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EFFICIENCY OF CROSSED HELICAL GEARS WITH WHEELS MADE OF SINTERED STEEL FE1.5CR0.2MO BY APPLYING THE SINTER-HARDENING TREATMENT

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Summary

High demands are placed on gears made from sintered steel regarding wear, scuffing, pitting load capacity, and efficiency. The structure of sintered steel can be influenced by sinter-hardening treatment in such a way that wear load capacity and meshing efficiency can have high values. The paper presents the research into overall efficiency, meshing efficiency, and the tooth friction coefficient of crossed helical gears with wheels made of Fe1.5Cr0.2Mo by applying the sinter-hardening treatment. A calculation method is also given for the determination of the tooth friction coefficient and meshing efficiency of crossed helical gears with wheels made of sintered steel. The results provide the first important information to product developers as indicators for the development of crossed helical gears.

Key words: *sintered steel, gear, wear, efficiency, power loss*

1. Introduction

Powder metallurgy mould parts made of ferrous materials are the main product group in powder metallurgy. They are very economical products for highly developed industries, such as automobile, machinery and general equipment industry, as well as for many other areas of the metal working industry. In comparison with other manufacturing processes, the share of powder metallurgical products is still not very high but it has achieved a steady growth. Sintered parts have evolved over the years from basic functional elements to critical machine elements, such as gears. These parts must produce higher, dynamic loads that can be transferred more frequently. These parts must carry high and dynamic loads.

The possible application of sintered steel parts can be expanded by using an additional treatment to optimize the properties of such components. According to [6], the wheel made of Fe1.5Cr0.2Mo sintered steel by applying the sinter-hardening treatment shows exhibits the greatest wear resistance among other tested specimens.

2. Test Conditions

The 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 through a magnetic particle brake. On each test bench, the engine and the gearbox, as well as the gearbox and the brake, were connected with a gear coupling. The measurement of the output torque was made on the transmission with a torque gauge bar through a slip ring transmitter. The speeds and output torques were controlled independently for each test bench. The test bench for crossed helical gears and the position of the measuring points are shown in Figure 1. The data of the test gear pair are given in Table 1.

The test transmission housing is made of aluminium. The axial sectional drawing of the worm shaft is given in Figure 2 on the left-hand side, while the axial sectional drawing of the wheel is shown on the right-hand side.

Table 1 Data of the test gear pair

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

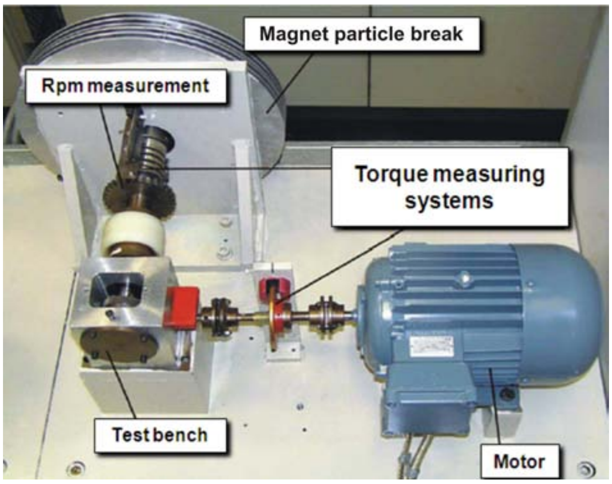

Fig. 1 Test bench

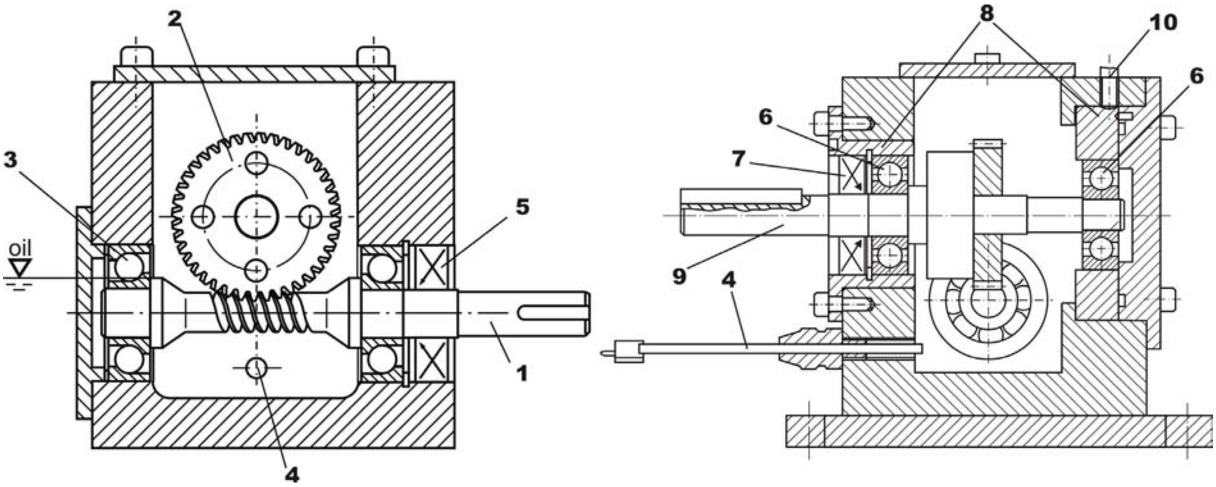

Fig. 2 Sectional drawings of test transmissions

The bearing of the worm shaft (1) is achieved with two angular contact ball bearings (3) in the X-arrangement. The worm (1) is below the wheel (2). The temperature in the oil sump is measured with a nickel-chromium-nickel thermal element (4). The thermal element (4) measures the sump temperature directly under the worm in the area of the gearing contact. The sealing of the test housing is provided by a shaft seal (5).

The wheel shaft (9) is supported by two groove ball bearings (6) (Figure 2 right). The grease lubrication of the bearings that are closed with seals allows sufficient lubrication. The entire worm is in the oil.

All test gears were measured before the experiment on the gear measuring center Klingelberg PNC 64. The gears were splash lubricated. Prior to each test, the gear housing was filled with oil up to the level that covered the entire worm.

The 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.

3. Experimental and Theoretical Research on Efficiency

3.1. Overall efficiency

The overall efficiency quantifies the efficiency of power transmission. The calculation of the overall efficiency for transmissions comes from torque. Equation (1) shows the relationship for crossed helical gears between the measured input and output torque as a function of the overall efficiency η_{ges}.

$$\eta_{ges} = \frac{P_2}{P_1} = \frac{T_2}{T_1 \cdot i} = \frac{P_1 - P_V}{P_1} = 1 - \frac{P_V}{P_1} \tag{1}$$

According to Equation (2), the total power loss P_V is the sum of the power loss of the gear pair P_{VZ}, the bearing power losses of the worm shaft P_{VD1} and the wheel shaft power loss P_{VL2}, and the power losses of the seals P_{VD1} and P_{VD2}.

$$P_V = P_{VZ} + P_{VL1} + P_{VL2} + P_{VD1} + P_{VD2} = P_{VZ} + P_{VL} + P_{VD} \tag{2}$$

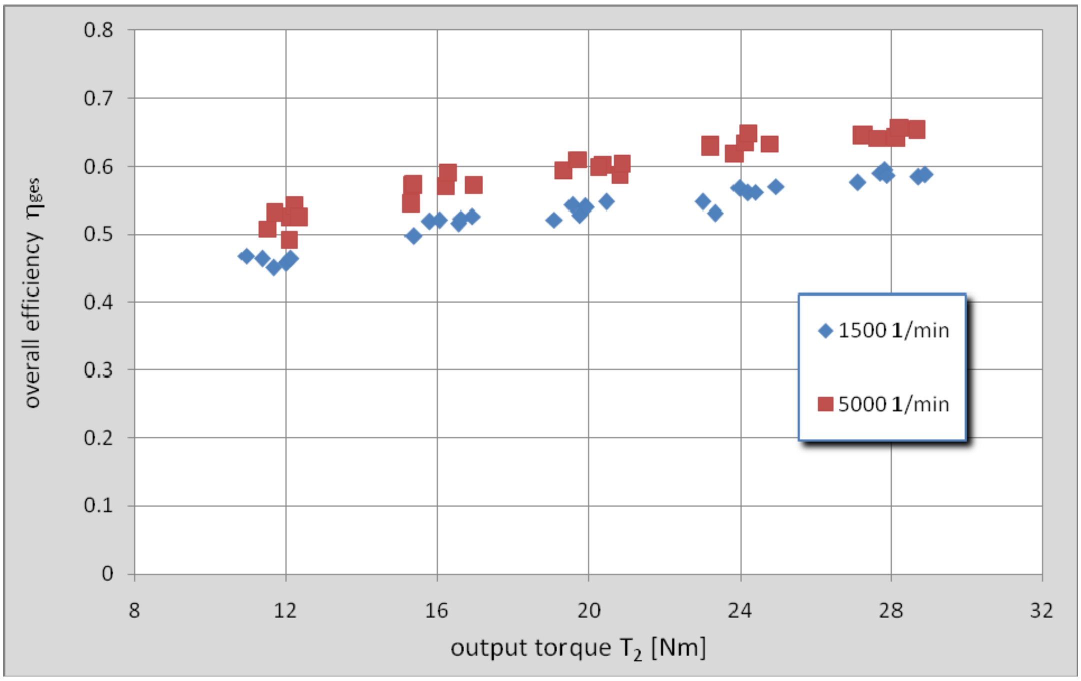

Fig. 3 Overall efficiency η_{ges} of crossed helical gears with wheels made of Fe1.5Cr0.2Mo sintered steel by applying the sinter-hardening treatment [1]

By measuring the output T_2 and the input torque T_1 (Fig. 1), the overall efficiency of the crossed helical gear η_{ges} can be determined. Figures 3 and 4 show the overall efficiency η_{ges} over a test period regarding the speed n_1 for the sintered steel wheels. The efficiency for all experiments is in the range from 45.1 to 65.7%.

The tests with the input speed $n_1 = 5000\ \text{min}^{-1}$ in the lower torque range provide the highest levels of efficiency, compared to the tests with the input speed of 1500 min^{-1}. In the tests with the input speed of 1500 min^{-1} and the output torque $T_2 = 32$ Nm at the load cycles $N_L = 0.54\times10^6$ strong scoring and pitting occur on the tooth flank (Fig. 5). The resulting high surface roughness leads to an increased friction in the contact area and higher temperatures and lower efficiency.

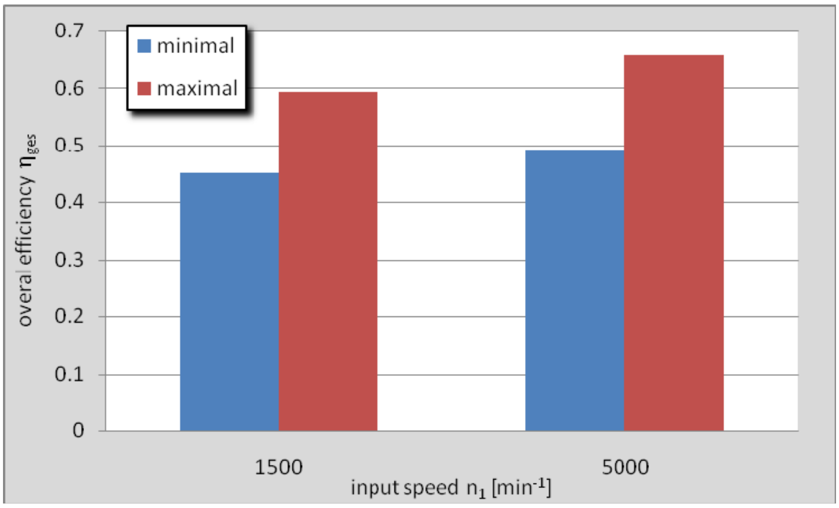

Fig. 4 Comparison of minimal/maximal overall efficiency of crossed helical gears with wheels made of Fe1.5Cr0.2Mo sintered steel by applying the sinter-hardening treatment

Fig. 5 Wear surface of wheel made of Fe1.5Cr0.2Mo sintered steel by applying the sinter-hardening treatment with $n_1 = 1500\ \text{min}^{-1}$, output torque $T_2 = 32$ Nm, number of cycles $N_L = 0.54\times10^6$ and mineral oil lubrication

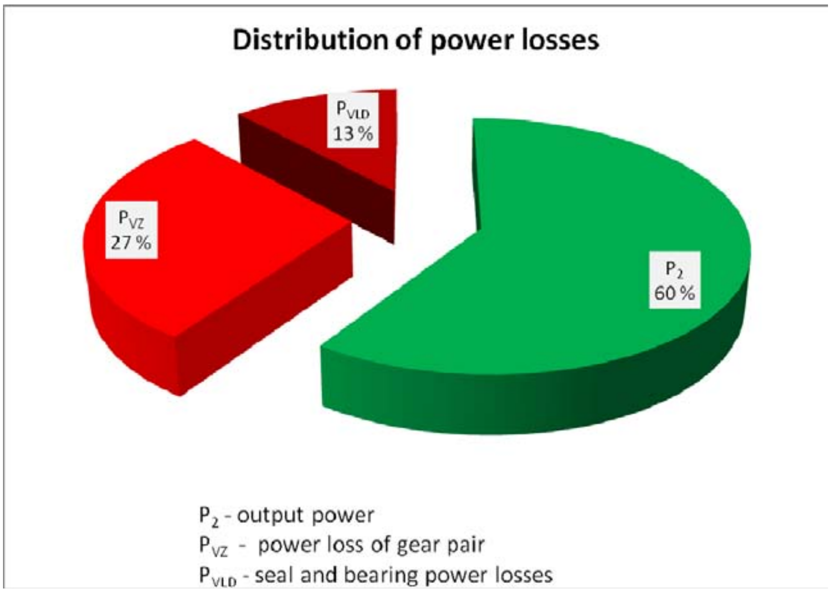

Fig. 6 Distribution of measured power losses for transmission with a speed of 5000 min^{-1} (average values)

Figure 6 shows the distribution of the measured power losses (average values) for sintered metal by applying the sinter-hardening treatment for the input speed of 5000 min^{-1}. The power losses in the bearing and seals were calculated according to the SKF procedure which is included in the program Schraubrad.de [1]. Compared with the bearing and seal power losses, the power loss of the gear pair for the input speed of 5000 min^{-1} is approximately 2.1 times greater.

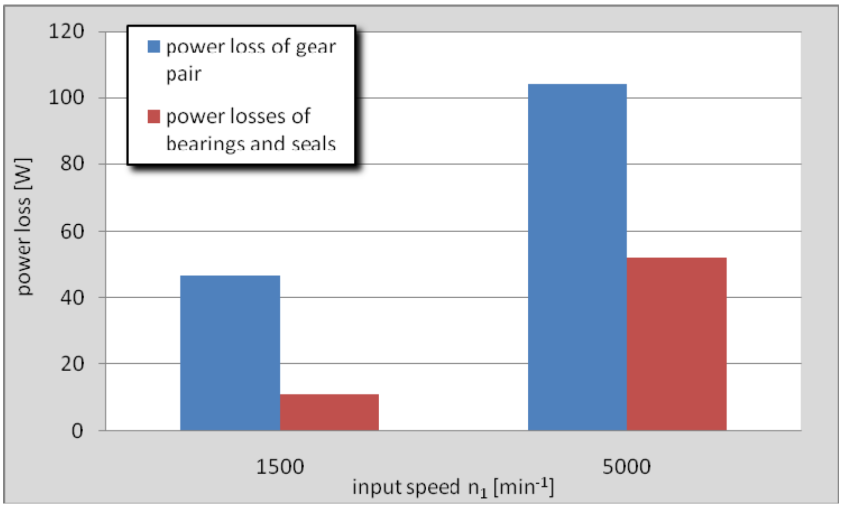

Fig. 7 Share of the total power loss in different operating conditions (average values)

Figure 7 shows the share of the total power loss (average values) for the input speeds of 1500 min^{-1} and 5000 min^{-1}. The power loss of the gear pair P_{VZ} for $n_1 = 5000$ min^{-1} in comparison with $n_1 = 1500$ min^{-1} is approximately 2.1 times greater. The power losses of the bearings and seals P_{VLD} for $n_1 = 5000$ min^{-1} compared to $n_1 = 1500$ min^{-1} are approximately 4.4 times greater.

3.2. Power loss of gear pair and tooth friction coefficient

The meshing efficiency η_z can be determined with the power loss of the gear pair P_{VZ} and the output power P_2.

$$\eta_z = \frac{P_2}{P_2 + P_{Vz}} \tag{3}$$

The meshing efficiency η_z can be also calculated with the averaged tooth friction coefficient μ_{zm} and the crossed axes angle $\Sigma = 90°$:

$$\eta_z = \frac{\tan\beta_{s2}}{\tan(\beta_{s2} + \arctan\mu_{zm})} \tag{4}$$

Thus, the tooth friction coefficient μ_{zm} is averaged:

$$\mu_{zm} = \tan\left(\arctan\left(\frac{\tan\beta_{s2}}{\eta_z}\right) - \beta_{s2}\right) \tag{5}$$

The relationship between the averaged tooth friction coefficient and the angle of friction is given in Equation (6).

$$\mu_{zm} = \tan\rho \;\;\Rightarrow\;\; \rho = \arctan(\mu_{zm}) \tag{6}$$

According to Wendt [3], the averaged tooth friction coefficient for the worm gear is made from the basic friction coefficient and the factor $Y_{T\rho}$.

$$\mu_{zm} = \mu_{0T} \cdot Y_{T\rho} \tag{7}$$

Equation (8) presents a calculation possibility for the dependencies of the basic friction coefficient μ_{0T} given in [1], [2] and [3]. The tooth friction coefficient depends on the helix angle of the wheel, the base circle of the wheel β_{s2}, the sliding speed and the load on the wheel base circle v_{gs}, the load that is present through the output torque T_2 in the calculation. The coefficient of difficulty of the equation is $R^2 = 0.95$.

$$\mu_{0T} = A_0 \cdot \sin\beta_{s2} + A_1 \cdot \left(30 \cdot \sin\beta_{s2} \cdot v_{gs}\right)^{A_2} + A_3 \cdot T_2^{A_4} + A_5 \cdot \left(\frac{v_{gs} \cdot T_2}{30}\right)^{A_6} + A_7 \cdot \left(\frac{T_2}{24{,}384 \cdot 10^{14}}\right)^{A_8} \tag{8}$$

The sliding speed on the wheel base circle v_{gs} is calculated according to Equation (9) and it depends on the wheel base circle r_{s1}, the angular velocity ω_1 and the helix angle of the wheel base circle β_{s2} according to Equation (9).

$$v_{gs} = \frac{r_{s1} \cdot \omega_1 \cdot \sin\Sigma}{\cos\beta_{s2}} \tag{9}$$

The non-linear regression analysis provides coefficients listed in Table 2 from A_0 to A_8.

Table 2 Coefficients of equation (8)

Coefficients	Values
A_0	101.9411
A_1	-0.424684
A_2	-0.368435
A_3	11.15655
A_4	0.030192
A_5	-37.23798
A_6	0.002729
A_7	4.933594
A_8	-0.027481

Figure 8 provides the statistical analysis of Equation (8). The ratio $\mu_{zm\ measured}/\mu_{zm\ calculated}$ includes the differences between the measured and the calculated values for the tooth friction coefficient μ_{zm}. In addition to the absolute frequency, the density function $\varphi(z)$ and the distribution function F(z) are applied.

Additional statistical data for Equation (8) are given in Table 3. According to the calculation, the expected value is 0.998 and the standard deviation from the value is 0.093459.

Table 3 Statistical data for equation (8)

Coefficient		Value
Expected value	μ	0.998
Standard deviation	σ	0.093459
Upper 95 % - limit		1.1515

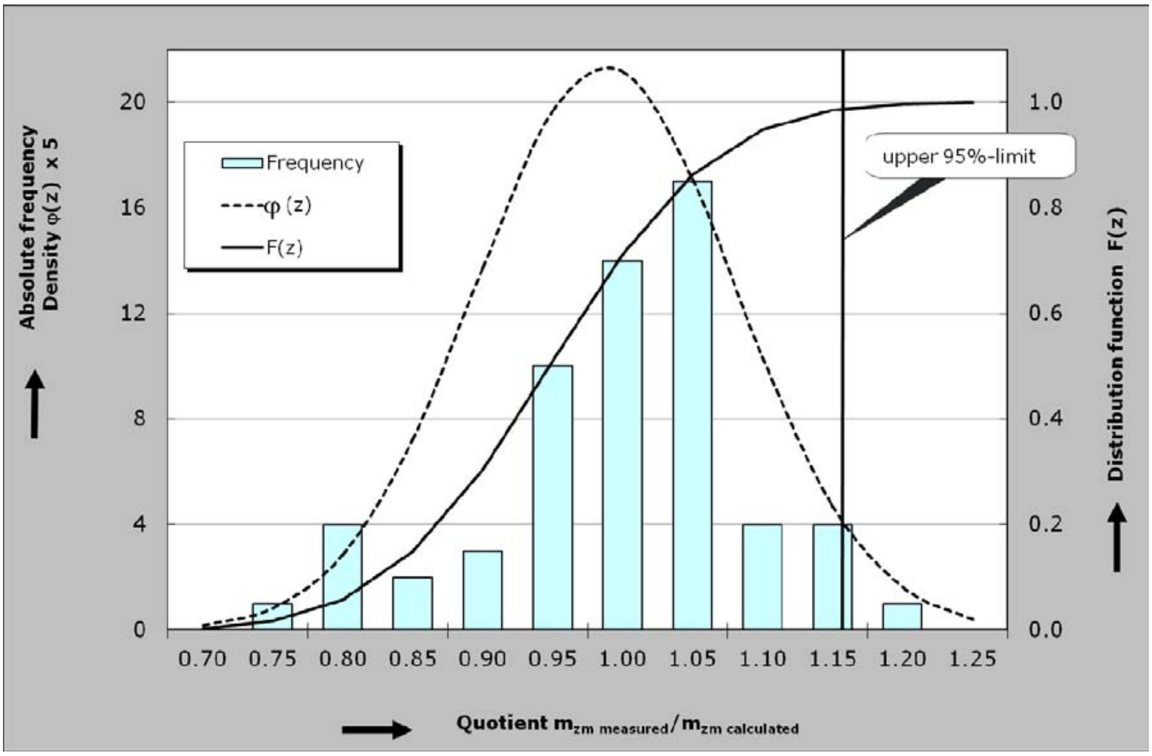

Fig. 8 Statistical analysis of the approximate equation for μ_{zm} according to Equation 8

Figure 9 shows the calculated meshing efficiency η_z as a function of the output torque T_2 for crossed helical gears with wheels made of Fe1.5Cr0.2Mo sintered steel and by applying the sinter-hardening treatment for $n_1 = 1500$ min^{-1} and $n_1 = 5000$ min^{-1}. Higher meshing efficiency η_z occurs at the input speed $n_1 = 5000$ min^{-1} and it is in the range between 58 % and 74 %. At the input speed $n_1 = 1500$ min^{-1}, the meshing efficiency η_z ranges between 54 % and 64 %.

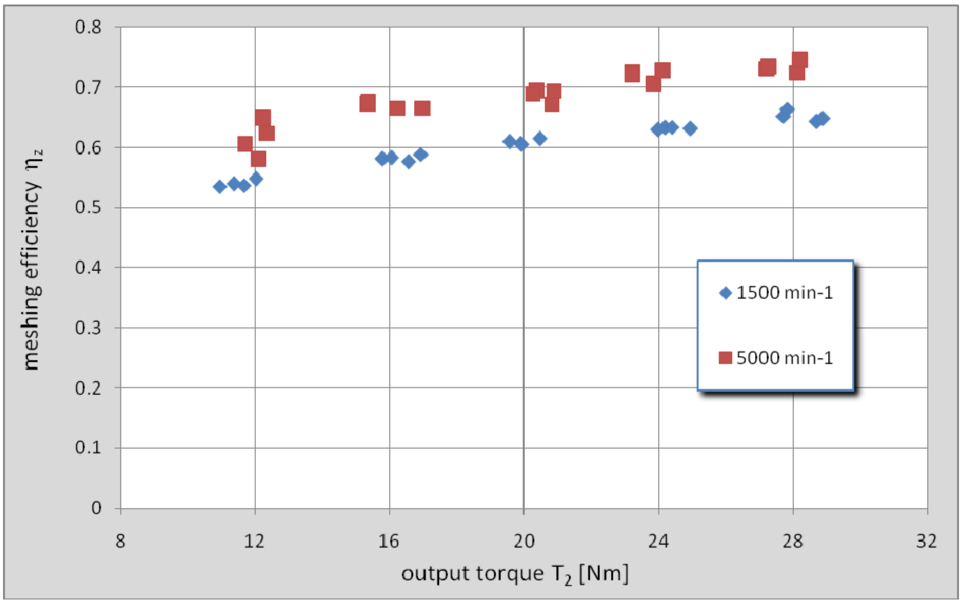

Fig. 9 The meshing efficiency η_z for wheels made of Fe1.5Cr0.2Mo sintered steel by applying the sinter-hardening treatment [1]

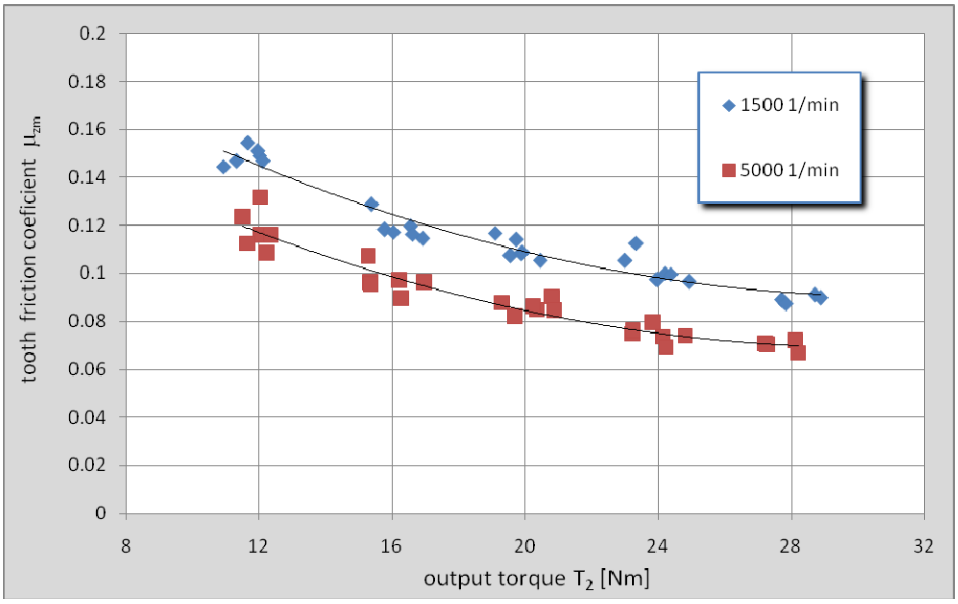

Fig. 10 The tooth friction coefficient μ_{zm} for wheels from sintered steel by applying the sinter-hardening treatment for $n_1 = 1500$ min^{-1} and 5000 min^{-1} [1]

Figure 10 shows the measured and, according to Equation 8, the calculated values of the tooth friction coefficient μ_{zm} for wheels made of Fe1.5Cr0.2Mo sintered steel by applying the sinter-hardening treatment for $n_1 = 1500$ min^{-1} and 5000 min^{-1}. The tooth friction coefficient decreases with increasing speed. The highest values (0.147) occur at the input speed of

1500 min^{-1} and the lowest output torque T_2 = 11.35 Nm. The lowest value (0.06) was measured at n_1 = 5000 min^{-1} and the output torque of 28.24 Nm.

An increase in the output torque also leads to smaller tooth friction coefficients. The influence of the load can be explained by an increased width of the wear surface. According to DIN 3996, the tooth friction coefficient depends on the parameter of the mean lubrication film thickness h^*. This value increases with the increasing width of the tooth contact and thus leads to a higher lubricant film thickness and smaller tooth friction coefficients.

4. Summary

This paper presents the experimental and theoretical research into overall efficiency, meshing efficiency, as well as the tooth frictional coefficient of crossed helical gears with wheels made of Fe1.5Cr0.2Mo sintered steel.

The experimental results of the overall efficiency range from 45.1 to 65.7%. The tests with the input speed n_1 = 5000 min^{-1} give the highest efficiency at lower output torque, compared to the tests with the input speed n_1 = 1500 min^{-1}. At the input speed of 1500 min^{-1}, the output torque T_2 = 32 Nm and the load cycles $N_L = 0.54\times10^6$, strong scoring and pitting occur on the tooth flank.

The tests show an increasing efficiency due to higher output torque. At higher wear rates during the experimental period, which occur at high torques, the temperature increases and the efficiency decreases.

The power loss of the gear pair P_{VZ} at n_1 = 5000 min^{-1} compared to the power loss of the gear pair at n_1 = 1500 min^{-1} is greater by about a factor of 2.3. The bearing and seal power losses P_{VLD} at n_1 = 5000 min^{-1} compared to the bearing and seal power losses at n_1 = 1500 min^{-1} are 4.4 times greater.

The tests show that the higher meshing efficiency η_z occurs at the speed n_1 = 5000 min^{-1} and it is between 58% and 74%. At the speed n_1 = 1500 min^{-1}, the meshing efficiency η_z ranges between 54% and 64%.

This paper gives a calculation for determining the tooth frictional coefficient and the meshing efficiency of crossed helical gears with wheels from sintered steel Fe1.5Cr0.2Mo by applying the sinter-hardening treatment. The comparison charts show that a good agreement exists between the experimental and calculation results of the tooth friction coefficient.

The tooth friction coefficient decreases with increasing speed. The highest value (0.147) occurs at the input speed of 1500 min^{-1} and the lowest output torque T_2 = 11.35 Nm. The lowest value (0.06) was measured at n_1 = 5000 min^{-1} and the output torque of 28.24 Nm. An increase in the output torque also leads to lower tooth friction coefficients.

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