Journal of Engineering Science *Vol. XXIX, no. 2 (2022), pp. 8 - 17*
Fascicle *Industrial Engineering* *ISSN 2587-3474*
Topic *Mechanical Engineering and Technologies* *eISSN 2587-3482*

https://doi.org/10.52326/jes.utm.2022.29(2).01
UDC 621.833:621.89

THEORETICAL AND EXPERIMENTAL MODELLING OF THE TRIBOLOGICAL BEHAVIOUR ASPECTS OF CONTACT ELEMENTS IN THE PRECESSIONAL GEARING (PG)

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Received: 02. 24. 2022
Accepted: 04. 22. 2022

Abstract. This paper deals with the theoretical-experimental model of the tribological behavior of the contact elements in precessional gears (PG) in the functions of conjugate profiles of linear velocities "*convex-concave*" or "*convex-convex*" for the corresponding geometric parameters (ρ_{k_i}, r, $\rho_{k_i}-r$) and the values of the kinematic force parameters (V_{E_1}, V_{E_2}, V_{a_i}) for each concrete position of the k_i points. The calculation of the constructive-functional and kinematic parameters of the "*roller-shoe*" and "*roller-roller*" elements of the tribocouple were adjusted for the experimental verification tests on the SMT-1 installation and adapted to the specific test conditions of the tribocouples with *conformal* contact of gear teeth made of plastics, which will be subjected to *sliding friction* and *rolling* processes of the concrete tribocouple elements.

Keywords: *precessional transmission, precessional gearing, contact load, tribomodel, sliding (rolling) friction, experimental modeling of the precessional gearing, tribological parameters, friction coefficient and force, power loss, mechanical efficiency.*

Rezumat. În lucrare este prezentat modelul teoretico-experimental al aspectelor de comportare tribologică a elementelor de contact în angrenaje precesionale (PG). În funcţie de vitezele liniare ale profilelor conjugate „convex-concave” sau „convex-convexe”, s-au dezvoltat parametrii (ρ_{k_i}, r, $\rho_{k_i}-r$) pentru geometria corespunzătoare şi valorile parametrilor forţei cinematice (V_{E_1}, V_{E_2}, V_{a_i}) pentru fiecare poziţie concretă a punctelor k_i. Calculul parametrilor constructiv-funcţionali şi cinematici ai elementelor tribocuplului *„rolă-sabot”* şi *„rolă-rolă”* au fost ajustaţi pentru încercările experimentale de verificare pe instalaţia SMT-1, adaptată la condițiile specifice de încercare a tribocuplurilor cu contact *conform* dinţilor roţilor din material plastic şi care vor fi supuse la *frecare de alunecare* şi *de rulare* a elementelor tribocuplului concret.

Cuvinte cheie: *transmisie precesională, angrenaj precesional, sarcină de contact, tribomodel, frecare de alunecare (rulare), modelare experimentală a angrenajului precesional, parametri tribologici, coeficient și forță de frecare, pierderi de putere, eficiență mecanică.*

1. Introduction

According to previous research, it is known that *unlike* classical gears [1-7], in Precessional Transmission (PT) [8-17] the profile of the teeth of the central gear is variable - a fact, which *leads* to the variation of the contact geometry of the teeth in the same gear, changing from one shape to another: from convex-concave - at the foot of the central gear tooth, to convex-rectilinear - towards the middle of the tooth and, convex-convex - towards the tip of the tooth.

The manner and limits of variation of the profile of the central wheel depend on the configuration of the fundamental parameters of the precessional gearing Z, r, δ and θ by which the reference front tooth meshing up to 100% is possible to ensure. *This type of gearing represents* a sliding rolling tribocouple with a specific tribological structure and behaviour, characterized by:

1. *Specific construction features of triboelements* (variable shape of the profile of the central wheel tooth; small difference between the radii of curvature of the conjugated surfaces);
2. *High degree of contact multiplicity*;
3. *Variation of contact shape* („*convex-concave*", „*convex-rectilinear*" and „*convex-convex*") for different teeth that are simultaneously in the gear;
4. *Relative spherospatial motion* of surfaces in the contact area;
5. *Variation in large limits of the peripheral linear velocities* of the surfaces of the satellite teeth and the central wheel in the contact area during a precession cycle;
6. *Variable relative sliding velocity* of the conjugated surfaces *for the teeth that are simultaneously in contact*;
7. *Approximately uniform load distribution of the conjugated teeth contact* (on the „convex-concave" portion, where the *angle of engagement* achieves values with small deviations from a constant value).

Due to the complexity and specificity of the given tribosystem, set out in points 1-7 listed above, a fundamental study of the tribological behaviour of the Precessional gearing *is difficult to conduct.* For this reason *we resort to* modelling on simple (standardized) tribosystems, *where tribological processes* similar *to tribological processes occurring in the contact area of the precessional planetary gear can be realized*. In order to meet this requirement, tribomodelling is carried out on the concrete case of the gear under investigation.

The evolution of friction force (coefficient), power (energy) losses and triboelement wear intensity for the couple of concrete material of the wheels and under contact load conditions similar to the working conditions of the precessional gear are followed by the research.

2. Methodological proposals for experimental investigations of tribocontacts in the Precessional Gearing

The experimental modeling of the precessional gearing is proposed to be carried out on the basis of the tribosystematization proposed by Professor **Ion Crudu** [16], performed on a generalized tribomodel and shown in Figure 1.

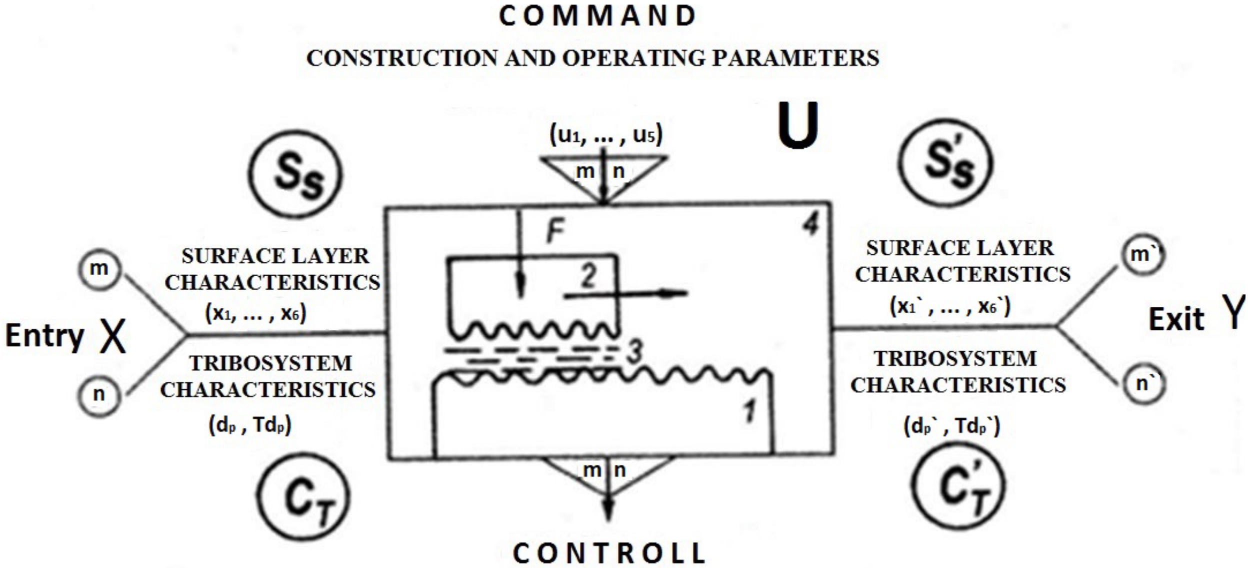


Figure 1. General structure of the physical tribomodel for the experimental recording of input and output data: 1-fixed triboelement; 2-mobile triboelement: 3-interposing medium; 4-working medium [16].

From a functional point of view, *the structure of the tribomodel is characterized by the following groups of parameters*:

1. X_i , Y_i – *input, output;*
2. U – *command;*
3. C – *control*

of which; m – *measurable sizes* and n – *immeasurable sizes*.

The basic *characteristics* are established for each of these size groups:

a. *Input - output sizes* $X_i \rightarrow Y_i$ *:*

a_1. ***Tribostate characteristics*** (S_{s0}, S_{s1}). These characteristics change over time and determine the behavior of the tribomodel over the test period. They are measurable *sequentially* - at the beginning and end of test sequence periods, or *singularly* - during a single run period. This category includes the following indicators:

X_1 - surface *roughness* in the contact zone $(\mathrm{R}_{Z0}, \mathrm{R}_{Z1})$;

X_2 – *macrohardness*$(\mathrm{HB}_0, \mathrm{HB}_1)$, $(\mathrm{HRC}_0, \mathrm{HRC}_1)$ or *microhardness* of contact surfaces $(\mathrm{H}_{\mu 0}, \mathrm{H}_{\mu 1})$;

X_3 – *stress state* $(\sigma_0,\ \sigma_1)$;

X_4 – *chemical composition of the surface layers, resulting from the influence of the environment and working conditions* (X_{40}, X_{41});

X_5 – *material structure* (metallurgical) (X_{50}, X_{51})

X_6 – *purity* (X_{60}, X_{61})

a_2. ***Tribomodel parameters*** (C_{T0}, C_{T1}). Defines the test duration, specific to each type of tribo-model and tribosystem.

b. *Command sizes* (U). Comprising two groups, with values, which are set at the beginning of the experimental tests:

b_1. **Tribo-model construction** $(\mathrm{U_c})$.

u_1 – *shape of the contact* (on flat surface, or cylindrical; *on line with different curvatures in the profile section*; point-shaped);

u_2 – *geometrical dimensions in the contact area*;

u_3 – *triboelement materials (material couples under test).*

b_2. *Test parameters*:

u_4 – *contact loading* (F_n) *and distribution on the triboform*;

u_5 – *relative velocity (sliding, rolling and combinations)*;

u_6 – *presence and type of lubricant*;

u_7 – *ambient temperature in the test chamber* (Θ);

u_8 – *environmental conditions (humidity).*

Control measurements (C), allow for the assessment of *wear level and other parameters by direct measurements*:

c_1 – *amount of material removed (worn)*;

c_2 – *evolution of the friction moment*;

c_3 – *temperature evolution in the contact area*;

c_4 – *evolution of vibration level, etc.*

As a result, based on the real precessional gearing, the main characteristics of the tribomodel are established for a group of sizes. Finally, the values of these quantities are sought from the condition of ensuring the similarity of tribological processes in the contact of the tribomodel and the precessional gear.

For experimental research the tribomodel was made on an installation of 2070 SMT-1 model manufactured according to GRSI RF: 7717-80 (Figure 2) [17].

The kinematic scheme of the 2070 SMT-1 installation is shown in fig.3 The SMT-1 machine (installation) can operate according to closed or open contour force schemes according to 3 kinematic schemes: the revolutions from the electric motor 19 - via belt wheel 1 and toothed belt transmission 5 - are transmitted simultaneously to the upper sample 13 via the belt wheel 6 and to the bottom sample 14 via the belt wheel 3.

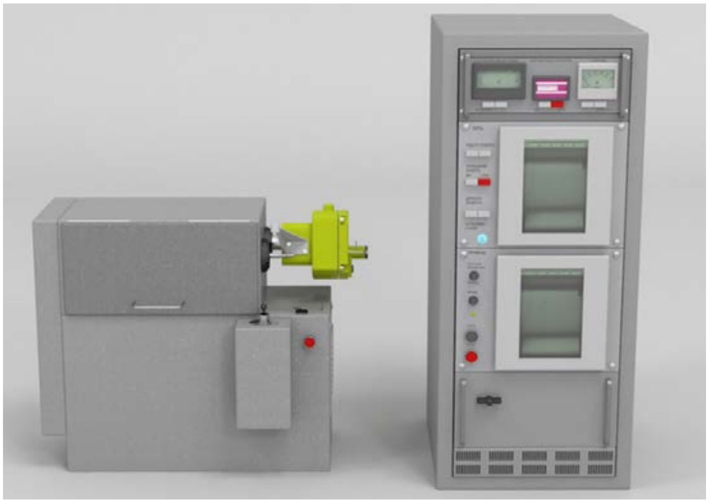


Figure 2. General view of the SMT-1 installation [17].

The sample 13 is installed on the shaft 12 of the folding trolley where the gear wheels 10 and 11 are located inside.

The folding trolley is balanced by means of the spring mechanism 8. An elastic sensor 7 is installed on the bottom sample shaft to record the friction moment through a non-contacting collector, from which signals are recorded on a recording board.

The loading (stressing) of the samples is carried out by means of the spring mechanism 16. The amount (magnitude) of the applied normal force is adjusted by turning the lever 15, and the transmission of the indications to the recording board is made via the flexible connection from the resistor 17.

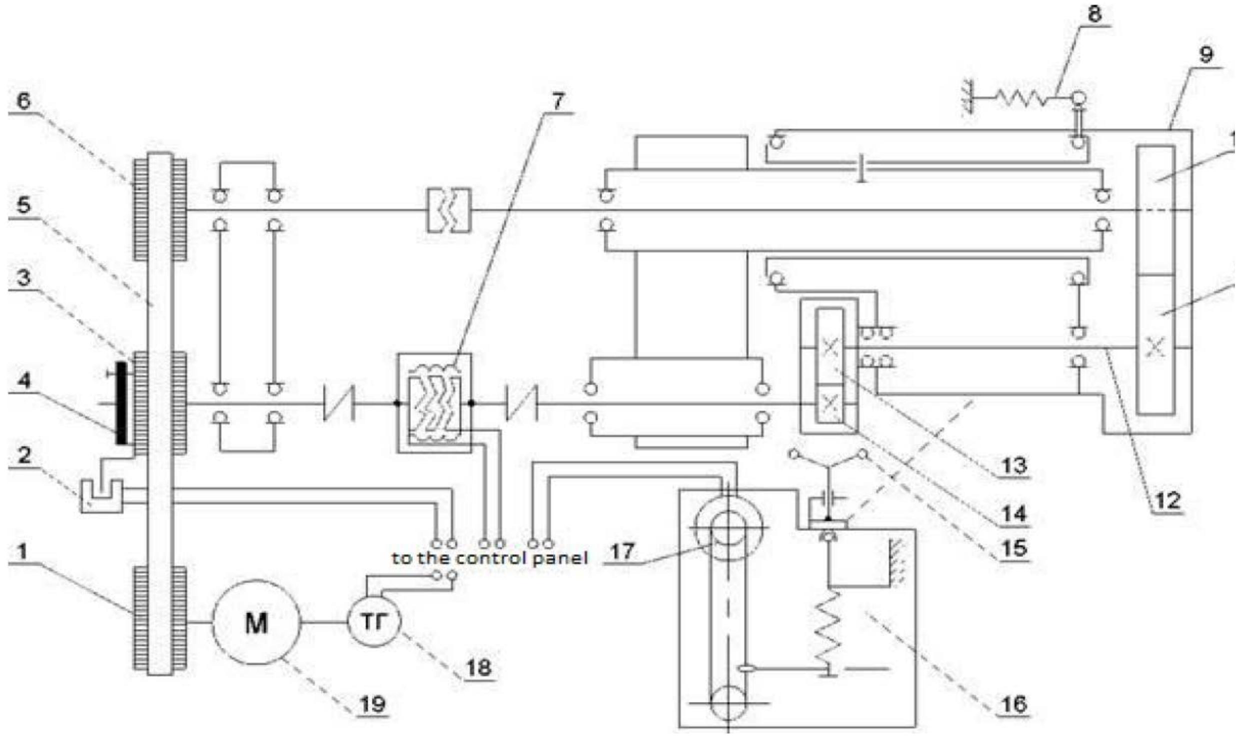
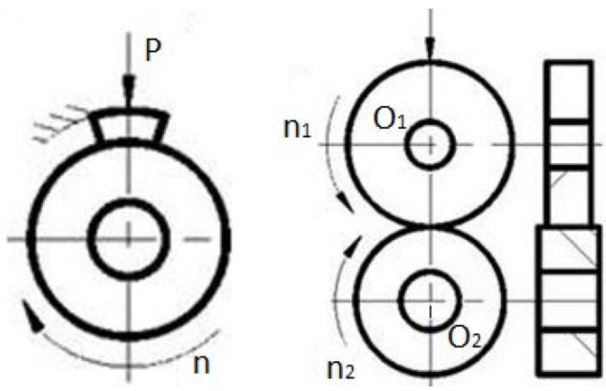


Figure 3. Kinematic scheme of the SMT-1 installation, adapted for experimental testing of the PG (with sliding or rolling motion of the rollers) [17].

The number of revolutions per minute of the electric motor shaft 19 is measured by means of the tacho-generator 18, and the number of revolutions per minute of the bottom sample (friction force) - by means of the non-contact transducer 2. The pin 4 is used to protect the machine (installation) from overloads.

For testing the „shaft-bushing" tribocouple, the process is carried out in a special chamber according to an open contour scheme. In this case the folding carriage 9 is lifted and fixed in this position (turned upside down). Devices, installed on the panel, *allow for the measurement of* the friction moment, the load force, the number of revolutions, the number of cycles (travel) and the temperature in the contact area of the samples in the tribocouple.



a) b)

Figure 4. Variants of performing the contact.

The contact modelling on the SMT-1 installation is carried out in two variants (fig. 4): a – „roller-shoe”; b – „roller-roller”.

The tribomodelling, adapted to the SMT - 1 installation, will be carried out for a precessional gear obtained by mathematical modelling, according to the following parametric configuration Z, δ,β, θ (Z_1 = 24, Z_2 = 25, θ = 3,5^0 , δ = 22,5^0 , r = 6,27 mm și R = 75 mm).

In the mathematical model, *the kinematic analysis was performed* for a series of contact points $k_0, k_1, k_2, \dots, k_i$ corresponding to crankshaft positioning angles for which the variation of the linear velocities V_{E1} of the contact points E_1 on the profile of the central wheel teeth, V_{E2} of the contact points E_2 on the profile of the satellite teeth and the relative sliding velocity between the flanks V_{al} was established.

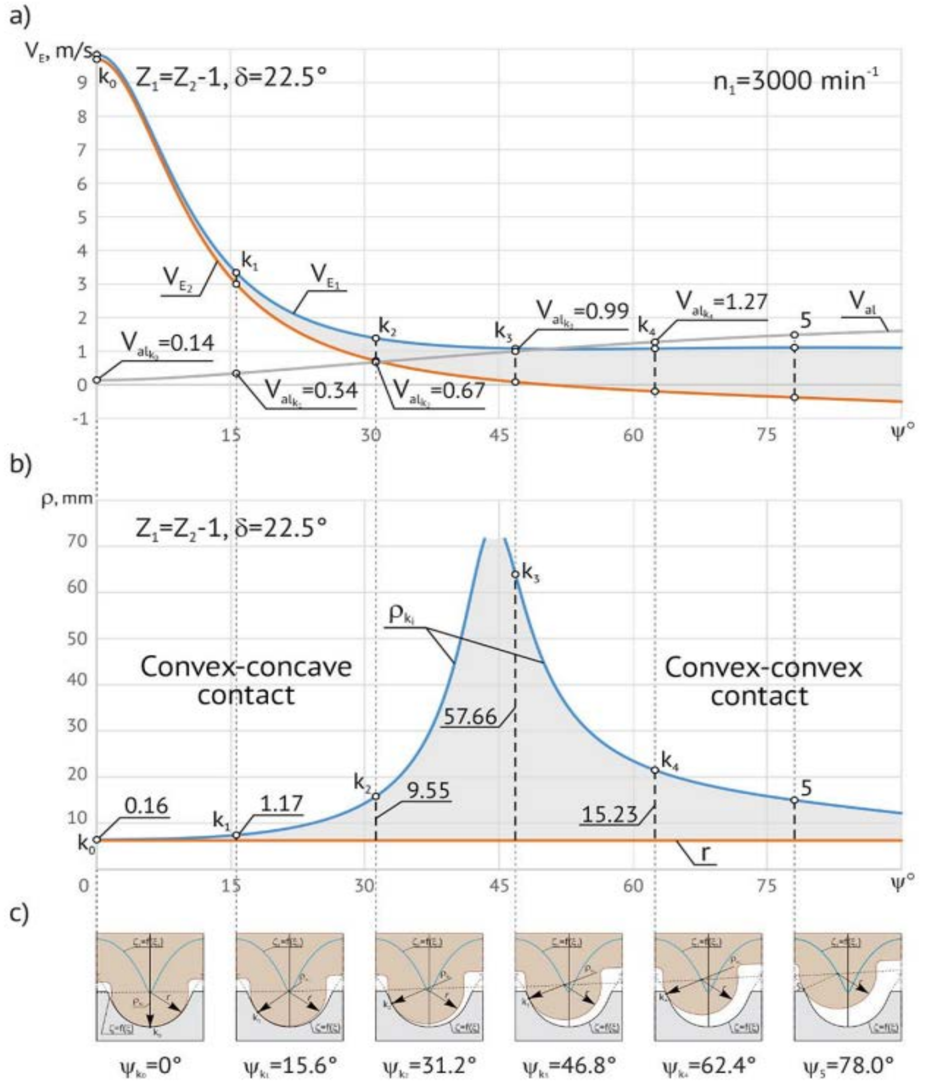


Figure 5. Linear velocities at the contact point V_{E1}, V_{E2}, V_{al} (a) and the difference in the radii of curvature ($\rho_{ki} - r$) (b) of the conjugated profiles in the contact k_i (c) as a function of ψ for $Z_1 = Z_2$–1 and δ = 22,5° (Z_1 = 24, Z_2 = 25, θ= 3,5°, δ = 22,5°, r = 6, 27 mm, R = 75 mm) [14].

The tooth contact geometry at the contact point areas is shown by the radii of curvature ρ_{ki} of the central wheel tooth profile and the satellite tooth profile r and their difference ($\rho_{ki} - r$). The tooth contact kinematics analysis is performed for the crankshaft speed frequency $n_1 = 3000min^{-1}$. When rotating the crankshaft by angle ψ, the contact of the teeth in gearing moves from point k_0 within $0^{\circ} < \psi < 43^{\circ}$ on the *convex-concave* phase, and then moves to the convex-convex phase.At each point on the tooth profile the geometric and kinematic parameters of the contact change taking concrete values at each position. Thus, for position k_0 of the contact, the linear velocity $V_{E1} = 1\,9{,}83$ m/ s, $V_{E2} = 2\,9{,}69$ m/ s, $V_{al} = 0\,{,}14$ m /s, and the radius of curvature of the profile of the central wheel teeth $\rho_{k0} = 6{,}43mm$, of the satellite teeth r = 6,27 mm and their difference ($\rho_{k0} - r$= 0,16 mm). The calculation results for positions $k_0, k_1, k_2, \ldots, k_i$ are shown in Figure 5. The realization of the experimental tribomodel for the precessional gear is related to the meeting of the constructive aspects of the contact by shape („*convex-concave*" or „convex-convex") and dimensions with the corresponding geometrical parameters (ρ_{ki}, r, $\rho_{ki} - r$) and the values of the kinematic parameters (V_{E1}, V_{E2}, V_{al}) for each concrete position of the point k_i.

3. Calculation of the constructive and kinematic parameters of the „roller-shoe" (Figure 6) and „roller-roller" (Figure 7) tribomodels

The tribomodel of the "convex-concave" portion of the contact, according to the kinematic possibilities of the SMT-1 installation, can be realized in the "**roller-shoe**" version (Figure 6) where: $\boldsymbol{r}$ – radius of the roller, $\boldsymbol{\rho_{ki}}$ – radius of curvature of the contact surface of the shoe.

Geometrically the contact is modelled according to the criterion ($\rho_{ki} - r$) - the difference in the radii of curvature of surfaces in the area of points k_i. As a kinematic modelling criterion only the sliding velocity V_{al} can be accepted. Based on these criteria, the values of the constructive and kinematic parameters of the model are concretized for each point k_i (Figure 5) determined by the angle ψ of the position of the crank of the precessional reducer satellite.

From a constructive point of view the roller and shoe triboelements, when interacting, must ensure linear contact with the corresponding difference in radii ($\rho_{ki} - r$) of curvature in the area of the contact points k_1 and k_2 (Figure 5), with corresponding values of the sliding velocity.

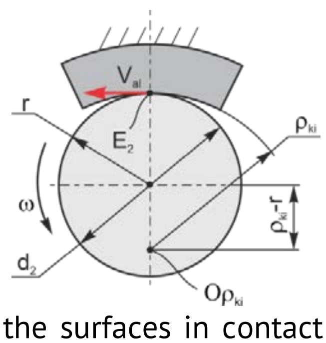


Figure 6. Interaction scheme of the surfaces in contact in the case of the experimental „*roller-shoe*" tribomodel for the „*convex-concave*" portion of the precessional gearing.

Based on the constructive features of the test assembly of the SMT-1 installation, it was accepted that the roller triboelement radius $r = 17mm$ (diameter $d_2 = 2r$).
The radius of curvature of the shoe triboelement is determined according to the relation:

$$\rho_{ki} = r + (\rho_{ki} - r) \tag{1}$$

The rotation speed (revolution) of the guiding roller shaft d_2, required to obtain the sliding velocity with values corresponding to the areas k_i and the contact points (V_{ali}), is determined from the relation:

$$n = {60V_{ali} \cdot 10^3}/{\pi d_2} \ (rot/min) \tag{2}$$

The results of the calculations are shown in Table 1.

Table 1

Constructive and kinematic parameters of the „roller-shoe" tribomodel

Point k_i	r(mm)	$d_2 = 2r$(mm)	$\rho_{ki} - r$(mm)	ρ_{ki}(mm)	V_{al}(m/s)	n(rot/min)
k_1	17	34	1,17	18,17	0,34	191
k_2	17	34	9,55	26,55	0,67	376

In the "roller-shoe" model the influence of the linear reciprocal rolling velocities V_{E1} and V_{E2} on the tribological behavior of the contact is lost.

The "convex-convex" tribomodel of the contact shape. The second variant of experimental modelling of the contact shape of the precessional gearing, possible to perform on the SMT - 1 installation, is the "roller-roller" model with a "*convex-convex*" external contact. In this case the shape of the model contact differs constructively from the shape of the precessional gear contact.

Accepting this compromise, it is possible to realize *the sliding rolling tribosystem model* with which the *tribological behaviour of the contact* under the action of linear reciprocal rolling velocities with relative sliding will be studied with. The sketch of the triboelement interaction in the case of the "roller-roller" contact model is shown in Figure 7, where: r, d_2– radius and diameter of the *driving* roller; ρ_{ki} and d_1 – radius and diameter of the *driven roller*; V_{E2} and V_{E1} – peripheral linear velocities of the reciprocal rolling surfaces of the driving roller d_2 and the driven roller d_1; ω – angular velocity of rotation of the shafts of the *driving* and *driven* rollers.

The constructive and kinematic parameters required for the experimental model of the precessional gear contact ($\rho_{ki} - r$), V_{E2}, V_{E1} and V_{al} for each point k_i, are shown in Figure 5. Constructively, the radius of the triboelement of the *driving roller* is accepted: $r = 17mm$ (diameter $d_2 = 34mm$). According to the accepted value of the diameter of the driving roller and the peripheral linear velocity V_{E2} the *angular velocity* ω is determined *for the contact position in the area of each point* k_i:

$$\omega = {2V_{E2} \cdot 10^3}/{d_2} \ (\frac{rad}{s}) \tag{3}$$

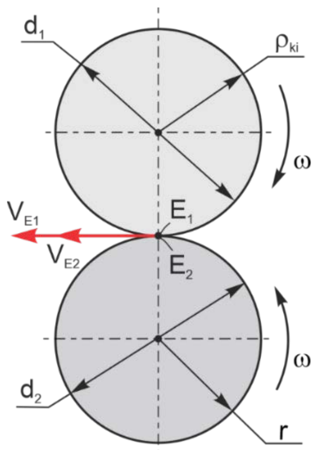


Figure 7. Scheme of the interaction of surfaces in contact for the experimental „*roller-roller*" tribomodel for the „*convex-convex*" shape of the contact.

Knowing the angular velocity, the *diameter of the driven roller triboelement* d_1 is determined:

$$d_1 = {2V_{E1} \cdot 10^3}/{\omega} \ (mm) \tag{4}$$

To achieve the kinematic regime in the contact area of the tribomodel, similarly to the kinematic regime in the contact area of the precessional gear, the *speed of the driving roller* is determined:

$$n = {60V_{E2} \cdot 10^3}/{\pi d_2} \ (rot/min) \tag{5}$$

The calculation results for points k_1 and k_2 are shown in Table 2.

Table 2

Constructive and kinematic parameters of the „roller-roller” tribomodel

Contact point	Roller radius r(mm)	Roller diameter d_2(mm)	$(\rho_{ki} - r)$ (mm)	d_{ki} (mm)	V_{E2} (m/s)	V_{E1} (m/s)	V_{al} (m/s)	ω (rad/s)	n (rot/min)
k_1	17	34	1,17	38,02	2,93	3,27	0,34	172	1646
k_2	17	34	9,55	66,55	0,7	1,37	0,67	41,17	393

Conclusions

1. It has been established that, *unlike classical gears, the Precessional Transmission (PT) can have a variable* profile of the central gear teeth, which *leads to* varying the tooth *contact geometry* within the same gear, *changing from one shape to another* (from “*convex-concave*” at the foot of the tooth of the central gear - to “*convex-rectilinear*” towards the middle of the tooth, and “*convex-concave*” towards the tooth tip), and the *manner and limits of its variation* (profile) depend on the configuration of the fundamental parameters of the precessional *gear* (*Z, r, δ* and *ϑ*) by which the *reference front tooth gearing* of *up to 100%* can be ensured.

2. It has been found that this type of gear represents a tribosystem with sliding rolling motion, with a *specific* constructive *tribological structure and behaviour* - *variable shape of the profile of the central wheel tooth*; small difference in the *radii of curvature* of the conjugated surfaces; high degree of contact multiplicity; contact shape variation – “CVX-CCV”, “CVX-RL” and “CVĂ-CVX” - for different teeth *simultaneously* in the gear; relative *spherospatial* motion of *surfaces* in the contact area; *variation* (within large limits) of peripheral linear velocities of the surfaces of the *satellite teeth* and the *central wheel* in the *contact* area during a precession cycle; *variable relative sliding* velocity of the conjugated surfaces for the teeth that are simultaneously in contact; approximately uniform *distribution* of the contact load of the conjugated teeth (on the “CVX-CCV” portion - where the angle of gearing *has values with small deviations from a* constant *value*).

3. From point 2 (*complexity and specificity of the given tribosystem*),described above, we note that a fundamental study of the *tribological behavior* of PT gearing is difficult to carry out, and for this reason we have resorted to modelling on simple (standardized) tribosystems, where tribological processes, similar to tribological processes which occur in the area of PP (precessional plane) gearing contact, can be realized. In order to meet this requirement, tribomodelling is performed on the concrete case of the gear under investigation.

It was proposed that the Experimental modelling of the gear *should be carried out* on the basis of the tribosystematization, developed by Prof. Ion Crudu, which is represented by a *generalized tribosystem* (with the setting of input, *command, control and output* parameters), *carried out* on a installation of the 2070 (SMT-1) model, produced according to GRSN RF – 80.

The devices installed on the panel, allow for the measurement of *the friction moment, the load force*, the number of revolutions, *the number of cycles covered* and *the temperature in the contact area* of the elements *("roller-shoe", "roller-roller") of the corresponding tribo-couple*. *Sample wear shall be assessed by the mass method on the VLA - 2000 electronic scale (accuracy - 0,0001 g).*

4. The constructive and functional parameters of the samples for the experimental tests of the tribocouple elements *"roller-shoe"* and *"roller-roller"* in their concrete test regimes have been calculated and adjusted to the SMT-1 installation.

Acknowledgements: The article was published by the authors by conducting scientific research under the *State Research-Innovation Project*, No. 160-PS of 31.01.2020 *"Increasing the competitiveness of precessional transmissions by developing and capitalizing on the gear with conform contact of the teeth and expanding their application area"*. *Project number 20.80009.700.24*, dated 31.01.2020. *Project leader* - Acad. Ion Bostan.

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