DOI: 10.21062/mft.2025.031 http://www.journalmt.com

Methodology for Comprehensive Testing and Optimization of Gears for Torsional Strength

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The article described a new methodology for testing the torsional resistance of a single-stage gear transmission used in agricultural machinery. The analysis encompassed the entire mechanical system rather than focusing solely on its individual components. The research identified three key ranges of structural resistance. The first range, with twist angles from 0° to 1.85° and torques up to 1050 Nm, was associated with the elimination of structural play and the alignment of contact surfaces. The second range, from 1.85° to 4.76° and torques up to 3450 Nm, confirmed the resilient behavior of the gearbox according to Hooke's law. In this range, the system worked stably and maintained repeatability of parameters. The third range, above 4.76° and 3050 Nm, showed the presence of permanent but local deformations. However, these displacements did not affect the functionality of the system in less demanding applications. The maximum torque of 5500 Nm did not cause macroscopic damage or oil leaks, which proves the high quality of the design and the effectiveness of material optimization. The developed method allows for an accurate determination of the safety factor and a detailed assessment of the strength properties. It can be used to optimize transmissions in various sectors such as agriculture, automotive and aerospace. The results also form the basis for further experiments, including fatigue tests and contact stress analyses. The proposed methodology enhances the predictive accuracy of gearbox durability under various load conditions. These advancements support the development of sustainable and efficient mechanical systems across multiple industries.

Keywords: gear, torsional strength, research methodology, structural optimization, design, gear diagnostics

1 Introduction

Gears are key components in agricultural machinery, responsible for transmitting the power and torque necessary for efficient operation. In response to increasing demands for performance, durability and sustainability, gear manufacturers are focusing on design optimization. This paper presents a case study carried out in cooperation with one of the leading manufacturers of agricultural transmissions in Poland, whose plant is located in Września. To meet global environmental standards, the manufacturer has implemented material, weight and construction optimization aimed at reducing energy consumption during production and operating costs.

The subject of the study was a single-stage gear transmission used in agricultural machinery. Its task is to increase the speed of the output shaft by a factor of 3.7 in relation to the drive shaft. The gearbox design has been optimised in areas such as the housing, bearing arrangement and shaft-gear connections. The selection of materials considered the principles of sustainable development. The aim of the research was a comprehensive torsional strength analysis, which included the evaluation of the transmission as a whole system and not just individual components. The analysis was aimed at identifying potential weak points and confirming the gearbox's resistance to applied torsional torques.

The literature on gears is very extensive. The authors of this paper, analyzing the available publications, have selected several items that present an interesting approach to the evaluation of the gearbox as an integrated system. The study by Zhang and co-authors analyzed the effect of changes in the stiffness of ball bearings due to thermal deformation on the dynamic characteristics of the gear transmission. The results of these studies indicate the need to take this phenomenon into account when designing high-performance electric drives, where periodic and chaotic movements have been observed, among others [1]. Jin and colleagues used the transfer array method to analyze the dynamic torsion of pinions in bevel-bevel gears in aerospace systems. These studies confirmed the possibility of accurate simulation of dynamic loads at different operating speeds [2]. In his research on torsional torque measurements, Garstish emphasized that modern dynamometers allow for precise evaluation of the efficiency of drive systems, and inaccuracies may result from errors in recording angular accelerations [3].

The work of Takacs and his team pointed out the need to consider dynamic moments of inertia. An iterative model for calculating the crank angle enabled an accurate estimation of the torque components, which is crucial for improving the energy efficiency of these systems [4]. Flodin conducted both laboratory tests and computer simulations to investigate the wear of the tooth surfaces of the gears. Analyses have shown that wear affects the distribution of contact stresses and the dynamic characteristics of the gearbox [5]. Automatic gear diagnostics based on vibration signals and machine learning techniques were described by Praveenkumar and co-authors. In their research, they demonstrated the high efficiency of carrier vector machines (SVMs) in identifying defects under various operating conditions [6]. Modelling and simulation of gear systems in the context of fault detection was presented in the paper of Zhang and colleagues, who highlighted the importance of this approach for large industrial systems [7,8]. The authors of this paper emphasize that research on the overall strength of gear systems is rarely described in the literature. To fill this gap, comprehensive torsion resistance tests were carried out, taking into account, m. among others, tooth clearances, mounting tolerances, lubrication and venting systems, and tooth geometry. The research results presented can be used in the design of advanced gear structures, as well as in the implementation of diagnostic and operational standards in the agricultural sector and other industrial sectors. Meingssaner, G. and colleagues dealt with the solution of new systems for reducing torsional vibrations with an impact on strength and noise. [9]. The author Marafona, J. and colleagues dealt with optimization is a flexible methodology for gear design since it allows for diverse approaches according to current demands [10]. The possibilities of additive technologies are significant. The authors Buonamici, F. And Yan, Z. and colleagues [11,12] focus on topological optimizations for the purpose of reducing vibrations, static and modal analysis of the components. The author Kozový, P. and colleagues dealt identification of residual stresses after machining a gearwheel made by sintering metal powder [13].

2 Materials and Methods

The main purpose of developing the methodology and then conducting the research was to experimentally verify the strength calculations and assess the quality of the components and their assembly. A brand-new single-stage gear transmission of the ROLMUS type was selected for testing: PMS(S)-21130370, which was used in agricultural machinery (Fig. 1).

The power of the tested gearbox was 40 kW, and its weight was 22 kg. Under operating conditions, this gearbox was driven by a hydraulic motor at a speed of 500 rpm, resulting in a speed of 1,850 rpm on the output shaft. Therefore, this gearbox was used in agriculture as a multiplier to increase the speed of agricultural machinery. The gearbox allowed for changes in rotational speed, direction of rotation (right and left rotations), and torque, which was reduced by a factor of 3.7 in relation to the engine torque. The gearbox was designed to work with a motor that could achieve a maximum torque of no more than 764 Nm and allowed for the connection of machines whose nominal torque value did not exceed 206 Nm.

A novel approach used by the authors and the developed research procedure was a comprehensive evaluation of the fully assembled gearbox as an integral system in which components made of different materials worked together under the load caused by torsional torque.

The authors of the study were particularly interested in:

- Examining the range of elasticity of the gearbox according to Hooke's law for gradually increasing torsional torque,
- Assessing the uniformity of deformations as a function of increasing torsional torques,
- Verifying the moment of permanent deformation,
- Checking whether exceeding the permissible value of the torsional torque of the gearbox would cause oil leakage, which was crucial when using the gearbox in field work (no oil should leak into the soil and crops),
- Performing a macroscopic assessment of the condition of the transmission components after the tests to evaluate gear wear.

The tested gear (Fig. 1) consisted of two interlocking gears:

- The drive wheel (z_1) had 52 teeth and was connected to the shaft of the hydraulic motor,
- The driven wheel (z_2) had 14 teeth and was an integral part of the output shaft (shaft to which agricultural machinery was connected).

The gear system was enclosed in a two-part cast iron housing sealed with a layer of high-temperature silicone. The gearbox body was fastened with screws ensuring adequate rigidity of the structure and protection against external environmental factors (such as rain and dust in field conditions).

The range of torques loading the gear shafts was determined in accordance with the PN-ISO 6336-1 standard [14], using mathematical relationships presented in Formulas 1 and 2:

$$M_1 = 9550\frac{P}{n_1} = 9550\frac{40kW}{500[\frac{obr}{min}]} = 764\ [Nm] \tag{1}$$

$$M_1 = 9550\frac{P}{n_2} = 9550\frac{40[kW]}{1850[\frac{obr}{min}]} = 206\ [Nm] \tag{2}$$

Where:
P …Hydraulic motor power [kW],
n_1, n_2 …Speed on individual rpm gear shafts respectively [rpm],
M…Torsional torque [Nm].

Fig. 1. *Gearbox multiplying the rotation of the hydraulic motor of agricultural machinery with a value of 3.7 (gear ratio u=0.27)*

The tests were carried out on a specially designed test stand (Fig. 2) at the Automotive Research and Development Institute BOSMAL Sp. z o.o. The gearbox was mounted in such a way that the output shaft was blocked. The input shaft was connected to a rotary hydraulic cylinder, ensuring a controlled increase in load. This solution generated torsional torque in the transmission. The measurement of the torsional torque and the angle of torsion of the entire gearbox was recorded over time. The measuring system recorded 100 values per minute. The measuring station allowed for the recording of the torsion angle with an accuracy of 14 significant digits after the decimal point and 13 for the torsional torque.

The tests were conducted in five stages, as described below.

- 1) The gearbox was installed on the test bench, and the connections of the housing and screws were marked with a pen to identify possible displacements during the examination.
- 2) The reliability of immobilization and the correct connection of the measuring equipment were checked.
- 3) The torsional torque generated by the rotary hydraulic cylinder was gradually increased while recording the torque and angle of rotation over time.
- 4) A series of tests were conducted in which the torsional torque was gradually increased to the following values: 250 Nm, 550 Nm, 1050 Nm, 1550 Nm, 2050 Nm, 2450 Nm, 3050 Nm, 3450 Nm, and 5500 Nm. After reaching each of these values, the torsional torque was gradually reduced to 0 and then increased again to the next set value. The procedure was repeated in cycles, starting at 0, then increasing to 250 Nm, decreasing to 0, increasing to 550 Nm, and so on until the maximum value of 5500 Nm was reached.
- 5) During the tests, macroscopic assessments of component displacement and leakage were performed. After the tests were completed, the gearbox was dismantled, and the surface condition of all parts of the tested gearbox was analyzed.

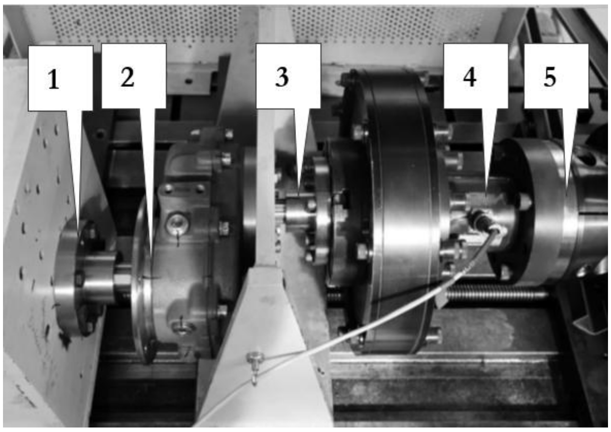

Fig. 2 *Gearbox multiplying the rotation of the hydraulic motor of agricultural machinery with a value of 3.7 placed on the torsional torque test stand*

Where:

1…The output shaft of the gearbox is immobilized by means of a coupling bolted to the immobilized body,

2…The marker of the displacement of the screwed elements,

3…The coupling mounted on the input shaft of the gearbox and connected to the measuring device,

4…The sensor recording the measurement,

5…The rotary cylinder generating the torque.

The test results were presented in the form of graphs and photos documenting the condition of the gearbox after the tests. Figs. 3-11 showed torsional torque vs. torsion angle diagrams for successive load levels, while Figs. 12-14 provided a macroscopic assessment of the gear structure after testing.

The conducted research provided valuable data on the gearbox's behavior under static torsional torque load. The developed research methodology enabled the collection of real results and was verified. Therefore, in the authors' opinion, it could be applied on a broader scale, not only for this specific gearbox but also for similar systems, as justified in the later sections of this paper.

3 Results and Discussion

The results of the measurements are presented in the form of diagrams of the dependence of the torsional torque on the angle of torsion of the gearbox for different load levels. Figs. 3-11 show the characteristics obtained for the torsional torque settings described in the previous section. The data from the measuring system was recorded in Excel files. The results of the measurements were used to develop the graphs and with the help of the "trend line" function, lines marked on the graphs with a dotted line and described with formulas and R2 values were determined.

The first actuation of the mechanism indicates slight deviations from the ideal linearity (Fig. 3). This phenomenon is typical of gear mechanisms, especially in systems that have not previously been subjected to cyclic stresses. Only after overcoming these initial shifts does the linear relationship stabilize, which is clearly visible when the value of about 0.3° is reached.

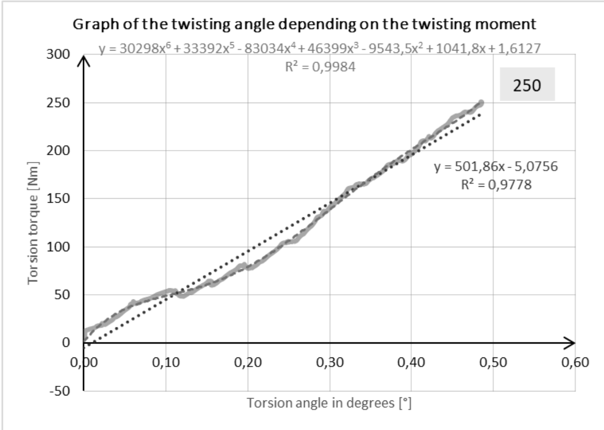

Fig. 3 *The dependence of the torsional torque (Nm) on the torsion angle (°) when the gearbox is loaded with torsional torque in the range from 0 to 250 Nm*

In the Fig. 3 the diagram shows two key characteristics: the sixth-degree polynomial function (green line) and the linear alignment of the measurement data (red line). The regression coefficient for the polynomial function is $R^2=0.9984$, indicating a very good fit of the model to the data over the full range of angles. Linear alignment ($R^2=0.9778$), on the other hand, confirms that in the range of up to about 0.3°, the gear system behaves almost perfectly elastic according to Hooke's law.

Fig. 3 illustrates the relationship of torsional torques to the torsional angle of the gearbox for a load of 250 Nm. Diagram analysis includes both linear and polynomial curve fitting. Linear alignment describes the equation $y=501.86x-5.0756$, and the coefficient of determination $R^2=0.9778$ indicates a high agreement of the data with the linear model. The sixth-order polynomial fit described by the equation $y=30298x^6+33392x^5-83034x^4+46399x^3-9543.5x^2+1041.8x+1.6127$ achieves an even higher coefficient of determination, $R^2=0.9984$, which suggests a better representation of local changes during deformations.

The graph does not start from a point (0.0), which can be explained by the presence of tooth clearances in the gear. This clearance is due to the need for minimal pre-shifts before the gears enter full meshing. Up to this point, the torsional torque is partially absorbed by the clearance equalization, resulting in a delayed onset of the linear strain relationship. This phenomenon is characteristic of mechanical systems and affects the initial parameters of the graph.

In addition, small fluctuations are visible on the curve, which can be caused by unevenness in the surfaces of the gear teeth in contact. During rotation, such unevenness can lead to temporary changes in load. Oscillations can also be the result of natural vibrations of the gearbox, dynamic interactions with the test machine or interference in the data due to slight measurement errors.

In terms of the analysis of the elastic range, the data indicate that in the range from 0 to 250 Nm, with the elimination of tooth clearance, the gear behaves according to Hooke's law. This means the proportionality of the deformations to the applied torsional torque in the elastic range. However, the fitting parameters indicate that greater precision is achieved for the polynomial function, suggesting that minor nonlinearities may be important in the full characteristics of the system.

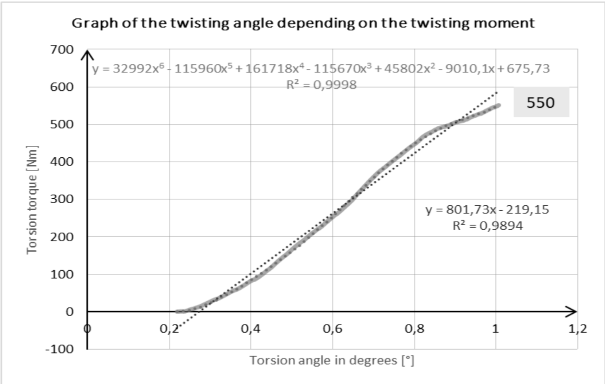

Fig. 4 *Dependence of the Torsional torque (Nm) on the torsion angle (°) at a load of 550 Nm*

In Fig. 4, with a load of 550 Nm, the diagram illustrates that the relationship between torsional torque and torsion angle is close to linear, as evidenced by the high coefficient of determination R^2=0.9894. Even though there are some slight deviations from linearity, it can be assumed that in this range of torsional torques, the system behaves elastically according to Hooke's law.

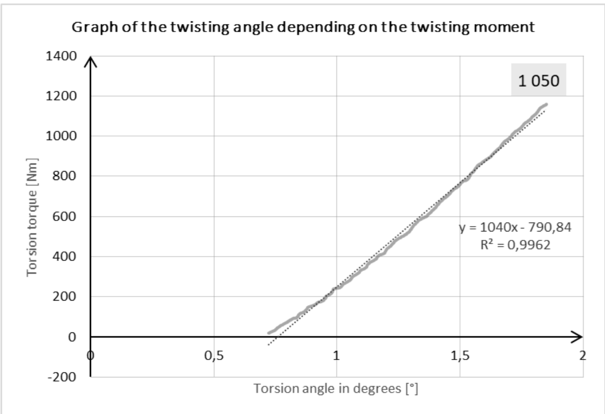

Fig. 5 *Dependence of the Torsional torque (Nm) on the torsion angle (°) at a load of 1050 Nm*

In the case of Fig. 5 (load 1050 Nm), the relationship remains almost perfectly linear, which is confirmed by the high R^2=0.9962 coefficient. The gear system continues to show full reversible elastic deformations. No signs of permanent deformities were observed.

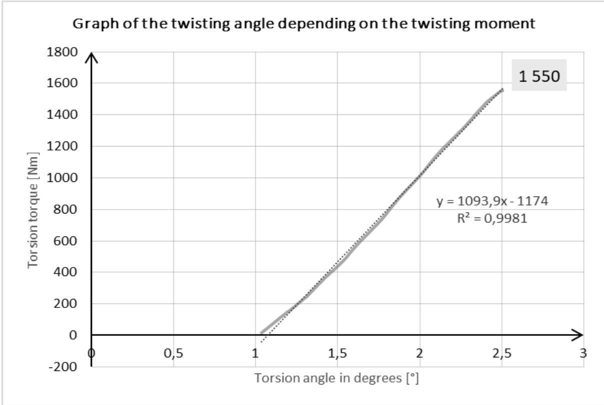

Fig. 6 *Dependence of the Torsional torque (Nm) on the torsion angle of the gearbox (°) at a load of 1550 Nm*

The same situation is shown in Fig. 6 at a load of 1550 Nm, despite the increased load, the gear system maintains a high linearity of the relationship with R^2=0.9981. The increase in the angle of torsion is proportional to the increase in torque, and its value suggests uniformity of deformations as a function of increasing torsional torque.

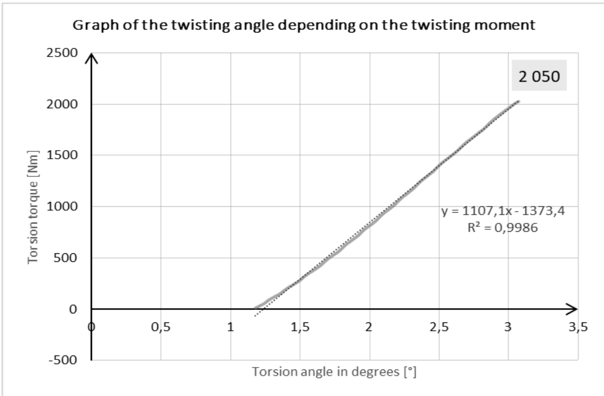

Fig. 7 *Dependence of torques (Nm) on the torsion angle (°) at a load of 2050 Nm*

Similarly, in Fig. 7 (load 2050 Nm) a similar situation can be seen. The regression coefficient is R^2=0.9986, indicating a perfect linear fit. In this torque range, no signs of irreversible deformation or unevenness of deformation were detected.

A similar situation can be found in Fig. 8 (2450 Nm), where the angle of twist increases in proportion to the moment, and this relationship is described by a linear equation with a coefficient of R^2=0.9988. Slight deviations may be the result of interdental clearance and the influence of shifts resulting from minor nonlinear deformations of structural elements.

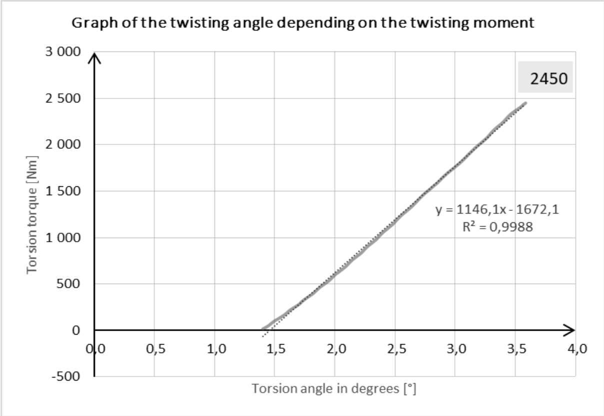

Fig. 8 *Dependence of the torsional torque (Nm) on the torsion angle of the gearbox (°) at a load of about 2450 Nm*

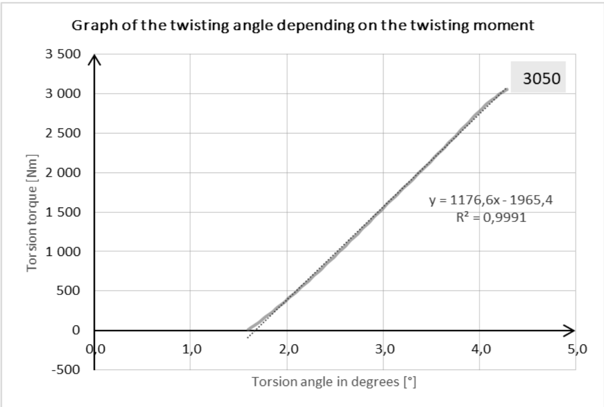

Fig. 9 *Dependence of the Torsional torque (Nm) on the torsion angle of the gearbox (°) at a load of about 3050 Nm*

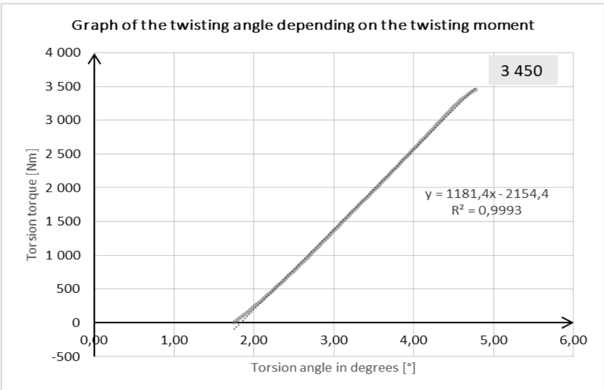

Fig. 10 *Dependence of the torsional torque (Nm) on the torsion angle (°) at a load of 3450 Nm*

Similar conclusions can be drawn from Figs 9 and 10, showing the tests for loads of 3050 Nm and 3450 Nm. High values of the coefficients (R^2=0.9991 and R^2=0.9993, respectively) confirm the elastic behavior, although the phenomenon of local deformations may occur in extreme load ranges. It is also worth noting the "shaking" of the graphs, which may be the result of imperfections in the fit of the transmission components, such as structural clearances or micro-shifts during loading.

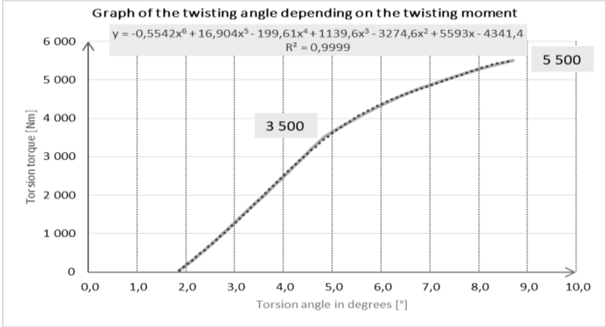

Fig. 11 *Dependence of the Torsional torque (Nm) on the torsion angle (°) at a load of 5500 Nm*

Fig. 11 shows the torsional torque dependence on the angle of torsion of the gearbox for a maximum load of 5500 Nm. This diagram shows a significant deviation from the linear relationship, which is confirmed by a more complex polynomial function of the sixth-degree $y=-0.5542x^6+16.904x^5-199.61x^4+1139.6x^3-3274.6x^2+5593x-4341.4$. For this polynomial, the coefficient of determination is $R^2=0.9999$, which indicates an exact fit of the curve to the experimental results, despite the apparent nonlinearities in the region of large moments. The graph clearly illustrates that above a load of about 3500 Nm, the behavior of the system changes. The angle of torsion increases faster in relation to the increase in torque, which indicates the appearance of permanent structural deformations in the transmission. This type of phenomenon is typical for exceeding the elastic limit of the constituent materials of the system, such as body components and gears. Deformations can result from local overloads, structural clearances, and axial displacements. At larger angles, the „trembling" of the curve is also visible, which may be the result of dynamic changes in stress during the examination. Analyzing the rest of the graph, until the torsional torque of 5500 Nm, there was no critical damage to the transmission. However, the observed deformations indicate a reduced uniformity of deformation. During the test, there was no oil leak, and no increased crackling or other disturbing noises were reported.

On the basis of the measurements carried out in the range from 0 to 3500Nm, Tab. 1 was developed, which illustrates the linear nature of torsion of the gear transmission depending on the torsional torque. Fig. 12 and Tab. 1 show the range over which the gearbox retains its elastic properties. At lower torsional torque loads, higher torsion angle values can be observed, which may be related to the initial compensation of micro-clearances on the contact surfaces of the gears and partial energy absorption during the pre-meshing stage. According to the authors, this process may be related to micro deformations and a gradual increase in the stiffness of the system after the elimination of these unevenness.

Tab. 1 *Characteristics of torsional strain parameters in the transmission system, showing the relationship between the torsion angle, applied torque, angular increments, and torque increments, as well as the calculated inclinations in linear ranges*

Torsion angle [°]	Torsional torque [Nm]	Angle Delta [°] $\Delta\theta = \theta_n - \theta_{n-1}$	Torque Delta [Nm] $\Delta M = M_n - M_{n-1}$	Tilt angle [Nm/°] $k = \Delta M / \Delta\theta$
0	0	0	0	-
0,49	250	0,49	250	510,2
1	550	0,51	300	588,24
1,85	1050	0,85	500	588,24
2,5	1550	0,65	500	769,23
3,08	2026	0,58	476	820,69
3,58	2450	0,5	424	848
4,29	3050	0,71	600	845,07
4,76	3450	0,47	400	851,06

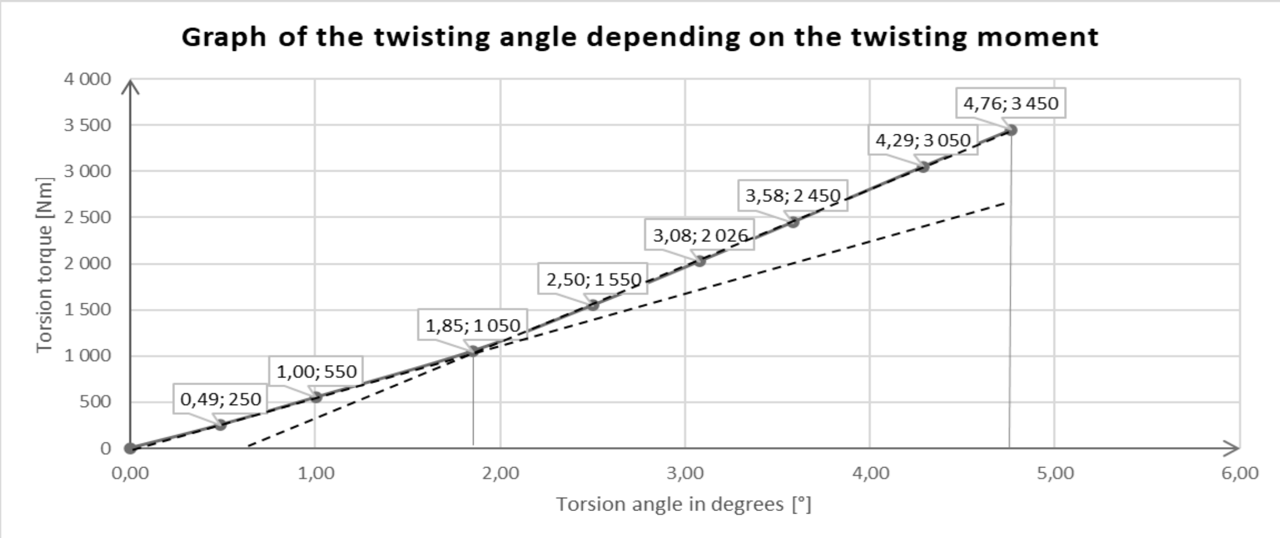

Fig. 12 *The dependence of the torsional torque (Nm) on the torsion angle of the gearbox (°) in the range from 0 to 3450 Nm*

In the Fig. 12 the solid line shows the actual measured values, while the dashed line illustrates the auxiliary linearity intervals of the system. The graph distinguishes two characteristic areas of the twist angle: from 0.00° to 1.85° and above 1.85° to 4.76°. In the first range, the larger increases in the torsion angle are the result of the elimination of structural clearances and the alignment of the contact surfaces in the transmission. This process accompanies the initial stage of the mechanism's operation. Once the gears are fully meshed, the system switches to a quasi-linear state, which is confirmed by the high fit factor of the trend function. This characteristic of the changes is consistent with the theoretical elastic model.

Assuming that the gearbox was calculated to be 764 Nm, it can be concluded based on the tests carried out that it meets linear relationships in the range from 0 to 3500 Nm.

$$X = \frac{M_m}{M_c} = \frac{3500\ \text{Nm}}{764\ \text{Nm}} = 4.58 \quad (3)$$

Where:

X…Actual static safety factor,

M_m…Torsional torque measured [Nm],

M_C…Calculated torsional torque [Nm].

The gear behaved elastically, returning to its original shape without permanent deformation when the load was removed. The mathematical model of this relationship is described by a linear equation (formula 4).

$$\theta = M \cdot k + C \quad (4)$$

Where:

θ…Gear angle [°],

M…Torsional torque [Nm],

k…Gear twist ratio [°/Nm],

C…Reference system correction factor (dimensionless).

The values of the k-factors in the individual diagrams are in the range from for the load to 3050 Nm, which indicates structural stability and a proportional increase in the angle of torsion in the elastic range.

The test bench's maximum torque of 5500 Nm did not displace or damage the transmission elements. Since the post-test assessment of the marker displacement showed no change except as indicated by the black arrows in Fig. 13. Observational tests showed that the displacement occurred on the flange attaching the gear to the rotary actuator of the measuring device. Fig. 13 shows this location with a red arrow. According to the authors of the paper, this displacement occurred after exceeding the torsional torque above 3500 Nm, which indicates a greater bend of the curve shown in Fig. 11. In the next version of the design, consideration may be given to increasing the number of bolts on the flange or the thread size. However, this is not a necessary condition, because the X coefficient calculated using formula 3 seems to be sufficient, as evidenced by the lack of leakage in the system.

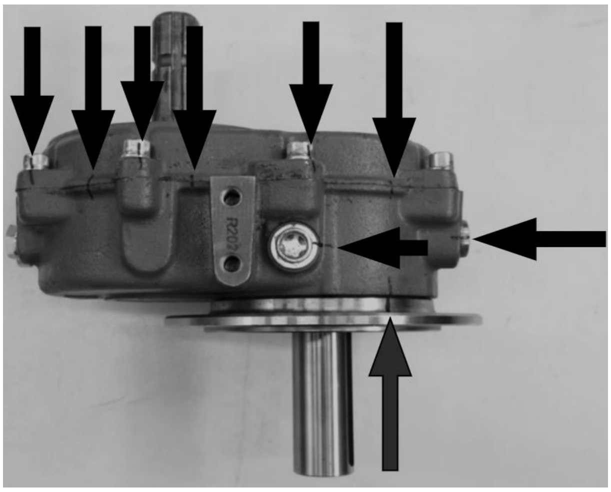

Fig. 13 *Visual macroscopic examinations after the completion of the torsional torque test at 5500 Nm*

In Fig. 13 black arrows indicate that there has been no rearrangement of the elements in the marked places of connections of the transmission elements and fasteners. A red arrow indicates a slight shift in the connection plate. Consequently, it can be inferred that the primary components of the gearbox remained structurally intact under the application of a torsional torque of 5,500 Nm imposed by the test machine.

After the tests on the outside of the gearbox were completed, the analysis of the internal components of the gearbox began. Fig. 14 shows a picture of the z_1 (drive) wheel assembly. The teeth of the input shaft showed no clear signs of overload except for minor surface defects. Fig. 15 illustrates the condition of the z2 (driven) wheel, where no mechanical damage was also observed. A photo of the z1 (large) gear and a photo of the z2 (small) gear after the load test in one transmission do not show serious damage that would disqualify the design solution and the adopted manufacturing technology. The z1 wheel shows small surface defects in the form of minor scratches and slight micro pitting, which did not affect its functionality. The z2 wheel, on the other hand, presents only mild abrasions, which may indicate even load transfer during the test. The absence of clear signs of overloading or serious damage on both wheels confirms the correct operating and lubrication conditions during the operation of the transmission. Under the influence of the maximum torque, the bolted connections did not loosen. No clearance was detected on the components linking the gearbox housing. Although there were local shifts of the connection flange, no critical structural weaknesses were detected.

The conducted tests confirmed that the tested gear meets the strength requirements. The structure maintained integrity even under loads exceeding nominal values. The gearbox housing deformed elastically up to 3500 Nm. Exceeding this value caused angular displacement of the fastening element. This did not affect the system's functionality in agricultural applications. No macroscopic damage was found on the housing or the meshing surfaces. This confirms compliance with the design assumptions. All threaded connections remained stable, verifying the correct selection of threads, screws, and washers. In the future, enlarging the threaded holes or increasing their number could be considered.

Fig. 14 *A photo of the z1 (large) gear after the load test*

In the Fig. 14 the teeth of the input shaft show only minor cavities on the work surface, which may indicate an initial stage of micro pitting, with no clear signs of overload or other serious mechanical damage.

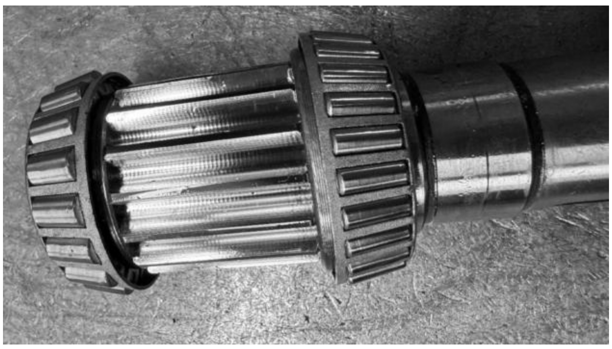

Fig. 15 *A photo of the z2 gear (small) after the load test*

In the Fig. 15 the tooth surfaces do not show clear signs of overload, but only minor abrasions characteristic of abrasive wear in the initial phase of operation.

The authors of this paper conduct an analysis of gear life in operating conditions. These works can be divided into 3 groups:

- 1. Long-term fatigue tests under variable torsional loads,
- 2. Extension of the analysis with numerical modeling to determine local stress concentrations,
- 3. Verification of the impact of operating factors such as lubrication and temperature on the durability of the system.

4 Conclusions

Research on the torsional strength of single-stage gear transmissions has yielded significant in-sights into the elastic properties and durability of mechanical structures. Experimental results indicate that Hooke's law can be utilized on a macro scale to evaluate the deformation of gear systems. Based on the results, three key deformation compartments were identified that could be useful in gear design in different industries.

The range of process clearances includes torsional angles from 0° to 1.85° and torsional torques up to 1050 Nm. Initial design inaccuracies such as tooth clearances and unevenness of contact surfaces are eliminated in this range. The results point to the need for precise alignment of components in applications that require high accuracy, such as control systems in surgical robots and aerospace systems.

The proportionality interval covers angles from 1.85° to 4.76° at moments from 1050 Nm to 3450 Nm. This relationship is important for drives used in precision industrial, space, and medical ap-plications.

The interval of local permanent deformations begins above the torsion angle of 4.76° at moments greater than 3050 Nm. Minor but noticeable displacements of structural elements such as bolted connections then appear. However, they do not have a significant impact on the functionality of the gearbox in applications where the required precision is moderate, e.g., in agricultural machinery.

Studies have shown that despite minor deviations from ideal linearity, the results obtained are useful for optimising gear design. They allow for a more accurate determination of the factor of safety and dynamic properties of the system. The methodology can be successfully used to test different types of gears in industries that require reliability and high precision.

The authors of the study emphasize that the presented research method is valuable for designers of precision positioning systems and manufacturers of industrial machines, robots, vehicles and propulsion systems in the aviation and space industry. They encourage further experiments including long-term fatigue tests, numerical analyses, and tests in real operating conditions. Further research could make a significant contribution to the creation of new design standards for responsible mechanical systems

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