

Load Capacity and Efficiency of Grease-Lubricated Worm Gears

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Varying installation requirements for worm gears, as, for example, when used in modular gear systems, can necessitate grease lubrication — especially when adequate sealing for oil lubrication would be too complex. Such worm gears are being increasingly used in outside applications such as solar power plants and slew drives. While knowledge about the operating conditions is often appropriate, the basic understanding for load capacity and efficiency under grease lubrication is quite poor. Investigations done at FZG and sponsored by FVA/AiF are shown here to give an impression of the basic factors of load capacity and efficiency. The results of the investigation indicate a satisfying quality of calculations on heat, load capacity and efficiency based on characteristic parameters of the base oil with only slight modifications to the methodology known from DIN 3996 or ISO TR 14521.

Introduction

Regarding some of a worm gearset's basic properties — high achievable gear ratio including the potential of self-locking — it seems quite worthwhile to use them in applications where the basic disadvantages — for lower to medium sizes primarily wear and a complex efficiency — are of minor importance. Worm gears are appropriate in applications absent high shaft speeds; should there be a need for easy sealing, lubricating the worm gearset with greases of higher viscosity (NLGI 2) will become increasingly attractive. Greases, on the other hand, deal with two major disadvantages — 1) a rather bad heat transfer, and 2) absence of cleaning efficiency. Therefore a precise knowledge of load capacity — especially of heat transfer and wear, as well as general information on appropriate greases — was subject to systematic verification. In this context results of worm gearsets with a center distance of $a = 65$ mm are shown to demonstrate the methodology and to indicate promising approaches for future investigation. Another experiment with a cast iron worm wheel indicates potential for greater optimization.

State of the Art

Calculating heat transfer in worm gears is based on the investigations of Neupert (Ref. 1) at stationary conditions that were included in the calculation method according to DIN 3996:2012-09 (Ref. 2). An approach for transient conditions was done by the analysis of Hermes (Ref. 3). Nevertheless all methods consider oil as a lubricant, resulting in consideration of the lubricant being of uniform temperature.

The calculation of wear is also based on the analysis of Neupert (Ref. 1) and has been sub-



Figure 1 Electrically braced test rig at FZG.

Table 1 Basic data for test gears

	Worm	Worm wheel
Material	16MnCr5	CuSn12Ni2-C-GCB
Number of teeth z	2	41
Flank form	I	
Module m	2.5 mm	
Quality	5	6
Heat treatment	58-62HRC, eht 0.4mm	
Roughness R_a	0.3 - 0.5 μm	

Table 2 Basic data for lubricants

Name	Viscosity @40°C [mm ² /s]	Viscosity @100°C [mm ² /s]	Thickener	Percentage of thickener	NLGI	Additive
Polyglycols						
PG1	131	21	Li	8	2	AO,EP
PG2	220	46	LiK	12	2	AO,EP
PG2-GÖ	220	46	-	-	-	
Mineral oils						
MIN1	220	16	LiK	14	2	AO,EP

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jected to several elaborations by, for example, Weisel (Ref. 4), Hermes (Ref. 3), Jacek (Ref. 5) and Nass (Ref. 6). The method is based on a specific wear-intensity-per-sliding-distance, taking into account material, type of oil lubrication, and type of oil, as well as basic operating conditions. Specific values for grease as a lubricant do not as yet exist. The calculation of efficiency as indicated (Ref. 2) is drawn from a base coefficient of friction considering the same influencing factors as for wear. Since all results showed the assignability of these methods, corresponding factors for grease lubrication were defined to be adaptable, as found in Reference 2).

Test Stand Operations

The tests were carried out with a center distances of $a=65\text{ mm}$, but with different greases. Figure 1 shows the used, electrically braced, test rig. Input/output torque, input speed and mass-temperature were measured continuously, as was the wear rate at given numbers of load cycles. The tests were done as load stage tests at different speeds. Cylindrical worm gears according to Table 1 were used.

Table 2 shows the matrix of the tested greases. In total, 21 greases with 12 corresponding base oils were tested, with compositions including polyglycols, polyalphaolefins, and mineral oils. In this paper, basically results of MIN1 and PG2—both sharing the same additive packages and thickeners—will be discussed. Additional lubricants are grease PG1 and PG2's base oil. Additional information can be found in Reference 7.

Test Results

Heat transfer. It is commonly known that greases do not display the thermal behavior of oils. Oils show uniform temperatures at a time being constantly agitated and thus transferring heat from the source to the housing by convection. Unlike that, NLGI-2 greases are conducting the heat.

Figure 2 shows the heat gradient in axis section of the worm, caused by a power loss of 218 W at 150 rpm at the worm. The mass temperature of the worm, being 110°C drops at a rate of almost $1^\circ/1\text{ mm}$.

The major factor to mass temperature is the power loss of the teeth. With a decreased ability of grease to transfer the heat to the housing, mass temperature under grease lubrication is significantly higher than the temperature of splash oil lubricated worm gears. By comparing temperatures of both oil and grease lubricated operational states with the same power loss, Figure 3 shows this fact. It can be seen, that the temperature of grease lubricated gears is approximately 10 to 20°C higher.

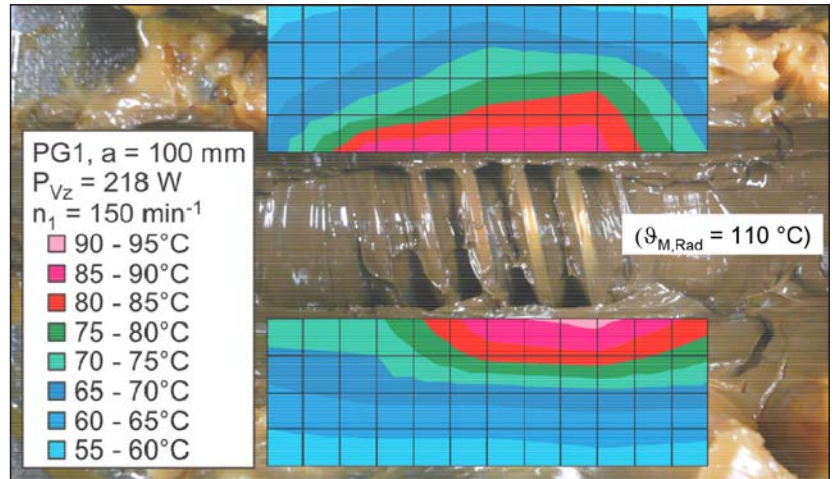


Figure 2 Heat gradient inside gearbox at worm section.

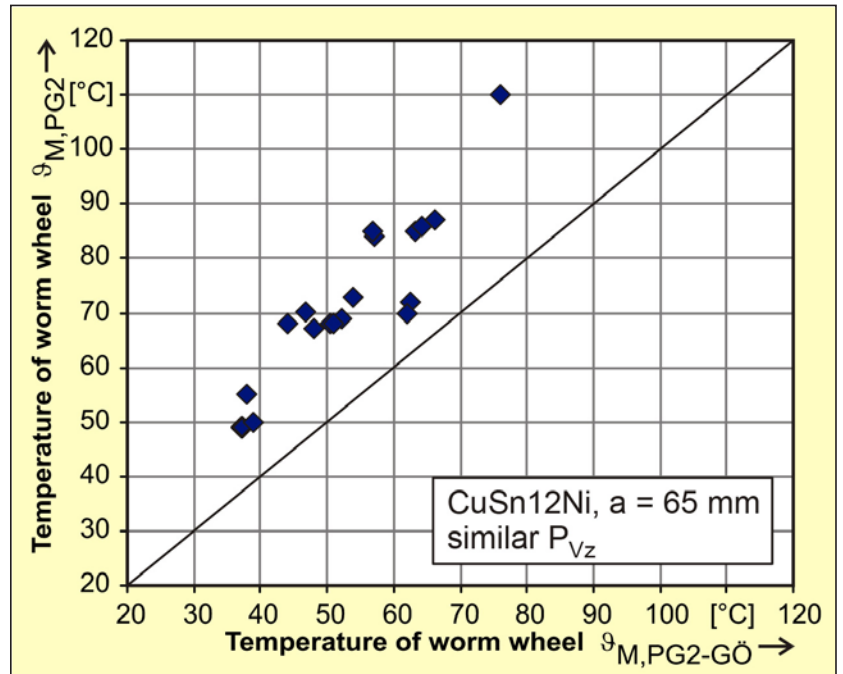


Figure 3 Mass temperature at same power loss for oil and grease lubrication.

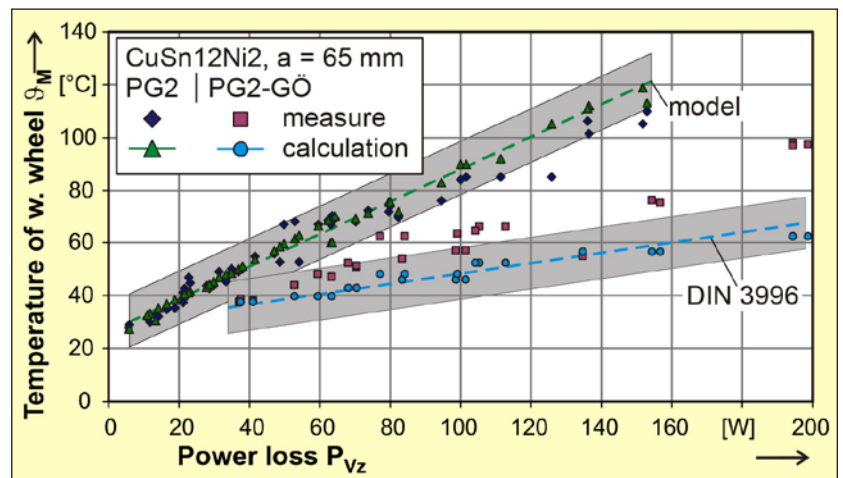


Figure 4 Mass temperature as function of power loss (grease PG2 and oil PG2-GÖ).

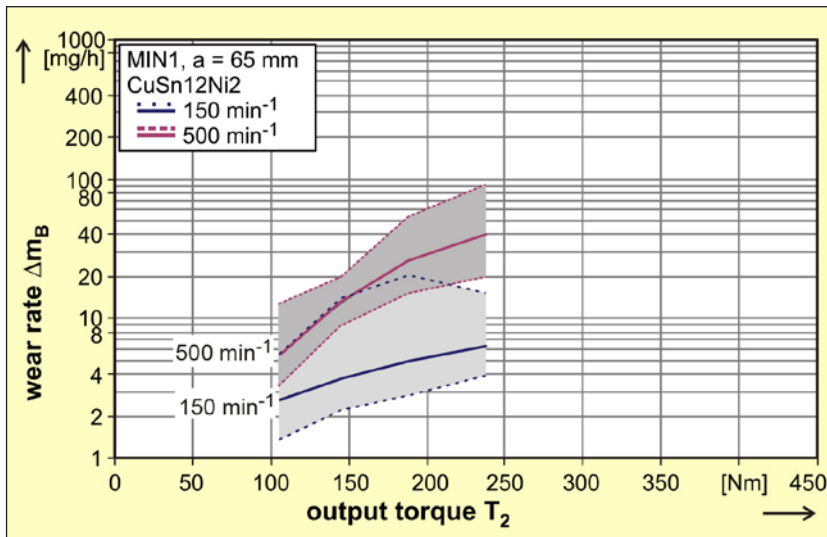


Figure 5 Wear rate for grease MIN1.

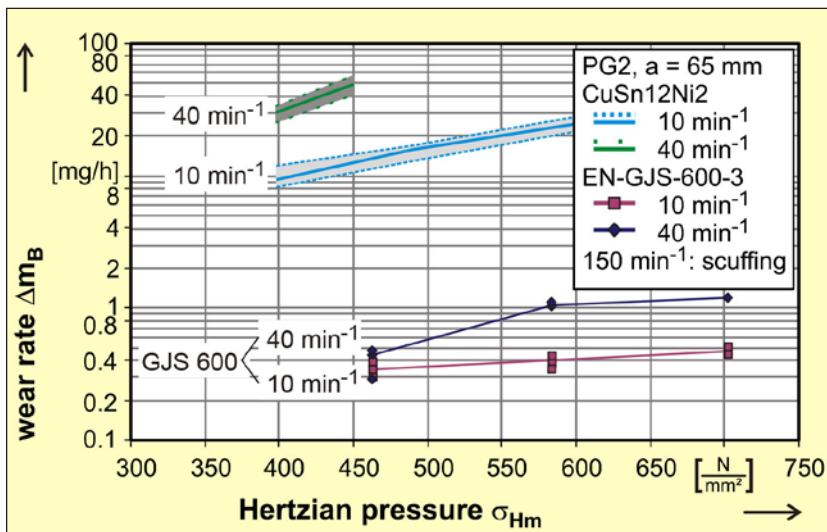


Figure 6 Wear rate for grease PG2, bronze and cast iron.

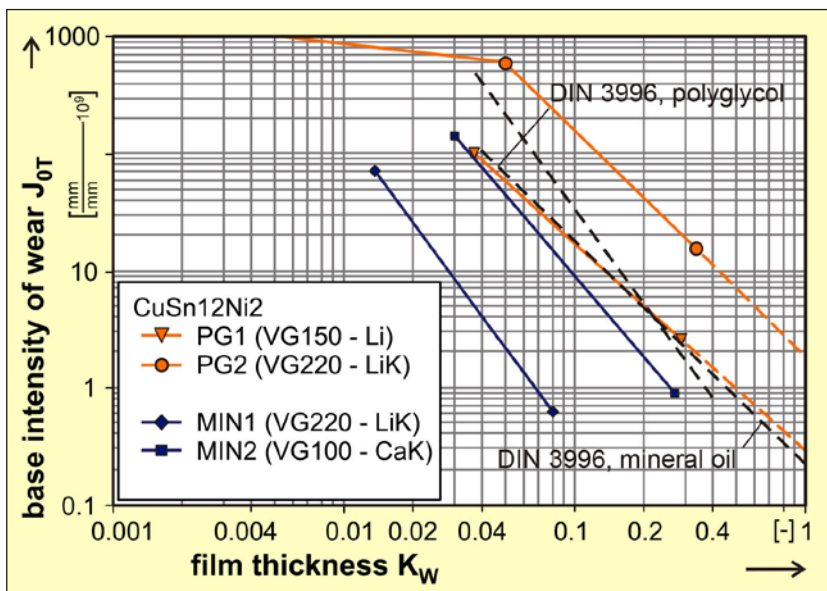


Figure 7 Base intensity of wear — all greases.

Figure 4 shows the comparison of mass temperature for oil and grease lubrication to a corresponding calculation model. While calculation with DIN 3996 (Ref. 2) shows a slight deviation from the measurements, a thermal homologous model calculating conduction over every element of the gearbox shows rather good concurrence. Thus an evaluation of mass temperature as the relevant temperature to calculate EHL film thickness is enabled.

Load Capacity

Figure 5 shows the wear rate in mg per hour as a function of output torque and input speed for grease MIN1 at stationary operating conditions. The results respond to characteristics similar to those of oil lubrication but including a rather high variance. This variance in wear rate is corresponding to a rather high variance in mass temperature. It is quite obvious that a real “steady state” is not reached as a raise and fall of mass temperature still can be found after more than twenty hours of running. The level of wear rate is quite high especially compared to oil lubrication.

Apparently, input speeds of 150 or 500 rpm lead to rather high wear rates. Slower speeds such as 10 or 40 rpm on the other hand show significantly lower wear. A change of the worm wheel material from bronze to cast iron shows significant improvements. Due to scuffing, higher speeds above 150 rpm weren't operable. Nevertheless, variance is much smaller than at higher speeds (Fig. 6).

To enable calculation of wear load capacity, the base wear rate J_{0T} is defined by the results. This method allows comparison of various lubricants — regardless of geometry and operating conditions — and based simply on a specific parameter representing the film thickness. Figure 7 shows these results for the greases MIN1, PG1 and PG2 — all based on Li or LiK-thickener. In addition, grease MIN2 is shown using CaK as a thickener and showing additional improvement of wear characteristics. The greases based on mineral oils show significantly better wear performance than calculated with DIN 3996 (Ref. 2) for oils.

Efficiency

Figure 8 shows the base coefficient of friction for grease PG2, its base oil PG2-GÖ, and the reference according to DIN 3996 (Ref. 2) as a function of mean sliding velocity. It was determined and duplicated that the grease is showing lower values at lower sliding speeds, as demonstrated in the reference.

Conclusion

- It has been proven that under specific operating conditions — especially when using low input speeds — greases may show lower wear rates than corresponding oils, as well as lower base coefficients of friction.
- Regarding the broad variety of lubricants that were subject to investigation (Ref. 7), grease MIN1 (LiK) happens to be a good compromise between wear and efficiency (Fig. 9)
- Taking MIN1 as a starting point, grease MIN2 (CaK) shows better wear characteristics, but higher friction coefficients and a smaller band of service temperature.
- Focusing on efficiency PG2 (LiK) proves to be the better choice, offering even less friction than MIN1. **PTE**

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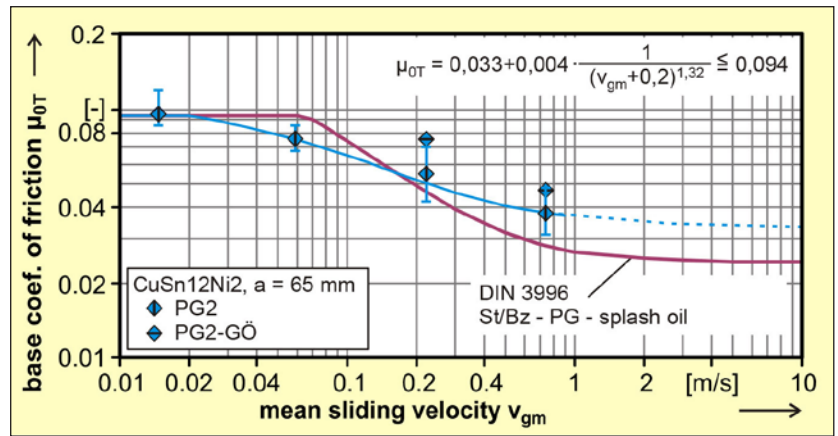


Figure 8 Base coefficient of friction for grease PG2 and corresponding base oil.

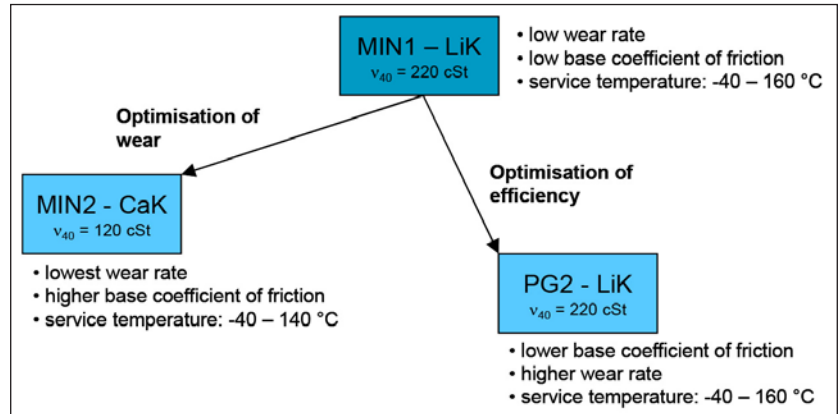


Figure 9 Potential for optimization.

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