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LOADING OF STRUCTURAL ELEMENTS OF LARGE-SIZED ROTATING AGGREGATES DURING LONG-TERM OPERATION

Methods for determining the teeth total skew angle crown engagement considering errors from manufacturing and from mutual arragement of pinion and coronal wheel of an open gearing are implemented. On the basis of experimental data, the possible range of the total skew angle has been determined for different conditions under which large-sized rotating aggregates are operated.

Algorithm and techniques of calculation operated for a long time rotating aggregates during transitive and steady-state regimes of electromechanical drive mechanism operation, non-uniformity of the turn angle of the coronal wheel were considered. Practical recommendations for design improvement of an open gearing and for a system of kinematic parameters adjustment of large-sized rotating aggregates drive mechanism are suggested.

Keywords: *rotating aggregate, gear crown wheel and pinion, drive mechanism, skew angles of teeth, weared out involute profil.*

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НАВАНТАЖЕННЯ КОНСТРУКТИВНИХ ЕЛЕМЕНТІВ ВЕЛИКОГАБАРИТНИХ ОБЕРТОВИХ АГРЕГАТІВ ПІД ЧАС ТРИВАЛОЇ ЕКСПЛУАТАЦІЇ

Реалізовано методику для визначення сумарного кута перекосу зубців вінцевої пари з врахуванням похибок виготовлення і взаємного розташування зубчастих коліс відкритої передачі. На основі експериментальних даних визначено можливий діапазон сумарного кута перекосу за різних умов експлуатації великогабаритних обертових агрегатів. Розроблено алгоритм і методику розрахунку навантаження конструктивних елементів тривало експлуатованих обертових агрегатів під час перехідних і стаціонарних режимів роботи електромеханічного приводного механізму з урахуванням нерівномірності кута повороту вінцевого колеса. Запропоновано практичні рекомендації з удосконалення конструкції відкритої зубчастої передачі і системи регулювання кінематичних параметрів приводу великогабаритних обертових агрегатів.

Ключові слова: *обертовий агрегат, зубчасті вінцеве колесо і шестерня, приводний механізм, кути перекосу зубців, зношений евольвентний профіль.*

Formulation of the problem. Large-dimensional rotating aggregates, such as rotary kilns, tube mills, drying drums, etc. are widely used in the construction, mining, power, metallurgy and chemical industries. The desire to get greater productivity of technological lines has led to increase in the overall dimensions of their casing and the required power of electric motors for its rotation. In such technological lines one or two-motor electro-mechanical drive with open gear (crown pair) whose module is within the range 20 ... 60 mm and the pitch diameter of a crown wheel is up to 12000 mm is used. In this case, installation and repair work become more complicated, and the operating conditions deteriorate: it concerns fastening of the crown wheel to the steel case and the housing of the crown pinion to the basement, roller supports, bandages and other components of the rotating assembly. Insufficient reliability of individual structural elements of the electromechanical system leads to unplanned stops of the technological line, which adversely affects the utilization rate of the equipment and reduces the expected effect from the use of large-sized aggregates.

The practice of electromechanical drive mechanisms long-term operation of large-sized rotating aggregates testifies that one of the least reliable links in this case is the assembly of an open gear pair [1]. Thus, the problem of determining loads in the structural elements of drive mechanisms of large-sized aggregates and increasing their durability due to design, technological and operational measures in the course of their long-termed operation is an urgent task.

Analysis of recent research and publications. The complexities of the technological processes that accompany the long-term operation of rotating aggregates of continuous operation are covered in the works [2, 3]. The authors [4, 5] consider the problem of determining the loads of large-sized electromechanical drive mechanisms elements of rotating aggregates. Advantages of using the finite element method in various studies of the theory of gear mesh are given in [6]. The authors [7] conducted a study of the change in the wheels load capacity of the toothed pair for different variants of their geometric sizes. The modelling of the toothed pair coupling was carried out using the finite element method. As a result of the research, the method of increasing the load capacity due to the value of the module and the number of contacting wheels teeth is suggested. Issues of quasi-static analysis in gear pairs under the action of a torque momentum are investigated in work [8]. In the process of large-sized rotating aggregates operation due to specific conditions of their work there is intensive wearing of the open drive transmission teeth, which causes extra dynamic loads in the engagement [9]. The authors [10] have established that in the two-motor drive mechanism of the drum mills, there is forced oscillations emerge due to the accumulated error of the pitch in the open gear transmission. The paper [11] presents the study results of the time dependences of the normal force in the engagement and the change in velocity. From the obtained graphs, it is seen that the velocity of the wheel varies about a certain mean value.

In the conducted studies [12], it is shown that the non-uniform distribution of the load over the length of the contact lines of the conjugate teeth has the most significant impact on the accuracy of the assembly work, elastic deformation of the structural elements and the butt beating of the tooth crown. It is established that the distribution of the load over the width of the teeth crown of the cylindrical transmission is linear. To determine the structural elements rigidity of the ball mill drive electromechanical system, authors [13] used a two-mass calculation model, calculated analytical parameters determining the rigidity of gears, drive shafts, and elastic couplings of drum ball mills. The studies results of displacements in the cross sections of the roller-supports of large-sized rotary kilns are given [14]. Calculations show that the elastic displacement of the kilns body can be a significant part of its total displacement, which negatively affects the operation of the open gearing. As an alternative to the existing structures of drive mechanisms of rotary units, it is possible to note the patents [15, 16] and the author's certificate [17], where the crown wheel and the pinion are made with two additional support rollers at their butts which contact each other and maintain a constant radial clearance in the engagement of the open gearing wheels.

Unsolved earlier part of the general problem. The conducted analysis of studies and publications showed that during long-term operation of large-sized rotating aggregates there are many problems that are associated with their design and operational features. Preferably, research relates to drive mechanisms with closed gearings. There is much less attention paid to the study of drives with large-modal open gear transmission operation, although such mechanisms are operated under more heavy conditions. A number of new designs of rotating aggregates drive mechanisms are suggested [15-17], but due to the large material and time costs required for modernization, as a rule, their implementation in the production process is not carried out.

Setting objectives. The development of the algorithm and the technique of calculating the loads of structural elements of long-term continued operated large-sized rotating aggregates, considering the errors of manufacturing, installation, parameters of their operation, and electromagnetic processes in the electric motors of the drive mechanism were performed.

Main material and results. The electromechanical part of large-sized rotating aggregates consists of a body with mounted bandages rested on rollers. The torque moment is transmitted from the main electric motor to the main gear unit, the coupling, the pinion gear, the teeth of which are in the engagement with the crown wheel teeth, which is secured rigidly with the help of special rods to the body of the rotating assembly. Under the conditions of installation, the crown wheel consists of two halves. For example, for a rotating unit of the diameter of the body equal to 5 m, the pitch diameter of the crown wheel is 7.74 m, the width of the tooth crown is 0.85 m and weighs is about 50 000 kg. During a long-term operation, the body of the rotating unit and, with it, the crown wheel simultaneously move in the vertical and horizontal planes due to the eccentricity of the fit of the crown wheel and the radial beats, the ellipticity of the near-bandages and support rollers, the non-straightness of the axis of rotation of the body of the rotating unit and its temperature deformations. Mounting and checking of the drive elements occurs during the complete stop of the technological line, and at that time the temperature of the housing of the rotating unit is equal to the ambient temperature. When the rotary unit is started in the technological regime as a result of temperature and force deformations, the mutual arrangement of the gear wheels of the crown pairs significantly changes.

The value of the total skew angle of the crown pair teeth depends on the accuracy of the manufacture and installation of the large module gears and wheels and the deviations of the axis of rotation of the unit from the straight line. The pinion and wheel of the crown pair have no common place, therefore, in the process of the rotating unit operation; their mutual arrangement can vary considerably. The estimation of the constructive and operational factors influence on the gear wheels work of the rotating aggregate crown pairs can be carried out with the help of the average probable value of their total skew angle.

The angle value of open transmission teeth technological skew is determined according the A. I. Petrusevich's formula, which is given in the paper [12]

$$\gamma_T = \sqrt{\Delta\beta_p^2 + \Delta\beta_g^2 + \gamma_\delta^2 \cos^2 \alpha_w + \gamma_n^2 \sin^2 \alpha_w}\,, \tag{1}$$

where $\Delta\beta_p$ and $\Delta\beta_g$ are the deviations from the given direction of the gears teeth angle and wheels inclination respectively;

γ_δ and γ_n are the angles of displacement and non-parallel of the pinion and wheels rotation axes respectively;

α_w are the angle of engagement.

For a rotating unit with a rectilinear axle of the body and an axial beating of the crown equal to the angle ξ_1 of inclination of the open gear wheels is of

$$\gamma' \approx \sin\gamma' = \frac{\xi_1}{d_{w2}}, \tag{2}$$

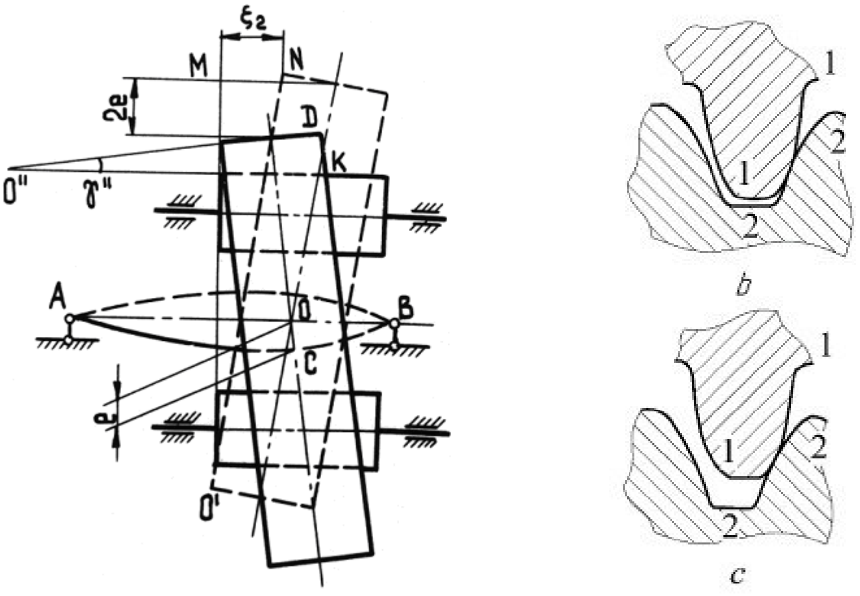
where d_{w2} is the pitch diameter of the crown wheel.

Movement of the crown wheel with the curved axis of the unit with two-motor drive rotation is shown in Fig. 1, where e is the value of the deflection of the unit body rotation axis in the section, which coincides with the position of the median plane of the crown wheel, ξ_2 is the axial displacement of the crown wheel beating which is caused by distortion of the unit body rotation axis.

At this case, the angle of inclination of the open gear wheels axes is determined depending on the value of its deflection in the section where the crown wheel is installed (Figure 1)

$$\gamma'' \approx \frac{e}{L}, \tag{3}$$

where $L = CB$ is the distance from the proximity of the rotating unit body rollers to the median plane of the crown wheel.



a

Figure 1 – Characteristic cases of the crown pair elements mutual arrangement in the case of the rotating aggregate body rotation curved axis

For the convenience of conducting further research, the angles of distortion the crown pair wheels axes are brought to the plane of engagement. Then, the dependencies for determining the variable components of the reduced angles of bias γ'_δ and the non-parallel γ'_n of the crown pair teeth from the axial beating of the crown ξ_1 in the function of rotation angle of the crown wheel ψ_K have the form

$$\gamma'_\delta = \frac{\xi_1}{d_{w2}} \cos\alpha_w \sin\psi_K; \tag{4}$$

$$\gamma'_n = \frac{\xi_1}{d_{w2}} \sin\alpha_w \cos\psi_K. \tag{5}$$

Dependencies for determining the variables of bias reduced angles components γ''_δ and the non-parallelness γ''_n of the crown pair teeth from the value the unit body rotation axis deflection e in the function of rotation angle ψ_κ have the form

$$\gamma''_\delta = \frac{e}{L}\cos\alpha_w \cdot \sin\psi_\kappa; \tag{6}$$

$$\gamma''_n = \frac{e}{L}\sin\alpha_w \cdot \cos\psi_\kappa. \tag{7}$$

Substituting (4) - (7) into (1), it is obtained the formula for determining the total skew angle of the crown pair teeth

$$\gamma_\Sigma = \sqrt{\Delta\beta_p^2 + \Delta\beta_g^2 + \left(\frac{\xi_1^2}{d_{w2}} + \frac{e^2}{L^2}\right)\left(\cos^2\alpha_w \sin^2\psi_\kappa + \sin^2\alpha_w \cos^2\psi_\kappa\right)}. \tag{8}$$

In the course of experimental studies, the axial beating of the crown wheel and the deflection of the rotation axis of the rotating unit body from the straightness were determined. Investigation of the operating conditions of the drive mechanism at different values of the errors of manufacturing and assembling the elements of rotating aggregates have showed that in real conditions of operation the total skew angle of wheels crown pairs teeth may exceed the recommended maximum value more than two or three times, there may be cases of incomplete contact of crown pairs of teeth wheel width. For rotating aggregates of the sizes of 5×185 m, 4.5×170 m, 4×150 m, the value of the maximum total skew angle of the wheel teeth under certain operating conditions lies in the range of $4.0 \cdot 10^{-4}$ to $1.0 \cdot 10^{-3}$ rad.

All this indicates that in the process of rotating aggregates embedding of the crown pair teeth wheels long-term operation is not fully implemented. Due to the specific conditions of their work (open gear transmission, the presence of abrasive media, large non-uniform distribution of load over the length of contact lines, etc.), there is an intense wear of the teeth pinion and wheels (Fig. 2). This causes the unevenness of the rotary motion of the crown pair wheels, resulting in extra dynamic loads in the engagement.

a *b*

Figure 2 – Profiles of the open gear wheels gear during their long-term operation:
a – the pinion and the crown wheel; b – pinion

The dependence of the variable component of the the crown wheel turn angle $\Delta\psi_\kappa$ in the function φ_m/u_2 of the reduced angle of the crown gear rotation during the time of one engagement of the conjugate teeth is approximated by the finite number of Fourier series terms [18]

$$\Delta\psi_{\kappa} = A_0 + \sum_{j=1}^{S_i} E_{ij} \sin\left(jz_2 \frac{\varphi_m}{u_2} + \varepsilon_{ij} \right), \tag{9}$$

where $E_{ij} = A_{ij} + B_{ij}$ is the total coefficient of the Fourier series;

A_0, A_{ij}, B_{ij} are the coefficients of the Fourier series;

u_2 is the gear ratio of the gear wheels of the crown pair;

z_2 is the number of crown wheel teeth;

ε_{ij} is the initial phase j is its harmonics;

S_i is the number of terms of the Fourier series which is necessary to provide the required accuracy of the calculation.

The mutual relation of the crown pairs teeth wheels rotation angles with the worn out profile of the teeth has the form

$$\psi_{\kappa} = \frac{\varphi_m}{u_2} + \Delta\psi_{\kappa}. \tag{10}$$

Given the expression (10), we obtain

$$\psi_{\kappa} = \frac{\varphi_m}{u_2} + A_0 + \sum_{j=1}^{S_i} E_{ij} \sin\left(jz_2 \frac{\varphi_m}{u_2} + \varepsilon_{ij} \right). \tag{11}$$

In this case, the variable transmission ratio of the wheels of the crown pairs of the worn out profile of the teeth has the form

$$\gamma = \left[\frac{1}{u_2} + \frac{1}{u_2} \sum_{j=1}^{S_i} E_{ij} jz_2 \, cos\left(jz_2 \frac{\varphi_m}{u_2} + \varepsilon_{ij} \right) \right]^{-1}. \tag{12}$$

To study the oscillation phenomena due to the kinematic errors of the crown teeth pair, due to the deviation of the teeth shape from the profile, the mechanical part of the rotating aggregates can be represented as a discrete calculation scheme (Fig. 3), where J_i*(i*=1, 2, …, *m)* are the moments of mass inertia of the drive mechanism; C_i*(i*=1, 2, …, *m*-1*)* are the elastic elements rigidity of the drive; v_i*(i*=1, 2, …, *m*-1*)* are the coefficients of linear resistance of the corresponding units. Metal construction of the rotating unit body by direct sampling is replaced by a set of masses with moments of inertia I_j*(j*=1, 2, …, *n*-1*)*; r_j*(j*=1, 2, …, *n*-1*)* and μ_j*(j*=1, 2, …, *n*-1*)* – respectively, the rigidity and coefficients of the linear resistance of the parts of the aggregate body; φ_i*(i*=1, 2, …, *m)* and ψ_i*(j*=1, 2, …, *n)* are the coordinates of movement of the corresponding masses of the drive mechanism and the body of the rotating aggregates.

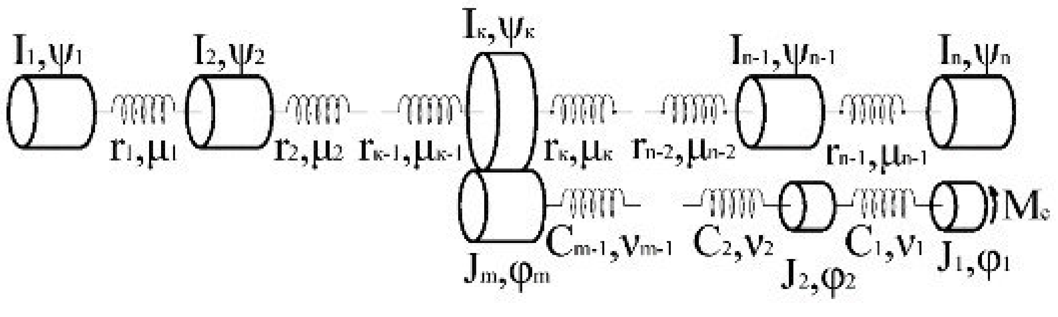


Figure 3 – Calculation scheme of a large-sized rotating aggregates

In the calculation scheme, the gear wheels of the crown pair are directly connected with the masses, which have moments of inertia J_m (for the pinion gear) and I_K (for the crown wheel). Assume that the values of the inertial, rigid and dissipative parameters of the drive elements are determined considering the reduction to the axis pinion gear rotation.

The equation rotating unit masses motion is made on the basis of the Lagrange equations of the second kind. Using expressions of kinetic and potential energies, Relay functions, and generalized forces, there can be obtained differential equations of motion.

For masses of the drive mechanism, these equations have the following form:

$$(J_1 p^2 + \nu_1 p + C_1)\varphi_1 - (\nu_1 p + C_1)\varphi_2 = M_e u_1; \tag{13}$$

$$\begin{gathered}[J_i p^2 + (\nu_i + \nu_{i-1})p + C_i + C_{i-1}]\varphi_i - (\nu_{i-1} p + C_{i-1})\varphi_{i-1} - (\nu_i p + C_i)\varphi_{i+1} = 0; \\ i = 2,3,\dots,m-1.\end{gathered} \tag{14}$$

where M_e is the electromagnetic moment of the motor;

u_1 is the gear ratio from the shaft of the pinion axle motor, which is equal to the gear ratio of the main gear unit;

p is the operator of differentiation with respect to time.

The equation of the open gear pair wheels motion directly is connected with the crown wheel of the rotating unit body section is represented by the dependence:

$$\begin{gathered}\left[\left(J_m + \frac{I_\kappa}{\gamma^2}\right)p^2 + \left(\nu_{m-1} + \frac{\mu_{\kappa-1}}{\gamma^2} + \frac{\mu_\kappa}{\gamma^2}\right)p + C_{m-1} + \frac{r_{\kappa-1}}{\gamma^2} + \frac{r_\kappa}{\gamma^2}\right]\varphi_m - \frac{2I_\kappa}{\gamma^3}\cdot\frac{d\gamma}{d\varphi_m}(p\varphi_m)^2 - \\ -(\nu_{m-1}p + C_{m-1})\varphi_{m-1} - \gamma^{-1}(\mu_{\kappa-1}p + r_{\kappa-1})\psi_{\kappa-1} - \gamma^{-1}(\mu_\kappa p + r_\kappa)\psi_{\kappa+1} = -\gamma^{-1}M_{OK},\end{gathered} \tag{15}$$

where M_{OK} is the moment of the forces resistant to motion, which is applied to the part of the rotating aggregates body connected with the crown wheel.

The equations of the body lumped masses motion of the rotating unit have the form:

$$(I_1 p^2 + \mu_1 p + r_1)\psi_1 - (\mu_1 p + r_1)\psi_2 = -M_{01}; \tag{16}$$

$$\begin{gathered}[I_i p^2 + (\mu_{i-1} + \mu_i)p + r_{i-1} + r_2]\psi_i - (\mu_{i-1}p + r_{i-1})\psi_{i-1} - \\ -(\mu_i p + r_i)\psi_{i+1} = M_{oi}; i = 2,3,\dots,\kappa-1,\kappa+1,\dots,n-1;\end{gathered} \tag{17}$$

$$(I_n p^2 + \mu_{n-1}p + r_{n-1})\psi_n - (\mu_{n-1}p + r_{n-1})\psi_{n-1} = -M_{on}. \tag{18}$$

where M_{oi} ($i = 1,2,\dots,\kappa-1,\kappa+1,\kappa+2,\dots,n$) are moments of motion resistance.

If the lateral gap in the engagement of the open gear is partially disclosed, the equation of motion of rigidity coupled masses with pinion and with a crown wheel takes the form

$$(J_m p^2 + \nu_{m-1}p + C_{m-1})\varphi_m - (\nu_{m-1}p + C_{m-1})\varphi_{m-1} = 0; \tag{19}$$

$$\begin{gathered}[I_\kappa p^2 + (\mu_{\kappa-1} + \mu_\kappa)p + r_{\kappa-1} + r_\kappa]\psi_\kappa - (\mu_{\kappa-1}p + r_{\kappa-1})\psi_{\kappa-1} - \\ -(\mu_\kappa p + r_\kappa)\psi_{\kappa+1} = -M_{OK}.\end{gathered} \tag{20}$$

It should be noted that considering non-rectlinearity of the body rotation axis and the specificity of the processed raw material movement along the axis of the rotating aggregate, the total moment of the body forces resistant to the motion is shown with the expression

$$M_{0\Sigma}^{*}=M_{0\Sigma}\cdot(1+\varepsilon\, sin\,\psi_{\kappa}),\tag{21}$$

where ε is the coefficient of resistance moment amplitude variation to the rotating aggregates body motion.

The dynamical properties of the asynchronous motor are considered by means of simultaneous integration of the rotating unit mechanical part motion equations with the equations of the motor electromagnetic state [19].

The mathematical model of an electric machine is described by the following equations

$$pi_s=a_s(u_s+\Omega_s\psi_s-r_si_s+b_s(\Omega_R\psi_R-r_R\cdot i_R);\tag{22}$$

$$pi_R=b_R(u_s-\Omega_s\psi_s-r_s\cdot i_s)+a_R(\Omega_R\psi_R-r_R\cdot i_R).\tag{23}$$

In these equations, the subscripts s and R denote that the corresponding parameters are related to the stator or rotor windings, respectively i_s and i_R are the matrix-columns of projections of currents on to the coordinate axes x, y; u_S is the matrix-column of voltage source; ψ_s, ψ_R are the matrix columns of complete flow connections; Ω_s, Ω_R are the angular velocity matrices; a_s, a_R, b_s, b_R are the constant coefficients; r_s, r_R are the stator and rotor windings resistances, respectively.

The electromagnetic moment of the motor is found by the formula

$$M_e=\frac{3}{2}p_0(i_{sy}\cdot i_{Rx}-i_{sx}\cdot i_{Ry})/\tau_i,\tag{24}$$

where p_0 is the number of pairs of magnetic poles;

τ_i is the inverse working inductance of the motor.

Depending on the location of the two neighbour teeth of the crown wheels, the total angle of the crown pinion turn, relative to the axis the crown wheel rotation , considering the choice of the lateral gap in the engagement of the open transmission χ is determined by the expression

$$\psi_{\kappa}=\frac{\varphi_m}{\gamma}+\Delta\psi_{\kappa}+(q-1)\cdot\chi/\gamma,\tag{25}$$

where χ is the angle where the teeth of the crown pinion is turned to chose the lateral gap between the teeth of crown pair.

The torque M_m which is transmitted by the crown pinion determined by the formula

$$M_m=C_{m-1}(\varphi_{m-1}-\varphi_m)+\nu_{m-1}p(\varphi_{m-1}-\varphi_m).\tag{26}$$

Determining the loads that emerge in the elements of the mechanical system of the rotating aggregate under different operating conditions, there can be carried out the following algorithm:

Variant 1: in the case where the lateral gap between teeth of a crown gear pairs is completely chosen $q = 1$; $M_m > 0$.

The mathematical model of the drive system of rotating aggregates is formed from the equations (13) - (15), (16) - (18), (22), (23) considering the relations (12), (24), (25).

Variant 2: in the case where the lateral gap between the crown pairs is maximal $q = 2$ $M_m < 0$.

The mathematical model of the drive system of rotating aggregates is formed from the equations (13) - (18), (22) - (23) considering the relations (12), (24), (25).

Variant 3: in the case where the lateral gap between teeth of a crown pairs is partially chosen $1 < q < 2$; $M_m = 0$.

There is constructed a mathematical model from equations (13), (14), (16) - (23) considering the relations (12), (24), (25).

According to the variants number (1, 2 or 3), the system of the corresponding differential equations is determined. Their numerical integration is performed by the Runge-Kutta method. The program provides an opportunity to identify efforts of meshed wheels crown pairs, moments in elastic links, dynamic factors and output calculation results with a given time step. Using modern PC allows us to optimize the parameters of the drive mechanism according the criterion of minimum dynamic loads.

As an example, the drive system elements load of six-support rotating kiln Ø 4.0×150 m with roller support on roller bearings and welded bandages is considered. Technological frequency of the kiln body rotation, $n_b = 0.55 \ldots 1.4$ rpm. As an electric motor of the main drive, the motor AKZ-12-35-6 with a power of 320 kW and speed of $n_m = 980$ rpm are chosen. The main gearbox A 600×900×1400 with gear ratio of $u_1 = 87{,}82$. Parameters of open crown pair gear: module $m = 45$ mm; number of teeth $z_1 = 19$, $z_2 = 150$.

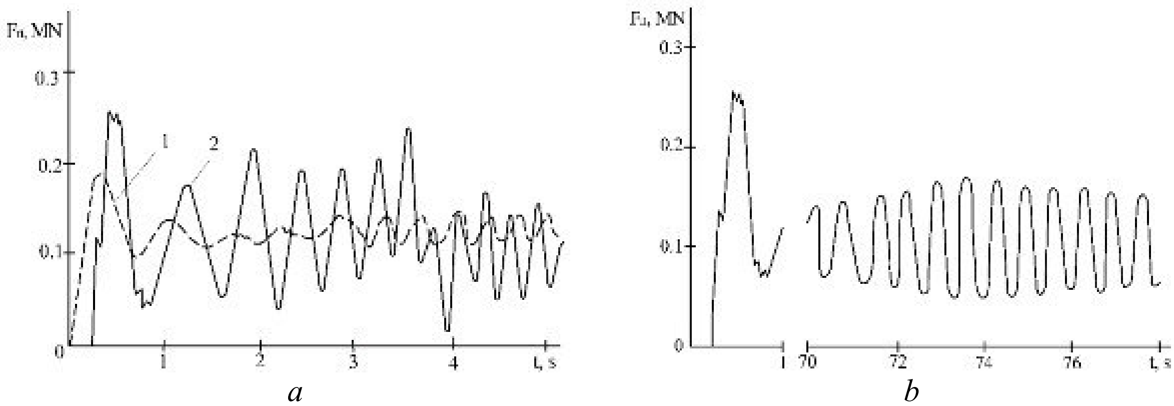


Figure 4 – Graphical dependences of the normal force change in the engagement of the open gear crown pair transmission of the drive mechanism of the rotating kiln Ø 4.0×150 m

Fig. 4, *a* and Fig. 4, b show the dependences of the normal force change in the engagement of the open gear crown pair transmission of the rotation kiln drive mechanism Ø 4.0×150 m during the transition and stationary modes of operation. Curve 1 in Fig. 4 a, represent the dependence of the change in the normal force in the engagement of an open gear crown pair with an ideal involute profile, the teeth ($\chi = E = 0$), the curve 2 in Fig. 4, *a* – provided that the worn-out involute profile of the teeth is when $\chi = 0.004$ rad and $E = 0.0005$ rad. Comparison of the graphic dependencies 1 and 2 in Fig. 4 *a* shows that the increase of the angle χ and the uneven angles of wheels rotation of the crown pairs for the considered case lead to an increase in the normal force in their toothed engagement by 27%.

Fig. 4, b shows the dependence of the normal force change on the engagement of an open gear crown pair during stationary mode under condition of a worn-out involute profile of the teeth when $\chi = 0.004$ rad and $E = 0.0005$ rad. From Fig 4, b, it can be seen that in the stationary mode of the rotating kiln at lower frequencies of the rotation of its housing, which is required in accordance with the technological regime of clinker annealing, resonant states may occur in the drive elements. It is due to the fact that the range of the rotational speed of the kiln body according to the technological regime of the clinker annealing lies in the limits of 0.55…1.4 rpm, which corresponds to the range of the crown wheel pair frequency which lies within (1.3…3.5) Hz. At the same time, the frequency teeth of the crown pairs of wheels can coincide with one of the inner frequencies of the rotating aggregate mechanical system.

Conclusions. 1. Analysis of structures and operational conditions for traditional drive mechanisms of large-sized rotating aggregates have indicated that the total skew angle open gearing teeth can reach the value of $4{,}0 \cdot 10^{-4}$ rad, sometimes up to $1{,}0 \cdot 10^{-3}$ rad. During long-term operation, embedding of crown gearing wheels teeth is not complete. This causes great non-uniformity of load distribution along lines of contact, intensive wheel and pinion teeth wearing out, and extra dynamic loads of the structural elements rotating aggregate.

2. Algorithm and technique of structural elements calculation of operated for a long time large-sized rotating aggregates are suggested. According to this technique, the determination of loading in mesh of an open gearing and torque moments in elastic links is conducted considering the mutual influence of mechanical and electromagnetic oscillation phenomena. The kinematic parameters of open gearing are presented in terms of non-linear dependences considering the errors from manufacturing and the parameters of wearing out.

3. The results of investigations, which have been carried out by means the numerical modelling method and by means of experiments, have indicated that during operation of large-sized rotating aggregates resonance states can emerge in elements of their drive mechanism at the expense of the open gearing teeth re-conjugation frequency coincidence with one of the lower natural frequencies of the mechanical system. It is expedient to use of transducers for monitoring the vibroparameters of the drive mechanism, the transducers signalize for correcting into the aggregate body rotary speed smooth adjustment system. Carrying out the adjustment of the kinematic parameters of the driving mechanism in due time makes it impossible to operate large-sized rotating aggregates in resonance regime.

4. The application of the drive mechanism design of large-sized rotating aggregates with mobile «floating» pinion of an open gearing is in prospect. In such a drive mechanism, the pinion and the crown wheel are made with two additional rollers at their butts. The additional rollers with the help of elastic elements are pressed against each others. The application of such drive mechanism design enables to maintain more constant mutual position of the wheels of an open gearing during its long-term operation, including such parameters as interaxial distance, lateral and radial gaps.

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