THERMAL STABILITY OF CROSSED HELICAL GEARS WITH WHEELS MADE FROM SINTERED STEEL

by

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A considerable slide exists between the flanks of worm and gear during the work, which results in flank wear and considerable loss of energy. The energy is, thereby, converted into heat, which leads to the warming-up of gear drive, compromising its correct operation, and to scuffing in critical cases. Oil temperature has an important role in thermal stability. Viscosity depends on temperature. Viscosity significantly affects the processes in the contact zone, i. e. the energy losses, and hence the temperature. Optimal lubrication can be provided only in the relevant field of temperatures. The paper presents experimental and theoretical research on the effect of temperatures and thermal stability of a worm and gear set with a gear made of sintered steel Fe1.5Cr0.2Mo on their ability and appearance of boundary conditions.

Key words: *thermal stability, oil operating temperature, gear, sintered steel*

Introduction

Worm and gear sets are used, for example, in automotive auxiliary drive units such as window lifters, seat adjustments, windscreen wipers, and in home appliances. Important advantages of worm and gear sets are their easy and inexpensive design, good noise performance, and high ratio that can be realized in one step. In these applications, combinations with pinions or worms made of steel and gear made of plastic are often used. High demands are set on gears made of sintered steel regarding wear and scuffing load capacity. Sintered steel has proven as a very good material for gears in worm and gear sets.

Oil is an important element for worm and gear sets that can provide optimal lubrication up to the critical temperature. Overheating can damage oil. The consequences are the rapid increase of wear at gear, further increase in oil operating temperature ϑ_S, damage of shaft seals, as well as the risk of scuffing damage. Oil operating temperature ϑ_S is important information for gear drive design. The choice of smaller gear drive is economically justified. Gear drive must work in boundary conditions of its load capacity. In particular, the short-term overloading of the worm and gear set must be thermally stable.

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Oil temperature is important for power losses of worm and gear sets. At constant exploitation work (constant input speed and output torque) power losses are constant. In that case, it is possible to establish a direct correlation between the increase in oil temperature and the power losses or friction losses in the contact zone. Oil viscosity affects the oil temperature also. Higher oil viscosity forms a thicker oil film in the zone, but it depends on high-flow which, as a result, has an increase in temperature and reduction of oil viscosity.

Literature review

Hochmann [1] determines the load capacity of material pair steel/steel with grease. The tested gears with grease have lower load capacity than the tested gears with oil. Worm and gear sets have different transmission conditions, therefore, these results cannot be used. The work by Kothoff [2] focuses on sintered steels as material that can have good wear resistance. Most of his experiments are conducted with two disks. He also conducts experiments with gears in which the results that can be compared with steel 16MnCr5 are obtained.

The lubricating film thickness, pitting and scuffing load capacity correlate significantly with the base oil viscosity of grease. The addition of a special synthetic graphite as solid lubricant shows increased wear in this research. The base oil viscosity is a decisive factor for calculating the lubricating film thickness of grease, as well as for calculating the pitting load capacity according to DIN 3990 [3]. Performance data of tested lubricants is available for calculating scuffing and pitting load capacity according to DIN 3990.

The work by Wendt [4] is the first one focusing on the study of load capacity for worm and gear sets with material combination steel/sintered metal. Wendt researches the influence of density on the load capacity of a worm and gear set with over 120 tests during a test period of 300 hours. His work provides approximate equations for the calculation of safety factors for tooth damage such as pitting and wear, while also yielding a lot of experimental results for oil operating temperatures under different loads.

The research conducted by the firm Hoganas AB. Sweden [5] deals with sintered steel Astaloy Mo (Fe0.85Mo) and Astaloy CrL (Fe1.5Cr0.2Mo) with additional treatment. The tooth root resistance of gears with m = 1.5875 mm, z = 18, α = 20° and b = 10 mm is tested. In addition, the rolls with diameters R_1 = 30 mm and R_2 = 70 mm are tested with respect to their fatigue resistance in rolling contact. The results show that sintered steel with the additional treatment can achieve tooth root resistance of steel gears with case hardening. The paper [6] is the first one publishing the results of a worm gear set with gears made of sintered steel Fe1.5Cr0.2Mo. It also focuses on wear damage as primary damage of the gear set.

Test conditions

Gear test rig

The practical tests were carried out by using five test benches with a center-to-center distance of 30 mm. The transmission of the asynchronous motor was mounted on the test bench and the output torque was applied via a magnetic particle brake. Measure points of test benches are shown in fig. 1.

The test transmitter was made of aluminum. The axial section of the worm shaft is shown on the left side of fig. 2, with the axial section of the gear shaft shown on the right. The bearing of worm shaft (1) was achieved with two angular contact ball bearings (3) in X-arrangement. The worm (1) was below the gear (2). The oil operating temperature was measured with a nickel-chromium-nickel thermal element (4). The thermal element (4) measured the operating temperature directly under the worm in the area of gearing contact. The sealing of the test gearbox was provided by a shaft seal (5). The gear shaft (9) was supported with two groove ball bearings (6) (fig. 2, right). The lubrication of the seals closed by bearing allowed sufficient lubrication. The entire worm was immersed in oil.

Table 1. Data of the test gear pair

Parameter	Data
Center distance	30 mm
Module	1.25 (1.252) mm
Transmission ratio	40
Pressure angle	20
Gear material	Fe1.5Cr0.2Mo
Worm material	16MnCr5
Speed	1500-10000 min^{-1}
Torque	12-36 Nm
Lubrication	Oil: Kluber GH6 1500

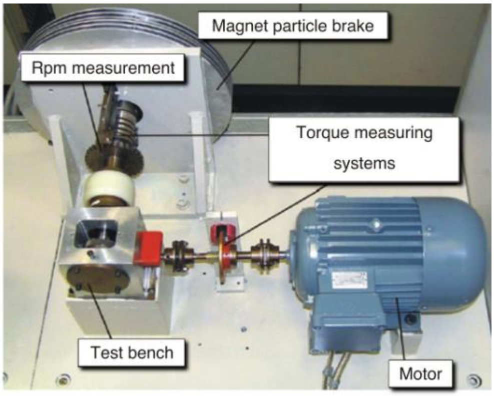

Figure 1. Gear test rig

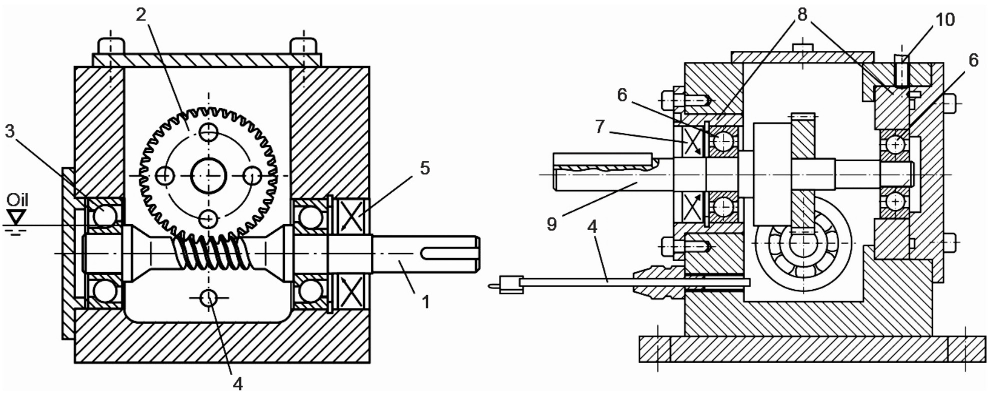

Figure 2. Sectional drawings of test transmission

The time duration of a test run was set to 40 hours. The output torque of the first test run was 12 Nm and for each next test T_2 increased by 4 Nm. After every 20 hours, the helix line was measured in order to determine the width of the wear surface. All data of test gear pair are given in tab. 1.

Material variants

The base of all investigation is the steel based powder Fe1.5Cr0.2Mo. Material variants are given in tab. 2. The properties of sinter steel depend on material density [7]. Pitting occurs in gears with high density . The main form of damage in gears with low density is wear.

Table 2: Material variants of sintered steel and 16MnCr5

	Additional treatment	Density [gcm^{-3}]
S1	Without	7.50
S2	Case hardening	7.49
S3	Case hardening + shot peening	7.49
S4	Pyrohydrolysis	7.50
S5	Sinter-hardening	7.43
S6	2% copper addition	7.43
	16MnCr5	7.85

Figure 3. Worm and gears of different material variants

Thermal stability of worm and gear set

Oil operating temperature

Oil operating temperature ϑ_S has great influence on the creation of oil film in the contact zone. The increase in oil operating temperature reduces its viscosity, while at same time reducing oil film thickness which can lead to the phenomenon of mixed or semi-fluid friction. Oil film thickness has a decisive role in wear and lubrication in the contact zone of a worm and gear set.

Oil operating temperature ϑ_S has great influence on power losses also. The size and characteristics of a contact pattern (roughness, material, accuracy of production) has influence on friction and oil operating temperature.

Temperature difference $\Delta\vartheta_S$ is the difference of oil operating temperature ϑ_S and ambient air temperature ϑ_0:

$$\Delta\vartheta_S = \vartheta_S - \vartheta_0 \tag{1}$$

This eliminates the temperature fluctuations of the environment. Figure 4 shows the change of oil operating temperature ϑ_S and ambient air temperature ϑ_0 during the operation time for synthetic oils D460EP (ν_{40} = 460 mm^2/s) and GH6 1500 (ν_{40} = 1500 mm^2/s), and for output torque T_2 = 24 Nm.

Figures 5 and 6 illustrate the influence of viscosity on the oil operating temperature rise. The temperatures shown here refer to experiments without load for synthetic (S) and mineral (M) oils with different viscosity. These values show the influence of the bearing and seal losses on the temperature, since there is no load, and power losses in the tooth contact are also minimal.

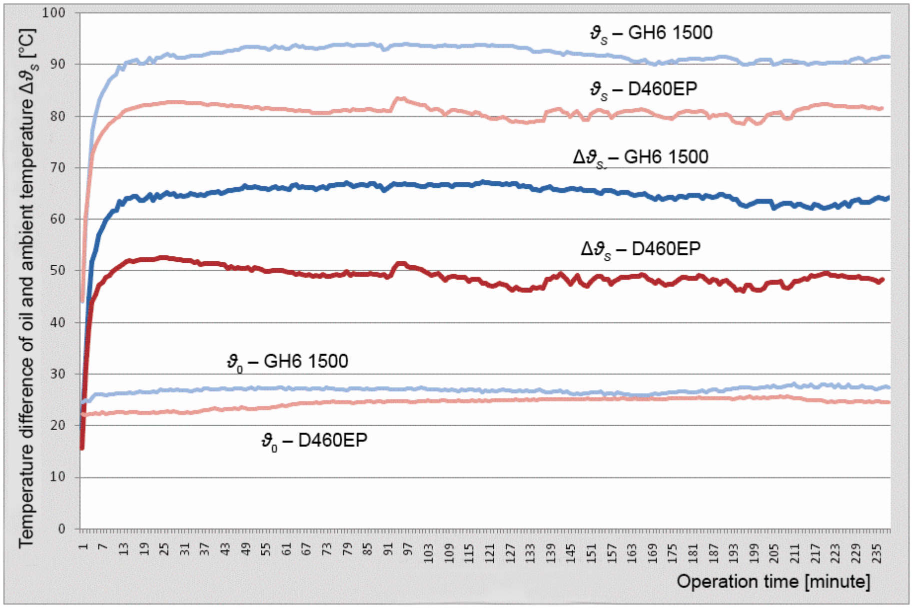

Figure 4. Temperature difference of oil and ambient air temperature for worm and gear set and T_2 = 24 Nm

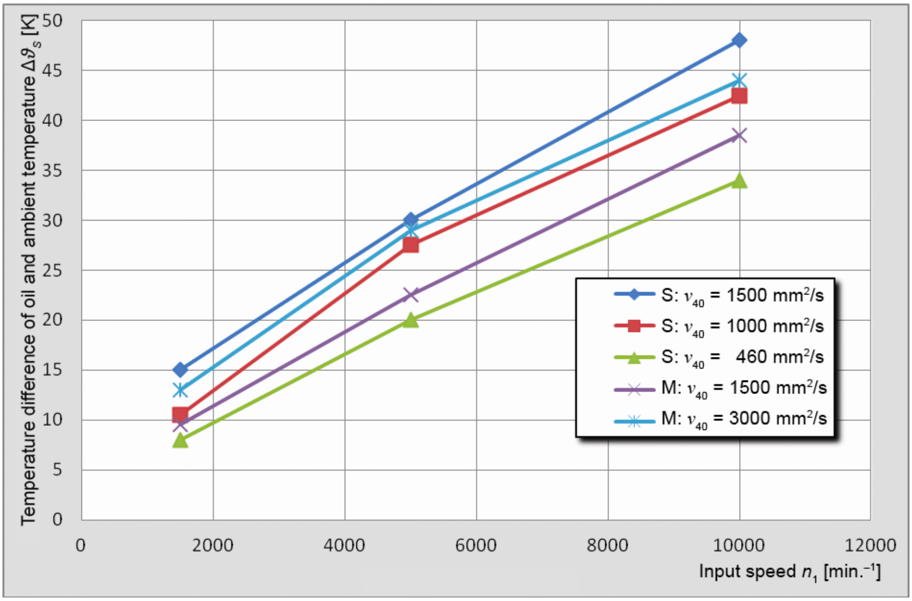

Figure 5. Temperature difference between oil and ambient air temperature for synthetic and mineral oil (S and M) depending on viscosity and input speed n_1

According to figs. 5 and 6, the highest temperature difference between oil and ambient air temperature $\Delta\vartheta_S$ for synthetic oils with the viscosity ν_{40} = 1500 mm^2/s occurs at 15 K to 48 K. The highest temperature difference between oil and ambient air temperature $\Delta\vartheta_S$ for mineral oils with the viscosity ν_{40} = 3000 mm^2/s occurs at 13 K to 44 K. The lowest temperature difference between oil and ambient air temperature $\Delta\vartheta_S$ for synthetic oils with the viscosity ν_{40} = 460 mm^2/s is in the range from 8 K to 34 K.

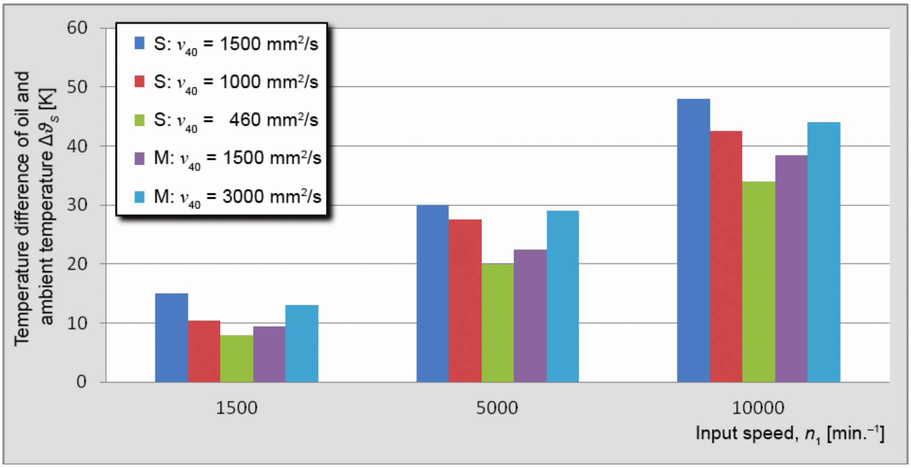

Figure 6. Temperature difference between oil and ambient air temperature for synthetic and mineral oil (S and M) depending on viscosity and input speed n_1

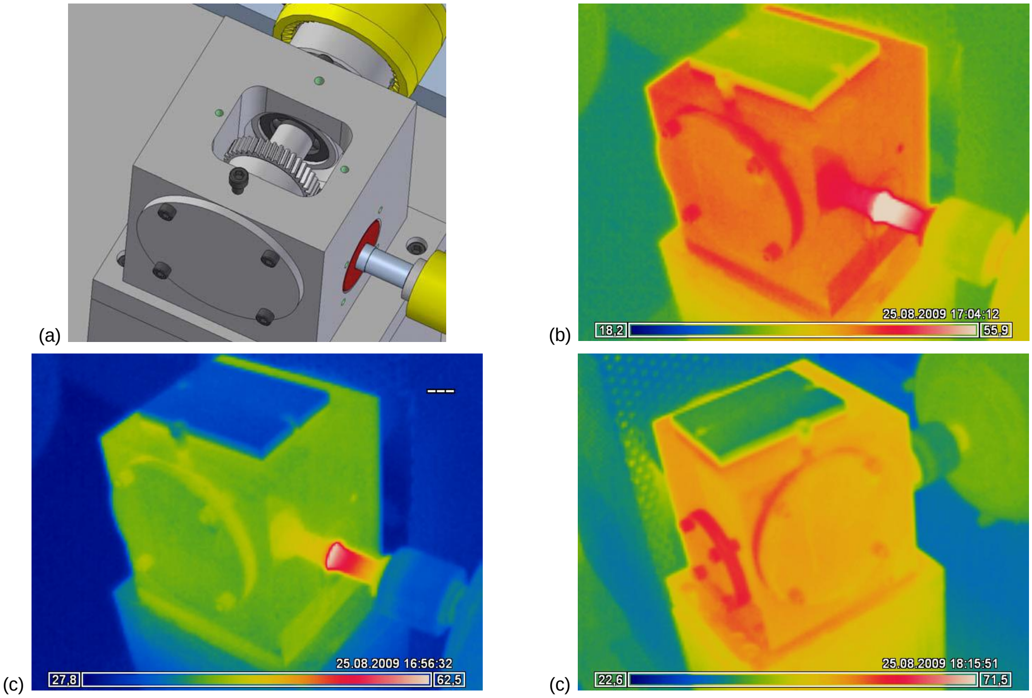

Figure 7. 3-D view and thermal image of the test gear drive [°C]

Figure 7 (a) shows a 3-D view of the test gear drive and fig. 7 (b, c, d) show thermal images of the test gear drive. The input speed at the beginning of the experiment is

5000 min.$^{-1}$. The highest local temperature outside of the teeth occurs at the drive-side shaft sealing ring, fig. 7 (b, c). On the opposite side of the drive shaft, the higher local temperature occurs at the cover on the worm shaft.

Temperature difference between oil and ambient air temperature

The lubrication of sintered steel gears in the experiments was carried out with oil. The oil was always between the teeth in contact, causing the occurrence of a better dissipation of heat flow from the teeth than in the case of grease lubrication. This resulted in significantly lower tooth temperatures and a reduction in wear. Temperature difference between oil and ambient air temperature $\Delta\vartheta_S$ can be determined by eq. 2. It depends on the power losses P_V [kW], the heat transfer coefficient k_g [Wmm^{-2}K^{-1})] and the surface of the gearbox A_g [mm^2].

$$\Delta\vartheta_S = \frac{P_V}{k_g A_g} = \frac{P_{VL} + P_{VD} + P_{VZ}}{k_g A_g} \tag{2}$$

The mesh power loss P_{VZ} can be determined as a function of the tooth friction coefficient μ_{zm}, the helix angle on gear β_{s2}, and the output power P_2 according to eq. 3:

$$P_{VZ} = \left[\frac{\mathrm{tg}\left(\beta_{s2} + \mathrm{arctg}\,\mu_{zm}\right)}{\mathrm{tg}\,\beta_{s2}} - 1 \right] P_2 \tag{3}$$

The tooth friction coefficient μ_{zm} can be calculated with eq. 4 [7]:

$$\mu_{zm} = A_0 \sin\beta_{s2} + A_1 (30 \sin\beta_{s2} v_{gs})^{A_2} + A_3 T_2{}^{A_4} + A_5 \left(\frac{v_{gs} T_2}{30} \right)^{A_6} + A_7 \left(\frac{T_2}{24.384 \cdot 10^{14}} \right)^{A_8} \tag{4}$$

The non-linear regression analysis provides coefficients of eq. (4), taken from [7], are: A_0 = 101.9411, A_1 = –0.424684, A_3 = –0.368435, A_4 = 0.030192, A_5 = –37.23798, A_6 = 0.002729, A_7 = 4.933594, and A_8 = –0.027481.

The sliding speed on the gear base circle v_{gs} [ms^{-1}] is calculated according to eq. (5) and it depends on the gear base circle r_{s1}, the angular velocity ω_1, and the helix angle of the gear base circle β_{s2} according to eq. (5):

$$v_{gs} = \frac{r_{s1} \omega_1 \sin\Sigma}{\cos\beta_{s2}} \tag{5}$$

The bearing and seal power losses can be calculated according to eq. 6:

$$P_{VLD} = P_{VL} + P_{VD} = B_0 \left(\frac{n_1}{60} \right)^B \left(\frac{\nu_{40}}{1000} \right)^B + B_3 T_2 \left(\frac{n_1}{60} \right)^B \tag{6}$$

The bearing and seal power losses can be calculated according to the new SKF-method [8]. But a direct calculation of bearing power losses with the SKF-method as part of the load capacity calculation is not possible because temperature data are necessary. Equation 6 is an approximate equation which allows the calculation of the temperature as a function of known parameters. The SKF-method was applied in the calculation of the bearing power losses from the test results. The approximate equation was thus based on values that were calculated with the SKF-method. The input speed n_1, output torque T_2 as well as the nominal viscosity of the lubricant v_{40} directly affected the power losses. The coefficients of eq. (6), taken from [7], are: $B_0 = 0.6829$, $B_1 = 0.8828$, $B_2 = 0.0288$, $B_3 = 0.3347$ and $B_4 = 2.33$.

The heat transfer coefficient depends on the input speed n_1 and it is obtained approximately from eq. 7 that is based on the experimental results. A dependence of the heat transfer coefficient on the viscosity is not represented, since the viscosity depends mainly on the temperature [6]:

$$k_g = 0.1771 \cdot (\log n_1)^{4,525} + 9.9131 \tag{7}$$

The surface of test gear drive is calculated according to the following equation:

$$A_g = 2(l_g h_g + b_g h_g) + l_g b_g \tag{8}$$

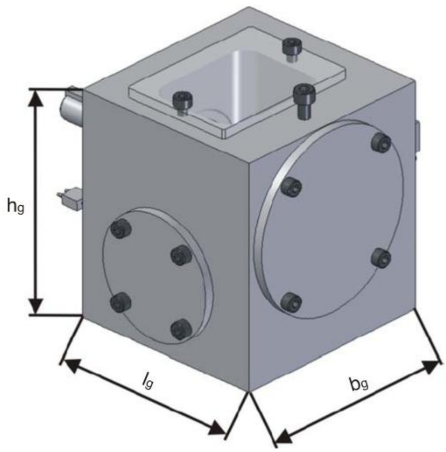

Figure 8. Dimensions of test gear drive

Figure 8 presents the test gear drive that was used in the experiments.

Figure 9 shows the temperature difference between oil and ambient air temperature $\Delta\vartheta_S$ over the output torque T_2 for all experiments with gears made of different materials. The figure shows that there is usually a linear relationship between the temperature difference between oil and ambient air temperature $\Delta\vartheta_S$ and the output torque T_2. Exceptions are the material variants S_2 and S_3.

Oil operating temperatures for increased output torque resulted from the higher power losses.

The lowest temperature difference between oil and ambient air temperature $\Delta\vartheta_S$ occurred, according to fig. 9, for the material pair steel/sintered steel material variants with structure of material with sinter-hardening additional treatment (47.6 K to 70 K), with pyrohydrolisys additional treatment (50.7 K to 61.5 K) and with 2% copper addition (56.2 K to 66.3 K).

The highest temperature difference between oil and ambient air temperature $\Delta\vartheta_S$ occurred for the material pair steel/sintered steel with case hardening as additional treatment (56.2 K to 81.7 K) and the additional treatments of case hardening and shoot peening (57.3 K to 76.5 K). Figure 10 shows the time course of temperature difference between oil and ambient air temperature for different material variants and $n_1 = 5000$ min.$^{-1}$. The highest temperature difference between oil and ambient air temperature $\Delta\vartheta_S$ and the highest increase occurred for the material pair steel/sintered steel with case hardening as additional treatment.

Temperature safety coefficient

The temperature safety coefficient is the safety against exceeding the critical temperature of lubrication. Critical oil temperature ϑ_{Slim} for oil lubricant Klubersynth GH6 1500 is 160 °C. The definition of temperature safety coefficient is:

$$S_T = \frac{\vartheta_{S\lim}}{\vartheta_S} > S_{T\min} \tag{9}$$

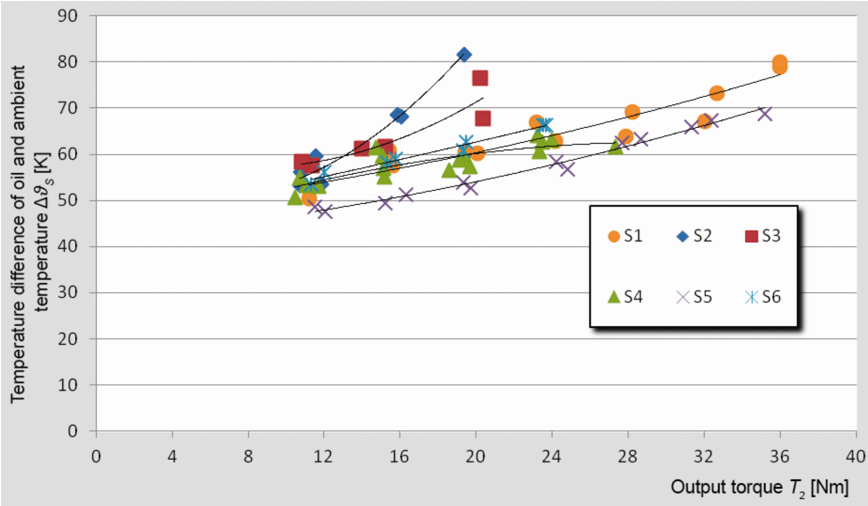

Figure 9. Comparison of measured temperature difference between oil and ambient air of worm and gear set with gear made of sintered steel with different additional treatments for n_1 = 5000 min.$^{-1}$

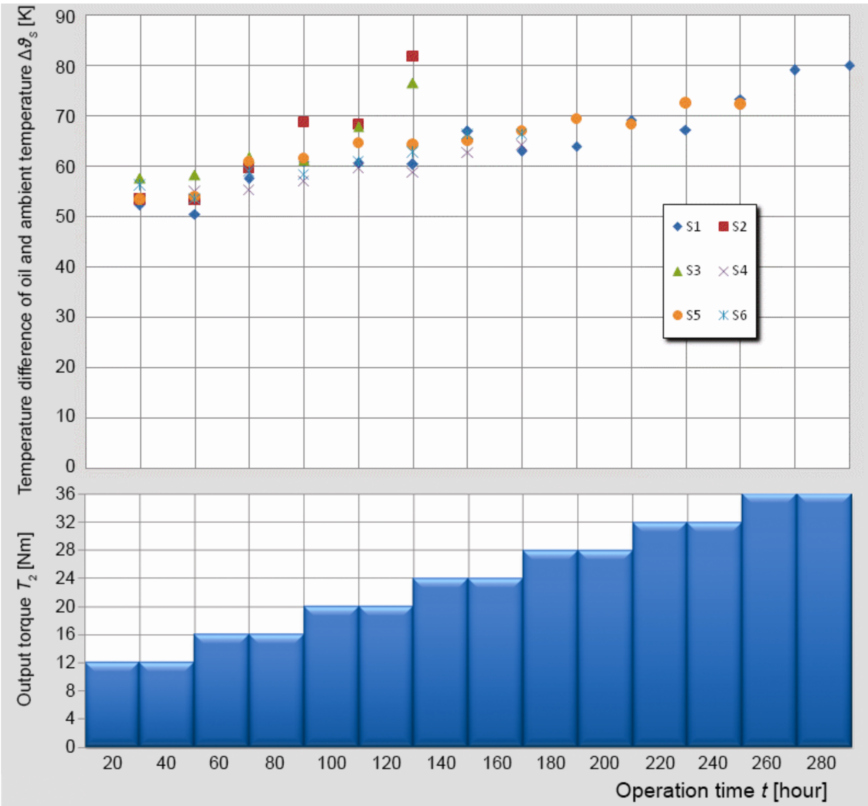

Figure 10. Temperature difference between oil and ambient air as function of operational time for different material variants and n_1 = 5000 min.$^{-1}$

The minimal temperature safety coefficient is obtained from the difference between calculated and measured temperature difference between oil and ambient air temperature. Figure 11 gives the distribution of quotient $\Delta\vartheta_{S\_measured}/\Delta\vartheta_{S\_calculated}$ for all experiments. Furthermore, the absolute frequency, the density function $\varphi(z)$, and the distribution function $F(z)$ are listed as well.

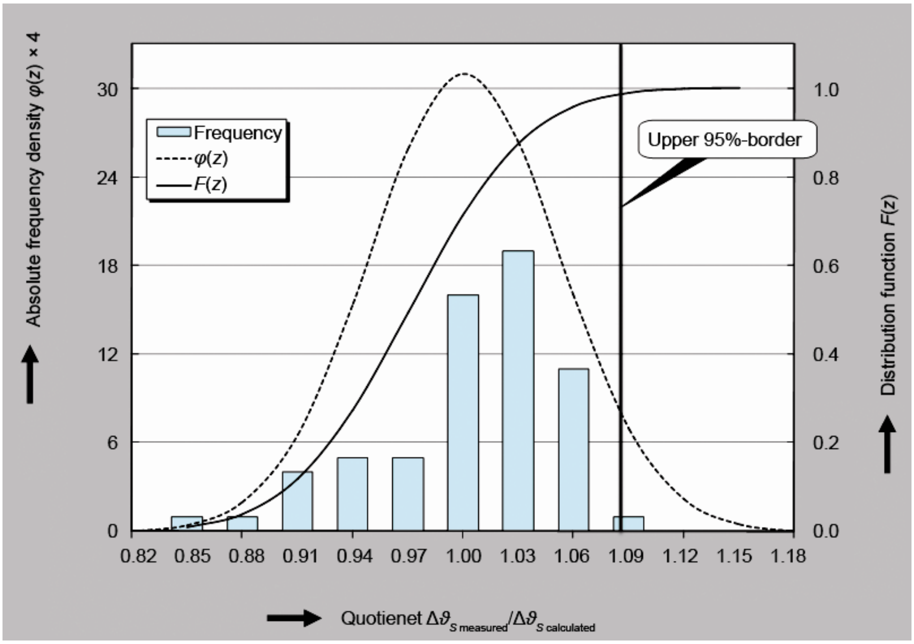

Figure 11. Distribution for determination of minimal temperature safety coefficient

The confidence border of 95% gives the minimal temperature safety coefficient. Other parameters for calculation of S_{Tmin} are given in tab. 3.

The minimal temperature safety coefficient S_{Tmin} = 1.1 is obtained from the determined data.

Table 3. Statistical data for eq. 9

Expected value	1.001
Standard deviation σ	0.05154
Upper 95 %-border	1.0856
Minimal temperature safety coefficient S_{Tmin}	1.1

Damages due to temperature instability

Wear

If during the work of a worm and gear set critical oil temperature is exceeded, oil viscosity and oil film thickness are reduced, which can lead to mixed or semi-fluid friction. Consequences are wear and the risk of scuffing damage of teeth flanks.

According to DIN 50320 [9], the wear describes the continuous loss of material from the surface of the basic body which has a relative movement with respect to a solid, liquid or gaseous mating with which it is in contact. Wear has exclusively mechanical causes. Different from hardness or tensile strength, wear is not a specific material property but a system property which depends on the particular tribological system. In our case, the elements of the tribological system are: the gear (basic body), the worm (opposed body) and the lubricant (intermediate component).

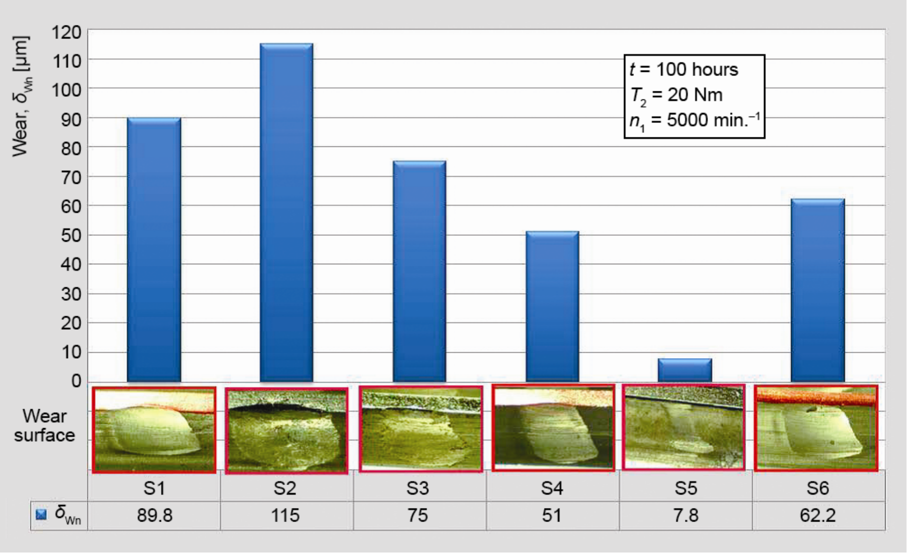

Figure 12. Wear rate δ_{wn} for all experiments with different material variants [10]

Worm and gear are in contact in a point. During operation, a change in the tooth flank of the gear appears due to wear. The worm forms on the tooth flank of the gear, a wear surface that has a shape that is identical to the worm gear flank. Wear progress widens the wear surface, which leads to a lower Hertzian pressure in the tooth contact. After a certain period of operation under intensive wear progress, the steady state occurs, where a necessary oil layer exists, so that the wear progress is minimal.

Figure 12 compares the wear rates of all experiments with gears made of different material variants after the test period of 100 hours and the output moment of 20 Nm. The maximal wear rate δ_{wn} occurred in the experiments with the gear made of material S2 – material variant with case hardening, and it was 115 μm. The minimal wear rate δ_{wn} occurred in the experiments with the gear made of material S5 – material variant with sinter-hardening as additional treatment, and it was 7.8 μm.

If the critical oil temperature is exceeded and mixed or semi-fluid friction appears, then the wear damage becomes critical.

Scuffing

There is a difference between cold and warm scuffing. Both damage types are caused by the lack of lubricant in contact between teeth. Cold scuffing is relatively rarely seen. It occurs mainly at low speed (<4 m/s) and between teeth that have relatively high hardness and rough quality of contact surfaces.

Warm scuffing occurs due to great pressure and high sliding velocity between tooth flanks. Under such a load, combined effects occur which lead to the increase in temperature that disrupts the lubricant film between tooth flanks, making the contact between tooth flanks direct and dry. This can cause a short local welding of the flanks which damages both flanks. Warm scuffing is characterized by strip-shaped bands in the direction of the tooth height, and with the strongest expression in the tooth addendum and tooth root. Scuffing on high-speed gears increases the temperature and tooth forces, eventually leading to shaft fracture due to high damage on tooth flanks.

With exception of S1, scuffing occurred on all material variants. Under output torque of T_2 = 28 Nm, and input speed n_1 = 5000 min.$^{-1}$ scuffing occurred for S4 and S6. Gear loads rose and, suddenly, the fracture of worm shaft occurred.

On material variant S4 "pyrohydrolysis", scuffing occurred on tooth root in trials with input speed n_1 = 5000 min.$^{-1}$ and output torque of T_2 = 16 Nm, and a very short running time of ~10 minutes. Scuffing also occurred on gear tooth flanks with speed n_1 = 5000 min.$^{-1}$ and output torque of T_2 = 36 Nm for material variant S5 sinter-hardening. On trials with input speed n_1 = 10000 min.$^{-1}$ scuffing also occurred under smaller output torque.

Conclusions

The research in this paper shows the importance of oil operating temperature and temperature stability of a worm and gear set for the normal work of gear drive. Operating temperature of oil depends on many factors, such as: oil type, flank roughness, sliding speed of contact surfaces, additional treatment of sintered steel, load or Hertzian pressures of contact surfaces, *etc*.

Increased viscosity of mineral and synthetic oil increases the operating temperature of gear drive. For the same viscosity of mineral and synthetic oil, the operating temperature is higher for synthetic oil.

Higher roughness of flanks in contact increases the power losses and oil operating temperature. Higher sliding speed of contact flanks increases oil operating temperature. The highest oil operating temperature occurred in experiments with the material combination steel/sintered steel with the additional treatment of case hardening and shoot peening. The higher trend of increasing oil operating temperature occurred with the higher loads for this material combination. The lowest oil operating temperature occurred for the material combination steel/sintered steel with the additional treatment of sinter-hardening.

The highest wear rate for identical test conditions occurred for the gear made of sintered steel with the additional treatment of case hardening and shoot peening. The lowest wear rate occurred for the gear made of sintered steel with the additional treatment of sinter-hardening.

With the exception of the gear made of sintered steel without additional treatment, scuffing occurred on all material variants as a result of higher operating temperature. The lowest scuffing resistance was displayed by the gears made of sintered steel with the additional treatment of pyrohydrolisys and with 2% copper addition.

The paper presents, based on conducted research, the calculation to determine the temperature safety coefficient of a worm and gear set with gears made of sintered steel.

Nomenclature

b – gear width, [mm]
b_g – width of test gearbox, [mm]
h_g – high of test gearbox, [mm]
l_g – length of test gearbox, [mm]
m – module, [mm]
S_T – temperature safety coefficient, [–]
z – number of teeth, [–]

Greek symbol

ν_{40} – viscosity at 40 degrees, [mm^{2}s^{-1}]

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