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Some aspects regarding planetary precessional transmissions dynamics

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Abstract. A mechanical system theoretically should not generate any vibrations and noise to be considered dynamically an ideal system, because vibrations mean a loss of energy. In planetary precessional reducers vibration can appear at bearings, gears, misaligned shafts, unbalanced rotors, couplings. Changes in dynamic process affects not only normal behaviors of mechanical systems but also the forces that act here. Regarding this aspects, regular vibration measurements during the operating speed provide information about any necessary repairs or maintenance. Here vibration research was made by using GUNT PT500 Machinery Diagnostic System and vibration signals was evaluated by using GUNT PT500.04 software which allow to perform correct Fast Fourier Transformation analysis. Data acquisition was made by using two piezoelectric accelerations sensor type IMI603C01 and one reference photoelectric sensor to record the shaft speed. Research concluded in this paper is made on precessional gearbox 2K-H Type. In first phase the crankshaft was installed with self-alignment on 2 radial bearings with cylindrical roller and second research phase was determination of the resonance domain and performing the spectral analysis by recording several vibrogram.

1. Introduction

In planetary precessional transmissions, vibration can appear at bearings, gears, misaligned shafts, unbalanced rotors, couplings. Changes in dynamic process affects not only normal behaviors of mechanical systems but also the forces that act here [1, 2].

Generation sources of vibration and its behaviors can conduct to dimensional distortions of sliding or rolling parts in direct contact, dimensional distortions at machine elements that performs rolling touching, dynamic or static imbalance and assembly errors because precessional reducer is a dynamical device where mechanical vibration can appear [2, 3].

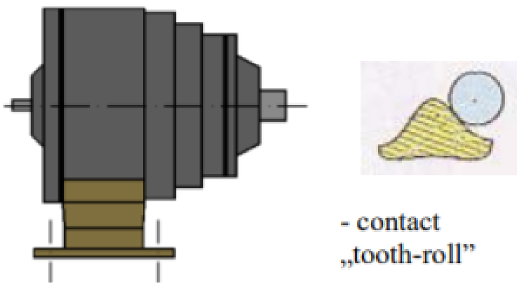


Figure 1. Researched precessional gearbox 2K-H Type.

Vibration analysis was carried out on power precessional reducer type 2K-H (figure 1) with the transmission ratio i=-10.5, nominal torque T=40 Nm and operational speed n=800 min^{-1}.

2. Constructive solution for minimizing assembly vibration errors in researched gearbox

Assembling process may occur some assembly errors (excessive clearance or too high grip) even if "satellite block and crank shaft" components node were balanced and executed perfectly. One of the most met assembly faults is the non-coincidence of the satellite precession center (point O in figure 2) with the intersection point of the O1-O1 axis of the crank (inclined part) and O-O of the drive shaft. An effective solution to compensate this error is to install the satellite and crank shaft, on radial bearings with cylindrical roller, which in turn is an additional source of vibration [2, 5].

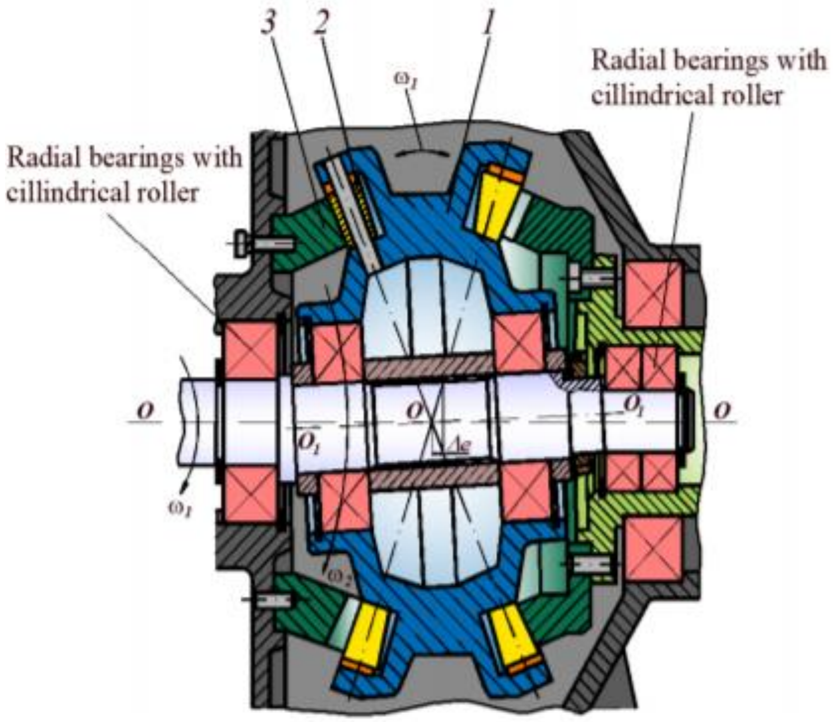


Figure 2. Constructive solutions.

In figure 2 it is shown one of the various constructive solutions regarding precessional transmissions where auto-aligning operation of satellite block 1 in this case can be described beneath. The constant component of the primary errors in the precessional planetary transmission elements, produce some deviation Δe among precession axis of the satellite block 1, which does not depend on the reciprocal position of the transmission parts. After the new position of the satellite block 1, the precession movement with the frequency rotation $\omega 1$ (figure 2) and the frequency rotation ω_2, trajectory is determined by the first harmonic of the actual error in the satellite block gearing (radial and frontal contact of the crowns): $\omega_2=\omega_1/z_2$ – where z_2 represents the number of rollers of the satellite crown, that contact with the fixed central toothed wheel 3. The possibility of self-alignment, which will allow to obtain compensation of the manufacturing and installation errors of the contacting parts, us result we obtain a reduction of the vibration and noise level, by installation of the satellite block 1 on radial bearings, with cylindrical roller [2].

To minimize static and dynamic imbalance was developed a new technical solution which will conduct to increase the reliability of the gearbox by reducing the dynamic load, in particular, on the bearings and gear elements. This solution is in the realization and patenting stage and for this reason will not be considered in this paper [4, 5].

3. Research on the vibration behavior of the precessional planetary 2K-H reducer

The experiments were performed on the test stand (figure 3) in the laboratory "Fine Mechanics" of the "Fundamentals of Machinery Design" department in a closed room with rigid floor. The experimental stand consists from the test table GUNT PT500.01 made of aluminum profile provided with T-shaped channels, in which is fixed electric drive with power $P = 0.36$ kW, precessional planetary gearbox 2K-H with transmission ratio $i = -10.5$ and the maximum torque $T_2 = 40$ Nm, the electromagnetic brake type ПТ-6 nominal torque T=40 Nm.

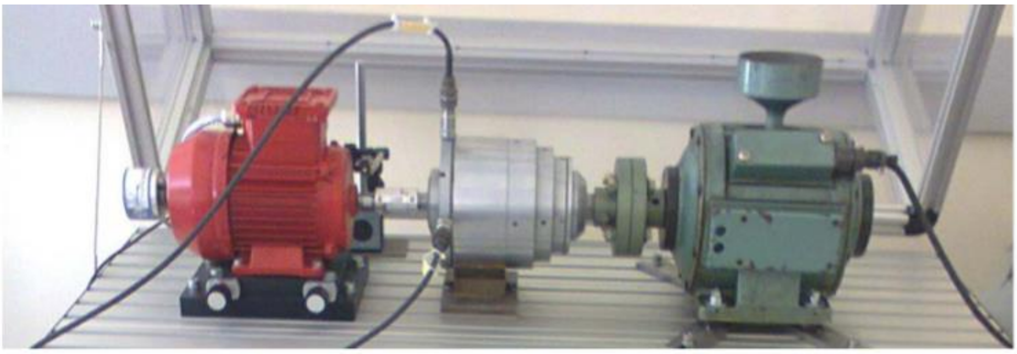


Figure 3. Researched planetary precessional gearbox 2K-H Type.

To compensate the execution and assembly errors, the crankshaft was installed with self-alignment on 2 radial bearings with cylindrical roller (figure 4).

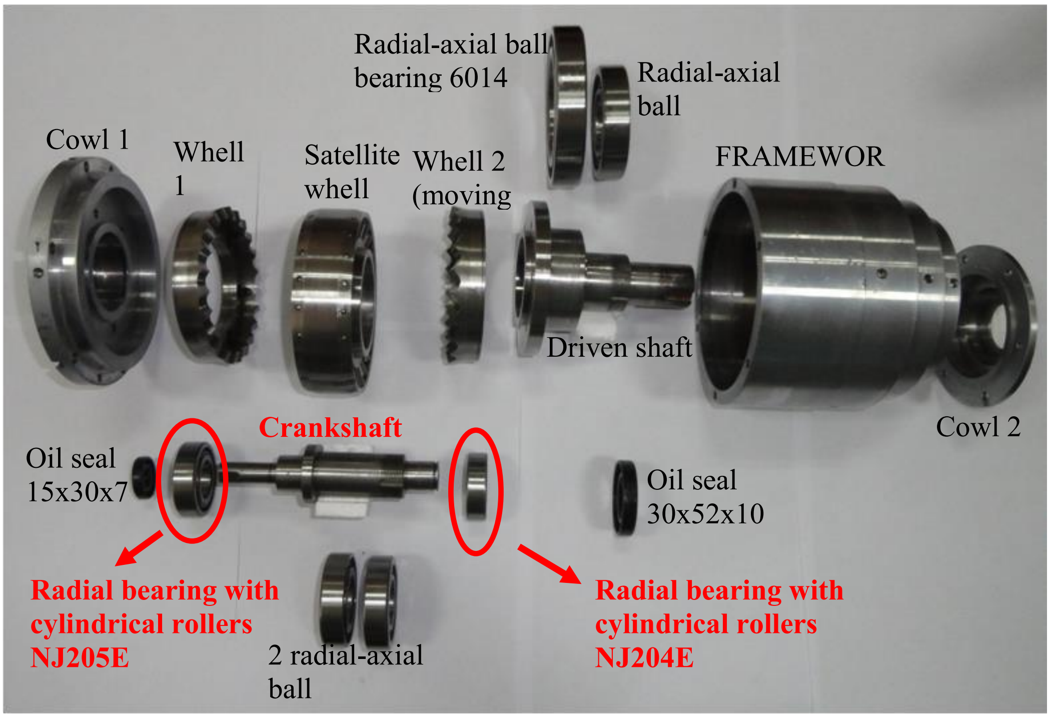


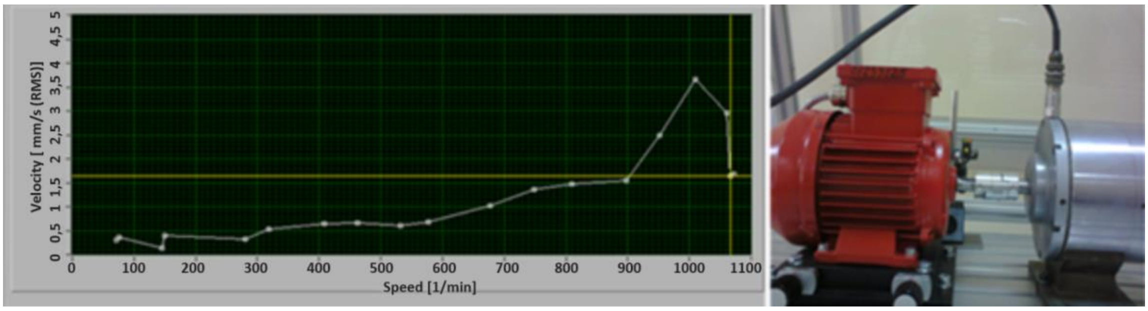
Figure 4. Component parts of the 2K-H precessional researched reducer.

4. Determination of the resonance domain

A trend in modern technology is to increase the speed of machines. This is justified by reducing the size of the machines to the same transmitting power. From the relation of the power P = F · v, it results that, at the same power, the force decreases with the increase of the speed and, consequently, the machine parts can have smaller dimensions.

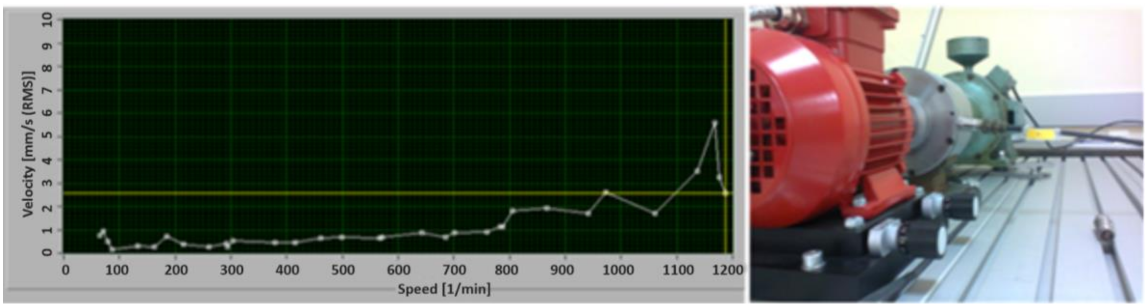
However, the increase in speed is limited by the inertial forces that occur during operation and which cause additional vibrations and stresses in the machine parts. It is known that any elastic body taken out of equilibrium oscillates with its own frequency, until the deformation energy is damped. The effect of centrifugal forces and vibrations is an important factor, which must be taken into account when we are designing gearboxes. The inertia forces vary periodically, resulting in vibrations in both the reducer and the foundation. If at some point, the frequency of forced oscillations caused by inertial forces (disturbing force) coincides with the frequency of the oscillations of an element, mechanism or machine, the phenomenon of mechanical resonance occurs, which can lead to crash of elements in the reducer or destruction of connections. among the elements of the mechanism. The vibration check in the case of reducers consists in determining the natural frequency (resonance range) and therefore the corresponding speeds which are called critical range. The purpose is that the operating speed does not coincide with any of the speeds that are found in the resonance range or is not too close to it. The coincidence of the critical speed with the operating speed is accompanied by the appearance of the resonance phenomenon, situation in which the amplitude of the own vibration can increase until the occurrence of damages or accidents [6].

The resonance range for the 2K-H precession reducer was determined on the 3 basic directions (vertical, horizontal and axial) by mounting the accelerometers at the respective points on the rolling bearing. In order to determine the resonance range, the diagnostic system of the PT 500 machine parts and the PT500.04 software “Traking Analysis” module of the German company GUNT were used. After starting the electric drive, the measurement starts, slowly adjusting the speed by accelerating from 0 to 1300 rpm, the image with the diagram is recorded from computer screen with the "Print Screen" key and saved in a file on computer, the measurement is continued, slowly adjusting the speed by braking from 1300 to 0 rpm, and on the obtained diagrams the resonance range and the critical speed are identified. Next three figures are shown the diagrams obtained in vertical direction (figure 5), in horizontal direction (figure 6) and, respectively, in axial direction (figure 7).



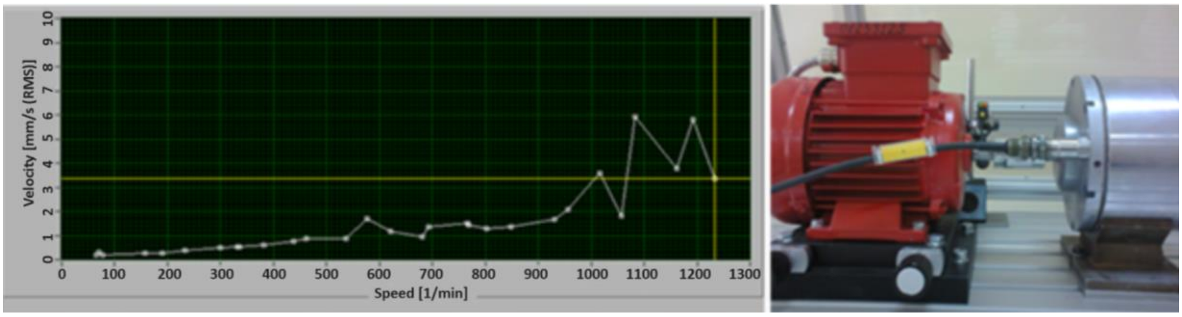
(a) (b)

Figure 5. Diagram in vertical direction: (a) - critical range 900 - 1200 min^{-1}; critical speed 1000 min^{-1}; (b) - vertical direction.



(a) (b)

Figure 6. Diagram in horizontal direction: (a) - critical range 900 - 1200 min^{-1}; critical speed 1100 min^{-1}; (b) - horizontal direction.



(a) (b)

Figure 7. Diagram in axial direction: (a) - critical range 900 - 1200 min^{-1}; critical speed 1100 min^{-1}; (b) - axial direction.

From the analysis of the resonance range, it was decided to choose the nominal speed at the regime phase n_{nom} = 800 min^{-1}, and the loading moment of the reducer was chosen maximal Tn = 25 Nm after brake calibration (figure 8), as much as 0.36 kW electric drive allowed.

Figure 8. Brake calibration.

5. Vibrodiagnosis (spectral analysis)

Spectral analysis is one of several powerfull techniques needed for analyzing and post-precesing researched and analyzed data. Analyzed data are acquisition data that have been taken in one, two, or three-dimensional space, and/or time. Spectral analysis is one of the most essential used methods for data research and analysis in oceanography, geophysics, astronomy, atmospheric science, metallography and last but not least in engineering too. In this paper vibration research, the spectral content of measured data gives information on the defects, wear and other information of mechanical parts under study [5].

The next step in the experimental research of the precessional reducer 2K-H was the spectral analysis of the reducer at the operating speed regime (n_{nom} = 800 min^{-1} and $Tnom$ = 25 Nm).

The PT500.04 GUNT software was used to draw up the vibrograms in the vertical (figure 10), horizontal (figure 11) and axial direction (figure 12), using the “Frequency Spectrum” module. To determine the degree of severity (damage index) of the vibrations, which must not exceed the value 4.5 according to the norms for evaluating the vibration level ISO 10816-1, the “Envelope Analysis” module was used (figure 9).

(a) (b) (c) (d)

Figure 9. Damage index: (a) vertical; (b) horizontal; (c) axial direction; (d) operating speed.

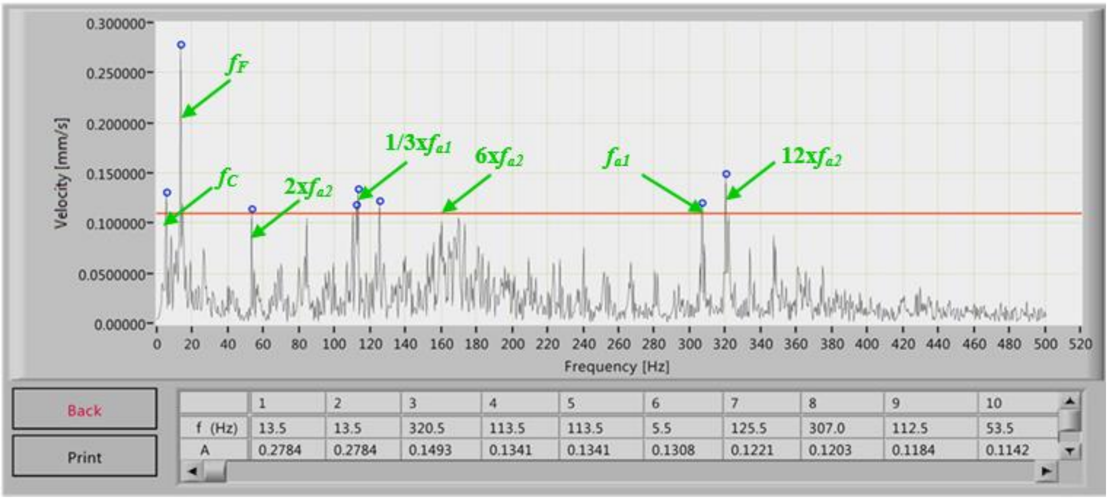


Figure 10. Vibrogram recorded in the vertical direction at $n_n = 800$ min^{-1} and $T_n = 25$ Nm.

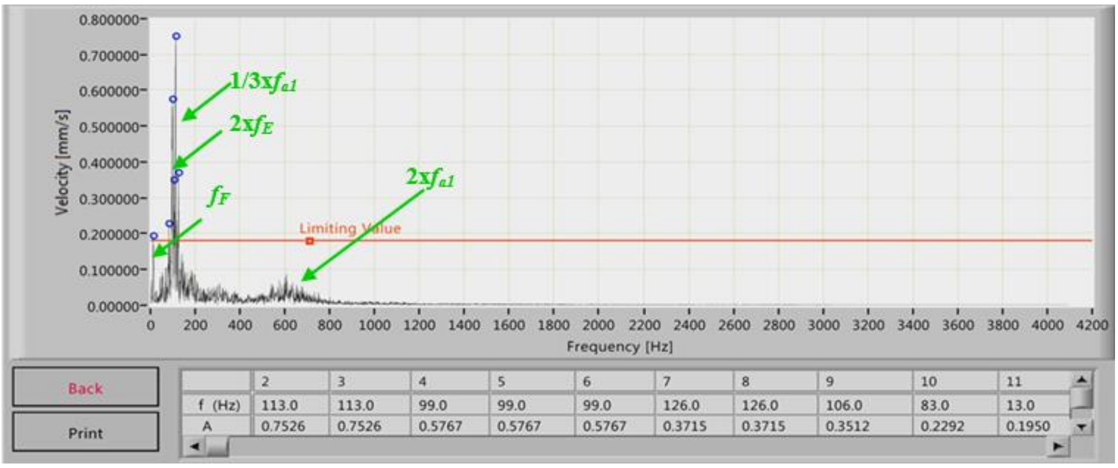


Figure 11. Vibrogram recorded in the horizontal direction at $n_n = 800$ min^{-1} and $T_n = 25$ Nm.

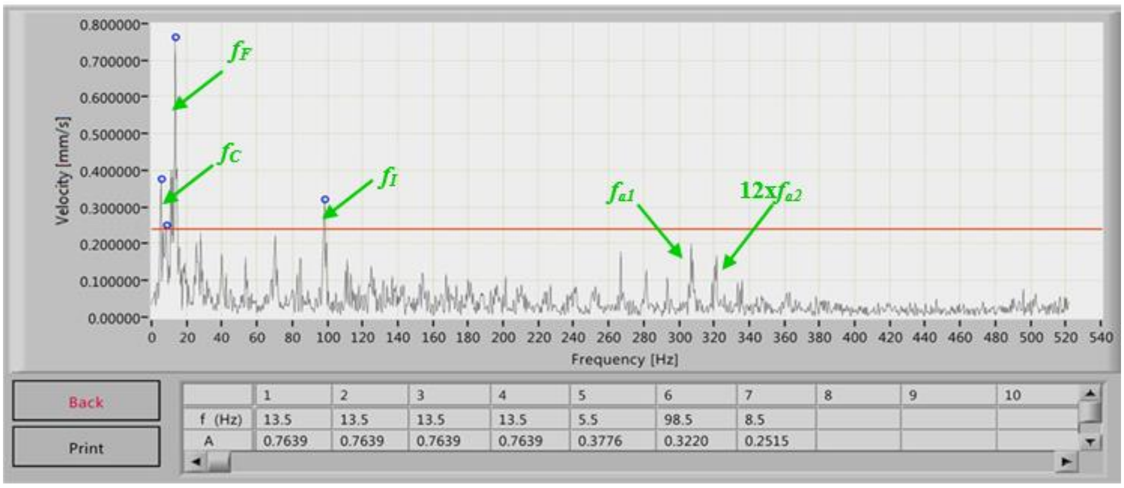


Figure 12. Vibrogram recorded in the axial direction at $n_n = 800$ min^{-1} and $T_n = 25$ Nm.

6. Conclusions and future research

Analyzing the vibration spectrum obtained on the 3 measuring directions, regarding the magnitude of the vibration amplitude, which did not exceed the value of 0.8 mm/s we can say with certainty that the researched 2K-H power reducer works optimally.

As we expected, the maximum amplitudes of the characteristic frequencies were obtained at approximately ***13.5 Hz*** - the fundamental frequency f_F; ***307 Hz*** - gearing frequency f_{a1}, between the the fixed precessional wheel with number of teeth z_1 = 23 and the crown 1 of the satellite with number of rollers z_2 = 22; ***26, 75 Hz*** - gearing frequency f_{a2}, between the crown 2 of the satellite with number of rollers z_3 = 22 and mobile precessional wheel with the number of teeth z_4 = 21. The specific harmonics of the gearing frequency detected in the frequency spectrum 2xf_{a2} = ***53.5 Hz***, 3xf_{a2} = *80.5 Hz*, 4xf_2 = ***107 Hz***, 12xf_{a2} = ***320 Hz***.

Also, in the frequency spectrum were detected the frequencies f_C = *5.5 Hz* specific to the oscillations of the bearing's cages and f_I = ***99 Hz*** specific to the oscillations of the inner rings of the bearings NJ205E and NJ204E on which the crank shaft is mounted with self-alignment.

Further research will be carried out on the same power 2K-H precessional gear box (figure 1) with the transmission ratio i=-10.5, nominal torque T=20 Nm and operation speed n=1000 min^{-1} regarding acoustical recording of noise curves using Brüel & Kjær Sound Level Meter Type 2250 Light.

Acknowledgments

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