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ASPECTS REGARDING OPTIMIZATION OF MECHANICAL CONTACT BETWEEN TEETH WITH NON-STANDARD PROFILE IN PRECESSIONAL GEAR

Valeriu DULGHERU, Stanislav SLOBODEANIUC

Abstract: *A basic problem in the operation of gears in conditions of insufficient lubrication is the intensification of the wear process of the teeth and the increase in the level of noise and vibrations, generated by the variability of the frictional forces. In precessional power transmissions, the teeth of the toothed crowns of the satellite block are executed in the form of conical rollers, ensuring the "tooth-conical roller" engagement, and in kinematic transmissions they are described by arc-shaped curves. The paper presents the real and compressed dynamic model of the precessional gear under the aspect of evaluating the limit frequency of the noise and vibration level. In the "bevel-roller" gear, rolling friction persists and approx. 4% sliding friction, and in the "tooth-tooth" gear, sliding friction persists, which can lead both to excessive wear of the teeth and to an increase in the level of noise and vibration generated by the spasmotic nature of the loads. To optimize the mechanical contact, a patented solution is proposed, which is based on the application of a superficial layer of elastomeric material with anti-friction and shock-absorbing properties on the working surface of the teeth . To ensure a major adhesion of the superficial layer to the coarse base of the teeth, the first layer of additive material is applied immediately after the last layer of the coarse print is applied. In the case of increased requirements regarding the vibro -acoustic behavior on the coarse surface of the teeth made of materials with high mechanical resistance, a layer of polymeric material with self-lubricating and shock-absorbing and adhesion properties is applied on the coarse surface of the tooth.*

Keywords: *precessional transmission , rollers, teeth, mechanical contact, optimization .*

1. INTRODUCTION

Noise pollution is becoming an increasingly pressing problem. In connection with this, the standards regarding the level of noise and vibration of the machines are becoming more and more rigid. In the production areas, the increased level of noise and vibrations can lead to the exacerbation of occupational diseases. But the lifetime of the machinery is also reduced, it can also lead to accidental, sometimes catastrophic, shutdowns. It should be noted that vibration analysis is the basis of predictive maintenance of machines, and a vibrogram actually represents the state of health of the machine, which must be diagnosed. In this context, the demands placed on car designers are becoming increasingly rigid in terms of minimizing the level of vibration and noise. The machines must be as quiet and durable as possible, ensuring minimum energy losses.
A research problem on the vibration and noise level of an installation includes:

- detecting the sources of vibrations and noise, the responsible elements;
- establishing the cause of the defect;
- taking measures (constructive, technological) to eliminate the identified problem.

It is known that about 80% of energy is transmitted through mechanical transmissions (in particular, gears). Therefore, better knowledge of the causes of vibration and noise in gears is the key to reducing their level. Among the sources, basic phenomena, which lead to the appearance of vibrations and noise in gears, are:

- tooth wear;
- overloading the teeth by excessive loading;
- runtime and assembly errors.

One of the solutions to reduce the wear in gears, which operate without lubrication, and the level of noise and vibration, is the transfer of the external sliding friction in the gearing of the teeth into internal friction by applying to the working surfaces of the teeth thin layers of elastomeric material . A solution to eliminate the sliding friction taking into account the inevitable geometrical friction through the layered deformation of the elastomeric material (rubber, plastic) is proposed for the involute gear (fig. 1,a,b) and the Wildhaber-Novikov gear (fig. 1 , c) [1]).

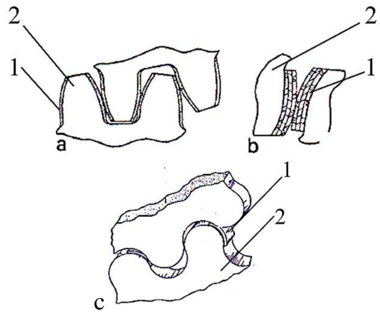


Fig. 1. Gear, in which the external sliding friction is transferred to the geometrical friction in the elastomer layers: 1, the elastomer layer; 2, the tooth.

The advantages of using thin layers of elastomer on the surfaces of conjugate teeth are:

- substantial reduction of dissipated energy, i.e. increasing efficiency (yield);
- removal of lubrication systems and seals;
- reduction of noise and vibration generation;
- elimination of games in kinematic couples.

Widespread use of very thin elastomer layers (0.01-0.5 mm) has been held back by a lack of information and technological difficulties in application. The emergence of additive technologies and new elastomeric materials can solve this problem. The purpose of research in the field of vibrations and noise in gear transmissions is to minimize the effect of vibration sources through constructive-technological measures (of the existing ones), but also through the development of new mechanical transmissions.

Planetary precessional transmissions are a new type of planetary transmissions, invented by Acad. Ion Bostan at the end of the 80s. The first invention patent was obtained in 1981, currently their number is approx. 200 inventions, which include: new gears; new manufacturing technologies; a wide range of kinematic structures and precessional drive mechanisms for various uses, including, in fine mechanics (spacecraft, avionics, robotics, automobiles, etc.) (Fig. 2 [2]).

The construction and operation specifics of the

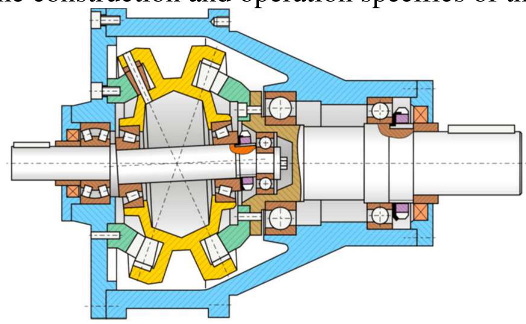


Fig. 2. Planetary precessional transmission type 2K-H.

planetary precessional transmissions allow the minimization of the influence of the sources mentioned above on the level of vibrations and noise:

- Reduction of wear and tear - by using the constructive-technological procedures for contact optimization;
- Reduction of tooth overload - thanks to the specific sphero -spatial movement of the satellite block, the high multiplicity of gearing is ensured (participation in the simultaneous load transmission of approx. half of the teeth of the gear wheel), which ensures the reduction of the load, which falls on one tooth (maximum load does not exceed 20-25% of the total load);
- Reducing the influence of execution and assembly errors - by compensating (at the design stage) and through constructive-technological processes of execution and assembly errors.

Researches of noise and vibration level and mechanical losses in precessional gear have been carried out by researchers [2-5].

2. THE REAL AND "COMPRESSED" DYNAMIC MODEL OF THE PRECESSION GEAR

Planetary precessional transmissions are complex mechanical systems from a structural and kinematic point of view - they possess three degrees of freedom (at least one node - the satellite block, performs sphero -spatial precession movement). As in ordinary transmissions, the vibration and noise spectrum includes frequencies generated by shafts, bearings, couplings, gear elements, casing, etc. As the gearing frequencies are predominant in the frequency spectrum of vibration and noise it is particularly important to analyze the sources, which cause these frequencies in the planetary precessional transmission. The other sources of vibration and noise are classic elements (shafts, bearings, couplings, casing) and are relatively studied. Unlike ordinary gears, in which the load is simultaneously transmitted by a pair of teeth (in the case of changes in the gear it can reach up to two pairs of teeth), the precessional gear is multiple. All teeth Z_r participate simultaneously in the engagement, the load being transmitted by [1]:

$$Z_f = \frac{Z_r - 1}{2}, \qquad (1)$$

where: Z_f is the number of teeth, which transmit the load; Z_r - the total number of teeth of the gear wheel.

It should be noted that the load is transmitted unevenly by the teeth simultaneously in gear. According to [3] the most requested tooth transmits appr. 20-25% of the total load. So, in the multiple precessional gear we face the same moments of inertia of the teeth (bevel rollers) I_k, but with stiffnesses e_k different.

In fig. 3, a half of the unfolded gearing of the gear with the conical rollers of the crown of the satellite block is shown, which simultaneously transmit the load forces F_{n1}, F_{n2}, F_{n3}, F_{nk-1}, F_{nk} are the normal forces, acting between teeth *1, 2, 3, k-1, k* simultaneously conjugate of the precessional gear . Also $F_{n1} > F_{n2} > F_{n3} > F_{nk-1} > F_{nk}$. The multiple precessional gear can be replaced by a dynamic model consisting of a successive series of stiffnesses e_1, e_2, e_3, e_{k-1}, e_k, and moments of inertia I_1, I_2, I_3, I_{k-1}, I_k which

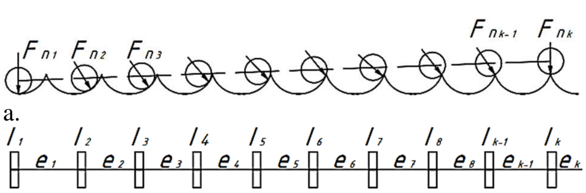


Fig. 3. Sprocket engagement layout with beveled satellite block crown rollers (a) and replacement dynamometer model (b).

represents the parameters of simultaneously engaged teeth. In fig. 3,b shows the replacement dynamic model of the multiple precessional gear with the stiffnesses e_1, e_2, e_3, e_{k-1}, i_k and the moments of inertia I_1, I_2, I_3, I_{k1}, I_k .

Due to the simultaneous participation in gearing of several pairs of teeth (multiple precessional gearing includes $(Z_r - 1)/2$ pairs of teeth, which simultaneously transmit the load) the dynamic model is particularly complex. To simplify the model, the compensation method [6] developed by Professor Rivin E will be used. Thus, according to the method for series dynamic models in the limits of frequencies $0 - f_{lim}$, the complexity of the dynamic system of the multiple precessional gear can be reduced without introducing essential errors in the natural frequency pattern. Thus the dynamic model of the precessional gear will be divided into subsystems (see fig. 4) provided that the natural frequency of the partial subsystem, with which it will be replaced, is higher than the upper limit f_{lim} of the frequency of the frequency range of interest. The natural frequencies of the subsystem shown in fig. 4 are:

$$n_k = \sqrt{\frac{e_{k-1} + e_k}{I_k e_{k-1} e_k}}, \qquad (2)$$

where:

e_{k-1} , e_k are the stiffnesses of elements *K* and *K-1;*

I_k – the moment of inertia of the element *K*.

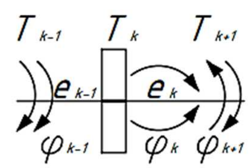


Fig. 4. The single table partial subsystem.

If vibration modes below f_{lim} must be kept between ±2dB in the elastomeruc layer, then the transformed subsystems must have natural frequencies $n_k \geq 3.5–4.0\ \omega_{lim}$, where $\omega_{lim} = 2\pi f_{lim}$. In fig. 5 this compression algorithm is presented. To begin with, the initial dynamic model (fig. 3,b) is divided into partial subsystems (fig. 5,a). Then for each partial subsystem in fig. 5, to calculate the natural frequency:

$$n_k^2 = \frac{1}{I_k^* e_k^*}, \qquad (3)$$

where:

$$I_k^* = I_k, \quad e_k^* = \frac{e_{k-1} e_k}{e_{k-1} + e_k}.$$

Subsequently, the subsystems, which have $n_k^2 \gg \omega_{lim}^2$(ie $I^* e^* \ll \omega_{lim}^2$), are replaced by equivalent subsystems of the opposite type:

$$I_k^{"} = \frac{e_{k-1} e_k}{e_{k-1} + e_k};\ I_{k+1}^{"} = \frac{e_{k-1}}{e_{k-1} + e_k} I_k;\ e_k^{"} = e_{k-1} + e_k. \tag{4}$$

Thus, after a series of possible substitutions, the dynamic system is rearranged. In fig. 5,b is

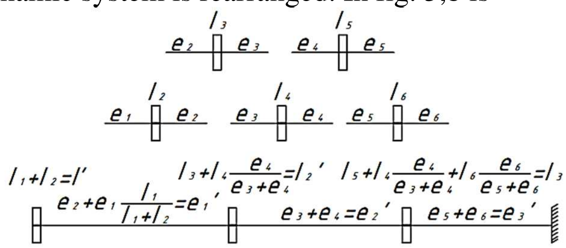


Fig. 5. " *Compression* " algorithm : (a), partial subsystems; (b), final version of the dynamic model *"tablet"*.

presented the final version of the "compressed" dynamic model . The obtained dynamic model can be further simplified, taking into account the specifics of the researched objective (the precessional gear):

- the teeth (conical rollers) are identical, therefore, the moments of inertia are equal to each other $I_1, = I_2, =I_3, =I_{k-1}, = I_k,$;

- between the normal forces, which appear between the conjugate teeth (F_{n1}, F_{n2}, F_{n3}, F_{nk-1}, F_{nk}), there is a connection, which can be established according to the geometric parameters of the precessional transmission.

3. MODELS FOR CONTACT OPTIMIZATION IN THE PRECESSIONAL GEAR

The transformation of motion and transmission of load in gears is a cyclic and pulsating process. In the precessional gear, the specific effect generated by the sphero -spatial (nutation) movement of the satellite block with two toothed crowns is also added . The effects, which occur in the contact of the teeth, depend on the mode of engagement, which can be of two types:

- "*tooth-bevel roller" gearing,* used in planetary precessional power transmissions;
- "*tooth-to-tooth*" gearing, used in kinematic (low-power) planetary precessional transmissions.

For each of the two types of gearing, contact optimization models were developed, based on which the principle of transferring the external sliding friction in the gearing of the teeth into internal friction was taken by applying thin layers of elastomeric material to the working surfaces of the teeth .

3.1 "*Bevel tooth-roller" gear*

The emergence of additive technologies and new elastomeric materials can successfully solve the reduction problem influence sliding forces in the gear as sources of power losses and of vibrations and noise generated by the spasmodic nature of the contact. For the correct choice of elastomeric materials and the required thickness of the layer, it is necessary to know the sliding speed in the gear and the sliding path. To reduce the sliding friction in the gear in planetary precessional power transmissions, the teeth of the toothed crowns of the satellite block are executed in the form of conical rollers installed on the axles with the possibility of rotation around them. It should be noted that during a complete precession cycle of the satellite block, the contact of the teeth will occur predominantly with rolling with the presence of an insignificant variable slip in contact. The value of the relative slip depends on the moment of rolling resistance, the resistance of the roll on the axle and the inertial forces of the roll. In fig. 6 presents itself „*bevel tooth*

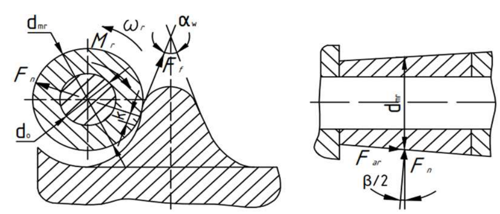


Fig. 6. Elements of the " *tooth - bevel roller"* precessional gear .

-roller" gearing elements with the geometric parameters necessary to assess the relative sliding speed.

According to [2] the sliding speed in the upper kinematic coupling can be determined with the relation:

$$v_a = K_1 \ln \frac{f_{\max}}{f_{\max} + k / r_{rm} - M_r / r_{mr} F_n}, \tag{5}$$

where: K_1 is the coefficient, which characterizes the operating conditions of the upper kinematic couple, the properties physical -mechanical properties of the materials and other parameters, which cannot be included in a separate calculation. In general, this coefficient takes into account the forced sliding and the lubrication conditions. For the operating conditions of precession gears $K_1=2.463$; f_{max} - the maximum coefficient of friction for the coupling materials under the given operating conditions; k - coefficient of rolling friction; M_r - the moment of resistance forces, which takes into account the friction between the conical roller and the axis and on the front surface of the conical roller; d_{mr} - radius of the roller in the middle section; F_n - the normal force in the upper kinematic couple.

The moment of the rolling resistance forces of the conical roller M_r can be expressed by the relation:

$$M_r = M_o + M_f + M_c, \qquad (6)$$

where: M_o is the moment of frictional forces on the roller axes; M_f– moment of friction forces on the front surface of the rollers; M_c – the rolling resistance moment of the conical roller on the flank of the teeth.

The moments of the frictional resistance forces on the axis of the rollers M_a and on the front surface of the rollers M_f is determined with the relations:

$$M_a = \frac{F_n f d_0}{2}, \quad M_f = \frac{F_a f (d_{mr} + d_0)}{4}, \qquad (7)$$

where: F_n is the normal force acting on the conical roller; F_a - axial force, determined from the relationship (see fig. 6): $F_a = F_{ar} \cos \beta/2 = F_n \sin \beta/2 \cos \beta/2 = F_n (\sin \beta)/2$, where β is the conicity angle of the conical rollers; f – coefficient of sliding friction of the friction coupling material; d_{mr} – diameter of the conical roller in the middle section; d_o – diameter of the axis of the conical rollers.

The rolling resistance moment of the roller on the flank of the teeth is determined from the relationship:

$$M_c = F_n k, \qquad (8)$$

where: k is the coefficient of rolling friction.

Taking into account relation (6) equation (5) will take the form:

$$v_a = K_1 \ln \frac{f_{\max}}{f_{\max} - \frac{\left[f d_o + 0{,}5 f \left(d_r + d_o \right) \sin \alpha_w \sin \left(\delta + \beta' \right) \cos \delta \right]}{d_{mr}}}, \qquad (9)$$

where: β' is the angle, which determines the position of the contact line of the "*tooth-bevel roller*" coupling; α_w - the engagement angle; δ – the angle of the conical axoid of the toothed crown of the satellite block.

The calculated sliding speed is the basis for the choice of the elastomeric material and the thickness of the layer for the compensation of the sliding friction in the "*bevel tooth-roller" gear* with the geometrical friction by the layered deformation of the elastomeric material (rubber, plastic).

After establishing the parameters of the elastomeric material layer , the next step is its technological realization on the flanks of the teeth. In the "*bevel-roller"* precessional gear , the application of the material layer elastomer can be made in two ways:

- applying the elastomer layer on the roller surface ;
- application of the elastomer layer on the working surfaces of the tooth.

Functionally there is no difference between the two ways of applying the elastomer layer. In this case, the quality of the layer, technology and application costs matter. In fig. 7 shows the gearing elements in the "*conical tooth-roller"* precessional gearing .

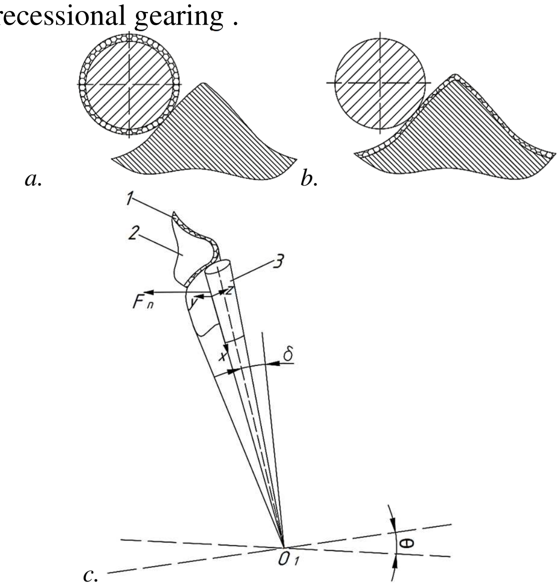


Fig. 7. Elements of the precessional gear "*tooth - conical roller"* (variants) (a,b); gear scheme (c).

In this plan, the authors developed procedures and devices for the technological application of the elastomer layer with programmed anti-friction and shock - absorbing properties on the flanks non- standard teeth through 3D printing [7,8]. The additive manufacturing process of gears made of polymeric material is based on the additive manufacturing device of fig. 8, which includes a housing 1, on which additive heads with nozzles 2 are installed , the forming bed 3, on which the gear wheel 4 is formed, the feeder with powdered polymer material 5. The process is directed by the computerized control module 6, which ensures the coordinated movements of the nozzle 2 and the forming bed 3. On the surface of the teeth 1 of the central gear wheel (fig. 9) a thin layer of plastic material 2 with shock load priming properties is applied through additive technologies, consisting of cellular units 3 (fig. 9,b,c) in structures 4 (fig. 9,b), located with the possibility of accepting some microdisplacements in the three *xyz* directions (fig. 7).

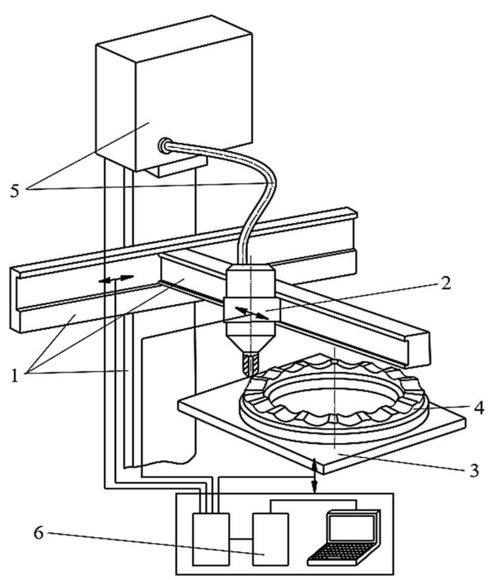
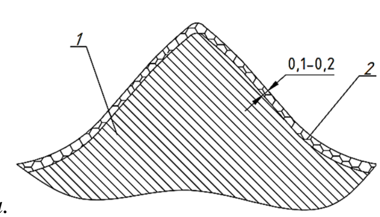
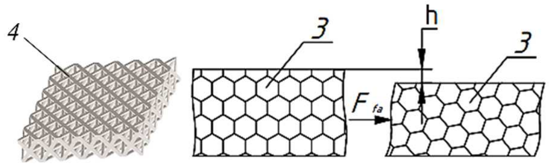


Fig. 8. 3D printing device .



a.



b. *c.* *d.*

Fig. 9. *a,* Elastomer layer applied on the flanks of the tooth; *b* , the structure of the elastomer layer; *c* , initial state of the elastomer layer; *d,* elastomer layer after application of normal and frictional force.

Due to the deformation capacity of the elastic material of layer 2 with the value $\boldsymbol{h}$ under the action of the normal force from the gearing $\boldsymbol{F_n}$ by ensuring the microdisplacements of the cellular units in the direction of the action of the frictional force F_a the relative displacement of the roller 3 with respect to the teeth 1 in the direction *OZ* is excluded , a fact that excludes the sliding friction between the rollers 3 and the teeth 1 and ensures reduction of sliding friction losses.

Due to the provision of microdisplacements of cell units 3 in the direction of action of the normal force $\boldsymbol{F_n}$ in the gear - ***OY axis*** there is the priming of shock loads in the gear, caused by execution and assembly errors and dynamic factors. Thus, the proposed technical solution allows reducing the level of noise and vibrations, reducing power losses due to sliding friction in the gear.

3.2 Gearing "*tooth-to-tooth*"

In the kinematic planetary precessional transmissions (of low power), thanks to dimensions small, the gear "*tooth-roller conical*" it is impossible to use. Therefore, for this field authors [9,10] elaborated „*tooth-to-tooth”* gear in various variants for kinematic transmissions use in various fields of fine mechanics (avionics, robotics, automobiles, medical equipment). Because of inability elimination slipping in the „*tooth -to – tooth”* gear power losses are more big according to various parameters of precessional gearing. And it also leads to growth the noise level and vibration generated by the character spasmotic of the contact.

Additive technologies allow partial reduction of sliding friction losses and noise and vibration levels by:

- application of the superficial layer with anti-friction and shock-absorbing properties on the already formed flanks of the teeth;
- the integral printing of the teeth by coarse printing of the teeth in material with increased mechanical resistance and the application in the last phase of the surface layer of elastomeric materials with anti-friction and shock-absorbing properties.

In the first case the process and device described previously are used (fig.8).

In the event integral printing by coarsely printing teeth from material with increased mechanical strength in the first phase with the application in the final phase of the superficial layer of elastomeric material with anti-friction and shock-absorbing properties, the process and device (fig. 10,[8]) use 2 or several heads of additive with nozzle (2, 2 ', 2'').

The next additive head (2', 2''...) will deposit the next layer of additive material immediately on the previously freshly deposited layer, which ensures better adhesion between layers and high productivity.

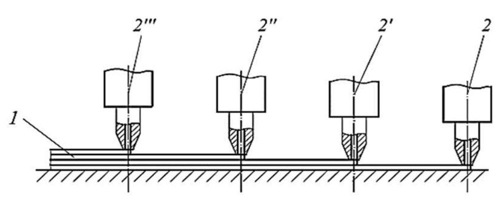


Fig. 10. 3D printing with several additive heads .

The proposed additive manufacturing process ensures the manufacturing of gears with a non-standard convex-concave profile and circular arc of the teeth (fig. 11, 12) with the application in the last phase of the surface layer of elastomeric material with anti-friction and shock-absorbing properties.

This is achieved by transferring the external sliding, which occurs between the conjugate profiles of the teeth, into thegeometric friction inside the elastomeric material through the layered deformation of the material.

Thus, the proposed technical solution allows reducing the level of noise and vibrations, the partial reduction of power losses due to sliding friction in the " *tooth-tooth" gear* .

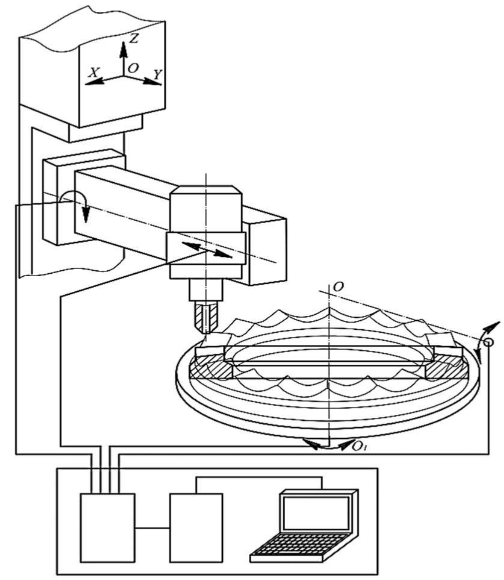


Fig. 11 . 3D printing device with additive head , which performs sphero -spatial (precessional) movement.

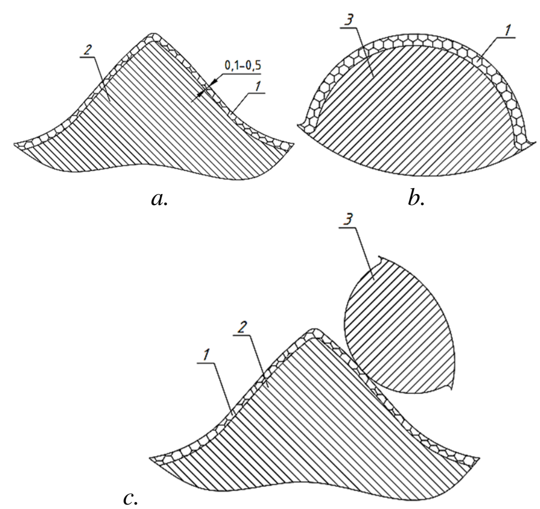


Fig. 12
. a, teeth with convex-concave profile; b, teeth with circular profile; c, the contact of the teeth with the elastomeric intermediate layer .

4.CONCLUSION

The pattern dynamically "*compressed "* obtained of the precessional gear enable evaluation noise level gearing frequency and vibration generous and development minimization a they.

The pattern elaborated gear "*tooth-roller conic"* allows evaluation sliding speed in the gear, that is use for arguments choice parameters elastomer layer with properties antifriction and shock absorption. Processes and printing devices elaborated additive ensures application on the flanks teeth of the surface layer of elastomeric material with anti-friction and shock-absorbing properties.

In the event kinematic precessional transmissions the elaborated additive processes and printing devices ensures print coarseness of the teeth in the material with high resistance mechanics with application in the last phase of the superficial layer of elastomeric material with anti-friction and shock-absorbing properties.

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Aspecte privind optimizarea contactului mecanic între dinții cu profil nestandard în angrenajul precesional

Abstract: *O problemă de bază în funcționarea angrenajelor în condiții de lubrifiere insuficientă este intensificarea procesului de uzură a dinților și majorarea nivelului de zgomot și vibrații, generat de variabilitatea forțelor de frecare. În transmisiile precesionale de putere dinții coroanelor danturate ale blocului satelit sunt executați în formă de role conice, asigurând angrenarea „dinte-rolă conică”, iar în trasmisiile cinematice sunt descriși de curbe în arc de cerc. În lucrare se prezintă modelul dinamic real și compresat al angrenajului precesional sub aspectul evaluării frecvenței limite a nivelului de zgomot și vibrații. În angrenajul „dinte-rolă conică” persistă frecarea de rostogolire și apr. 4% de frecare de alunecare, iar în angrenajul „dinte-dinte” persistă frecarea de alunecare, care poate duce atât la uzarea excesivă a dinților cât și la majorarea nivelului de zgomot și vibrații generat de caracterul spasmotic al sarcinilor. Pentru optimizarea contactului mecanic se propune o soluție brevetată, care se bazează pe aplicarea pe suprafața de lucru a dinților a unui strat superficial de material elastomeric cu proprietăți antifricțiune și de amorsare a șocurilor. Pentru asigurarea unei aderențe majore a stratului superficial cu baza grosieră a dinților primul strat din material aditiv se aplică îndată după aplicarea ultimului strat al printării grosiere. În cazul creșterii cerințelor privind comportamentul vibro-acustic pe suprafața grosieră a dinților executați din materiale cu rezistență mecanică înaltă se aplică un strat din material polimeric cu proprietăți de autolubrifiere și amorsare a șocurilor și de aderență pe suprafața grosieră a dintelui.*

Key words: *transmisie precesională, role, dinți, contact mecanic, optimizare.*

Valeriu DULGHERU, PhD Dr.Sc, Professor, Technical University of Moldova, Fundamentals of Machine Design. valeriu.dulgheru@bpm.utm.md

Stanislav SLOBODEANIUC, PhD student, Technical University of Moldova. stanislav.slobodeaniuc@tran.utm.md