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Constructive Design and Dynamic Testing of the Double Harmonic Gear Transmission

The paper presents the design construction and functioning of a new type of harmonic gear transmission named double harmonic gear transmission (D.H.G.T.). In the second part of this paper is presented the dynamic testing of the double harmonic gear transmission, which is based on the results of the experimental researches on the D.H.G.T. The authors present the results of experimental research, enabling a scientific interpretation of the dynamic behaviour of the D.H.G.T., on the mechanical efficiency and the stress state of flexible wheel.

Keywords: *double harmonic gear transmission, flexible toothed wheel, constructive design, dynamic testing, stress, efficiency.*

1. Introduction

In the last decade researchers have turned their attention towards the improvement of existing gear transmissions, which lead to the emergence of new more efficient systems based on gear transmissions.

In this category of new transmissions, we may include double harmonic gear transmission, whose operation is essentially different from that of traditional gears, as it permits the transmission and transformation of the kinematic and dynamic parameters of the rotary motion by elastic deformations (waves), that propagates on the periphery of a flexible element, with a certain frequency, by a harmonic law (where the name comes).

This transmission presents a series of advantages such as: very high kinematic accuracy, great transmission ratio (i = 50…150), reduced dead course, extremely small loose motions, small weight and dimensions, small inertia moments, coaxial and modular construction, offering them a larger and larger range of applicability in: the actuator mechanism of industrial robots, machine tools, servo-mechanisms, spacecraft, airplanes and helicopters construction, radar antennas and nuclear reactors, etc [1], [2], [3], [5], [8].

The functional performance and durability of the double harmonic gear transmission are greatly influenced by the durability of the flexible toothed wheel, as it is the most powerfully requested element of the transmission.

The flexible toothed wheel of the D.H.G.T. is in a state of complex stress, which depends on many factors such as: the type of wave generator, the geometric shape of the flexible toothed wheel, the torque transmitted and the coupling of the flexible toothed wheel with the output shaft.

This has imposed the need for studying the state of tension of the flexible toothed wheel in assessing the dynamic behavior of D.H.G.T.

The paper presents the constructive design, functioning and some dynamic aspects of the D.H.G.T.

2. Constructive design and functioning of a D.H.G.T.

A double harmonic gear transmission (Figure 1) is made of: a waves generator with cam (1) - as input link, a flexible toothed wheel (2) - with the form of a short circular tube with thin wall, open at both ends and having at each end a crown gear (z_2 external and z^1_2 internal), the rigid fixed wheel (3) and the rigid mobile wheel (4) - as output link [5].

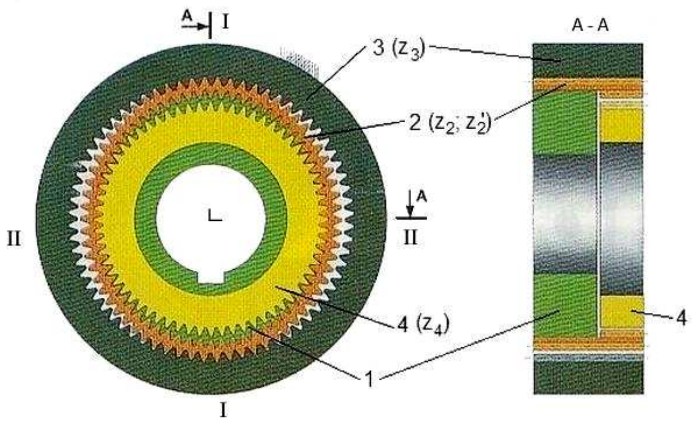


Figure 1. Scheme of a double harmonic gear transmission

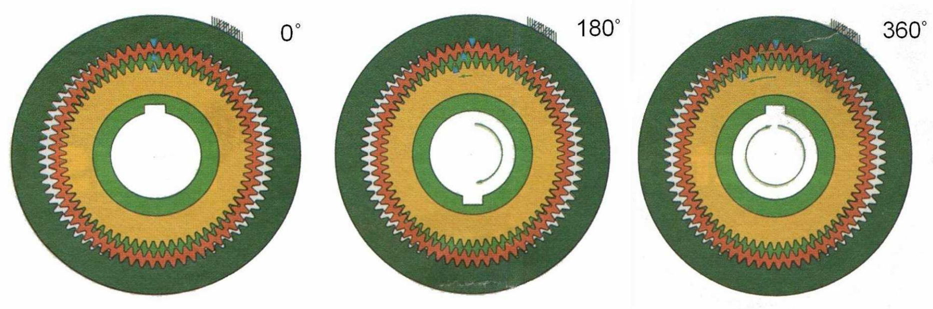
It is based on the same working principle as the simple harmonic transmission, but one must correlate the number of teeth of the conjugate wheel, in order to obtain a relative movement between the flexible toothed wheel (2) and rigid fixed wheel (3) and between the flexible toothed wheel (2) and rigid mobile wheel (4).

$$z_3 = z_2 + 2\,;\ \ z_2^{/} = z_4 + 2 \tag{1}$$

The waves generator is forcefully mounted inside the flexible toothed wheel which deforms elliptically. The flexible toothed wheel will have four equidistant gearing areas: two with the rigid fixed wheel having internal teeth (first step, I-I: z_3 / z_2) and two with the rigid mobile wheel having external teeth (second step, II-II: z'_2 / z_4). Between the two pairs of opposing driving zones (I-I and II-II) there is a 90° angle.

The deformation of the flexible toothed wheel provides the input external teeth of the flexible wheel in the gaps between the teeth rigid fixed wheel (in opposite areas located near the major axis of the ellipse) and the inner teeth of the flexible wheel in the gaps between the teeth rigid mobile wheel (in areas near the small axis of the ellipse).

At baseline of operation (φ_1 = 0° - Figure 2. a) one finds in contact on the opposed vertical positions (I-I) the external teeth (z_2) of the flexible wheel with the internal teeth (z_3) of the rigid fixed wheel, and in the position disposed at a 90° angle to the vertical axis in the other front plan of the flexible wheel (II-II) one can find in contact the internal teeth (z'_2) of the flexible wheel with the external teeth (z_4) of the output rigid mobile wheel.



a. **b.** **c.**

Figure 2. Double harmonic gear transmission kinematics

The clockwise rotation of the waves generator, forces the successive gearing of the teeth flexible wheel disposed toward the direction of rotation of the waves generator, which will come out of gearing of the teeth flexible wheel which are engaging in the opposite direction of rotation of the waves generator.

By rotating waves generator where the φ_1 = 180° (Figure 2. b), the gearing areas will retain their places, but will be rotated relative both the flexible wheel and the rigid mobile wheel in the opposite direction rotating from waves generator.

Flexible wheel will rotate with a tooth to the rigid fixed wheel and rigid mobile wheel will rotate all of a tooth to flexible wheel i.e., two teeth to the rigid fixed wheel.

Thus, at a full rotation of the waves generator (φ_1 = 360° - Figure 2. c), the rigid mobile wheel will rotate in reverse four teeth to the rigid fixed wheel, and the flexible wheel will rotate with two teeth in the sense of rotation of the rigid mobile wheel.

3. Dynamic testing of the double harmonic gear transmission

Dynamic testing of D.H.G.T. sought to determine experimentally the state of deformations and stresses of the flexible wheel body and to study the factors that influence the mechanical efficiency of D.H.G.T., [4].

To determine the state of deformation and tensions in the wall of the flexible wheel, from the two zones of „harmonic" driving of the D.H.G.T., the resistive electric tensometry method is used, [2].

The experimental researches have been done on the stand shown in Figure 3, and D.H.G.T. tested is presented in Figure 4, which is characterized by the following parameters: the transmission ratio, i = 48,2; the wave generator with eccentric disk, the maximum radial deformation, w_0 = 0,3 mm; the teeth modulus, m = 0,3 mm; the number of teeth, z_3 = 202; z_4 =188; z_2 = 200; z^{1}_{2} = 188; the length of the flexible wheel, l = 30 mm; the wall thickness of flexible wheel, s = 0.6 mm.

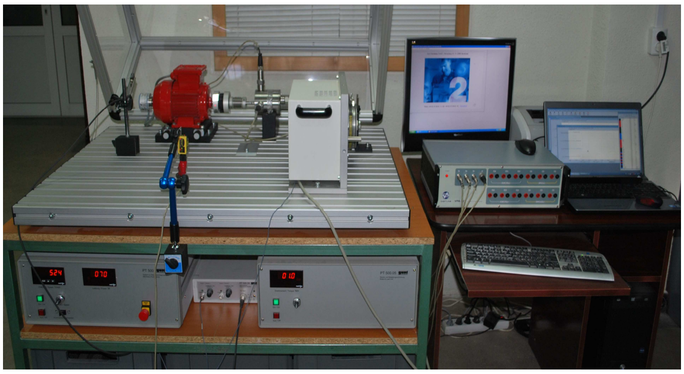


Figure 3. Stand for dynamic testing

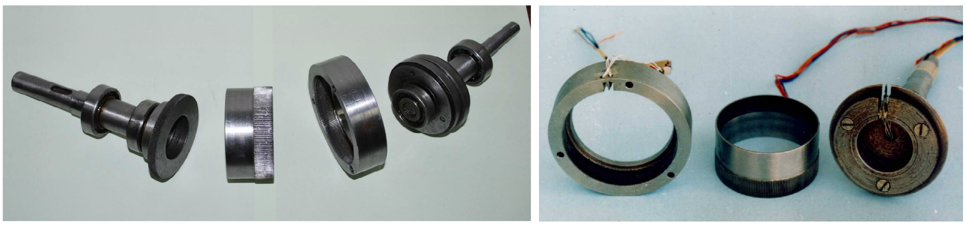


Figure 4. D.H.G.T. elements

To study the factors that influence the mechanical efficiency of D.H.G.T., the influence of torque variation of the output shaft (M_{t4}) vas monitored at a constant input speed (n_1), and the influence variation in input speed (n_1) at a constant torque output (M_{t4}).

4. Experimental results

In Figure 5 are shown diagrams of variation of strains, respectively the principal tensions in the two sections considered of the flexible toothed wheel (curve 1 - for section I-I; curve 2 - for section II-II) for the case: M_t = 50 N·m and n = 500 rpm.

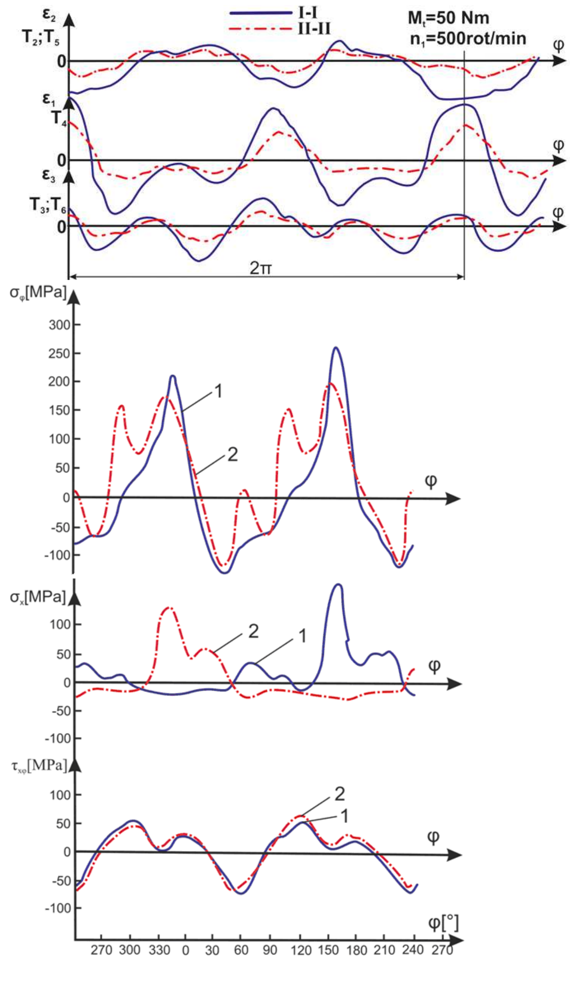


Figure 5. Diagrams of strains and stress

It is observed that the diagram of the variation of the peripheral tension $\sigma_{\varphi I} = \sigma_{\varphi I}(\varphi)$ - curve (1) in the section I-I of flexible wheel (area of the waves generator) is similar to that seen in long flexible wheel from simple harmonic gear transmissions, [6, 7].

In section II-II of flexible wheel, the peripheral tension curve $\sigma_{\varphi II} = \sigma_{\varphi II}(\varphi)$ – curve (2) contains several peaks, but its maximum tension ($\sigma_{\varphi IImax}$= 198 MPa) does not exceed the maximum peripheral tension of section I-I ($\sigma_{\varphi Imax}$= 267 MPa), for the dynamic regime considered.

The presence of additional peaks is due to the change of the form of the deformation of the flexible wheel following the change of character of distribution of tasks on teeth are in gearing in this section.

The analysis of axial tension diagrams finds that these have different character variation in the two sections. In section I-I, the maximum axial tension (σ_{xImax}= 167 MPa) appears in the major axis of the waves generator and section II-II maximum (σ_{xIImax}= 43 MPa) appears in the minor axis of waves generator.

Unlike in the case of axial tensions, in the tangential tensions the two curves are almost identical in variation, while preserving the character of variation and their sizes ($\tau_{x\varphi Imax} \approx \tau_{x\varphi IImax} \approx 64$ MPa).

In Figure 6 there were diagrams of variation of mechanical efficiency of the H.D.G.T. depending on input speed (n_1), by keeping constant the torque at the output (M_{t4}).

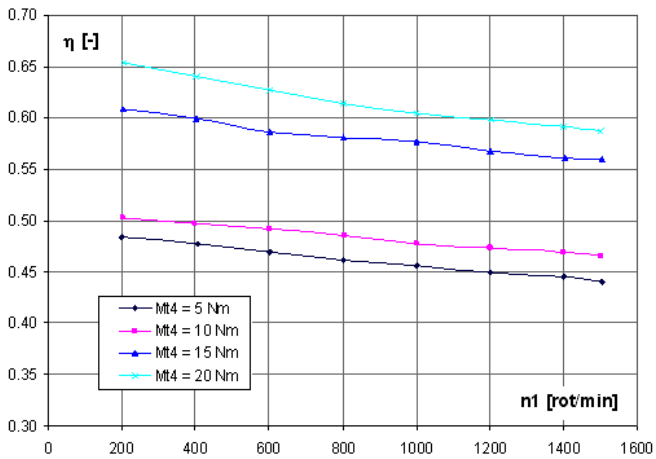


Figure 6. Diagram mechanical efficiency of the H.D.G.T., $\eta = \eta(n_1)$

Studying diagrams mechanical efficiency, $\eta = \eta(n_1)$, it is observed that the increase in input shaft speed H.D.G.T. leads to a lowering of the mechanical efficiency of the transmission. Regressive character is preserved for all four stages of the output torque.

Also, it is found that for a given input speed specification n_1, the mechanical efficiency of the D.H.G.T. increases with increasing load. Thus, at a speed n_1 = 500 rpm, transmission efficiency will increase from 47.3% (for loading M_{t4} = 5 N·m) to 63,4% (for loading M_{t4} = 20 N·m).

In Figure 7 there were diagrams of variation of mechanical efficiency of the H.D.G.T depending on output torque (M_{t4}), for different stages of input speed (n_1).

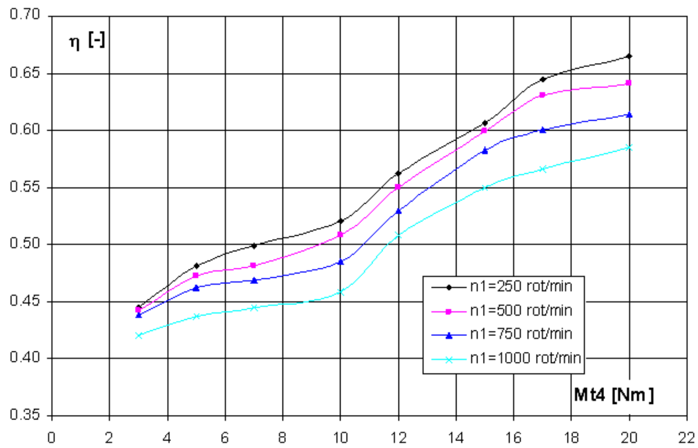


Figure 7. Diagram mechanical efficiency of the H.D.G.T., η = η (M_{t4})

The analysis of these diagrams shows that with increasing speed to input shaft of the D.H.G.T. lower values for efficiency are obtained, while to increase of the output torque will increase the mechanical efficiency of the transmission.

The experimental results can by verified, because for the same engine speed, n_1 = 500 rpm, we find about the same range of variation of the mechanical efficiency of the double harmonic gear transmission (47,3% ÷ 64,1%), when the load varies in the range (5 ÷ 20) N·m.

4. Conclusions

The paper presents the construction, functioning and dynymic testing of the double harmonic gear transmission with the waves generator with eccentric disk.

The analysis of experimental research finds that peripheral tension (σ_φ) has a decisive influence on the sustainability of the flexible wheel because it has the highest value.

It was also found that section I-I of flexible wheel of the D.H.G.T. requested is stronger gearing than the section II-II, because in this section of the flexible wheel is close (in the major axis) between the rigid fixed wheel and the waves generator.

The analysis of mechanical efficiency of the D.H.G.T. can make a statement that it increases with the increase in the output torque, that decreases with increasing input speed.

The theoretical results confirm the experimentally obtained results [2, 5], the deviations being in the tolerable limits.

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