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EXPERIMENTAL RESEARCH OF PRECESSIONAL TRANSMISSIONS WITH MULTIPAIR CONVEX-CONCAVE GEARS: DATA ACQUISITIONS, EXPERIMENTAL SAMPLES AND RESULT

Maxim VACULENCO, Ion BODNARIUC, Oleg CIOBANU, Radu CIOBANU, Iulian MALCOCI, Nicolae TRIFAN, Dumitru VENGHER

Abstract: *The paper presents aspects of experimental research on the functional characteristics of kinematic and low-power precessional transmissions with multi-pair convex-concave gears with straight and inclined teeth. The energy losses in the gears are investigated depending on the relative sliding with friction between the flanks of unitary or simultaneously multipair conjugated teeth, as well as the bearing capacity of the convex-concave contact of the teeth depending on the difference in flank curves, including starting moment and the torsional stiffness of the precessional gear. Gear teeth are manufactured from plastic by roll molding on CNC machine tools adapted to the Gleason system, as well as by additive 3D printing technologies. The teeth of the central and satellite wheels are made of various pairs of plastic-metal-plastic, metal-metal-metal, plastic-plastic-plastic materials. The parametric characteristics of kinematic gears samples and low-power transmissions, experimental stands for data acquisition and computerized processing are described, research results are analyzed. The comparative analysis of the functional characteristics of precession transmissions in relation to other mechanical transmissions is presented.*

Key words: *Mechanical transmissions, functional characteristics, precessional gears, convex-concave tooth contact, mechanical efficiency, bearing capacity, torsional stiffness.*

1. INTRODUCTION

Mechanical gear transmissions fabricated by the specialized manufacturing companies, as a rule, are divided into type dimensions depending on the transformed and transmitted power *N* [kw], depending on torque *T* [Nm], angular speed ω [s^{-1}], applied to gear shafts and by the kinematic parameter expressed by the ratio of transmission *i*.

Among the functional characteristics of precessional transmissions which, as a whole, define their technical level and competitiveness in relative to other types of mechanical transmissions it can be take in consideration following parameters:

- The mechanical efficiency that expresses energy losses, in particular, in tooth gearings, %.
- Bearing capacity of tooth in contact $\sigma_H \leq \sigma_H{}'$, MPa.
- The specific consumption of material expressed by transmission mass in kg, relative to the transmitted moment of torque, kg/Nm.
- Noise and vibration emission, dB.
- Torsional rigidity, Nm/rad.
- Kinematic precision, sec. angle.
- Moment of inertia, kg m2
- Starting moment, g cm.
- Stop time, sec.
- Thermal stability expressed by intensity of heat release, kcal/m^2h degree [4,5,7,8].

2. THEORETICAL ASPECT OF RESEARCH

Experimental research of mentioned functional characteristics are performed on physical models according to the nominal moment of torque T, Nm approved in the design calculation stage at the contact pressure of the median diameter d_m for $A_{CX-CV}^{D(\beta)}$ precessional gearing where (considered as the generalizer parameter) according to the relation [1,6]:

$$d_m = 53{,}3\sqrt[3]{\frac{T_4 \cos\beta_m(\rho_4 - r_3)k_{HP}k_{H\beta}k_{HV}}{\left(\sigma_{HP}'\right)^2 \cos\alpha_\omega \psi_{bd} Z_\varepsilon tg\beta_3\rho_4}}, mm \quad (1)$$

for $A_{CX-CV}^{D(\beta)}$ precessional gears with bevel teeth with angle βm and

$$d_m = 53{,}3\sqrt[3]{\frac{T_4(\rho_4 - r_3)k_{HP}k_{H\beta}k_{HV}}{\left(\sigma_{HP}'\right)^2 \cos\alpha_\omega \psi_{bd} Z_\varepsilon tg\beta_3\rho_4}}, mm \quad (2)$$

for spur precessional gears A_{CX-CV}^{D} , where

σ_{HP}' - the permissible stresses in the contact of the teeth with consideration of relative sliding with friction and rolling;

α_ω- the pressure angle between the flanks conjugated teeth;

$\psi_{bd} = b_\omega / d_m$ - the relative length teeth coefficient;

Z_ε- the number of simultaneous pairs of conjugates teeth;

β_3 - the taper angle of the crown satellite wheels' teeth;

ρ_4 - the radius of curvature of the teeth with a convex-concave profile of the central wheel with Z_4 teeth;

$(\rho_4 - r_3)$- the difference in the curves of the tooth flanks central wheel ρ_4 and crown wheel-satellite with Z_3 teeth;

β_m- angle of inclination of the teeth;

k_{HP} - load distribution factor of non-uniformity between pairs of simultaneously conjugated teeth;

$k_{H\beta}$ - load distribution factor of non-uniformity along the length of the teeth;

k_{HV} - load dynamics factor.

The permissible stresses σ_{HP}' are considered with the presence of relative sliding with friction V_{al} between the flanks of the teeth conjugate simultaneously at the points of contact *k0...k5*, determined by numerical computation of the function $\zeta = f(\xi)$ for gearing with geometric parameters and distinctive kinematics of the transmission related to the physical model.

The permissible stresses σ_{HP}' in the contact teeth for $A_{CX-CV}^{D(\beta)}$ precessional gears at the current research stage is determined by analysis their similarities with involute bevel gears and for the A_{CX-CV}^{D} precessional gears through analysis of the contact of the kinematic coupling in the CYCLO gears transmissions.

To identify tooth contact particularities in A_{CX-CV}^{D} precesional gears related to the function $\zeta = f(\xi)$ we determine the linear velocities in contact points V_{E1} and V_{E2} and their difference to them, which represents the relative speed of slippage in the *E* contact point of the teeth.

Mechanical transmissions functional characteristics for $A_{CX-CV}^{D(\beta)}$ and A_{CX-CV}^{D} toothed precessional gears vary within certain distinctive limits individualized for each characteristic separately depending on the transmitted load, as a rule, the torque applied to the driven shaft, in the case of 2K-H type precessional transmissions, expressed by T_4, Nm.

The moment of nominal torque T_4, Nm is determined from expression (3), in the case of gears with tilt teeth $A_{CX-CV}^{D(\beta)}$ and from expession (4) in the case of gears with straight teeth A_{CX-CV}^{D} for a specific number of pairs for simultaneously conjugated teeth Z_ε, depending on the elasticity constant of the material Z_M teeth, contact geometry and shape of teeth αw, α_ω, ψ_{bd}, β_3, ρ_4, $(\rho_4 - r_3)$ including load factors k_{HP}, $k_{H\beta}$ and k_{HV}.

In this case for $A_{CX-CV}^{D(\beta)}$ precessional gears with tilt teeth with an angle β_m the torsional nominal moment T_4, Nm at the driven shaft will be

$$T_4 = \frac{d_m^3 \left(\sigma_{HP}'\right)^2 \cos\alpha_\omega \psi_{bd} Z_\varepsilon tg\,\beta_3 \rho_4}{53,2^3 \cos\beta_m \left(\rho_4 - r_3\right) k_{HP} k_{H\beta} k_{HV}}, Nm \qquad (3)$$

and for A^{D}_{CX-CV} precessional gears with straight teeth

$$T_4 = \frac{d_m^3 \left(\sigma_{HP}'\right)^2 \cos\alpha_\omega \psi_{bd} Z_\varepsilon tg\,\beta_3 \rho_4}{53,2^3 \left(\rho_4 - r_3\right) k_{HP} k_{H\beta} k_{HV}}, Nm. \qquad (4)$$

In our case, the physical model of 2K-H precessional transmissions was researched experimental for the concrete type, we determine the nominal torque for driven shaft T_4, Nm split into several intervals $T_{4,i}$, for example, $T_{4,1}=0$; $T_{4,2}=0.25$; $T_{4,3}=0.5$; $T_{4,4}=0.75$ and $T_{4,5}=T_4$, for which was determined the corresponding values of torques $T_{1,i}$ applied tothe main shaft according to the expression $T_{1,i_{1...5}} = \frac{T_{4,i_{1...5}}}{\eta \cdot i}, Nm$.

The paper presents the results and comparative analysis for functional characteristics of small power kinematic precessional transmissions obtained experimentally related to physical models for A^{D}_{CX-CV} precessional gears with straight teeth and median diameter $D=42\ mm$ shown in figure 1 and for $A^{D(\beta)}_{CX-CV}$ precessional gears with inclined teeth with median diameter $D=84\ mm$ shown in figure 2. [3]

The precessional transmission shown in figure 1, the conjugate teeth of the two crowns of the satellite-wheel 3, on the one hand contact with the teeth of the fixed central wheel 1 assure co-ratio $Z_1=Z_{2-1}$, and on the other hand with the teeth assure contact with movable central wheels 2 with co-ratio $Z_4=Z_3+1$, thus constituting the A^{D}_{CX-CV} gearing (or $A^{D(\beta)}_{CX-CV}$ gearing), which provides convex-concave contacts between conjugate teeth with little difference in curves flanks.

For tooth gearing (Z_3–Z_4), the mobile central wheel 2 has one more tooth than the crown satellite-wheels 3 with which it is conjugated, what cinematically favors the operation of the gear in reduction mode.

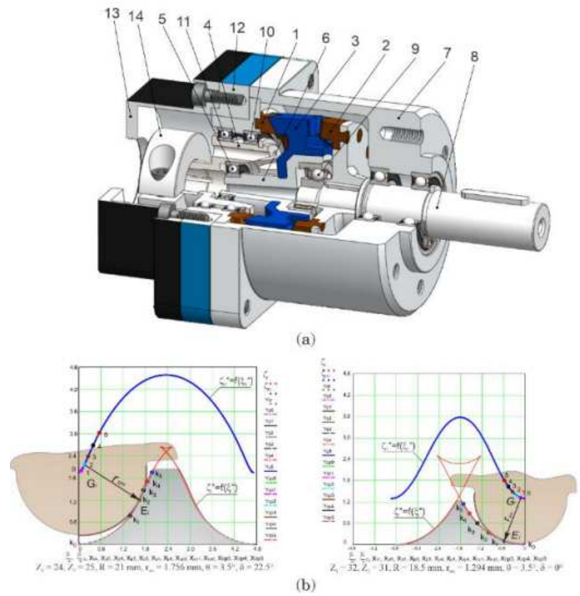


Fig.1. 2K-H type precessional transmission ($i=+14.29$) for precesional gears A^{D}_{CX-CV} with the ratio $Z_1=Z_{2-1}$, $Z_4=Z_{3+1}$, number of tooth for semi-crank satellite, with conical axoid angles $\delta_{(1-2)}=22°30`$ and $\delta_{(3-4)}=0°$ (a); tooth profiles in gears (Z_1-Z_2) and (Z_3-Z_4) (b).

To facilitate operation of the transmission in this mode at small sizes values of the transmission ratio i, gearing with the tooth ratio $Z_4=Z_3+1$ and $Z_2>Z_3$ is projected with the angle of the conical axoid $\delta_{(3-4)}=0°$.

At the same time, to reduce relative slipping friction between the flanks, the number of pairs of teeth was reduced at the same time in frontal gear up to $\varepsilon_f=1...2$.

2K-H type precessional transmission with A^{D}_{CX-CV} gearing, with the configuration of numbers of teeth $Z_1=Z_2-1$, $Z_4=Z_3+1$, $Z_2>Z_3$ and the angles for conical axoids $\delta_{(1-2)}>0°$ and $\delta_{(3-4)}=0°$, in this case the number of teeth varies $15\leq Z_{1,2,3,4}\leq 60$, ensures varying transmission ratios in range $+8.3\leq i\leq +30.3$, meaning that with shafts rotation in the same direction.

At the same time, it must be emphasized that the gearing (Z_3–Z_4) with the co-ratio of the number of teeth $Z_4=Z_3+1$ is projected with the angle of the conical axoid $\delta_{(3-4)}=0°$ and $Z_2>Z_3$ to ensure convex-concave geometry of tooth contact and tooth contact kinematics point favorable to the operation of the transmission in the reduction regime.

From figure 1(b) we find that the angle of pressure α_ω between the flanks of the teeth (Z_3–

Z_4) with co-ratio $Z_4=Z_3+1$ is greater than in gearing (Z_1–Z_2) with the co-ratio $Z_1=Z_2-1$. Therefore, to reduce energy losses generated by the gearing (Z_3–Z_4) with the gear ratio $Z_4=Z_3+1$, it is recommended to reduce the numberof pairs of simultaneously engaged teeth up to1...2 pairs under load in front gearing and the teeth are recommended to be made inclined, with the provision of a higher rate of longitudinal coating by rolling.

The teeth in gears (Z_1–Z_2) and (Z_3–Z_4) conjugate in multi-pair contacts with convex-concave geometry, formed by the profiles with variable curvature curves of the center wheels 1 and 2, and in the circular arc of the crowns of the satellite wheel 3. Rotational movement of the crankshaft 4, by means of the support 5 mounted on the end of the semi-axle 6 of the satellite wheel 3, turns into its sphero-spatial movement. Z_2 and Z_3 teeth have crowns of satellite 3 engage corresponding to teeth Z_1 and Z_4 of the center fixed wheel 1, mounted in the housing 7, and movable wheel 2, mounted on the driven shaft 8, thus achieving transmission ratio $i=-Z_2Z_4/(Z_1Z_3-Z_2Z_4)$.

The defining constructive feature of the kinematics precessional transmission consists in the fact that first gears (Z_3–Z_4) is executed with co-ratio of teeth numbers $Z_4=Z_3+1$ and the angle of the conical axoid $\delta_{(3\text{-}4)}=0°$ and the second gearing (Z_1–Z_2) – with the co-ratio $Z_1=Z_2-1$ and $\delta >0$. So for the physical model of the precessional transmission 2K-H type with gears shown in figure 1 the nominal moment of torque T_4, Nm applied to the driven shaft will be determined for the following constructive and geometrical gearing parameters:

- the nominal diameter of the gearing $d_m = 42$ mm;
- limit of contact with consideration relative sliding in tooth contact simultaneously conjugate in gearing, $\sigma_{HP}{}'=1100\ MPa$ (for steel wheels approximating the influence of relative slip between flanks);
- the average pressure angle between the flanks teeth simultaneously conjugated in contacts k_i – $\alpha_\omega=20°$;
- the coefficient of the relative width of the wheel teeth flat plants $\psi_{bd}=0.2$;
- the number of pairs of teeth simultaneously conjugated according to the profile modification of conjugate teeth $Z\varepsilon=4$ ($Z\varepsilon=3$);
- the taper angle of the teeth of the crowns satellite wheels $\beta_3=3.5°$;
- the radius of curvature of the teeth flanks of the central wheel ρ_4 and the difference in radii of curvature (ρ_4–r_3) of the flanks of the equivalent pair of teeth determined by the equations $\zeta = f(\xi)$, depending on the precession angle ψ: $\rho_4=1.789\ mm$, $(\rho_4-r_3)=0.696\ mm$;
- load distribution factor between the simultaneously conjugate teeth $k_{HP}=1.4$;
- the load distribution factor along the length teeth $k_{H\beta}=1.2$;
- the load dynamism factor in gearing $k_{HV}=1.5$.

For the physical model of the kinematic precessions transmission presented in figure 1 the nominal torque applied to driven shaft T_4, Nm is determined from formula (4).

The transmission (fig. 1) has the median diameter $d_m=42\ mm$ and transmission ratio $i=+14.29$, with convex-concave gears manufactured on CNC machine tools from 4140 steel with tooth number co-ratio $Z_1=Z_2-1$ and $Z_4=Z_3+1$.

For precessional gearing A^{D}_{CX-CV} with three pairs of simultaneously conjugated teeth, torsional moment $T_4=21.7\ Nm$, and with four pairs of teeth $T_4=29.1\ Nm$.

Table 1 shows the torque values T_4, Nm divided into five equal intervals $T_{4,i}$ applied to the driven shaft with four pairs of teeth simultaneously conjugate and the torques $T_{1,I}$ calculated at the main shaft with the relation:

$$T_{1,i} = \frac{T_{4,i}}{\eta \cdot i}, Nm. \tag{5}$$

Table 1

Torque values on the main and driven and main shafts

Nr	The intervals of the nominal torque $T_{4,i}$, Nm	Torque on driven and main shifts	
		$T_{4,i}$, Nm	$T_{1,i}$, Nm
1	*0*	*0*	*0*
2	*0,25 T_4*	*7,275*	*0,6364*
3	*0,50 T_4*	*14,55*	*1,2732*
4	*0,75 T_4*	*21,82*	*1,9113*
5	*1,00 T_4*	*29,10*	*2,5455*

The majority's experimental research performed on functional characteristics of the precessional transmission with the median diameter $d_m=42\ mm$ is achieved with loading of the driven shaft with the nominal torque $T_4=29.1\ Nm$ divided into intervals $T_{4,i}$ and the reactive moment at the driving shaft calculated from expression (5), for geometry tooth flanks designed according to ζ=f(ξ) and the transmission ratios *i (, -45, -124, , -360, +164, +14.29* etc).

The 2K-H precessional transmission shown in figure 2 has the transmission ratio $i=-164$ with $A_{CX-CV}^{D(\beta)}$ precessional gears with following parameters $Z_1=40$, $Z_2=41$, $\beta_2=3.2°$, $Z_3=33$, $Z_4=32$, $\beta_3=3.5°$, $\delta_{(1\text{-}2)}=\delta_{(3\text{-}4)}=15°$, $\theta=3.5°$ and differs constructive than the one in figure 1 by the particularities of the transformation mechanism of the rotary movement of the driving shaft 5 in sphero-spatial movement of the wheel-satellite 2. The main shaft 5 with crank function is equipped with a portion inclined under the nutation angle θ relative to the axis - bearings 8 and 9. On the inclined portion of the drive shaft 5 is mounted satellite-wheel 2 on radial-axial bearings 6 and 7. Axles of bearing pairs 8, 9 and 6, 7 are concurrent in the precession center *O*.

Thus, when turning the crankshaft 5 in radial bearings 8 and 9, satellite wheel 2 installed in radial-axial bearings 6 and 7 perform sphero-spatial movement around the precession center *O* with nutation angle θ and floats axially between the central wheels 3 and 4.

It can be mentioned that the coincidence point of intersection of the axes of the pairs of bearings 6 and 7, 8 and 9 with the center of precession leads to the displacement of the center of mass components of the precessional node in relation to the center of precession and implicitly at the imbalance dynamically with additional requests of bearings and crankshaft, defined by Euler's dynamic equations.

At the same time, the coincidence of the point of intersection of the crankshaft-shaft axes with the center of precession can lead to displacement the contact patch between the flanks outside the length assets of multiparous conjugated teeth.

Functional characteristics of the 2K-H type precessional transmission shown in figure 2 experimentally is investigated at the nominal torque *T4 = 164 Nm* calculated for $A_{CX-CV}^{D(\beta)}$ precesional gearing with median diameter $d_m=84\ mm$, transmission ratio $i = -164$ achieved with metal wheels made of steel 4140 with co-ratio and the number of conjugated teeth $Z_{1(4)}=Z_{2(3)}-1$, with the radius of curvature $\rho 4$ of the teeth central gears determined for a pair of teeth equivalent $\rho 4=8.94\ mm$ and the difference of the radii of curvature of conjugate teeth $(\rho_4-r_3)=6.712\ mm$. [3]

Torques applied to the driven shaft $T_{4,i}$ *Nm* and the reactive moments formed at the driving shaft is calculated according to expression (5) the values of which are presented in table 2.

Table 2

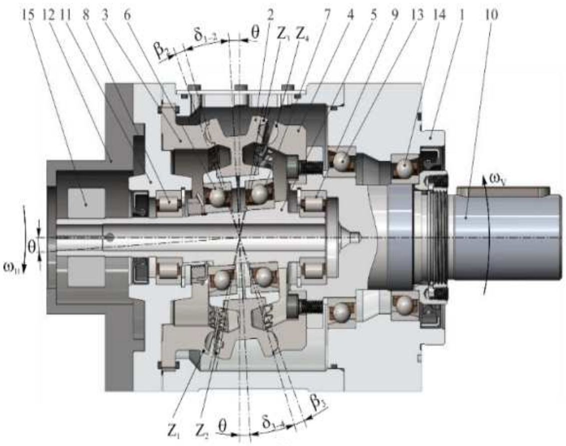
Torque values on the main and driven and main shafts

Nr	The intervals of the nominal torque $T_{4,i}$, Nm	Torque on driven and main shifts	
		$T_{4,i}$, Nm	$T_{1,i}$, Nm
1	*0*	*0*	*0*
2	*0,25 T_4*	*41,0*	*0,3125*
3	*0,50 T_4*	*82,0*	*0,6250*
4	*0,75 T_4*	*123,0*	*0,9375*
5	*1,00 T_4*	*164,0*	*1,25*

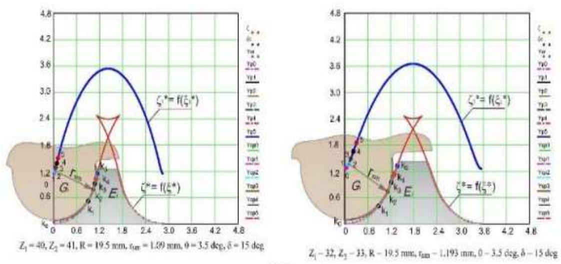
In precessional transmission with $A_{CX-CV}^{D(\beta)}$ gearing with generated circular arc curvilinear teeth by the Gleason system, the functional characteristics are more advanced in comparison A_{CX-CV}^{D} gearing with straight or rectilinear teeth [2].

In context, it should be noted that the summary length of contact lines l_Σ of pairs of simultaneously conjugate teeth $Z\varepsilon$ in $A_{CX-CV}^{D(\beta)}$ gearing with inclined teeth is determined with the consideration that the relative length of the teeth in relation to the median diameter of the plane wheel according to the relation

$$l_\Sigma = \frac{bZ_\varepsilon}{\cos\beta_m} = \frac{\psi_{bd} d_m Z_\varepsilon}{\cos\beta_m}. \qquad (6)$$



(a)



(b)

Fig 2. Precessional transmission 2K-H (i=*-164*) with $A_{CX-CV}^{D(\beta)}$ gears with diameter d_m=*84 mm*: with circular teeth and satellite-wheel mounted axially floating on the crankshaft with concurrent axes in the center of precession under nutation angle θ (a): the flank profiles of the conjugate teeth in contacts k_i (b).

At the same time, the expression $\sqrt{\frac{\cos\beta_m}{\psi_{bd}Z_\varepsilon}}$ from the formula (1) represents the total length of the lines of contact of simultaneously conjugate inclined teeth under the angle β_m. In reality, in $A_{CX-CV}^{D(\beta)}$ gearing with curvilinear teeth, depending on the curves shape in circular arc summary length of the contact of simultaneously conjugated teeth determine from the expression

$$l_{cg} = \frac{\pi d_m \gamma_{cg}}{360^\circ}, \qquad (7)$$

where γ_{cg} is the angle at the center O_c of the curve directory of the flanks of the teeth.

It has been found [2] that where the difference l_Σ calculated by relations (6) and (7) does not exceed 5% of the length l_Σ, then the calculation can be carried out by the relations accepted for $A_{CX-CV}^{D(\beta)}$ gearing with straight teeth inclined with the angle β_m.

3. EXPERIMENTAL RESEARCH FOR FUNCTIONAL CHARACTERISTICS OF PRECESSIONAL TRANSMISSIONS

Functional characteristics of precessional transmissions with $A_{CX-CV}^{D(\beta)}$ gearing were researched on the physical models of the transmissions, shown in Figures 1 and 2.

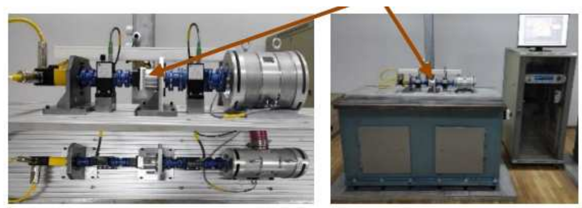


Fig.3. Experimental research stand for the functional characteristics of 2K-H kinematic precessional transmissions with d_m=*42mm* with acquisition and processing of computerized data

Stands for experimental research shown in Figures 3 and 4 were designed with computerized data acquisition and processing, with the provision of universal testing possibilities developed in accordance with the prescribed requirements in specific experimental testing methods for each functional characteristic.

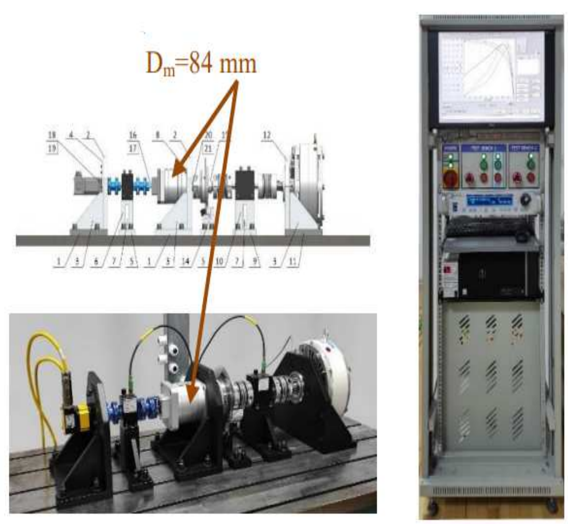


Fig.4. Experimental research stand for the functional characteristics of of 2K-H kinematic precessional transmissions with d_m=*84mm* with acquisition and processing computerization of data

3.1. Relative weight dependence for tooth gearing related to drive shafts torque

The dependencies shown in figure 5 are built according to standard GOST 31592-2012. Reductory obșemașino - stroitelinogo primenenia. Obşie tehniceskie uslovia. 19s. UDK 621.83:006.354. Moskva, Standartinform, 2014 for various types of transmissions (curves 1 – 10), and curve 11 is constructed in Table 3 according to results presented in [1] for $A_{CX-CV}^{D(\beta)}$ gearing and 12 – for A_{CX-CV}^{D} gearing.

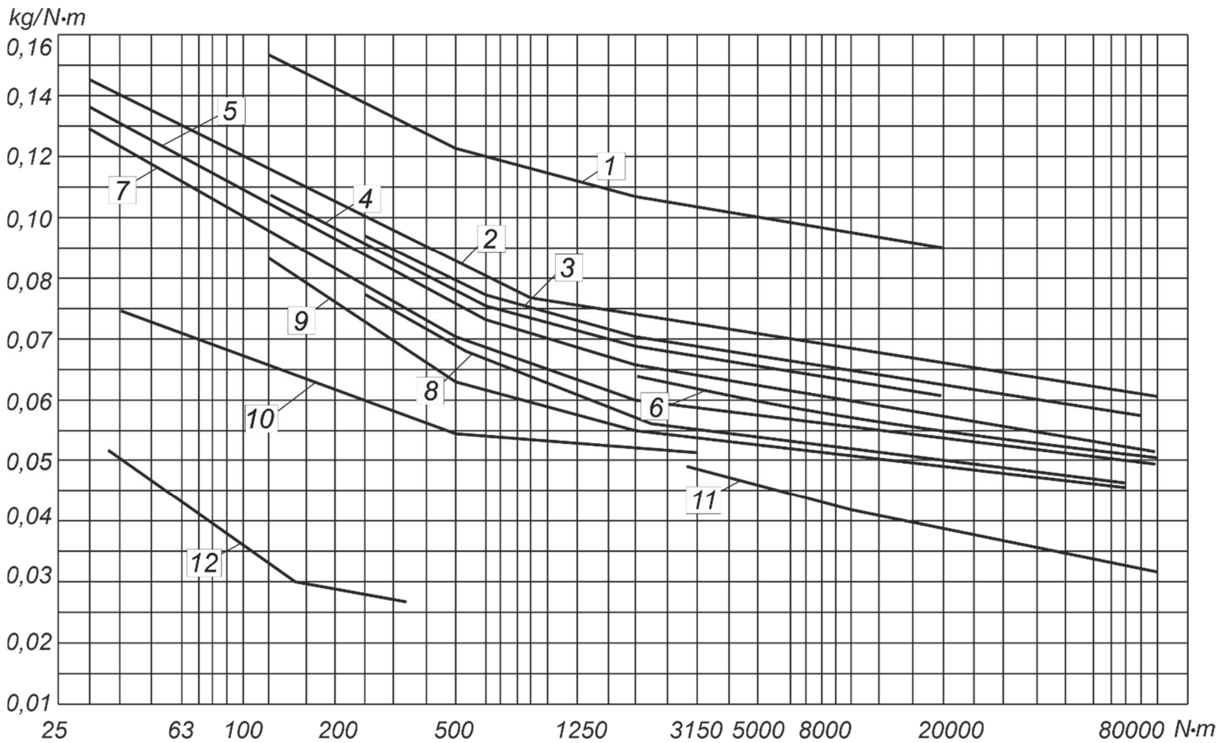


Fig.5. Dependence of the relative weight of the reducer by torque at the output shaft: 1 – conical reducer; 2 – cylindrical-conical reducer in two-stage, single-stage auger; 3- reducer cylindrical-conical in three stages, cylindrical worm in two steps, cylindrical in three steps with asymmetric scheme; 4 – snail; 5 – cylindrical in two steps with diagram asymmetric, worm, globoidal, two-stage worm; 6 – cylindrical in three stages with a symmetrical scheme; 7 – cylindrical in one step; 10 – harmonical; 11 – 2K-H type precessional transmission with A_{CX-CV}^{D} gearing; 12 - 2K-H type precessional transmission with $A_{CX-CV}^{D(\beta)}$

Table 3

Relative weight dependence for tooth gearing related to drive shafts torque [1]

i	T, Nm	m, kg	γ, kg/Nm
-527	225000	7875	0,035
-405	2950	145	0,049
-279	15000	681	0,045
-144	9010	391	0,042
-144	54312	2822	0,051
-575	370000	8225	0,022
	29	0,478	0,0165
	164	4,894	0,0298
	32	1,7	0,053

3.2. Mechanical efficiency

The mechanical efficiency of mechanical transmissions express energy losses in all the kinematic couples contained in the structure the transmission.

For mechanical transmissions based on new principles of transformation and transmission of movement and load, including precessional transmissions, energy losses in gears represent special importance for scientific and practical interest.

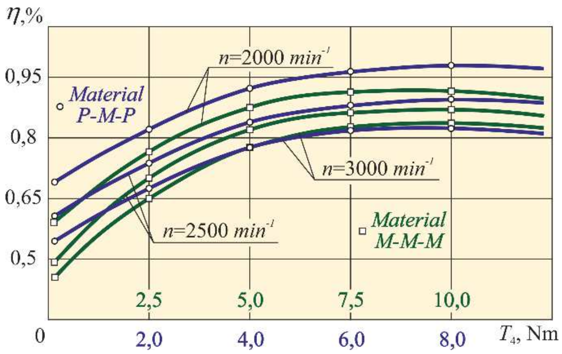


Fig.6. The mechanical efficiency of the 2K-H type kinematic precessional transmission with the median diameter of the gear d_m=42 mm depending on the torque T4, Nm for different frequencies of revolutions n, min-1 of the driving shaft (wheels in pairs Metal- Metal-Metal and Plastic-Metal-Plastic, plastic lubricant, etc.)

For this purpose, the $A^{D(\beta)}_{CX-CV}$ and A^{D}_{CX-CV} gearing of the precessional kinematic and low power transmissions were made from different tribological couples of steel-steel, steel-plastic and plastic-plastic materials using the technologies of casting under pressure from the masses plastics, through additive technologies with 3D printing and on multi-axis machine tools with numeric command.

Figures 6 and 7 show the dependence of the mechanical efficiency η for the 2K-H transmission in Metal-Metal-Metal and Plastic-Metal-Plastic pairs, from 4140 carbon steel and Glass-Filled PEEK materials manufactured on CNC machine tools with metal-to-metal wheels made of steel 4140 on CNC machine tools depending on torque T_4 for the work frequency $n=(1000\text{-}3000)\ min^{-1}$, the lubrication medium with grease plastic and liquid with oil.

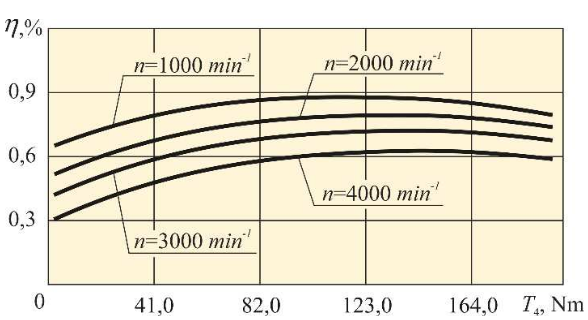


Fig.7. The mechanical efficiency of the low-power 2K-H type kinematic precessional transmission with the median gear diameter d_m=84 mm depending on the torque moment T4, Nm for different frequencies of revolutions n, min-1 of the driving shaft (metal wheels with inclined teeth, liquid lubricant, etc.)

It was found that the maximum value of the mechanical efficiency η, % corresponds to the maximum value of the nominal moment applied to the driven shaft, and the increase in the frequency of revolutions of the shafts n, min-1 leads to the decrease of the mechanical efficiency of the transmission.

At the same time, it is found that with the increase of the transmission ratio i, the mechanical efficiency of the precessional transmission decreases as in all mechanical transmissions, expressing close similarities with harmonic transmissions.

3.3. Torsional rigidity

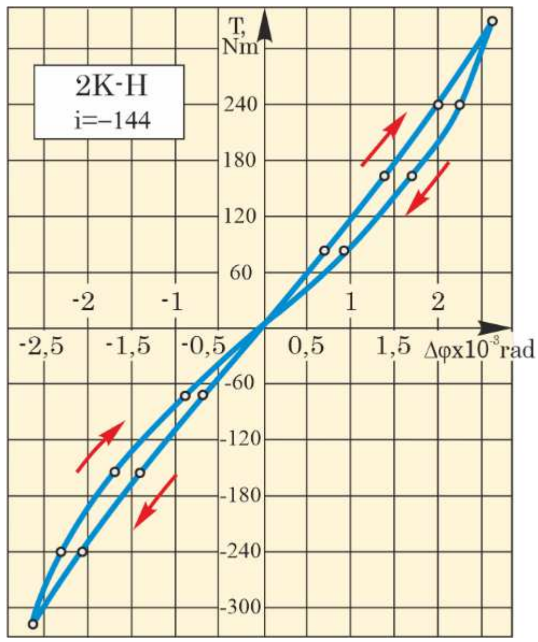
The tested power reducers [1] were used for torsional stiffness research, the stands being completed with the necessary equipment. In most machines and mechanisms, the torsional moment is applied to the driven shaft, a fact that conditions the experimental research of torsional rigidity by loading the driven shaft with the locking of the driven shaft. However, this methodology cannot be carried out in reducers with a high transmission ratio, due to the significant difference between the torque values created at the input and output precessional shafts. Creating a large torque on the driven shaft with the help of weights significantly complicates the experiment by introducing an error in the measurement system, generated by the bending of the driven shaft. In these cases, the driving shaft is torqued and the driven shaft is locked with the reducer housing.

Based on the research results of two 2K–H reducers with the transmission ratio $i=-144$ and $i=-78,\ 8$ the function diagrams $\Delta\varphi=f(T)$ were developed shown in figure 8.

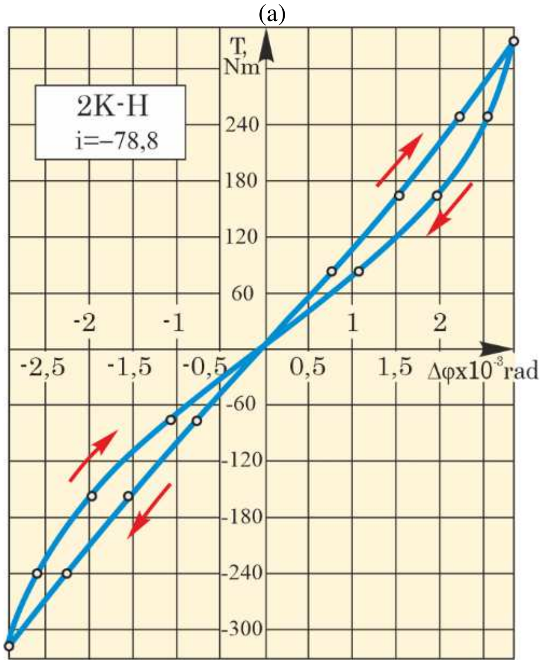
4. CONCLUSION

In conclusion we can mention the fact that precessional transmission provide relative good performance in comparison with another mechanical transmission related to mechanical efficiency and relative weight dependence for tooth gearing depending on shafts torque.

For further experimental research our research team will focus on measuring and determine vibration and sound level that will be compared with classical transmission and standard prescriptions.



(a)



(b)

Fig.8. The torsion angle depending on the torsion moment T for the 2K−H reducer: i=−144 (a) and i=−78.8 (b)

Research is to obtain torsional stiffness of processional gear respectively moment of inertia.

Due to the fact that past research was done for power precessional transmission, the future research will be carried out especially for kinematical precessional transmissions, where will be tested, researched and measured the mechanical efficiency, torsional rigidity, tribological aspect of gearing (teeth contact) when central wheels and satellite is made from different types of material, like Metal-Metal-Metal and Plastic-Metal-Plastic and research regarding vibrational and sound behavior of precessional transmissions.

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Cercetarea experimentală a transmisiilor precesionale cu angrenări dințate convex-concave multipare: achiziții de date, mostre experimentale și rezultate

Rezumat: În lucrare sunt abordate aspecte ale cercetărilor experimentale privind caracteristicile funcționale ale transmisiilor precesionale cinematice și de mică putere cu angrenări convex-concave multipare cu dinți drepți A^{D}_{CX-CV} și în arc de cerc $A^{D,\beta}_{CX-CV}$. Sunt cercetate pierderile energetice în angrenări în funcție de alunecarea relativă cu frecare între flancurile dinților conjugați unitar sau simultan multipar, precum și capacitatea portantă a contactului convex-concav al dinților în funcție de diferența curburilor de flanc, inclusiv consumul specific de material, rigiditatea torsională a angrenării precesionale, momentul de inerție și oprire. Danturile roților dințate sunt fabricate din plastic prin turnare, pe mașini unelte CNC adaptate la sistemul Gleason, cât și prin tehnologii aditive cu imprimare 3D. Danturile roților centrale și ale roții-satelit sunt fabricate din diverse cupluri de materiale plastic-metal-plastic, metal-metal-metal, plastic-plastic-plastic. Sunt descrise caracteristicile funcționale ale transmisiilor, ale standurilor experimentale cu posibilități de achiziție și prelucrare computerizată a datelor. Se prezintă analiza comparativă ale caracteristicilor funcționale ale transmisiilor precesionale în raport cu alte tipuri de transmisii mecanice.

Maxim VACULENCO, PhD Associate Professor, Technical University of Moldova, Design Faculty, Studenților str., 9/8, bl. 6, office 6-403, Chișinău, Republic of Moldova, maxim.vaculenco@dip.utm.md

Ion BODNARIUC, PhD Associate Professor, Technical University of Moldova, Mechanical Engineering and Transports Faculty, Studenților str., 9/8, bl. 6, office 6-421, Chișinău, Republic of Moldova, ion.bodnariuc@bpm.utm.md

Radu CIOBANU, PhD Associate Professor, Technical University of Moldova, Mechanical Engineering and Transports Faculty, str. Studenților 9/8, off. 6-414, Rep. Moldova, city Chișinău, radu.ciobanu@bpm.utm.md

Oleg CIOBANU, PhD Associate Professor, Technical University of Moldova, Mechanical Engineering and Transports Faculty, str. Studenților 9/8, off. 6-414, Rep. Moldova, city Chișinău, oleg.ciobanu@bpm.utm.md

Iulian MALCOCI, PhD Associate Professor, Technical University of Moldova, Mechanical Engineering and Transports Faculty, str. Studenților 9/8, off. 6-412, Rep. Moldova, city Chișinău, iulian.malcoci@bpm.utm.md

Nicolae TRIFAN, PhD Associate Professor, Technical University of Moldova, Mechanical Engineering and Transports Faculty, str. Studenților 9/8, off. 6-108, Rep. Moldova, city Chișinău, nicolae.trifan@bpm.utm.md

Dumitru VENGHER, PhD Student, Assistant professor, College of Technical University of Moldova, Rep. Moldova, city Chișinău, avn. Ștefan cel Mare 168, dumitru.vengher@adm.utm.md