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PROPERTIES OF A THREE DISC CYCLOID GEAR AS A RESULT OF BENCH TEST

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Abstract

The most common design solution are cycloidal gearboxes with two discs, in which the reaction forces of bearings from the working discs create a bending moment on the high-speed input shaft, additionally affecting bearings. In the article are presented the results of tests of the cycloid gear prototype with three discs. Instead of the second disc, the introduction of two side discs, 180^0 from the centre disc, allows you to reduce this moment to zero. To compensate for the unbalance of the shaft, the side discs have half the width of the centre disc. Each disc works with its own, separate set of bronze rollers, separated by Teflon washers, which should reduce the friction forces. In the article are presented the results of the tests of the cycloid gear prototype determining basic parameters such as efficiency, torque fluctuations on the input and output shaft, housing vibrations in three directions. FFT analysis of registered parameters showed high compliance of the designated frequencies for different measured signals. The assumed advantages, i.e. high efficiency and low vibration level in the tested range, were confirmed. The results are presented in the form of three-dimensional graphs as functions of speed and torque. High efficiency (80-90) % was obtained for load moments above 150 Nm, which is practically independent of the rotational speed with which the cycloid gearing works. The oil temperature during the tests was maintained in the range of 36°C ± 2°C. The maximal torque of the cycloidal gear was 500 Nm.

Keywords: *cycloidal gear, efficiency, vibrations*

1. Introduction

The term "cycloidal gearbox" comes from the method of the satellite 1 teeth forming, Fig. 1, which shape utilizes the epicycloid curve above the pitch circle and hypocycloid curve below the pitch circle [1, 2, 5]. The use of these curves forms a tooth with a relatively wide top and the base, what significantly affects its durability, even allowing for 5 times torque overload.

Internal gear meshing consists of: moving gear 1 and stationary gear 2. The cycloid curve in gear 2 is usually replaced by rollers (lowering friction). This solution has low influence on the nature of meshing, but significantly reduced costs.

Because of eccentric "e" on the shaft 3, it is possible to move the satellite 1 around the gear 2. The trajectory of this move consisting of relative rotational movements.

Flange 4 with the pins and the rollers (rolling friction) cooperates with holes in the gear 1. The ratio of this mechanism is equal to 1, so $\omega_1=\omega_4$ and $M_1=M_4$.

Properly designed geometry and selected tolerances of gear 1 working surfaces and the material from which it is made, have a significant impact on the cycloid gear working behaviour. Cycloid drive can be described by the following kinematic equation:

$$\omega_1 r_1 - \omega_2 r_2 + \omega_3 e = 0. \tag{1}$$

For cases presented in Fig. 1, substituting the appropriate angular velocity $\omega_i=0$ and so:

$$\begin{aligned}
&\text{if } \omega_3=0 \text{ then } \omega_1 = \omega_2 \frac{r_2}{r_1} \text{ and the basic ratio } i_o = \frac{\omega_1}{\omega_2} = \frac{z_2}{z_1}\\
&\text{if } \omega_2=0 \text{ then } \omega_1 = -\omega_3 \frac{e}{r_1} \text{ and the ratio } i_{31} = \frac{\omega_3}{\omega_1} = -\frac{r_1}{e} = \frac{1}{1-i_o}\\
&\text{if } \omega_1=0 \text{ then } \omega_2 = \omega_3 \frac{e}{r_2} \text{ and the ratio } i_{32} = \frac{\omega_3}{\omega_2} = \frac{r_2}{e} = \frac{i_o}{i_o-1}.
\end{aligned} \tag{2}$$

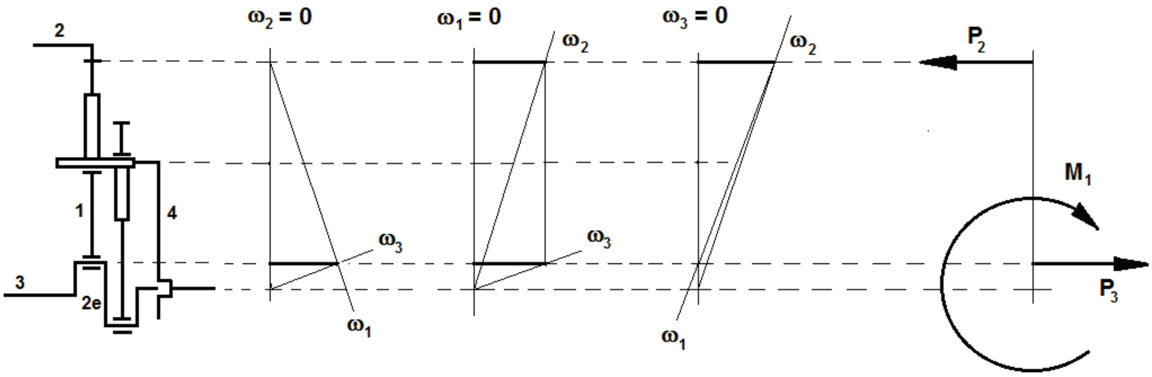

Fig. 1. Cycloid gear kinematic

From the torques equilibrium conditions, acting on the three gear shafts follows the relationship:

$$P_3 e - P_2 r_2 + M_1 = 0. \tag{3}$$

For the most common solution, ω_2=0, it obtained:

$$M_1 = -P_3 e \, i_{31} \eta_o, \tag{4}$$

where:
η_o – gearbox efficiency.

Torque acting on a single gear in the gearbox of "n" planet gears is "n" times smaller.

$$M_1' = \frac{M_1}{n}. \tag{5}$$

The literature about the cycloid drive geometry and the conditions to be met during the design process is wide [1-9]. The Polish literature is based on the monograph [1].

2. Research object

The most common design solution are gears with two discs, in which the reaction forces of bearings from the working discs create a bending moment on the high-speed input shaft, additionally affecting bearings. The introduction of a third disc, shifted by 180° from the centre dial, allows this moment to be reduced to zero, Fig. 2. To compensate for the unbalance of the shaft, the side discs have half the width of the centre disc. Each shield works with its own, separate set of bronze rollers, separated by Teflon washers, which should reduce the friction forces.

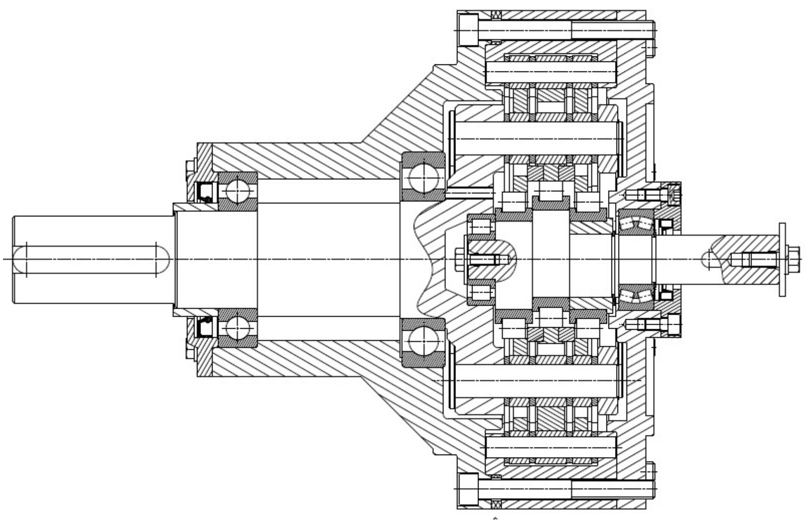

Fig. 2. Three Cycloidal Disc Gearbox Preparation – 2D Drawing of the Gearbox

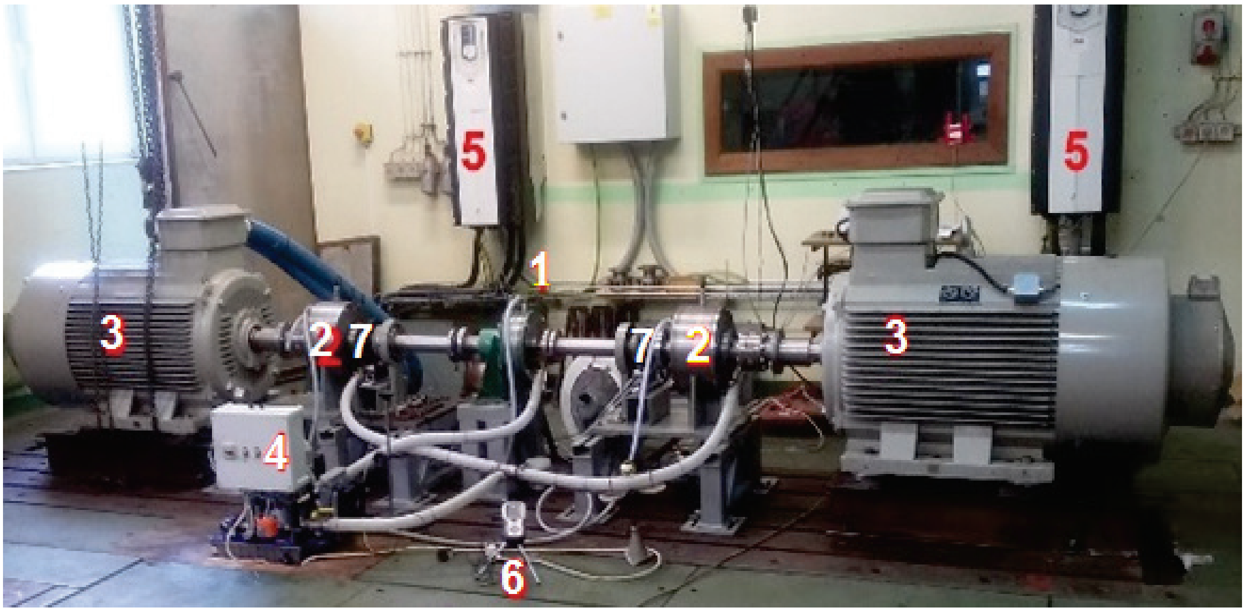

Fig. 3. Test Bench of Cycloidal Gearbox: 1 – cycloidal gearbox with three discs, 2 – planetary gear set, 3 – electric motor end generator, 4 – oil supply system, 5 – converter, 6 – sound meter, 7 – torque end speed sensor

The test bench equipment (Fig. 3) consists of:

1. Two electrical motors “3” (200 kW), which are characterized by reversible work, i.e. can both operate as a motor and as a generator. The common DC voltage cable allows the transmission of electrical energy from the generator to the engine. In this way, the external power supply is charged only to cover energy losses much smaller than appear in the working drive system. This concept of the test is used at the Department of Vehicles and Fundamentals of Machine Design of Lodz University of Technology since 1995 [5].
2. Two ABB frequency converters “5”, which in the power supply have the rectifier bridge and input filter to reduce harmonic distortion from an external power supply.
3. The steering computer, which allows to plan and execute an experiment. Changing the parameters of electrical machines can be realized by step, trapezoidal or sinusoidal function. A limitation of electrical parameters is frequency converters “5” and motors “3”.
4. The acquisition system, Fig. 4 and HBM's measurement cards, enabling capture, archive and compile data from the test bench. This unit operates separately from the CPU computer to prevent mutual interference and increase safety.
5. A lubrication and cooling oil system 4 is adapted to the test object. The viscosity of the oil is checked on the separate test bench before and after the test. To keep the desired temperature inside the cycloid housing an oil cooling system were used. All three gearboxes were supply from the same oil tank. To execute the oil flow a pump unit “4” was installed in cooling circuit.
6. In order to reduce the vibration of test rig components, a set of laser sensors “ShaftAlign” made by Prüftechnik was used to eliminate misalignment of all shafts and clutches in the driveline. After reaching the required accuracy in setting the cooperating connections, the test can be started.

Schematic view of the data acquisition hardware is shown on the Fig. 4. Catman data acquisition software (HBM) was used for acquisition, visualisation and analysis of measurement data.

Tests were carried out at constant torques on the output shaft of the tested gear. The speed was changed from the assumed minimum value to the maximum value and then back to the minimum again. Test bench control system was monitored by computer, which software allows for setting up the test cycles. The desired values were transmitted by the controller, inverters “5” to the respective electric machines “3”.

Measured values were archiving by the separate computer.

The following results apply for a cycloidal gear with:

– 3 Discs CYCLO Gearbox – prototype equipped in new output shaft and its bushes;
– Work mode: Reducer;
– Input speed (mean): 0-1800-0 rpm;
– Output torque (mean): 494.5 Nm;
– Type of oil: ATF M3 Plus;
– Temperature of oil: 37.4^{o}C.

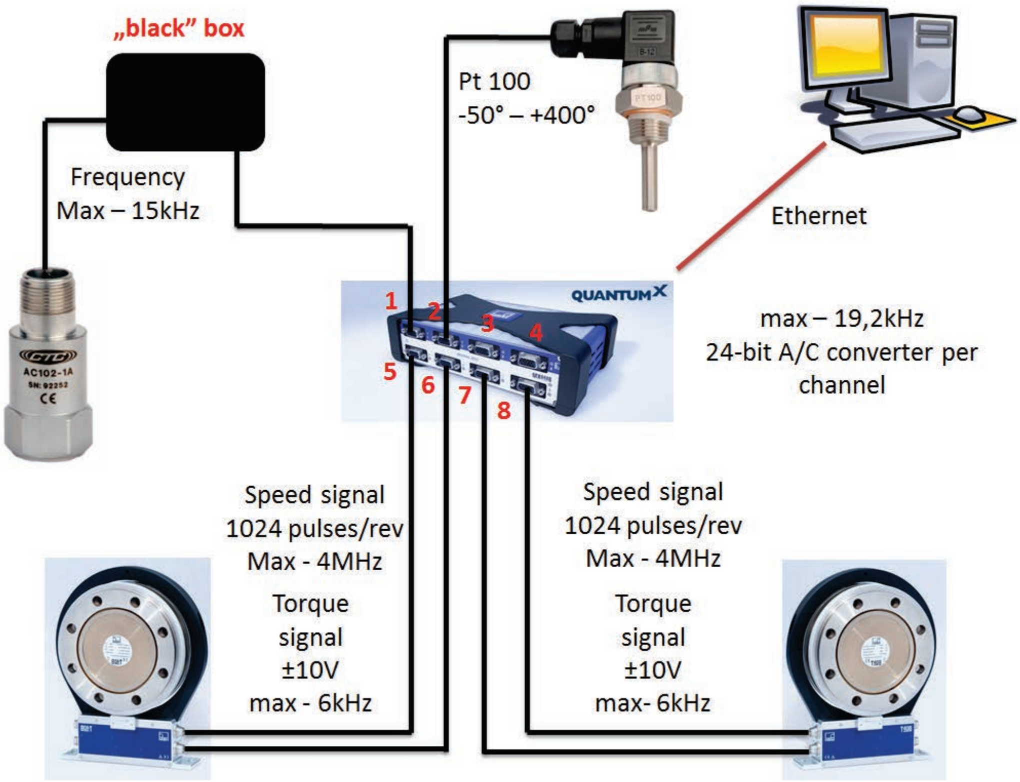

Fig. 4. Schematic view of the data acquisition hardware

As shown in Fig. 5, the test cycle consisted of the acceleration phase, steady speed and deceleration at constant torque on the brake. The acceleration and deceleration phases were carried out with the lowest possible accelerations, which allowed the analysis of vibrations as a function of rotational speed. The preliminary comparison of the curves from the measurements indicates the existence of characteristic speeds at which there are resonant oscillations overlapping the vibrations resulting from the kinematics of the cycloidal gear. The most obvious are the vibrations of the torque on the transmission shafts. The basic feature is the same nature of moment changes, which is confirmed by the theory that resonant frequencies are characteristic for a given transmission.

Another feature is the almost full compliance of the shape of the output torque and body oscillation vibrations (reaction torque) registered by a laser sensor. The input torque is about 19 times smaller, by the ratio. As is known, the observation of body vibrations is used in the diagnostics of machines with rotating elements.

There has also been a change in the noise level of the entire station in the resonant ranges, despite the measurement with a low accuracy microphone. The noise level of the cycloidal gear itself is certainly lower than registered.

Quiet operation and low vibrations when transferring large torques at variable rotational speeds are one of the basic features of cycloidal gears.

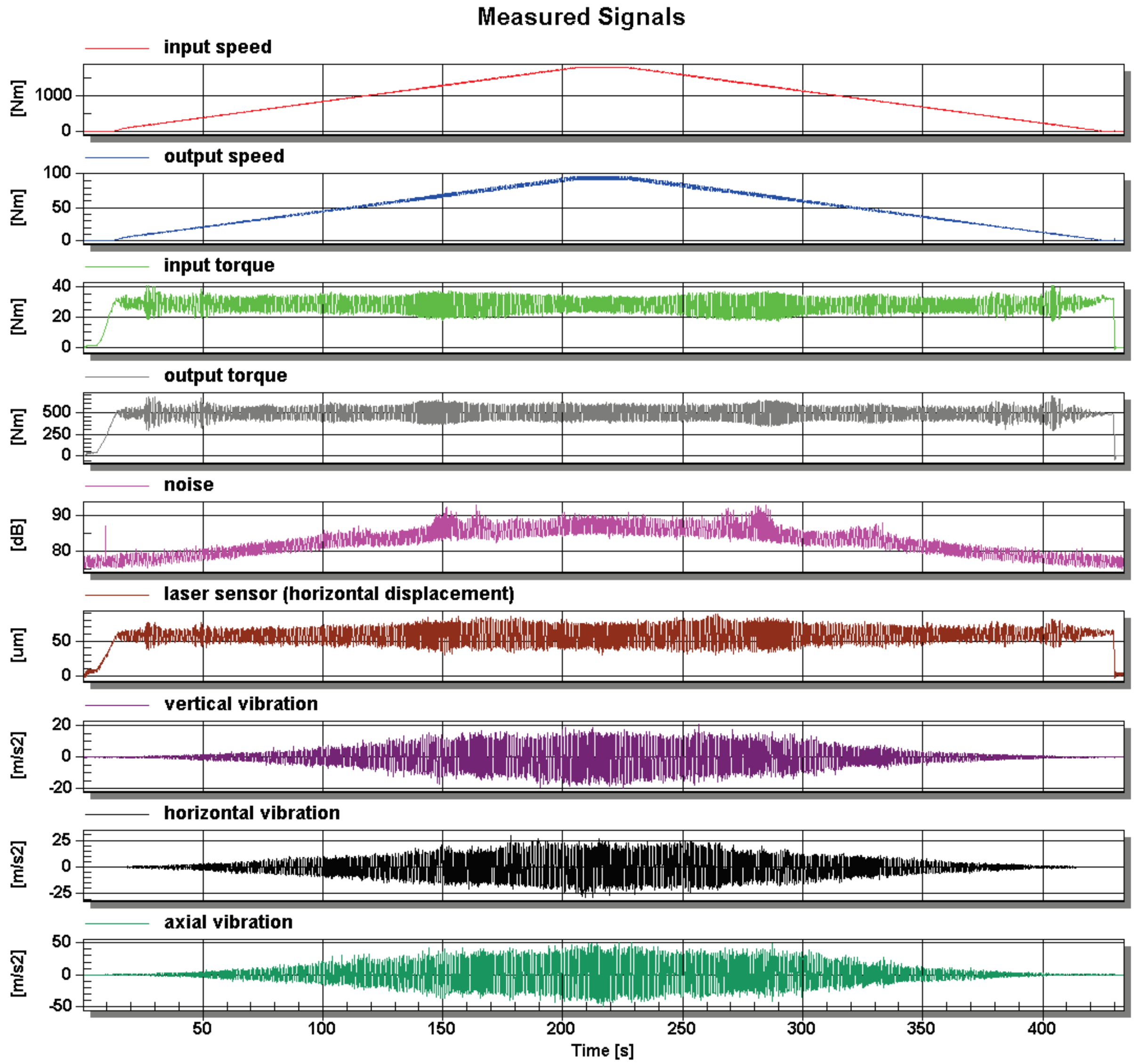

Fig. 5. Sample results from one sample

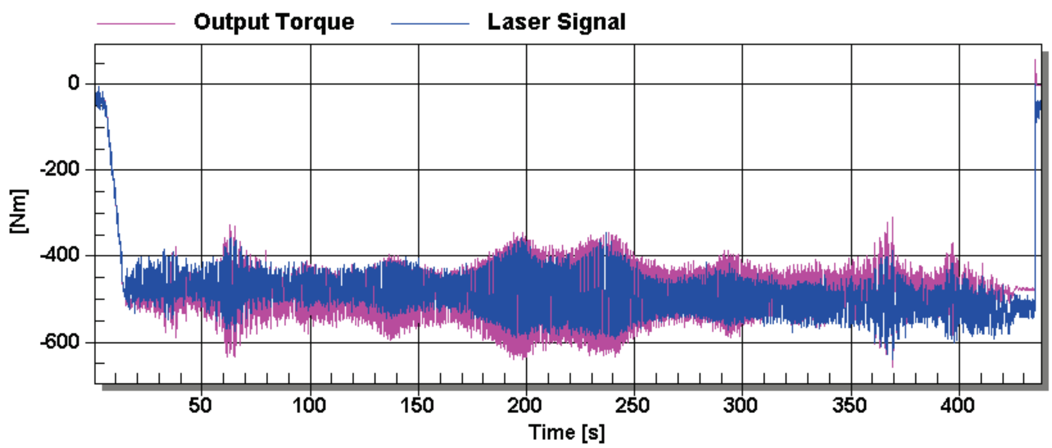

Fig. 6. Comparison of signals

The FFT analysis concerned of signals that were recorded at a constant speed. The nature of the analysed signals is more visible with the help of ZOOM of the measured time range, Fig. 7. The number of samples of the analysed signals was 20439 per one size.

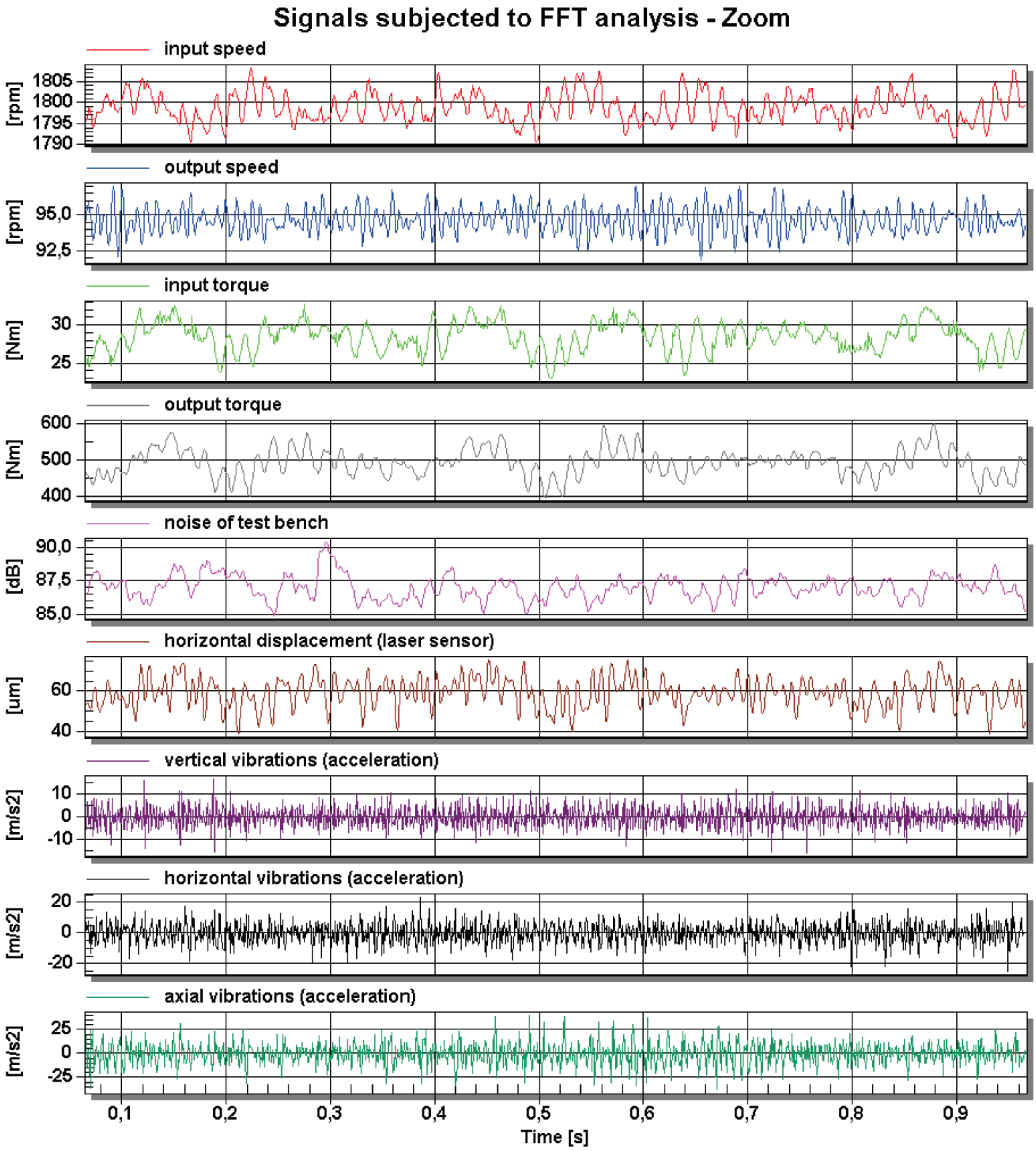

Fig. 7. Exemplary parameter waveforms during steady speed subjected to FFT analysis

You can see almost "copied" waveforms of the torque at the input and output of the cycloidal gear.

The FFT analysis of the measured signals was made with 1024 point for RMS amplitudes, Fig. 8. The other parameter sampling (or capture) frequency = 1200 Hz for input and output speeds, input and output torques, laser signal, and noise. It is 2400 Hz for all piezoelectric signals (vibrations). Therefore, spectrum frequency ranges are accordingly (0-600) Hz and (0-1200) Hz respectively. Frequency resolutions (spacing between frequency lines) are accordingly 1.17 Hz and 2.34 Hz.

Noteworthy is the high compatibility of the frequency characteristics of the compared parameters and so:

1. In the frequency range up to 60 Hz, six parameters show such compatibility. The FFT waveforms for the torque on both shafts and the input speed are almost identical. For the

rotational speed of the output shaft, less compliance was obtained, probably due to 19 times smaller values and hence higher measurement errors;

2. Above 60 Hz, the indication of piezoelectric vibration sensors can be considered correct. Low frequencies are skipped due to the principle of operation of these sensors. FFT waveforms determined for three axes are almost identical and consistent with the results obtained for shaft rotation speeds;
3. The widest range of characteristic frequencies allows determining the registration of rotational speeds and amplitude of body oscillations.

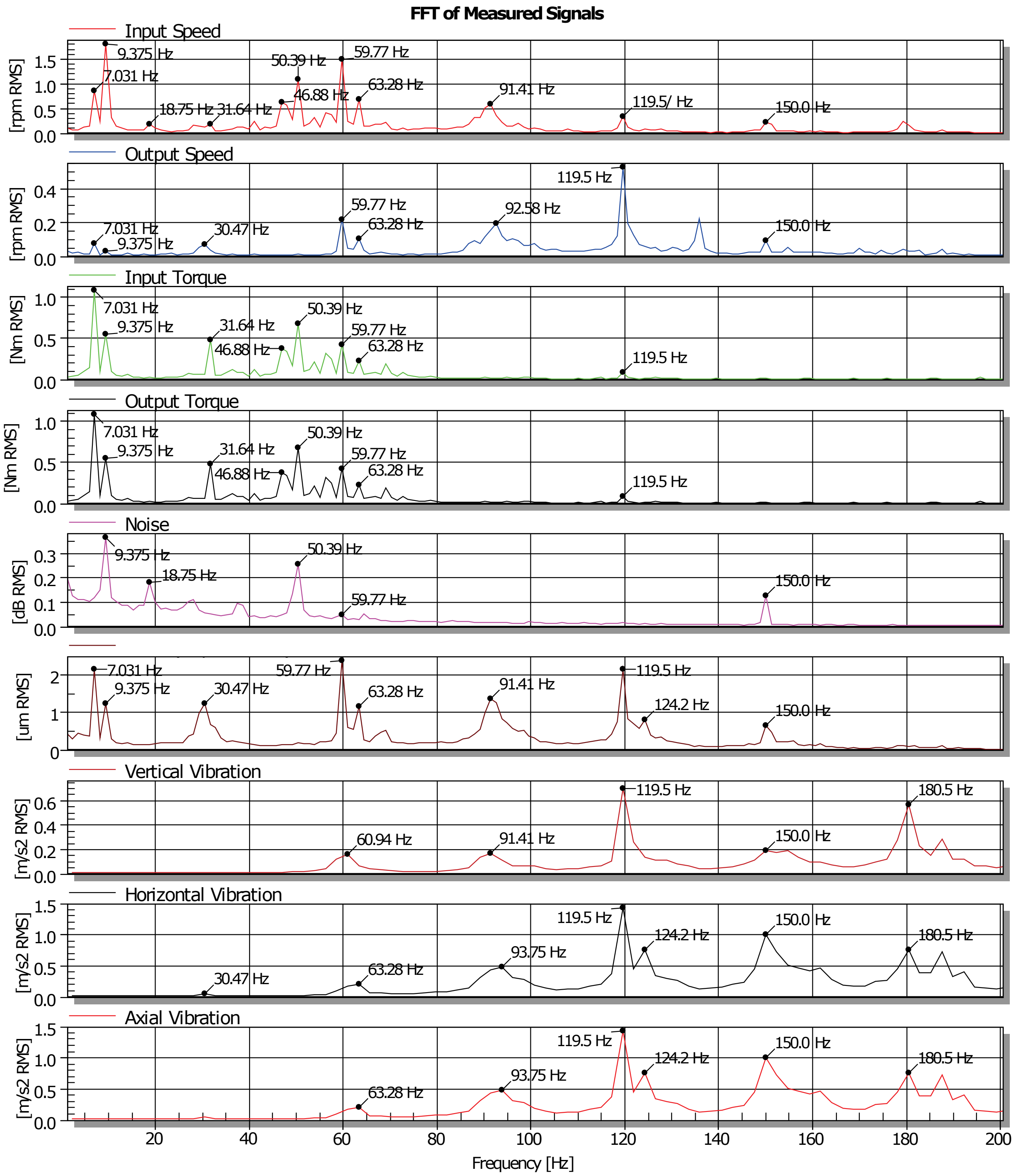

Fig. 8. Set of FFT results for the trial of Fig. 7

The frequencies of table 1 are theoretical, significant frequencies for a cycloidal gear. The obtained spectra of the analysed signals should look for frequencies close to these values.

Tab. 1. Theoretical values of significant frequencies for High Speed 1798 rpm and Low Speed 94.6 rpm

	DMF [Hz]	*FH of HS [Hz]*	*FH of LS [Hz]*
1 th	31.5	30.0	1.6
2 th	63.1	59.9	3.2
3 th	94.6	90	4.7
4 th	126.2	119.9	6.3
5 th	157.7	149.8	7.9
6 th	189.3	179.8	9.5

Meaning of abbreviations in the table with theoretical values of significant frequencies:
DMF – Disc Mesh Frequency;
n th – n harmonic of frequency;
FH of HS – Fundamental Frequency of High Speed;
FH of LS = Fundamental Frequency of Low Speed.

3. Discussing the Results

1. As low frequencies in the spectrum, there are values of 7.031 Hz and 9.375 Hz. These frequencies approximately correspond to the 5th (7.9 Hz) and 6th (9.5 Hz) harmonics of the output shaft speed. They occur in the spectra of speed, torques, laser sensor readings and noise signals (without a frequency of 7.031 Hz for noise).
2. The frequency source 18.75 Hz (occurring in the input speed and noise spectra), has not been identified. The reducer and multiplier are characterized by different parameters.
3. Frequencies of 30.47 Hz and 31.64 Hz correspond respectively to the fundamental frequency of the input shaft speed (30 Hz) and the disk meshing frequency (31.5 Hz). The fundamental frequency of the input speed appeared in the spectra of the output speed, laser signal and horizontal vibrations. In contrast, the disk meshing frequency appeared in the input speed, input torque and output torque spectra.
4. The frequency sources 46.88 Hz and 50.39 Hz (occurring in the all signals spectra), has not been identified. The reducer and multiplier are characterized by different parameters.
5. Frequencies of 59.77 Hz and 63.28 Hz correspond respectively to the 2nd harmonic of frequency of the input shaft speed (59.9 Hz) and the 2nd harmonic of disk meshing frequency (63.1 Hz).
6. Frequencies of 91.41 Hz (92.58 Hz) and 93.75 Hz correspond respectively to the 3rd harmonic of frequency of the input shaft speed (90 Hz) and the 3rd harmonic of disk meshing frequency (94.6 Hz).
7. Frequencies of 119.5 Hz and 124.2 Hz correspond respectively to the 4th harmonic of frequency of the input shaft speed (119.9 Hz) and the 4th harmonic of disk meshing frequency (126.2 Hz).
8. The spectra of the analysed signals contain further frequencies corresponding to the higher harmonics of the input speed of the cycloid gear.
9. Frequencies above 130 Hz originate from planetary gears that work as reducer and multiplier.

The values of the torques given on the right side of Fig. 9-12 correspond to the loads applied on the brake (generator). Taking into account the multiplier ratio i_m = 4.4063 and its efficiency, the actual loading moment is obtained. As can be seen from the above statement, the oscillatory loads that change their waveforms are additionally superimposed on the fixed values. The ranges with a distinctive resonant character are visible.

The input shaft additionally overlaps the own rotational resistance of the cycloidal gear. The same type of oscillatory loads can also be seen here. Their FFT analysis was presented earlier.

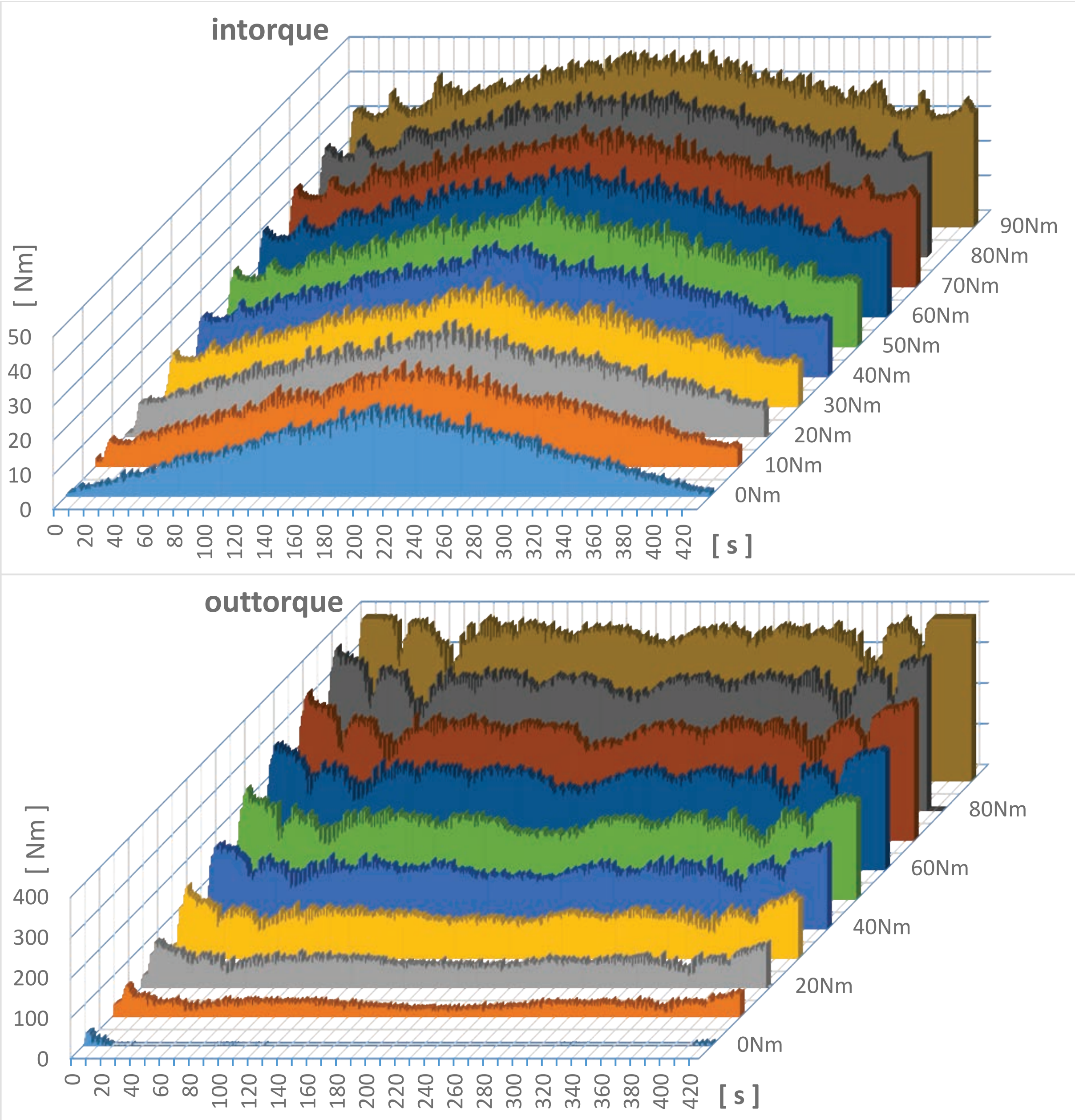

Fig. 9. Waveforms on the transmission shafts

In the acceleration courses in Fig. 10, no significant differences were recorded. Acceleration of horizontal and vertical vibrations is almost the same. They depend clearly on the speed and, to a small extent, on the torques transmitted by the transmission.

Accelerations in the axial direction are slightly larger than vertical and horizontal, and slightly increase with increasing torque of the load.

The deflection of the gearbox body does not depend on the angular velocities and torques, and its value indicates a very rigid construction of the fixings to the foundation (cast-iron position board).

The noise level increases with the increase of angular velocities and does not depend on the torque being transmitted. The values determined relate to the entire position and in this case constitute a very general assessment.

Noise measurement was made using DT-8852 noise level meter, Fig. 3. The measurement conditions did not meet the standards in this regard. Hence, the results give only a general view of its level.

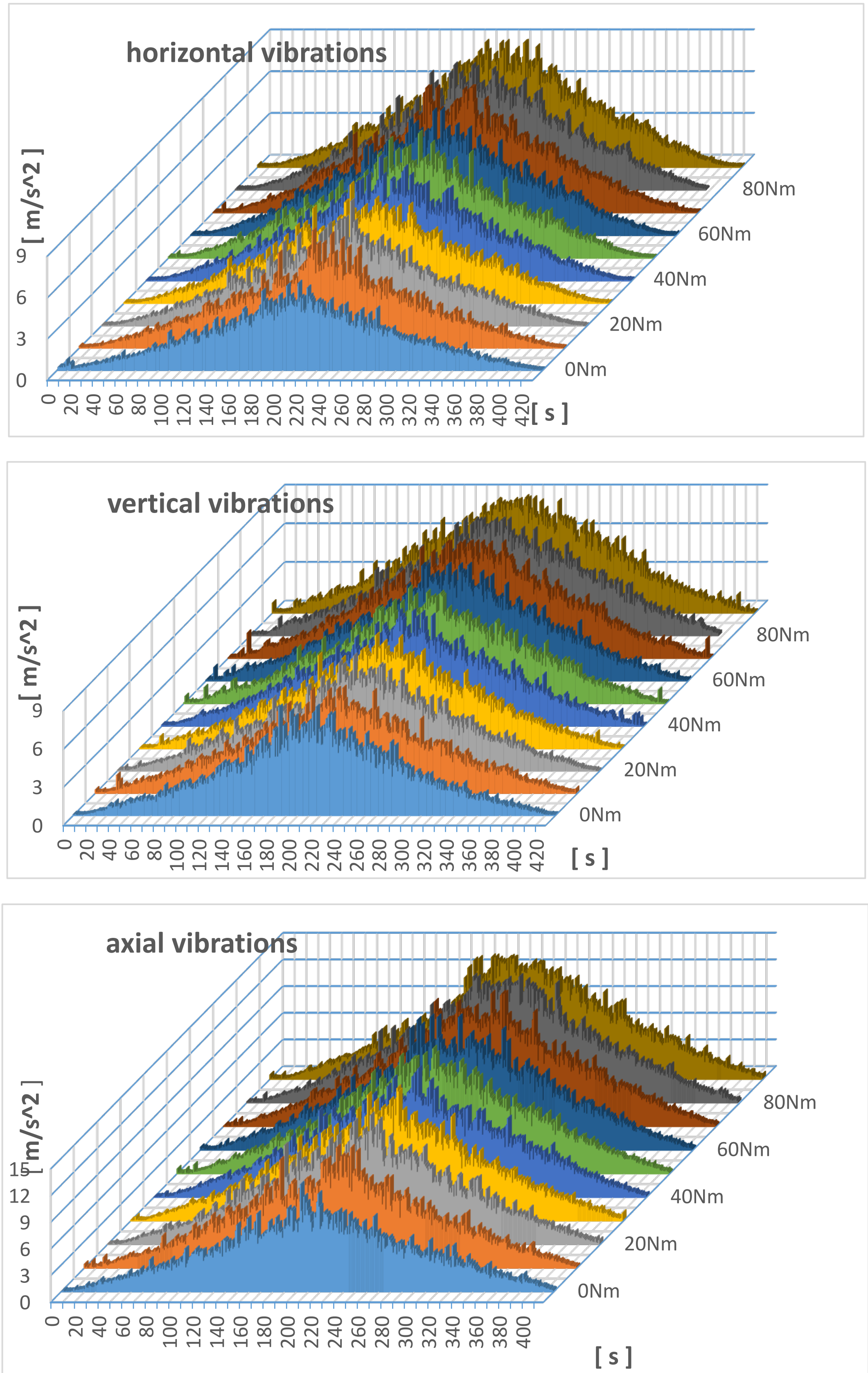

Fig. 10. Comparison of vibration acceleration in three directions

Figure 12 shows the total noise level generated by all station equipment during tests. The noise generated during standstill by the fans: motor, generator and inverters reached 73-74 dBA and only through the oil system is 64-65 dBA.

Figure 13 shows the course of efficiency as a function of speed and torque. It can be considered that the efficiency is slightly dependent on the rotational speed.

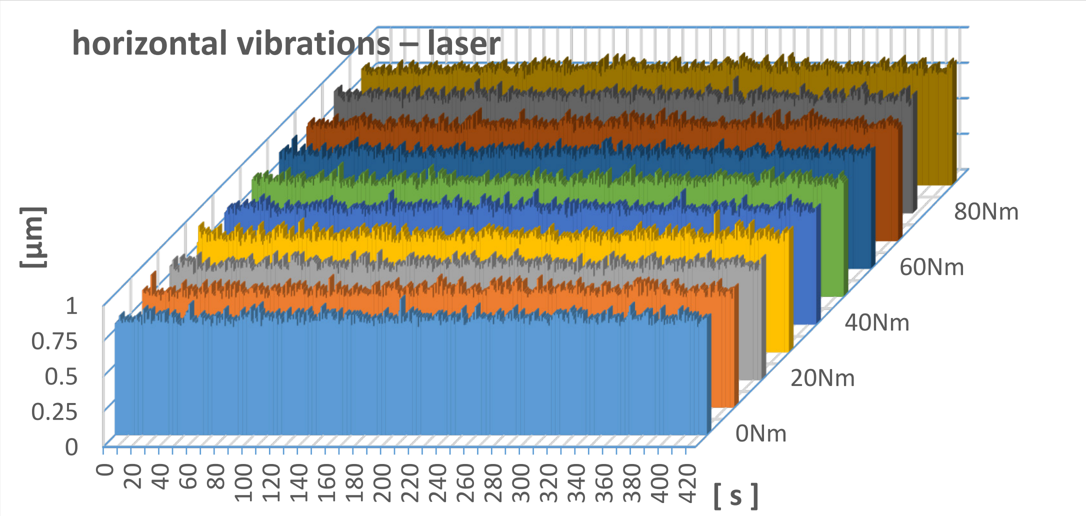

Fig. 11. Comparison of the amplitude of the body vibrations

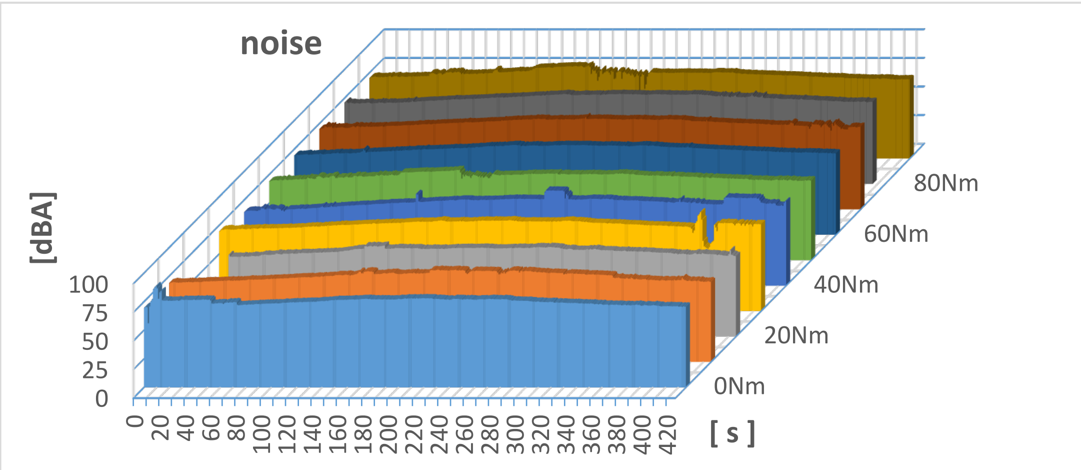

Fig. 12. Comparison of the noise level of the position

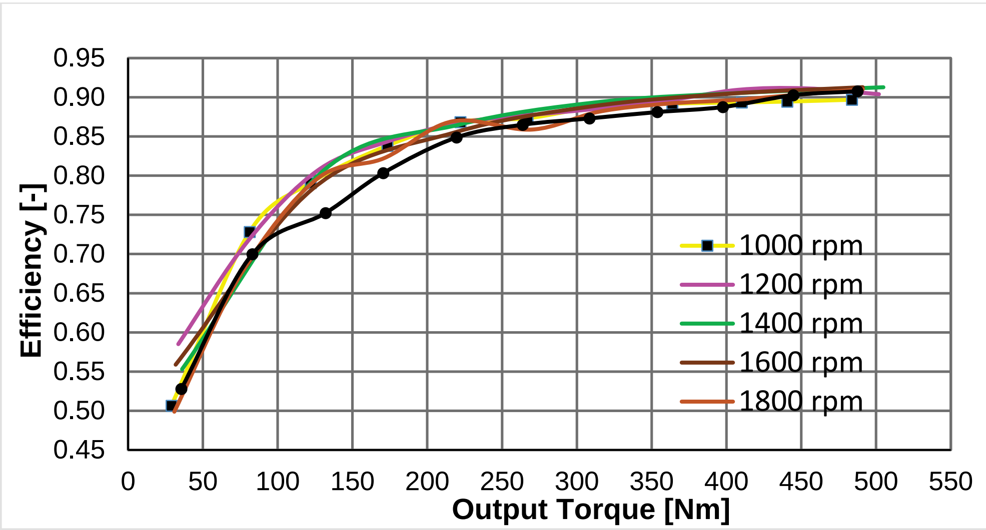

Fig.13. Gearbox efficiency as a function of speed and torque

4. Conclusion

Experimental studies have confirmed the following features:

1. The results of the FFT analysis of the measured signals indicate with high probability the misalignment of the input shaft of the cycloidal gear.
2. Based on the obtained results, it is difficult to say whether the output shaft of the cycloidal gear is misaligned. The solution to this problem can be to reanalyse the FFT of the measured signals with a lower frequency resolution.
3. High efficiency 80% – 90% for load moments above 150 Nm, practically independent of rotational speeds. The oil temperature during the tests was maintained in the range of 36°C ± 2°C. The transmission torque is 500 Nm.
4. Low amplitude vibrations and accelerations in the whole operating range are slightly dependent on torque.
5. Relatively low noise level of the entire station.
6. Narrow hysteresis of speed and torque taking into account the amplitude of vibrations of these values.

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