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CONTRIBUTIONS TO THE GENERATION OF PRECESSIONAL GEAR TEETH BY PLASTIC DEFORMATION

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Abstract: Gears are machine parts that occupy an essential place in the operation of various mechanical constructions. The execution of the gears at a high quality level and at a low cost, puts in front of the specialist multiple problems. From the design stage onwards, the specialist must take certain measures to achieve a high reliability of the designed gears. Starting from the correct choice of materials, the designer must adopt constructive solutions of the gears that can be made in the most economical conditions. The cost of manufacturing gears decreases significantly when combining the operations of generating teeth and changing the properties of the surface layer of the teeth. One of these new methods of manufacturing the gears of the precessional gear, in which the formation of the profile of the teeth is not done by cutting, is to obtain the teeth by plastic deformation by rolling.
Keywords: plastic deformation, precessional gear, precessional multiplier, satellite

INTRODUCTION

The gear material is chosen according to the conditions in which the gear works. From this point of view, it is necessary to know well the forces that require the gear, the peripheral speed, the character of the demands (constant or with shocks), the operating conditions without noise, the environmental conditions, etc.

For the processing of gears, very different materials are used, such as: steels, cast iron, bronzes, brass, plastics, etc.

Depending on requests, it is recommended:

≡ for gears with low loads and peripheral speeds, between 0.5 and 2 m / s, it is recommended to use non-ferrous alloys based on zinc or copper, thermoplastic materials and ferrous alloys: steel and cast iron;

≡ for gears with loads and average peripheral speeds, between 2 and 8 m / s, non-alloy or low-alloy semi-hard steels are recommended, as well as cast iron. These wheels are found in large gearboxes, in some lifting and transporting machines, in agricultural machines, mining combines, etc.

≡ for heavy-duty gears with high peripheral speeds, between 8 and 16 m/s, with high loads on the tooth, with shocks in operation, alloy and non-alloy steels are recommended. In cases of particularly high stresses, it is recommended to use high alloy steels, cementation, Cr-Ni, Cr-Ni-Mn, Cr-Ni-W, etc.

These wheels are found in gearboxes from machine tools, ships, vehicles, airplanes, turbines, etc.

Table 1 shows the recommended steel brands for gears. In order to replace the expensive and deficient materials, metallic powders of cheaper materials are used, the semi-finished products being obtained through powder metallurgy. Powders of pure metals, chemical compounds of metals or metal mixtures are used, which by pressing and sintering ensure the formation of workpiece products for gears (Bostan I, 2019; Bostan I, 2018; Bostan I, 2016; Grămescu T., et all, 2000; Grămescu T., et all, 1993).

Table (1). Materials used in the construction of gears and their mechanical properties

The kind of material	Mark	Recommendations for use
General purpose steels for construction	OL 50 OL 60 OL 70	Very low load gears at low peripheral speeds. Very low load gears at moderate peripheral speeds
Alloy steels for heat treatment, intended for machine building	20 MoNi 35 21 TiMnCr 12 TiMnCr 12	Heavy-duty gears at high peripheral speeds and shock loads
Cast steel in pieces	OT 40-3 OT 50-3 OT 60-3	Large gears, very little required

ARGUMENTATION FOR CHOOSING THE DIMENSIONS OF THE WORKPIECE PRODUCT

The profile of the teeth of the central wheels of the precessional gear is variable depending on the values of the angle of the bevel axoid δ, the taper angle of the rollers β, the nutation angle θ, the number of teeth of the gears Z_1 and the number of rollers of the satellite Z_2, and the correlation between them. Depending on these parameters, the height of the teeth will be different, which will influence the height of the workpiece product that will be subjected to plastic deformation (Irani M., et al, 2014; Malcoci I. et al, 2019; Trifan N., 2014; Trifan N., 2016).

The correct choice of the dimensions of the semi-finished product is one of the main problems, the solution of which depends on the reduction of material and energy consumption, as well as the quality of the wheels obtained by plastic deformation. Referring to the manufacture of conical wheels with convex-concave profile of the teeth by knurling, the height of the tooth of the workpiece product is determined by the condition of equal volume of metal moving from the gaps between the teeth to their tip during plastic deformation by rolling (figure 1).

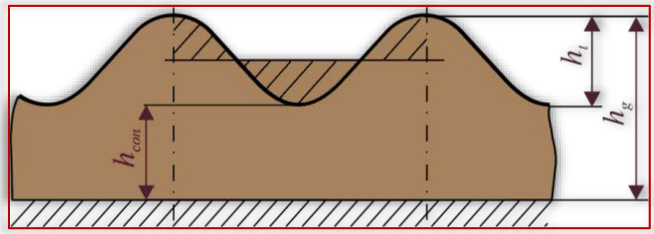


Figure 1. Schematic for determining the height of the gear

$$h_g = h_t + h_{con} \quad (1)$$

were h_{con} is the constructive height of the gear, [mm];h_t - tooth height in normal section, [mm]; h_g - gear height, [mm].

If the workpiece product has been previously deformed and has no porosity, being further subjected to cold deformation, by the accumulation of dislocations, vacancies and microcracks, its volume increases. If the semi-finished product of the gear has been previously deformed hot and another hot deformation follows, the variation of its volume is negligible. In this case it can be said that the volume of the workpiece product in the deformation process remains constant.

In this case it can be said that the volume of the workpiece product in the deformation process remains constant.

Thus, based on the constancy of the volume, for the annular cylindrical workpiece product the volume of material can be written:

$$V_{wp} = \pi h_{hw} \cdot [(R_{out})^2 - (R_{ins})^2] \quad (2)$$

were: h_{hw} is height of the workpiece product subjected to plastic deformation, [mm];

R_{out} - the outer conical radius of the workpiece, [mm];

R_{ins} - inside conical radius of the workpiece, [mm].

It is necessary to specify that according to relation (3) the volume of material necessary to form Z teeth of the gear from the precession gear is determined (figure 2 (a)).

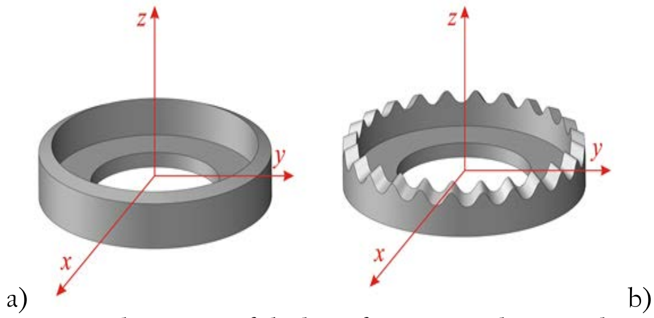


Figure 2. Observance of the law of constant volume at plastic deformation of the teeth in the precessional gear by plastic deformation: a) initial workpiece product; b) plastic deformed gear

The volume of material for forming a gear tooth from the precessional gear (figure 2 (b)) will be determined from the relation:

$$V_t = \frac{1}{3} \cdot b_w \cdot (S_{ins} + \sqrt{S_{out} \cdot S_{ins}} + S_{ins}) \quad (3)$$

where: b_w is the width of the tooth, [mm];

S_{out} – the area of the outer surface of the tooth, [mm^2];

S_{ins} – the area of the inside surface of the tooth, [mm^2].

Based on the computer model of the plastic deformation device of the gears, the working strokes (vertical feedrate of the table), the speed of the main shaft of the machine tool were analyzed to ensure the deformation speed prescribed by the specialized literature.

The increase of the tooth height depending on the vertical advance of the plastic deformation node, in which the plastic deformation rollers are fixed, was determined according to the relation:

$$\Delta h_{pc} = s_{pc} \cdot \cos(\delta + \beta + \theta) \quad (4)$$

where: s_{pc} is the advance of the plastic deformation node to a precession cycle, mm;

δ - the angle of the conical axoid, [º];

β - angle of conicity of the rollers, [º];

θ - the angle of nutation, [º].

CHOOSING THE OPTIMAL PROFILE OF THE TEETH

Certain kinematic structures of precessional planetary transmissions at the correct choice of the basic geometric parameters of the gear can operate efficiently in multiplication mode- figure 3 (Bostan I., 2019; Bostan I., 2019; Ciobanu R., 2014).

Kinematically, the precessional gear represents a Hooke joint. This imposes some conditions on the generation of non-standard profile teeth of the central wheel. From the kinematic analysis performed, some conditions regarding the optimal design of the precessional multipliers are highlighted.

They can be divided into two distinct groups:

≡ conditions related to the reasoned choice of the tooth profile;
≡ conditions related to the reasoned choice of the structural scheme of the multiplier and of the type of the connection mechanism (coupling).

The choice of the optimal profile of the teeth in the precessional gear of the multiplier must comply with the following conditions:

≡ maximum resistance condition - the ability to transmit maximum loads in small dimensions;
≡ the condition of avoiding self-locking;
≡ the efficiency condition in terms of minimizing energy losses in the gear;
≡ the condition of observing the uniformity of the rotational movement of the driven element $\omega_{const.}$ by compensating for the schema error.

The maximum resistance condition is based on self-excluding aspects. The tooth must be as massive as possible to be able to transmit large loads, but at the same time, to have a minimum height and extended profile to ensure a maximum engagement angle, which creates optimal conditions for the transformation of the rotational movement of the the precession moving element of the satellite in multiplication mode. In this case, the designer must ensure the optimization of the choice of the profile of the teeth in order to consider to the maximum the two conditions that self-exclude.

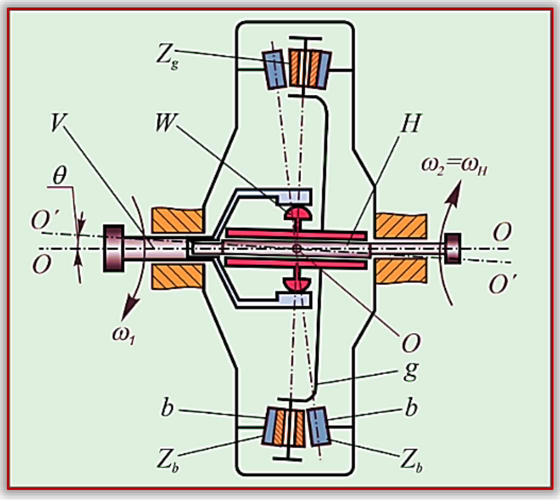


Figure 3. K-H-V type kinematic structure with two central wheels

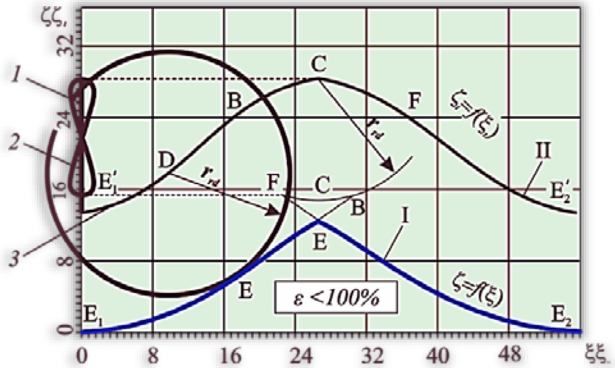


Figure 4. Curves of the trajectory of the movement of the "tooth-roller" contact point

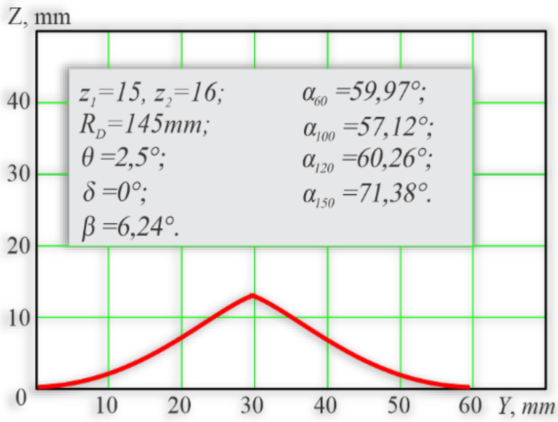
The angle of nutation angle θ (inclination of the crank H) must be as large as possible to ensure a higher load-bearing capacity (a torque applied to the larger conductive element T_{hi}) and, at the same time, as small as possible to ensure minimum height and extended profile of the teeth.

Since the precessional gear is a Hooke joint, which generates the so-called schematic error, this error must be compensated. In the case of the articulated connection of the satellite with the housing, the trajectory of the movement of the contact point E "tooth-roller" should represent a straight line (line I, figure 4). In reality, due to the existence of the scheme error, the trajectory of the movement of the contact point represents a closed octoidal curve (curve 2, figure 4).

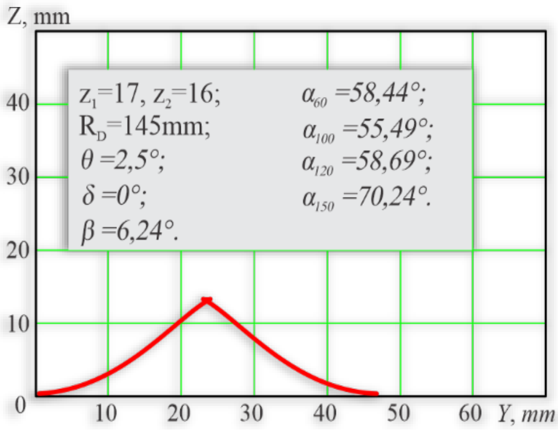
In the case of the articulated connection of the satellite with the driving element, the trajectory of the contact point represents a curve, on which the octoidal curve of the scheme error is superimposed (curve 3, figure 4). This speaks to the need to change the profile of the center wheel teeth at the stage of their generation to compensate for this schematic error.

Obtaining the optimal operating parameters of the precessional multipliers in addition to the basic parameters of the precessional gear is also influenced by another group of conditions related to the reasoned choice of the structural scheme of the multiplier and the type of connection mechanism (coupling)

Based on the resistance calculations performed of the chosen tooth profile, using the computerized principle of creating the toothed solid based on the parametric equations describing the tooth profile [6, 7, 8], the profiles of the teeth of the central wheels were obtained (figure 5 (a, b)) and generated the toothed crown of the wheel and subsequently created the 3D model of the gear (figure 6).



(a)



(b)

Figure 5. Profiles of central wheel teeth with number of teeth: a) z_1=15, z_2=16; b) z_1=17, z_2=16

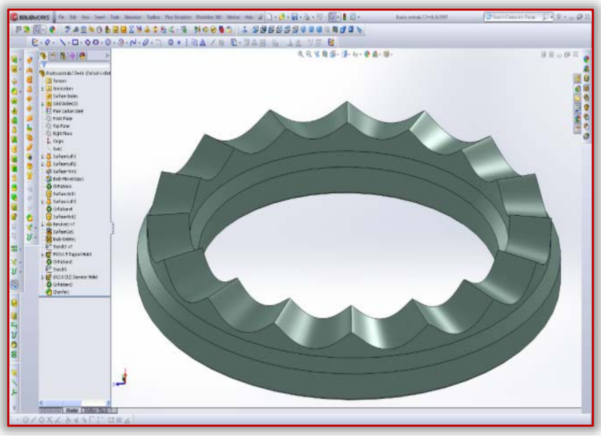


Figure 6. Computerised model of the sun gear

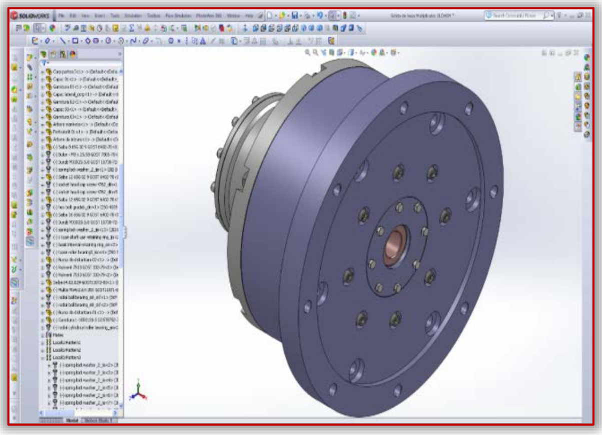


Figure 7. 3D model of the multiplier

DESIGN OF THE EXPERIMENTAL PROTOTYPE OF THE PRECESSIONAL PLANETARY MULTIPLIER K – H – V WITH THE TRANSMISSION RATIO I =-16

When designing any transmission, the designer must ensure to the maximum the satisfaction of the ever-increasing requirements regarding the bearing capacity, compactness, mass and dimensions, low production cost, etc. and, in particular, with respect to the kinematic characteristics, structural compatibility with other aggregates of the machine, etc. Precessional planetary transmissions correspond to these ever-increasing requirements of manufacturers and consumers of reducers and multipliers due to the constructive-kinematic peculiarities presented in the previous chapters.

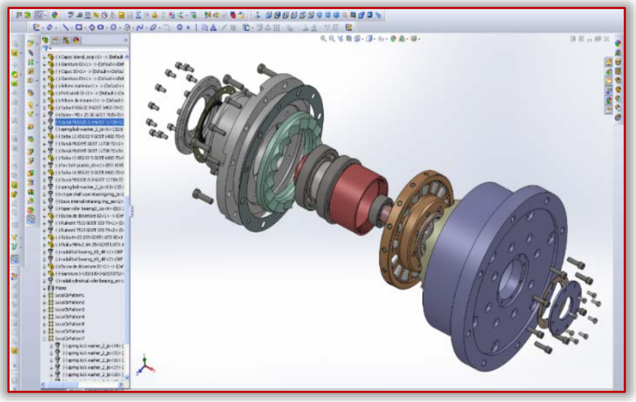


Figure 8. 3D model of the multiplier in assembly perspective

Based on the kinematic structure (figure 3), the 3D model of the processional planetary multiplier presented in general view (figure 7), and in assembly perspective (figure 8) was elaborated. The 3D model of the processional multiplier in the developed state is informative both in terms of structure and assembly process (succession of assembly phases).

RESEARCH OF THE MECHANICAL EFFICIENCY OF THE PRECESSIONAL MULTIPLIER

Experimental research has the primary role of validating theoretical results. The basic energy parameters of a multiplier are the mechanical efficiency, which determines the power losses in the kinematic torques of the multiplier and the starting moment, which in some cases establishes the functionality of the working machine. For example, in the case of the wind turbine, the starting moment of the multiplier determines the operation of the wind turbine at low wind speeds (Bostan I., 2019; Bostan I., 2019; Bostan I., et al, 2019; Ciobanu R., 2014).

The test method comprises the whole complex of operations on the multiplier, in order to assess its efficiency in operation. The tests are performed under normal environmental conditions. For the test of the precessional transmission with reducer and multiplier operation, the open flow test stand of the power flow from the Mechanical Transmission Testing Laboratory of the "Basics of Machine Design" department was used (figure 9, 10).

The mechanical efficiency of the multiplier is determined according to the formula:

$$\eta = \frac{T_2}{T_1 \cdot i}, \tag{5}$$

where T_1-the torque on the output shaft of the multiplier, [Nm]; T_2 - torque on the multiplier input shaft, [Nm]; i - the transmission ratio of the multiplier.

The values of the torques on the input and output shafts of the multiplier T_1 and T_2 are determined according to the indications of the indicators of the dynamometers 5, 8 using the pricing graphs). The measuring devices of the stand ensure the measurement of the torques T_1 and T_2 with the corresponding accuracy 0,5 ÷ 1%. The stability of the T_1 and T_2 torques at each of their values was investigated within 1-2 hours of uninterrupted tests.

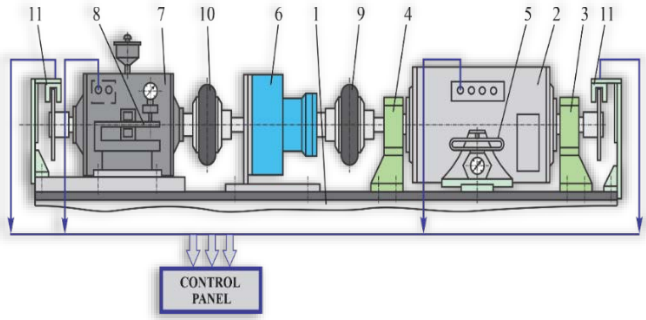


Figure 9. Scheme of the experimental stand for the test of the precessional multiplier:
1 - Rigid stand; 2 - DC electric motor with a power of 8.0 kW; 3, 4 - support; 5 - force dynamometer with indicator; 6 - the precessional reducer; 7 - electromagnetic brake with metal powders; 8 - force dynamometer with indicator; 9,10 - compensating couplings with elastic elements; 11 - speed measuring transducer

Figure 10. Experimental stand for testing the precessional multiplier

The research of the mechanical efficiency depending on the load and the number of revolutions took place from a methodical point of view similar to the case of operation of the transmission in reducer mode, with the modification of the load regime imposed by the stand possibilities: 2 Tn; 0.4 Tn; 0.6 Tn) and 3 speed regimes which represent the speed accepted at the reducer load divided by the transmission ratio.

To ensure the same kinematic regimes, the multiplier was tested at 40min-1 speeds; 50min^{-1}; 60min^{-1}. The gearbox was charged gradually, the load being increased from 0.2 of the nominal torque value to 0.6 Tn, as allowed by the electric motor. Based on the obtained results, the graphs of the mechanical efficiency were constructed depending on the torque for the speeds n = 40min^{-1}; n = 50min^{-1}; n = 60min^{-1} (Figure 11).

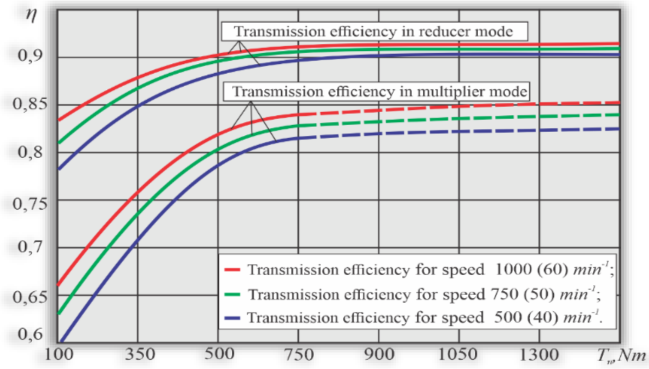


Figure 11. Mechanical efficiency depending on the torque in reducer and multiplier mode

Because the graphs for reducer operation show that from the load of 0.6 Tn the efficiency of the reducer stabilizes, the graphs of the efficiency of the multiplier were continued until the moment of loading 1.0 Tn by similarity. The comparative analysis of the yield graphs in reducer and multiplier mode shows that at low values of the charging moment the multiplier efficiency is much lower than the reducer efficiency. This is explained by the fact that in multiplier mode the starting moment is higher than in reducer mode, measurable with the charging moment at low loads. Then the initial total charging moment is equal to the sum of the charging moment and the starting moment (Bostan I., et al, 2019; Ciobanu R., 2014).

CONCLUSIONS

Based on the optimized conceptual schemes of the precessional gear and the connection mechanisms, the structure of the precessional multiplier K – H – V with two central wheels was developed with a connection mechanism in the form of coupling with teeth or tapered rollers, which ensures high load capacity and total compensation. of the axial force generated in the precessional gear.

It has been established that the mechanical efficiency of the multiplier is determined by the power losses in the tooth-roller gear, in the coupling and the bearings. To reduce the power loss in the coupling, the coupling with tapered rollers has been proposed, in which the sliding friction is replaced by rolling friction.

Experimental research of the precessional multiplier has shown that the mechanical efficiency of the multiplier is relatively high about 85%, but at low loads the yield is low, explained by the fact that at low values the load is commensurable with the starting moment, which in multipliers is higher.

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