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# NONCIRCULAR-SCREW GEARS

*Mathematical model of synthesis of circular-screw gears with asymmetric function of transmission ratio based on the offered main and additional conditions of synthesis is stated in the article. The advantage of application of transmissions by noncircular gears for struggling against resonance oscillations that allows extending the possibility of their application is shown in this work.*

***Key words:*** *noncircular gears, variable transmission ratio, struggling against resonance oscillations.*

**Itroduction, Purpose and research problems statement.** Creation of reliable and durable transmission gears is the important scientific and practical problem of modern machine-building industry which possible to be solved on the basis of gear transmissions improving by toothing synthesis. One of the ways of gear transmission development by toothing synthesis extending their functional capabilities is designing of gears with a variable transmission ratio (transmissions by noncircular gears).

Experience of such kind of gears implementation created on the base of involute mesh showed the advantage of their using in chain mechanisms and drives of machines for equalization of chain link speeds and elimination of their internal dynamic loads.

In full measure, it is reasonable to apply this method for improving the anti-resonance stiffness of circular-screw gears which have the high load-carrying capacity and widespread in reduction gearboxes of heavy engineering industry. Practice shows that 3 – 5 % of reduction gears failures concerned with any types of vibrations and resonance phenomenon.

However, the development of gearing by synthesis of efficient toothing geometrics with variable transmission ratio providing the assigned transformation law of motion demands the decision of a number of questions such as selection of transfer function ratio, determination of main and additional conditions of transmission by noncircular gears, elaboration of mathematical model of synthesis of efficient geometrics of circular-screw toothing and estimation of their influence upon gearing working capacity etc.

At present time the circular-screw gears which offer the high load-carrying capacity become prevalent for using in reduction gearboxes of heavy engineering industry; development of these gears can be done by means of synthesis of efficient toothing geometrics.

**Analysis of Publication, Materials.** The scientific works of M.L. Novikov, R.V. Fedyakin, V.A. Chesnokov, A.F.Kirichenko, A.V. Pavlenko, V.A. Krasnoshekov, V.N. Sevruk, V.M. Gribanov, V.P. Shishov and others dedicate to the problems of synthesis of circular gears with circular-screw toothing.

N.I. Mercalov, M.A. Skuridin, O.A. Pyj, N.A. Gaevskiy, N.I. Kolchin, F.L. Litvin, R.S. Varsimashvili, N.L. Ututov, D. Gunter, B. Raingard, M. Kanchiti, I. Kisuko and other made a considerable contribution in research of gearing with variable transmission ratio. They laid the foundation for developing of gearing with noncircular gears and considered the examples of their practical use.

The scientific works of B.M. Abramov, E.L. Airapetov, M.D. Genkin, A.I. Petrusevich A.P. Fillipov, V.K. Grinkevich, S.S. Gutyrya, T. Toshima, K. Masan, D. Wallas, A. Seireg, G. Opits and others dedicated to studying of problems of vibroactivity reducing of gearings with circular gears. It is shown in these works that existing various methods of resonance vibration control of reduction gearboxes (designation of supercritical and subcritical shaft rotational speeds; rise of manufacturing accuracy of gearing production and assembling; modification of construction of gears, housings and shafts; application of special covering of reduction gearbox parts; using of dynamic dampeners and so on) lead to rise in price of construction, increase of mass and size, and for many cases these methods are ineffective and unreliable [1; 3; 4].

**Objects and problems.** It is determined by investigations that the solvation of anti-resonance stiffness problems of circular-screw gears is possible by using the variable transfer function which allows extending of noncircular gear application including the resonance vibration control of gears. In this case the function of transmission ratio has to have the asymmetric law of variation.

One of the types of asymmetric function of transmission ratio, which provides assigned transformation law of motion, can be obtained as following:

$$i(\varphi_1) = \frac{r \cdot \left[\xi + \cos(j_1 \varphi_1)\right] + B \cdot \sin(j_1 \varphi_1)}{u \cdot r \cdot \left[\xi + \cos(j_1 \varphi_1)\right] - B \cdot \sin(j_1 \varphi_1)}, \tag{1}$$

which has three main indexes of asymmetric function of transmission ratio: $\zeta$, $j_1$ and $B$ characterize the degree of asymmetry, frequency and magnitude of transmission ratio changing respectively.

In function (1) $i$ and $u$ are the transmission ratio and transmission number of noncircular gear; $r$ is the mean radius of driving gear centrode; $\varphi_1$ is the turning angle of driving noncircular gear; $j_1$ is the coefficient of asymmetric function of transmission ratio which equals the quantity of maximum values of centrode radius of driving noncircular gear.

Fig. 1 represents the transmission ratio $i$ – turning angle of driving gear $\varphi_1$ diagram.

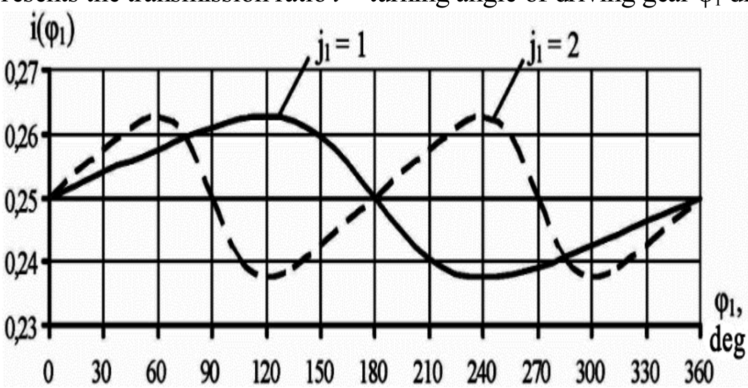


Fig. 1. Charts of asymmetric function of transmission ratio

Mathematical analysis $i(\varphi_1)$ (see Fig. 1) shows the following: function (1) is asymmetrical in regard to it's one-half period under $\xi > 1$ and recommended value $\xi$ should be 2; $j_1$ is whole number. It is recommended to take the quantity of maximum values of centrode radius $j_1 \geq 2$ in order to avoid the mass imbalance.

Making the mathematical analysis of index $B$ which characterizes the variation value of transmission ratio it is possible to determine the dependence of index $B$ from the transmission number of gearing $u$, the center-to-center spacing $a_w$, and the coefficient of nonuniformity of mechanism motion $\delta$:

$$B = \frac{a_w u\sqrt{3}}{\delta\cdot(u+1)}\cdot\left(\sqrt{u^2+2u+\delta^2+1}-u-1\right). \qquad (2)$$

Numerous investigations show that for existing dimension-type reduction gearboxes the values $B$ lie in the range $0 \le B \le 13{,}74$ mm.

Thus, under the assigned parameters u and aw the relation (2) allows to select the efficient value $B$ subject to required coefficient $\delta$ from additional synthesis criterion $B \le B_\delta$ where $B_\delta$ is index of asymmetric function for required $\delta$.

Figure 2 shows the gearing centrodes which radii described by the equations:

for driving gear
$$r_1 = r + \frac{B\sin(j_1\varphi_1)}{2+\cos(j_1\varphi_1)}; \qquad (3)$$

for driven gear
$$r_2 = u\cdot r - \frac{B\sin(j_1\varphi_1)}{2+\cos(j_1\varphi_1)}. \qquad (4)$$

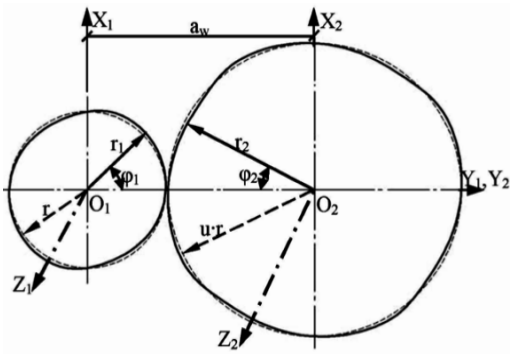


Fig. 2. Gearing centrodes in a fixed coordinate system $X_1Y_1Z_1$ and $X_2Y_2Z_2$ under $j_1$ 2: $= r$ and $u{\cdot}r$ are the mean radii of centrodes of driving and driven gears

In order to estimate the strength factor of gearing we determine the sizes of contact area by the tooth length:

$$C = \frac{\left\{r\left[2+\cos(j_1\varphi_1)\right]+B\sin(j_1\varphi_1)\right\}\left(\varphi_1^*-\varphi_2^*\right)}{\left[2+\cos(j_1\varphi_1)\right]\sin\beta}, \qquad (5)$$

where $\varphi_1^*$ and $\varphi_2^*$ are the angles which count out from straight lines normal to lines which connect the gear's centers.

The analysis of relation (5) shows that contact patch moves around of tooth on the constant distance along it's height and maximum change of contact patch sizes in toothing does not exceed 4,2% from value of other circular gears.

Using the geometrical-kinematic criterions and forced factors it is possible to realize the theoretical estimate of working capacity of synthesized circular-screw gearings with asymmetric function of transmission ratio by means of their comparison with other types of gearings which have a constant transmission ratio.

Dependance of absolute value of motion relative speed of mesh point from turning angle of driving gear describes by equation:

$$V_C = \frac{\sqrt{K_{VC}}}{r^2\left[2+\cos\left(j_1\varphi_1\right)\right]^2 (u+1)^2 \left\{u \cdot r \cdot \left[2+\cos\left(j_1\varphi_1\right)\right] - B\sin\left(j_1\varphi_1\right)\right\}^2}, \quad (6)$$

where $K_{VC}$ is the coefficient of absolute value of motion relative speed of mesh point.

Mathematical analysis shows that traverse speed of mesh point along contact line is the variable quantity and depends on the turning angle of gears and value $B$; changing of value $V_C$ does not exceed 5,7% relative to values for circular gears.

In order to evaluate the wear we generate relations for determination of teeth slip coefficients $\vartheta_1$ and $\vartheta_2$:

for driving gear $$\vartheta_1 = \frac{\sqrt{K_{VCX}^2 + K_{VCY}^2}}{\left\{u \cdot r \cdot \left[2+\cos\left(j_1\varphi_1\right)\right] - B\sin\left(j_1\varphi_1\right)\right\}\sqrt{K_{VK1}}};$$

for driven gear $$\vartheta_2 = \frac{\sqrt{K_{VCX}^2 + K_{VCY}^2}}{\sqrt{K_{VK1}}}, \quad (8)$$

where $K_{VK1}$ is the coefficient characterizing absolute value of motion relative speed of mesh point of teeth along contact line of driving gear; $K_{VCX}$, $K_{VCY}$ are the coefficients of absolute value of motion relative speed of mesh point relatively the axes of coordinates.

Changing of slip coefficients $\vartheta$ on the driving and driven gears is equally and directly proportional to value $B$, and does not exceed 4% relative to values $\vartheta$ for circular gears.

Taking into consideration that rotational speed of driving gear $\omega_1$ and moments of inertia of gear's reduced mass $I_{red.1}$ and $I_{red.2}$ are constant, the equation of motion of machine with noncircular gears [5] becomes as following:

$$T_{mot.} = \left[T_{u.r} + T_{add.}\right] \cdot i\left(\varphi_1\right), \quad (9)$$

where $T_{mot.}$ is the moment of motive force on shaft of driving noncircular gear; $T_{u.r.}$ is the moment from forces of useful resistances on shaft of driven noncircular gear; $T_{add.}$ is the additional moment caused by variability of transmission ratio and determined by relation

$$T_{add.} = I_{red.2}\varepsilon_1 \frac{r \cdot \left[2+\cos\left(j_1\varphi_1\right)\right] + B\sin\left(j_1\varphi_1\right)}{u \cdot r \cdot \left[2+\cos\left(j_1\varphi_1\right)\right] - B\sin\left(j_1\varphi_1\right)} + \frac{I_{red.2}\omega_1^2 B j_1 r\left(1+u\right)\left(1+2\cos\left(j_1\varphi_1\right)\right)}{\left[u \cdot r \cdot \left(2+\cos\left(j_1\varphi_1\right)\right) - B\sin\left(j_1\varphi_1\right)\right]^2} +$$
$$+\frac{1}{2}\frac{dI_{red.2}}{d\varphi_1}\left[\frac{\omega_1\left\{r \cdot \left[2+\cos\left(j_1\varphi_1\right)\right] + B\sin\left(j_1\varphi_1\right)\right\}}{u \cdot r \cdot \left[2+\cos\left(j_1\varphi_1\right)\right] - B\sin\left(j_1\varphi_1\right)}\right]^2, \quad (10)$$

where $\varepsilon_1$, $\varepsilon_2$ are the angular accelerations of rotation of driving and driven gears.

Analysis of results represented on Fig. 3 shows that change of additional moment $T_{add.}$ per one rotation of driving gear does not exceed 5,8% from the value of external loading moment.

Equation for normal force determination in toothing for driving and driven circular-screw gears looks as the following:

$$F_{N1(N2)} = \frac{T_{Z11(Z22)}\left[2+\cos\left(j_1\varphi_1\right)\right]\sqrt{K_{1N(2N)}}}{p\left\{r\left[2+\cos\left(j_1\varphi_1\right)\right] + B\sin\left(j_1\varphi_1\right)\right\}\sqrt{A_1}\sin\lambda_{1(2)}}, \quad (11)$$

where $T_{Z11}$ and $T_{Z22}$ are the total moments acting upon driving and driven gears respectively; $K_{1N}$ and $K_{2N}$ are the coefficient of normal vector scalar for teeth surface, $\lambda_1$ and $\lambda_2$ are the turning angles of tool tips under cutting of driving and driven gears; $p$ is the helix parameter.

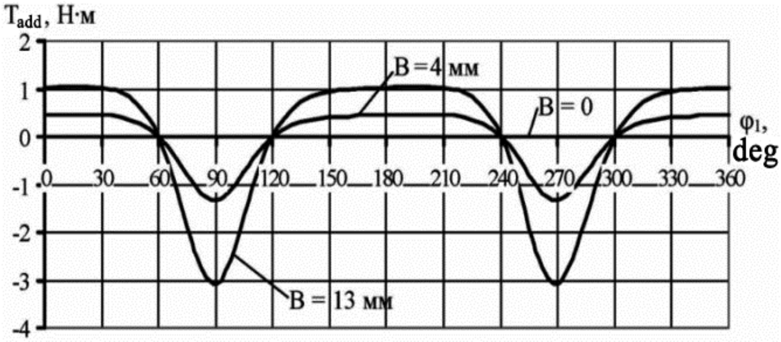


Fig. 3. Additional moment $T_{add.}$ – turning angle $\varphi_1$ relation

Mathematical analysis shows that in gearing with asymmetric function of transmission ratio the normal forces in gears mesh under $T_{mot.}$ = const and $T_{u.r.}$ = const have variable values; at the same time a change of $F_N$ in gears mesh per one revolution of driving gear becomes no more than 3,7% from the magnitude for circular gears.

According the results of conducted comparative analysis for synthesized gearing with circular gear transmissions we conclude that there is a possibility to use the circular-screw gearing with asymmetric function of transmission ratio for application in reduction gearboxes of heavy engineering industry.

Field experience [1; 3; 4] confirms that impulse excitation (teeth concussion at the time of input and output out of mesh) is the main reason for vibration onset provoking in gearing. In reduction gearboxes the first pass of gearing is a most vibro-active zone. Under the coincidence or multiplicity of frequency of natural and forced vibrations the resonance occurs.

Analytical dependence for finding a frequency of natural vibrations of transmission by noncircular gears has the following type:

$$f_c = 3{,}15 \cdot 10^5 \cdot \frac{(1+u)\cdot\sqrt{1+u^2}}{2a_w u}. \qquad (12)$$

Equation for determination of tooth mesh frequency of forced vibrations of transmissions with asymmetric function of transmission ratio under the impulse excitation obtained as follows:

$$f_z = \frac{\omega_1 a_w \left\{ r\cdot\left[2+\cos(j_1\varphi_1)\right] + B\sin(j_1\varphi_1)\right\}}{286{,}5\cdot m\cdot(u+1)\cdot r\cdot\left[2+\cos(j_1\varphi_1)\right]}, \qquad (13)$$

where $\omega_1$ is the rotational speed of driving gear; $m$ is the toothing module.

Taking into account (12) and (13), the resonance (critical) frequency of noncircular gear expressed by relation:

$$\omega_{1\,crit.} = 8{,}8\cdot 10^4 \cdot \frac{m\cdot r\cdot\left[2+\cos(j_1\varphi_1)\right]\cdot(1+u)^2\sqrt{1+u^2}}{a_w^2 u\cdot p\cdot\left\{r\cdot\left[2+\cos(j_1\varphi_1)\right]+B\sin(j_1\varphi_1)\right\}}. \qquad (14)$$

The graph of $\omega_{1crit.}$ – $\varphi_1$ curve is represented on the figure 4.

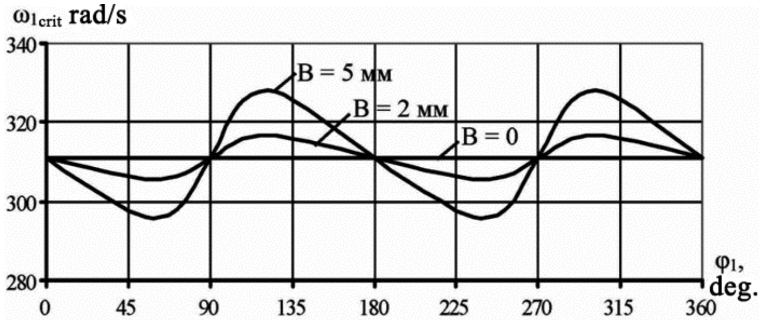


Fig. 4. Resonance rotational speed of driving gear from impulse excitation

The analysis of equations (12) – (14) shows that critical rotational speed of shaft $\omega_{1res}$ with noncircular gear changes for its one revolution, B is the variation value of critical rotational speed (see Fig. 4). It is determined that in transmissions by means of noncircular gears the tooth mesh frequency $f_z$ of forced vibrations is variable value, and does not coincide and multiple its natural frequency $f_{nat}$ of vibrations. Conducted theoretical investigations of resonance vibrations of transmissions by means of noncircular gears came to conclusion that asymmetric law of changing of transmission ratio function prevents from resonance occurrence.

In order to decrease the resonance [3] risk of rotational speed $\omega_1$ of driving noncircular gear it is necessary to follow the criterion:

$$\begin{cases} \omega_1 \geq (1+K)\cdot \omega^o_{1crit} \\ \omega_1 \leq (1-K)\cdot \omega^o_{1crit} \end{cases}, \tag{15}$$

where $K$ is the coefficient which define margin of resonance origin zone, $\omega^o_{1crit.}$ is the resonance rotational speed of driving gear of transmissions by means of noncircular gears, which determined by relation:

$$\omega^o_{1\,crit} = 8{,}8\cdot 10^4 \cdot \frac{m\cdot(1+u)^2\sqrt{1+u^2}}{a_w^2 u\cdot p}. \tag{16}$$

Considering (14) – (16) and using mathematical transforms, the equation for $B_{crit.}$ definition will have the type:

$$B_{crit} = 0{,}82\cdot K\cdot \frac{r\cdot\left[2+\cos(j_1\varphi_1)\right]}{\sin(j_1\varphi_1)}. \tag{17}$$

Therefore, value $B$ has to satisfy conditions of additional synthesis matter: $B \geq B_{crit}$.

Thus, value B of function $i(\varphi_1)$ which characterises variation value of transmission ratio, is chosen from condition $B_{crit} \leq B \leq B_\delta$.

Experimental investigations for determination of coefficient $K$ and comparative tests of gearing by means of noncircular gears with asymmetric function of transmission ratio and transmissions by means of circular gears were conducted. Experimental investigations provided with the purpose of practical approbation of results and conclusions which derived under theoretical study of toothing and contain the following: control of transmission ratio, comparative assessment of vibration resonance of transmissions by noncircular gears with asymmetric function of transmission ratio (under the different meanings of coefficient $K$) and transmissions by circular gears. For these:

– device for gear-milling machine 5К32 which makes the teeth cutting of noncircular gear is engineered and manufactured;

– according results of theoretical calculations, the experimental noncircular gears with circular-screw toothing for double-reduction gearbox are synthesized and produced;

– measurement complex including the stand for inspection of centrode production accuracy and transmission ratio accuracy is prepared;

– complex for vibration measurement of gearbox under rotational speed of driving shaft up to 356 rad/sec is prepared;

– procedure of experimental investigations of resonance vibrations of gearing is developed;

– bench tests of experimental gearings and circular gearings, as well as their comparison characteristics are presented.

Experimental gearings made by steel 40X GOST (State Standard) 4543-71. Heat treatment: pinion gear – refining up to HB 269…302, gearwheel – refining up to HB 235…262. Characteristic of transmissions is represented in Table. 1.

*Table 1*

**Characteristic of experimental transmissions**

<table>
<tr><td rowspan="2">Designation of parameter</td><td colspan="4">1st pass</td><td colspan="4">2nd pass</td></tr>
<tr><td colspan="3">Non circular</td><td>Circular</td><td colspan="3">Non circular</td><td>Circular</td></tr>
<tr><td>Normal module $m_n$, mm</td><td colspan="3">3,0</td><td>3,0</td><td colspan="3">3,0</td><td>3,0</td></tr>
<tr><td>Transmission ratio $u$</td><td colspan="3">2,0</td><td>2,0</td><td colspan="3">2,0</td><td>2,0</td></tr>
<tr><td>Number of teeth:<br>–pinion gear $z_1$<br>– gearwheel $z_2$</td><td colspan="3"><br>21<br>42</td><td><br>21<br>42</td><td colspan="3"><br>32<br>64</td><td><br>32<br>64</td></tr>
<tr><td>Axle base $a_w$, mm</td><td colspan="3">100</td><td>100</td><td colspan="3">150</td><td>150</td></tr>
<tr><td>Coefficient $K$</td><td>0,06</td><td>0,08</td><td>0,15</td><td>0</td><td>0,06</td><td>0,08</td><td>0,15</td><td>0</td></tr>
</table>

On the Fig. 5 the transmission with noncircular gears is obtained.

Estimation of transmission's transmission ratio of experimental gearings and gearbox in whole was conducted under the testing (Fig. 6).

Fig. 5. Double-reduction gearbox with noncircular gears

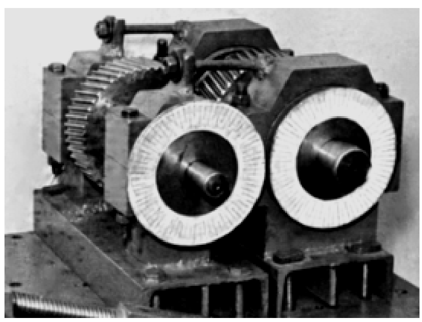

Fig. 6. Stand for transmission ratio inspection

In result of conducted experiment the following conclusion was made up: change of transmission ratio strictly corresponds to relative change of centrode radius of noncircular gears. At the same time, the maximum discrepancy between theoretical and experimental results of transmission ratio values of gearbox amounts to 6%.

In order to define the vibration level in gearbox with noncircular gears, the stand with closed-loop power was engineered; vibration-measuring apparatus ВИ6-6ТН together with self-recording instrument Н 327-3 was used for experiments (Fig. 7). On the housing of research gearbox the following objects were set up: vibration detector ДВ-1СГ for detection of vibration in horizontal plane and ДВ-1-СВ – for vertical plane. Testings were carried out in regime of smooth variations of rotational speed of driving shaft $\omega_1$ from 0 to 356 rad/sec, and vibrations recorded by using self-recording instrument Н 327-3. According results of conducted experiment the following things were determined: with the same character of vibrations, the vibration amplitude in vertical plane is much greater of vibration amplitude in horizontal plane; in gearbox with circular gears under $\omega_1$ = 298 rad/sec, the sharp increase of vibration amplitude $Y$ (Fig. 8, *a*) has been stated; in gearbox with noncircular gear under $K$ = 0,15 on the full scale of rotational speed of driving gear, the increase of vibration amplitude has not been observed (Fig. 8, *d*); under $K$ = 0,08 the maximum magnitude of vibration amplitude comparing with magnitude under $K$ = 0,15, has increased 1,5 times in speed range $\omega_1$ from 272 up to 323 rad/sec (Fig. 8, *c*); under $K$ = 0,06 the maximum magnitude of vibration amplitude comparing with magnitude under $K$ = 0,15, has increased 3,5 times in speed range $\omega_1$ from 281 up to 315 rad/sec, and at the same time the maximum values of amplitudes approached to amplitude values for circular gears (Fig. 8, *b*).

Graphs analysis (Fig. 8) showed that border of resonant zone appearance observes under $K$ = 0,08. It allows giving the recommendations by definition of index $B$ in asymmetric function of transmission ratio.

Realized testing of two-stage gearbox with noncircular gears and common constant transmission ratio for recommended value $K$ = 0,08 under $B$ from 2 up to 7 mm has showed the following: at a full range of rotational speed of driving shaft there was no increase of vibration amplitudes to be observed, which conforms the theoretical background.

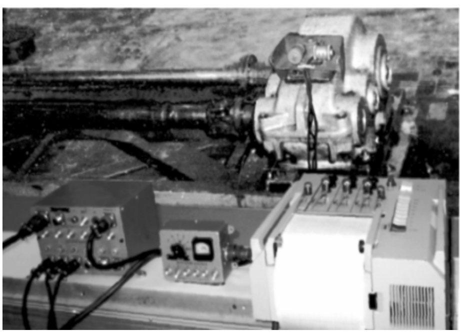

Fig. 7. Stand for measurement of gearing vibrations.

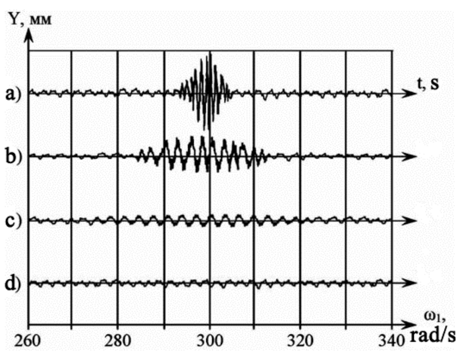


Fig. 8. Oscillogram of vibrations in gearboxes:
a) with circular gears;
b) with noncircular gears under $K$ = 0,06;
c) with noncircular gears under $K$ = 0,08;
d) with noncircular gears under $K$ = 0,15

**Conclusions.** Submitted results of conducted theoretical and experimental researches which let us to come to following conclusions:

1. Mathematical model of synthesis of circular-screw gears with asymmetric function of transmission ratio is developed. The efficient geometrical parameters of toothing (upon proposed supplementary conditions of synthesis) which secure the operating regime of gearing with no resonance effect ($B \geq B_{crit}$) and required coefficient of nonuniformity of motion δ ($B \leq B_{\delta}$) are determined on basis of synthesis task solution.

2. Theoretical analysis of working capacity of synthesised gearings by means of comparing with gearings with constant transmission ratio is carried out.

3. Estimation of resonance vibrations of transmissions with noncircular gears with asymmetric function of transmission ratio from impulse excitation is carried out. Dependances for definition of border of resonant vibrations zone appearance are detected.

4. Experiment-calculated works for the purpose of transmission ratio control and estimation of resonance vibrations of gearings by noncircular gears with circular-screw mesh are carried out. As a result of testing it was determined that in gearbox with noncircular gears with symmetric function of transmission ratio in a whole range of $\omega_1$ for recommended value *K*, the increasing of vibration level has not been observed.

5. One of the ways of circular-screw gears development by synthesis of rational mesh geometrics with asymmetric function of transmission ratio which guarantees the specified low of energy conversion and expanding the functional capabilities of application of transmissions with noncircular gears for resonance vibrations struggling is proposed.

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## НЕКРУГЛІ ГВИНТОВІ ЗУБЧАСТІ КОЛЕСА

В статті описано математичну модель синтезу круглих гвинтових коліс з асиметричною функцією передавального відношення на основі пропонованих основних і додаткових умов синтезу. Переваги застосування передач з некруглими зубчастими колесами для попередження резонансних коливань, дозволяють, як показано в роботі, розширити можливості їх застосування.

***Ключові слова:*** некруглі зубчасті колеса, змінне передавальне відношення, попередження резонансних коливань.

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