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PERMISSIBLE TEMPERATURES AND THERMAL STABILITY OF WORM GEARS

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Abstract: In comparison to other types of transmission devices, worm gearsoffer a number of advantages which allow fora wide scope of applications in the fields of power and movement transmission. A deficiency of worm gears is a relatively low level of efficiency, especially in extreme operating conditions primarily related to the high transmission ratio and high rotation frequency. Namely, between the coupled flanks of a worm screw and worm wheel there is a significant amount of sliding which results in wear. Consequently, a substantial amount of mechanical energy is converted into heat, i.e. the thermal stability of a worm gear is disrupted as a result of warming. Thereby, the lubricating features of oil are changed, which subsequently results in the destruction of teeth and damage of seals. By means of an experiment and analysis the authors of the paper have determined the operating temperature of worm gears and performed an analysis of their thermal stability.
Keywords: worm gears, thermal stability, oil sump temperature

1. INTRODUCTION

Worm gears offer a number of advantages in comparison to other types of transmission, which allows for a wide scope of applications in the fields of power and movement transmission. They are widely used in machine tools, transport equipment, in vehicles, primarily for the purposes of power transmission, as well as in fine-tuning and precision devices for movement transmission. They are widely used as actuators of the short-term propulsion of small devices.

Thermal stability of worm gears plays an essential role in their proper operation and compliance with the anticipated operating functions, and it mostly depends on a lubricant. Between the coupled flanks of a worm screw and worm wheel there is a significant amount of sliding. Thereby, a substantial amount of mechanical energy is converted into heat. In the event of energy losses, an important role is played by the applied oil and its operating temperature. Oil viscosity depends upon temperature. However, the oil viscosity itself significantly influences the processes in the contact zone, and therefore the temperatures. The optimum lubrication for the chosen oil can be secured only within the appropriate range of the operating temperature.

2. OIL SUMP TEMPERATURE AND POWER LOSSES

The oil sump temperature ϑ_S exerts the greatest impact over the formation of an oil film in the contact zone. An increase in the oil sump temperature results in a decrease of oil viscosity and, simultaneously, in a reduction of the oil film thickness, thus running a risk of mixed or semi-liquid friction. Hence, the thickness of the oil film is decisive in the events of wear and lubrication in the contact zone of a worm gear.

The oil sump temperature ϑ_S also influences the energy losses to the entire transmission. The size and characteristics of the contact surface in the coupling zone (roughness, type of materials, production accuracy) have impact on friction, and therefore on the operating temperature.

The oil sump over temperature $\Delta\vartheta_S$ is the result of the difference of the oil sump temperature ϑ_S and ambient temperature ϑ_0 (Figure 1):

$$\Delta\vartheta_s = \vartheta_S - \vartheta_0 \quad (1)$$

Variations in ambient temperature also affect the operating temperature, i.e. the oil viscosity. Lubrication of worm gears is most frequently performed with oil, because it smoothly reaches the contact zone and easily produces the required oil film. This leads to wear reduction.

For the purposes of determining energy losses in a worm gear pair it is necessary to primarily determine the losses in bearings and seals. If the losses in bearings and seals are marked with P_{GLD} then total energy losses in a worm gear are as follows:

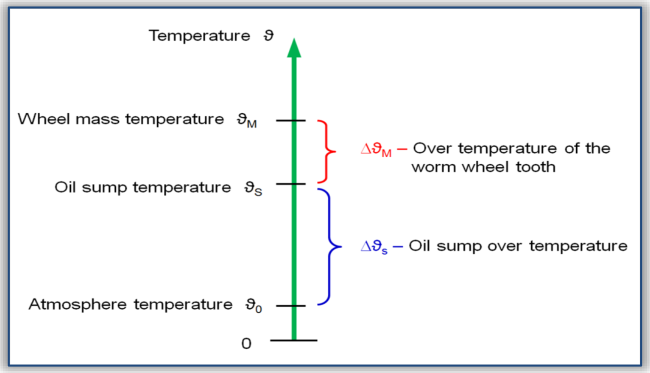

Figure1. Temperatures relevant to the calculation

$$P_G = P_{GO} + P_{GLD} + P_{Gz} \quad (2)$$

where P_{G0} stands for idling energy losses, and P_{Gz} stands for energy losses in a worm gear pair.

An oil sump over temperature $\Delta\vartheta_S$ is a consequence of power losses in a worm gear pair, as well as in bearings and seals. In order to maintain thermal stability of a worm gear, the power losses in a worm gear pair are transferred to the environment by means of oil and casing, i.e. the following applies:

$$P_G = \Delta\vartheta_S \cdot k_G \cdot A_G \quad (3)$$

In the expression (3) k_G indicates the heat transfer coefficient, and A_G is the casing surface relevant to the transfer of heat into the environment.

Accordingly, an oil sump over temperature $\Delta\vartheta_S$ can be determined according to the following formula:

$$\Delta\vartheta_S = \frac{P_G}{k_G \cdot A_G} \quad (4)$$

During the coupling as a result of friction, the coupled teeth of a worm gear pair heat up. The wheel mass temperature is, therefore, higher in comparison to the oil sump temperature for $\Delta\vartheta_M$ (Figure 1), i.e. it is as follows:

$$\vartheta_M = \vartheta_S + \Delta\vartheta_M \quad (5)$$

An increase in the gear tooth temperature $\Delta\vartheta_M$ depends on the power losses in the worm gear pair P_{Gz}. According to the DIN 3996 [3], it is determined as follows:

$$\Delta\vartheta_M = \frac{P_{Gz}}{\alpha_L \cdot A_R} \quad (6)$$

Symbols: P_{Gz}-Power losses in the worm gear pair; α_L- The heat transfer coefficient; A_R - The surface relevant for cooling down of the worm gear pair

The power losses in the worm gear pair P_{Gz} have been experimentaly determined depending on the load, i.e. they can be determined if the total power losses in the worm gear are reduced by the losses in bearings and seals:

$$P_{Gz} = P_1(1 - \eta_\Sigma) - P_{GLD} \quad (7)$$

According to the DIN 3996 [3] the heat transfer coefficient α_L is determined depending on the input rotation speed n_1, according to the following formula:

$$\alpha_L = c_k \cdot (1940 + 15 \cdot n_1) \quad (8)$$

The coefficient c_k equals $c_k=1$, if the worm wheel is immersed, i.e. it equals $c_k=0.8$ if the worm screw is immersed in oil.

The relevant cooling surface of the worm gear pair A_R is determined depending on the width b_{2R} and the medium circle diameter d_{m2}, according to [3]:

$$A_R = b_{2R} \cdot d_{m2} \cdot 10^{-6} \quad (9)$$

In order to secure reliable functioning of a worm gear it is necessary to maintain the oil sump temperature within acceptable limits. A thermal overload results in oil damage. As a consequence, one faces a sudden increase in a worm gear wear, a further oil sump over temperature ϑ_S, damage to the shaft seal, as well as an increased risk of scuffing. In practice, this implies damage and impairment of a worm gear. It is the knowledge of oil temperature ϑ_S that is essential when defining a worm gear. A

worm gear needs to operate within its permitted capacity. This especially applies to mobile operations with short-term overloads, when it is necessary to ensure the thermal stability of a worm gear.
The oil temperature is relevant even in the event of energy losses in worm gears. In stationary conditions (constant speed and constant torque) energy losses are constant as well. In that case, it possible to establish a direct correlation between the oil sump over temperature and the ambient temperature and energy losses, i.e. the losses due to friction in the contact zone. The oil viscosity itself has impact over the oil sump temperature. The oils with a larger degree of viscosity form a thicker oil film in the lubrication zone, but since it is caused by a high flow this also results in an increased temperature and lower viscosity.

3. EXPERIMENTAL DETERMINATION OF PERMISSIBLE TEMPERATURES

Transmission Gear Data

For the purposes of the experiment the authors used a worm gear with axial distance of 30 mm. The axial cross-section of a worm gear has been shown in Figure 1b. The worm shaft was positioned under the worm gear. The test bed was driven by a 2.5 kW induction motor. The output torque was transmitted via magnetic coupling which can endure up to 160Nm torque. Both input and output transmission connectors were of a curved-tooth gear coupling kind. The output torque was measured by a measuring shaft with slip rings. This experimental research on worm gears was conducted regarding operational conditions with transmission ratio of i = 40 and input speed n_1 = 5000 min^{-1} (Table 1).
The measurement of the worm screw temperature at the input side of the gear is performed by means of infra-red thermal radiation (infra-red thermocouple). The oil sump temperature in the worm gear is measured according to the Ni-Cr-Ni thermocouple. The acquisition of measuring data is performed via Dolphin technology.

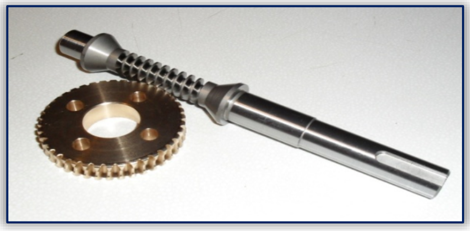

Figure 2a. Worm pair

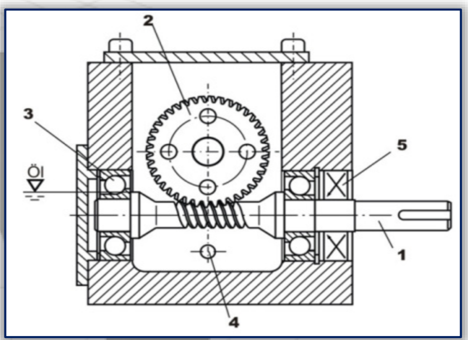

Figure 2b. Test transmission: 1 – worm shaft; 2 – worm gear; 3 – angular contact ball bearing; 4 – thermocouple; 5 – worm shaft seal

Table 1. Transmission gear data

Geometrical size	Values
Central distance a [mm]	30
Worm gears type	ZI- DIN 3975
Transmission ratio	40
Module m_x [mm]	1,2608
Number of teeth z_1/ z_2	1/40
Wheel material	CuSn12Ni2-C-GCB
Worm material	16MnCr5
Input speed [min^{-1}]	5000
Synthetic oil GH6-1500	ν_{40} = 1500mm^2/s; ν_{100}= 232mm^2/s

Results

On the basis of the measured values of torques and rotation speeds at the input (n_1, T_1) and output worm shafts (n_2, T_2), as well as on the basis of the measured oil temperature ϑ_S and ambient temperature ϑ_0, Table 2 presents the results of relevant parameters of the worm gear. The power losses in bearings and seals are determined according to the SKF calculation procedure [1], which provides rather sound calculation results. Other values have been determined according to the previously presented expressions (1) - (9). The calculation has been performed for eleven load levels whose values are given in Table 2.
In the experimental worm gear the coupling zone between the worm screw and worm wheel is constantly immersed in oil, so one may assert that lubrication and heat transfer are satisfactory.

Table 2. The results obtained by means of experiment and analysis

No	P_1	P_2	P_G	T_1	T_2	P_{GLD}	P_{Gz}	ϑ_0	ϑ_S	$\Delta\vartheta_S$	$\Delta\vartheta_M$	ϑ_M
	[W]	[W]	[W]	[Nm]	[Nm]	[W]	[W]	[°C]	[°C]	[°C]	[°C]	[°C]
1	257.20	132.73	124.46	0.49	10.14	21.97	102.49	23.55	71.27	47.72	4.065834	75.33583
2	272.08	146.21	125.87	0.52	11.17	23.02	102.85	25.44	76.30	52.86	4.079894	80.37989
3	266.62	160.35	106.26	0.51	12.25	24.02	82.25	29.78	77.48	44.7	3.262619	80.74262
4	306.83	193.08	113.75	0.59	14.75	26.59	87.16	24.55	77.32	52.77	3.457493	80.77749
5	303.81	198.97	104.84	0.58	15.20	27.12	77.72	30.20	81.33	51.13	3.083185	84.41318
6	327.60	212.32	115.28	0.63	16.22	28.11	87.17	27.41	77.30	49.89	3.457782	80.75778
7	356.48	233.39	123.09	0.68	17.83	29.85	93.24	30.07	82.84	52.77	3.69879	86.53879
8	369.09	243.60	125.49	0.70	18.61	30.69	94.80	24.85	80.65	55.8	3.760541	84.41054
9	379.45	255.12	124.33	0.72	19.49	31.53	92.80	29.18	83.61	54.43	3.681299	87.2913
10	392.30	269.39	122.91	0.75	20.58	32.63	90.28	31.20	85.52	54.32	3.581163	89.10116
11	404.88	285.88	118.99	0.77	21.84	33.89	85.10	31.17	86.07	54.9	3.37591	89.44591

4. GRAPHICAL REPRESENTATION OF THE RESULTS

The change flow of the measured values of oil temperature ϑ_S and ambient temperature ϑ_0 is presented in Figure 3. In this Figure one can discern that, as a result of change in the output torque within the limits of T_2=10 Nm to T_2=22 Nm, the oil sump temperature is changed within the limits of ϑ_S=71°C to ϑ_S=85°C. Hence, with the increase in load, energy losses increase as well, and consequently the oil sump temperature.

For the purposes of further consideration, the heating of oil during operation plays the essential role, i.e. the oil sump over temperature in comparison to the ambient temperature $\Delta\vartheta_S$, Figure 1, i.e. the expression (1).

The change flow in the increase of oil temperature $\Delta\vartheta_S$ for different values of the output torque T_2 is presented in Figure 4. The trend in the change flow lines shows that there is a linear correlation between the oil sump over temperature $\Delta\vartheta_S$ and the output torque T_2. In the event of an increase in the output torque T_2, the oil temperature $\Delta\vartheta_S$ increases as well, as a result of increased power losses. The greatest value of an increase in the oil temperature $\Delta\vartheta_S$ = 55°C has been obtained for the value of the output torque T_2=22Nm.

Figure 5 shows a correlation between the measured values of the oil sump over temperature $\Delta\vartheta_S$ and the total power losses of the worm gear P_G. The diagram shows that there is a linear correlation between $\Delta\vartheta_S$ and P_G. The increase in torque causes the increase in forces in teeth and bearings, as well as the increase in total power losses.

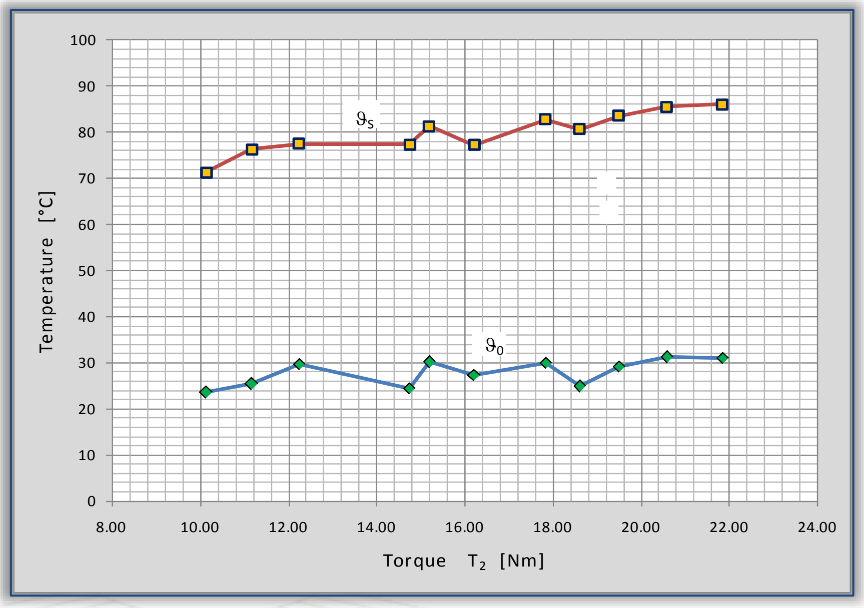

Figure 3. The change flow in oil temperature ϑ_S and ambient temperature ϑ_0 obtained by means of an experiment for the rotation speed of n_1=5000min^{-1} and for different values of the output torque T_2

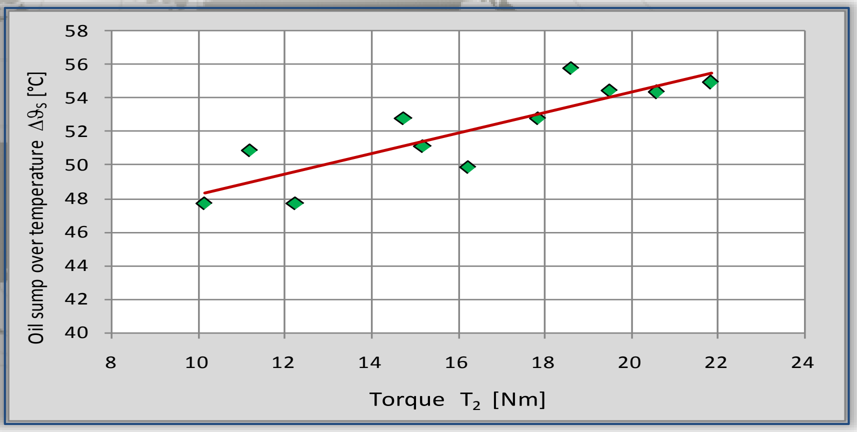

Figure 4. The change flow in the increase of oil temperature $\Delta\vartheta_S$ obtained by means of an experiment for the rotation speed of n_1 = 5000 min^{-1} and for different values of the output torque of T_2

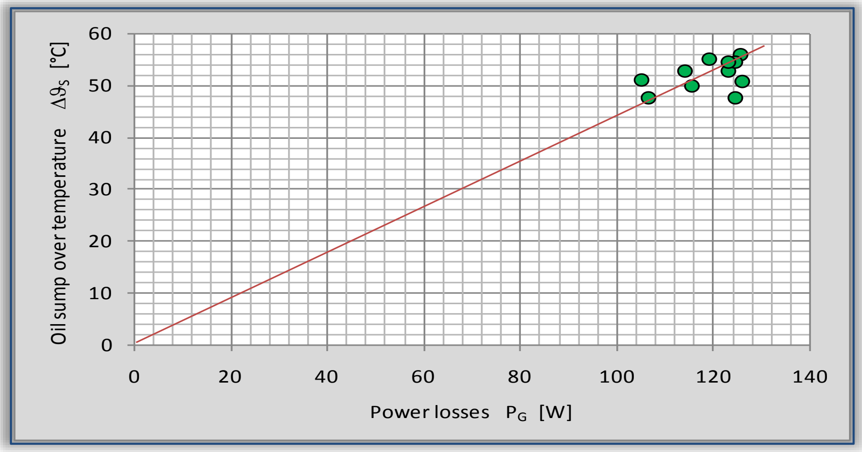

Figure 5. The change flow in the increase of oil temperature $\Delta\vartheta_S$ obtained by means of an experiment for the rotation speed of n_1=5000min^{-1} and for different values of the output torque T_2

The change flow in the wheel mass temperature ϑ_M, obtained according to the expression (5) for different values of the output torque T_2,is shown in Figure 6. One can discern that there is a linear correlation between the wheel mass temperature ϑ_M and the output torque T_2. In the event of change in the output torque within the limits of T_2=10 Nm and T_2=22 Nm the wheel mass temperature changes within the limitsof ϑ_M=75°C and ϑ_M=89°C. In the event of an increased output torque T_2, as a result of increased power losses, the wheel mass temperature ϑ_M increases as well.

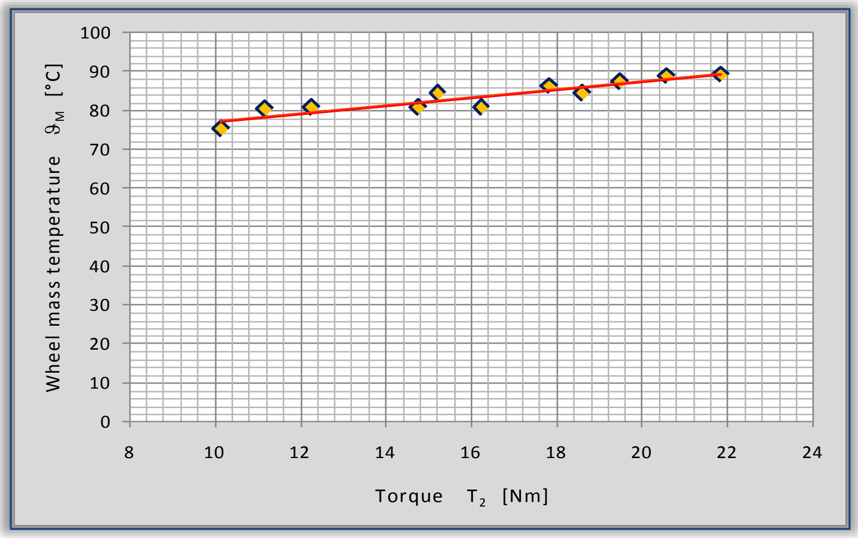

Figure 6. The change flow in the wheel mass temperature ϑ_M for the rotation speed n_1=5000 min^{-1} and for different values of the output torque T_2

5. CONCLUSIONS

In order for a worm gear to operate in a reliable fashion it is necessary that oil sump temperature be maintained within permissible limits. Should one fail to provide the aforementioned, the damage of a worm gear would be imminent. Hence, it is essential to determine the values of oil sump temperature ϑ_S.

There is a correlation between oil sump temperature ϑ_S and the efficiency degree of a worm gearη_Σ, i.e. the friction of contact surfaces of a worm gear. Oil viscosity also influences the oil sump temperature. Higher viscosity oils generate a thicker oil layer between the coupled flanks. However, these types of oil generate greater losses due to a lubricant flow (churning), which increases the operating temperature and, in turn, reduces oil viscosity.

The paper presents the results of analytical and experimental research into thermal stability of worm gears. In order to determine changes in the output torque within the limits of T_2=10Nm and T_2=22Nm the authors have measured the oil temperature ϑ_S and ambient temperatureϑ_0, and the obtained results are presented in Figure 3. The Figure shows that the increase in load results in the increase in energy losses, as well as in the increase in oil sump temperature.

From the research perspective the heating of oil plays an important role during the operation, i.e. the oil sump over temperature in comparison with the ambient temperature $\Delta\vartheta_S$. is essential. The linear correlation between the oil sump over temperature $\Delta\vartheta_S$ and different values of the output torque is presented in Figure 4. Hence, the increase in the output torque T_2 causes an oil sump over temperature $\Delta\vartheta_S$ as a result of increased power losses. The greatest value of the oil sump over temperature $\Delta\vartheta_S$ = 55°C is obtained for the values of the output torque T_2=22Nm.

There is a linear correlation between the measured values of the increased oil temperature $\Delta\vartheta_S$ and the total power losses of the worm gear P_G (Figure 5).

The correlation between the wheel mass temperature ϑ_M and the output torque T_2 is presented in Figure 6. One can observe that there is a linear correlation between the wheel mass temperature ϑ_M and the output torque T_2. Due to increased power losses, with the increase in the output torque T_2, the wheel mass temperature ϑ_M increases as well.

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