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Precessional gear-box research regarding vibration activity behaviour

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Precessional gear-box research regarding vibration activity behaviour

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Abstract. An ideal dynamic system in general and precessional transmissions in particular should not generate any vibration, because vibration is equal with loss of energy and bigger sound level emission. In our previously research we were focused on different vibro-acoustical aspects regarding kinematical and power precessional planetary gear-box when precessional reducer is minimal and maximal filled with cooling and lubrication liquid at different operational speed (power precessional transmission K-H-V Type) and vibro-acoustical behaviour for kinematical precessional transmission 2K-H Type. In this paper I will analyse vibration behaviour for power precessional gearbox 2K-H Type regarding vibrations that occurs at bearings and main shaft misalignment. Vibration research and analysis will be made using GUNT PT500 Machinery Diagnostic System. Data acquisition will be made using acceleration sensor type IMI603C01 for FFT analysis in GUNT PT500.04 software. Obtained results will be compared with German standard VDI-2058 "Limit value for vibration severity".

1. Introduction

The purpose of modern-day machines in general and gearbox in particular is to carry out needs-based on maintenance, good working conditions and to minimize the repair and other servicing downtimes and dysfunctionalities. This increases the overall equipment effectiveness and optimizes the cost. The mechanical condition of a machine or a machine component can be accurately diagnosed from the nature and extend of vibration they generate. The aim is to detect damage as it occurs, allowing scheduled repairs or maintenance to be carried out in time [1].

Vibration in planetary precessional gear box occurs at bearings, gear wheels, misaligned shafts, imbalance rotating parts, couplings. If damage occurs, not only the dynamic processes change, but also the forces that act on system components [4].

2. Vibration generation sources in the planetary precessional gear

Precessional planetary transmission is a mechanical system subjected to mechanical oscillations generating vibrations and noise and can be divided into the following groups: dimensional deviations of sliding contact elements, dimensional deviations at machine parts that make rolling contact, static and dynamic imbalance and assembly errors vibration [1].

2.1. Dimensional deviation of sliding contact elements

Precessional gear units that engage with the engaging frequency $\omega_z = z \cdot \omega_1$ where z teeth number and ω_1 the angular velocity of the main shaft, are a source of vibration and noise and a transmission quality indicator. The roles of the satellite, which form a kinematic twist-axle roller, are also a source of vibration:

- dimensional and shape deviations of the rollers and axles;
- changing the position of the rollers in the solicited and unsolicited area of the satellite crowns as well as the stiffness of the contacts;
- uneven motion of tapered rollers as a result of different stresses depending on the position.

2.2. The Dimensional deviation at machine parts that make rolling contact

Precessional planetary transmission typically includes 3 pairs of bearings: the main shaft bearing, the crankshaft, which support the precession and rotation movement of the satellite shaft, and the driven shaft. The separate level of vibration and bearing noise influences the general level of pre-planetary planetary transmission and its competitiveness. The bearings act through different effects (direct or indirect) at the vibration and noise level of other elements (shafts, satellite block, housing) [4].

The roller bearing contact may have the following causes:

- changing the position of the rolling elements (rollers or balls) in the loaded or unloaded area of the bearings, in relation to the dimensional deviations and the number of rolling elements. Also, periodically modifying the stiffness of the contacts and the system by favouring the occurrence of parametric type vibrations;
- non-uniform motion of the rolling body due to different and complex loads (especially in the satellite block bearings) depending on the position; this uneven motion and strain also leads to friction and collisions of the wheel bodies or cages;
- making roll-over contacts on surfaces with dimensional, shape and position deviations: diameters, frontal or radial tread patterns, eccentricity, ovality, corrugation, roughness;
- the movement of rolling body over impurities placed on contact surfaces or localized defects or damage (Pitting tipping, abrasive wear, fingerprints, etc.)
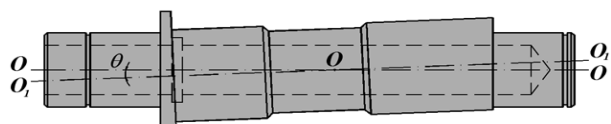
2.3. Static and dynamic imbalance

When the centre of mass of a moving or rotating element does not coincide with the centre of rotation (or precession), an imbalance is created, which may be in a plane called static or multi-plane imbalance, known as dynamic imbalance. Generally, in any machine, the shafts have a small imbalance that will generate a sinusoidal vibration with a frequency equal to that of the rotation motion.

The central deviation of the masses of the satellite block from the precession centre "O" with an "a" value generates phasic vibrations (Figure 1). Solution: static balancing of the satellite block at the design stage.
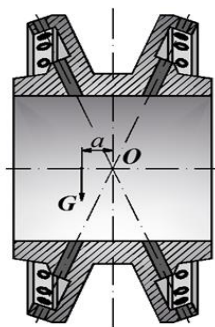
**Figure 1.** Satellite block imbalance   **Figure 2.** The crank shaft imbalance

In precessional transmissions, the imbalance is generated by the satellite and the crank shaft. The functional slope of the O_1-O_1 axis with the angle θ to the O-O axis of the precessional planetary transmission crank (Figure 2) also leads to the generation of vibrations. Solution: balancing the crank shaft. One of the solutions is shown in figure 2, when a cavity is made in the crank-shaft, the axis of which is inclined at an angle γ greater than the angle θ, and hole diameter is determined with formula

$$d = D\sqrt{tg\theta \: / \: tg\gamma}$$

2.4. Assembly vibration errors

The components of the "satellite block and crank shaft" node can be executed perfectly and balanced but assembling may cause different assembly errors (excessive play or too high grip). One of the most common assembly faults is the non-coincidence of the satellite precession centre (point O in figure 3) with the intersection point of the O_1-O_1 axis of the crank (inclined part) and O-O of the drive shaft. An effective solution to compensate for 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.
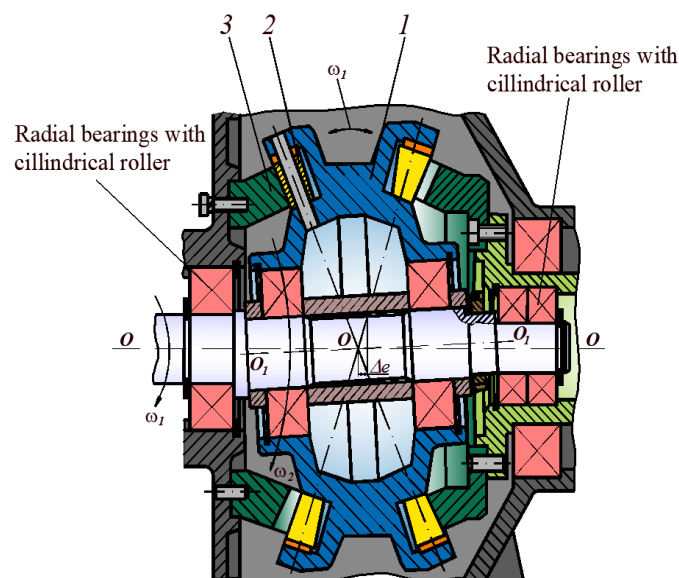
Figure 3. Satellite block and crank shaft node

There are various constructive solutions, next in figure 3 it is shown one of them [4]. The auto-aligning process of satellite block 1 in the precessional planetary transmission can be presented in the following way. The constant component of the primary errors of the precessional planetary transmission elements, which does not depend on the reciprocal position of the parts in the transmission, generates some deviation *Δe* of the satellite block 1 from the precession planetary transmission axis. Regarding this new position of the satellite block 1, the precession movement with the frequency ω_1 (Figure 3) and the frequency rotation ω_2, after a trajectory determined by the harmonic first of the actual error in the satellite block gear (radial and frontal impact of the crowns): $\omega_2 = \omega_1 / z_2$ – where z_2 represents the number of rollers of the satellite crown, in contact with the fixed central toothed wheel 3.

The installation of the satellite block 1 on radial bearings, with cylindrical roller give the possibility of self-alignment, which will allow to obtain compensation of the manufacturing and installation errors of the engaging elements, which perform in reduction of the vibration and noise level.

3. Precessional transmission vibration frequencies identification

3.1. Planetary precessional transmission type 2K-H

Precessional planetary transmissions, like any gear with toothed wheels, are characterized by various excitatory forces that generate various oscillations. We will then examine two structural types of 2K-H and K-H-V planetary planetary transmissions, which have different kinematic structures and different vibration generators [4].

In order to detect vibration sources in the precessional planetary transmission, it is necessary to determine the oscillations of the dynamic elements, which would allow the processing and analysis of vibrograms. The crank shaft rotates with the input frequency ω_1, so the crankshaft assembly and assembly errors will be pronounced around these frequencies:

$$\omega_1 = \frac{\pi n_1}{30}(s^{-1}).$$ (1)

The satellite block will also perform a spherical-space motion around the precession center (the O-O crankshaft intersection point (Figure 4) with the O_1-O_1 axis of the crankshaft inclined axis and the conical shaft axes of the crowns) with the frequency ω_1 and rotating motion around its geometrical axis with angular velocity:

$$\omega_2 = \frac{\omega_1}{i_1} = \omega_1 \cdot \frac{z_2}{z_1 - z_2},$$ (2)

where z_1 is the teeth number of the central wheel 1; z_2 - the rollers number of the satellite crown block.
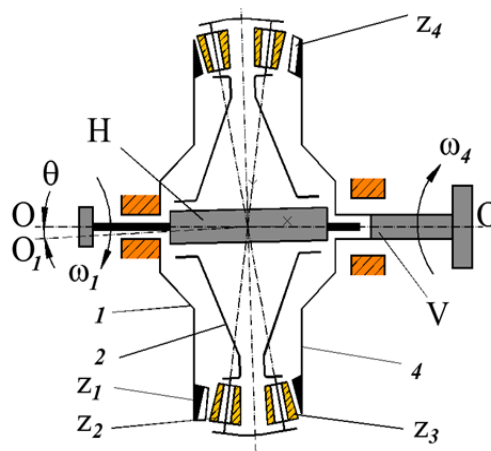

Figure 4. 2K-H kinematic scheme

The bearing rings of the "satellite-crankshaft" block will rotate with the frequencies:
- the inner ring, rigidly installed on the crankshaft, will rotate at the crankshaft angular speed ω_1;
- the outer ring, rigidly installed in the satellite block, will rotate at the angular speed ω_2^*.

Case 1: If the crank shaft and the satellite block rotate in the same direction $z_1 = z_2 + 1$ and $\omega_2^* = \omega_1 - \omega_2$.

Case 2: If the crank shaft and the satellite block rotate in different directions $z_1 = z_2 - 1$ and $\omega_2^* = \omega_1 + \omega_2$.

In this case z_1 is the teeth number of fixed central wheel 1, z_2 - the number of crowns 2 of the satellite block, which contact the fixed wheel 1.

The driven toothed wheel 4 will rotate at the angular speed:

$$\omega_4 = \frac{\omega_1}{i_2} = \omega_1 \frac{z_2 z_4 - z_1 z_3}{z_2 z_4},$$ (3)

where z_3 is the number of rolls of the second crown of the satellite block 2, which engages the driven toothed wheel 4, z_4 – the teeth number of the central gear wheel 4.

3.2. Planetary precessional transmission type K-H-V

From structural point of view, the K-H-V precessional transmissions (figure 5) include an additional vibration generator node, namely the connecting mechanism V, which can be made in the form of synchronous or fingers couplings [4].
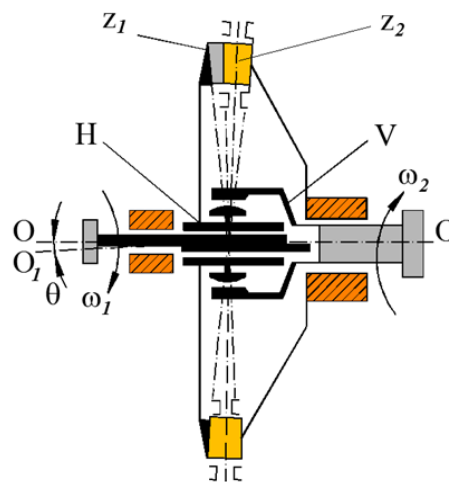
Figure 5. K-H-V kinematic scheme

The crankshaft rotation frequency is analogous to the 2K-H scheme, with only one precession gear (one toothed wheel and one satellite roller). The rotation frequency of the driven shaft is determined with the relation:

$$\omega_2 = \frac{\omega_1}{i} = \pm \omega_1 \frac{z_2}{z_1 - z_2},$$ (4)

where z_1 is the teeth number of central wheel; z_2 - the rollers number of satellites crowns.

Case 1: $z_1 < z_2$, $\omega_2 = -\omega_1 \dfrac{z_2}{z_1 - z_2}$.

Case 1: $z_1 > z_2$, $\omega_2 = \omega_1 \dfrac{z_2}{z_1 - z_2}$.

In vibro-acoustic diagnostic practice there are various methods to minimize the vibration and noise level for technical and energetic machines. The mechanical transmissions used in various machines, installations are sources of vibration and high frequency noise. The most efficient but also the most expensive method for obtaining silent transmissions is the method of processing the parts with very high precision or the static and dynamic balancing of the mobile elements. For precessional planetary transmissions we developed and research several methods for decrease vibration and sound level [1, 2, 4].

a) Power K-H-V precessional transmission

b) Kinematical 2K-H precessional transmission

c) Power 2K-H precessional transmission
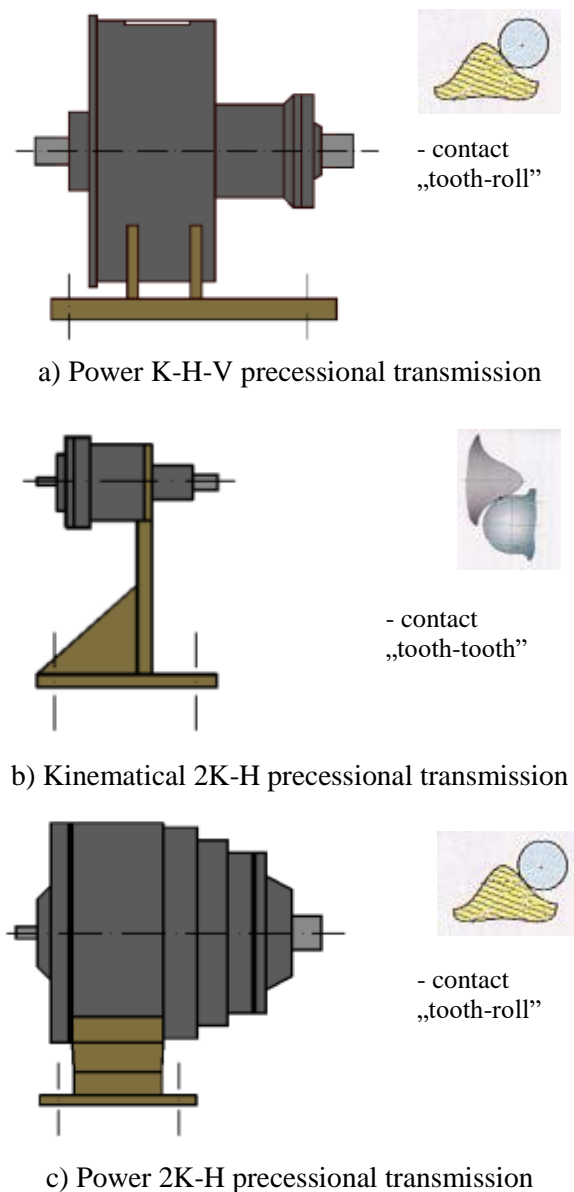
Figure 6. Researched precessional gearbox

4. Performed and future research

For the experimental part of the research, three types of precessional planetary reducers were selected, one of which was a 2K-H kinematic reducer in two satellite constructions made of metal powders and plastic mass [2,3] and two 2K-H and KHV power reducers [1]. The noise level was measured with the Brüel & Kjær analyzer Type 2250 and the severity of the vibrations was measured using the PT500 and PT500.04 software of the German company GUNT [5].

precessional reducers have demonstrated a good and silent operation at different loading regimes and speed [2,3], in accordance with the standards [6], with specification that precesional reducer with satellite wheel from plastic material is more silent.

Further vibro-acoustical research will be carried out on power 2K-H precessional transmission (figure 6c) where the transmission ratio *i=-10,5*, nominal torque *T=20Nm* and operation speed $n=800min^{-1}$.

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