investigations using dip lubrication. In the field, however, applications with spray lubrication are also common.

Currently further research work is being done on the effect that spray lubrication has on the lubrication supply mechanism and the load-carrying capacity of grease-lubricated gears.

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