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A.O. Kychma, Ye.V. Kharchenko
Lviv Polytechnic National University

DETERMINATION OF VIBROPARAMETERS IN ELECTROMECHANICAL DRIVES WITH LARGE-SIZED OPEN GEARS

Запропонована математична модель для визначення перехідних режимів роботи обертових печей, сушильних барабанів і млинів з урахуванням електромагнітних процесів у асинхронному двигуні, механічних коливань елементів приводного механізму і довгомірної металоконструкції корпуса, а також зношення зубців відкритої зубчатої передачі. Наведено результати числових розрахунків.

A mathematical model of determining the transient regimes of operation of rotary furnaces, heating drums, and mills with taking into account electromagnetic processes in asinchronous motor, mechanical oscillations of elements of drive mechanism, and long-sized metal bodies, as well as wearing of the teeth of an open gearing is suggested. Results of numerical calculation are presented.

Introduction. Electromechanical drives find their wide application in technological machines (heating drums, mills, rotary furnaces, and other analogues), which are complex systems with high vibroacoustic activity. In the general case, they can be considered as oscillation systems with accidental oscillations which are caused by impulsive force between two solids, as well as by periodical vibration which are caused by disbalance of rotating machine parts, ovality of shafts journals, variable rigidity, and form of involute profile of teeth of large-sized open gears and reduction gears. The control and removal of high vibration of driving mechanisms of such aggregates is a necessary condition of their effective operation, this facilitates the work of service personnel.

Practice of operating the driving mechanisms of rotary furnaces, mills, and heating drums shows that the assembly of open gearing of drive mechanisms is one of the least reliable links of such drives [1]. Problems of longevity of large-sized gearings of drive mechanisms of rotary furnaces, mills, and heating drums are considered in the works [2, 3]. In the course of operation of large-sized rotary furnaces, mills, and heating drums, the teeth of the open gears of the drive intensively wear because of specific condition of operation, which causes extra dynamic load in their meshing [4–6].

The main causes of excess vibration of aggregates are errors of manufacturing, inaccuracies of assembly of structural components, as well as intensive abrasive wearing of open gearings. Under a certain operating speed, the frequency of oscillations of forces of interaction between individual parts of the drive mechanism may coincide with the natural oscillation frequency of a machine of the considered type. The harmful influence of resonance phenomena which emerge in such case is described in the work [7].

Experiments that we have done on operating machinery of cement and chemical factories of Ukraine confirm that vibrodisplacement of elements of driving mechanisms reaches considerable value, and the level of noise near kiln rotary furnaces and drying drums reaches a value of 80 - 95 dB.

Thus, the problem of control of vibroparameters of electromechanical drives with large-sized open gearings and the development of measures for their reduction is an urgent problem.

The aim of this paper is creation of a mathematical model for investigation of dynamic processes in an electromagnetical drive with taking into account electromagnetic processes in asynchronous motor mechanical oscillations of elements of the drive mechanism and long-sized metallic structure of the body as well as wearing of teeth of the long-sized open gearing (crown gearing).

Theoretical background and method. Calculation scheme of the electromechanical drive system is represented in Fig. l, where L_1, L_2 …, L_{m-1} are the moments of inertia of parts of driving mechanisms; I_1, I_2,…, I_{k-1}, I_{k+1},…, I_n are the moments of inertia of the body; L_m, I_k are the moments of inertia of wheels of a large-sized open gearing; l_1, l_2 …, l_{n-1}- are the lengths of segments with distributed masses. Differential equations of motion for masses of the driving mechanism are the following:

$$\left(L_1 p^2 + u_1 p + C_1\right) j_1 - \left(u_1 p + C_1\right) j_2 = M_e U_1, \tag{1}$$

$$\left[L_j p^2 \left(u_{j-1} + u_j\right) p + C_{j-1} + C_j\right] j_j - \left(u_{j-1} p + C_{j-1}\right) j_{j-1} - \left(u_j p + C_j\right) j_{j+1} = 0,$$
$$\left(j = 2,3,...,m-1\right), \tag{2}$$

where p is the operator of differentiation with respect to time; M_e is the electromagnetic moment of the motor; U_1 is the gear ratio from the motor shaft to the pinion axis; j_1, j_2 …, j_{m-1} are the coordinates of motion; C_1, C_2 …, C_{m-1} are the equivalent torsion rigidity of elastic elements; u_1, u_2 …, u_{n-1} are the coefficients of linear resistance of the corresponding links. Values of inertial, rigid, and dissipative parameters of elements of the drive were determined with taking into account their reduction to the pinion axis.

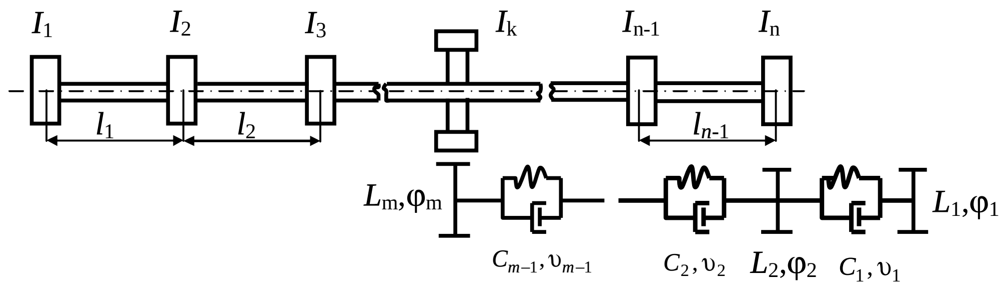

Fig. 1. Calculation scheme of the drive system of technological machines

The interrelation of motion coordinates γ_k of a crown wheel, is a periodical function of motion coordinates φ_m of gear pinion with taking into account the errors of manufacture and the value of wear of the open gear teeth [8].

$$g_k = j_m / U_2 + A_0 + \sum_{j=1}^{S_i} E_{ij} \sin(j z_2 j_m / U_2 + e_{ij}) + (\Delta - 1) d, \tag{3}$$

where U_2 is the gear ratio of the open crown gearing: A_0, E_{ij} are coefficients of Fourier series; s_i is the number of the series terms; z_2 is the number of the teeth of the crown wheel; e_{ij} is the elementary phase of the j-th harmonic; Δ is the index which indicates mutual disposition of gears; d is the value of the angle gap of the gearing.

The equations of torsional oscillations of the body are of the form:

$$\left(a_j^2 d^2 g_j / dx_j^2\right) + \left(m_j a_j d^3 g_j / dx_j^2 dt\right) = d^2 g_j / dt^2, \quad (j = 1, 2, ..., n-1), \tag{4}$$

where g_j is the section turning angle of the j - th segment of the body; $a_j = \sqrt{G / r_i}$ is the speed of elastic wave propagation (G and r_i – are the shear modulus and the generalized density of the material; m_j is the coefficient of inner friction in the material; x_j is the longitudinal coordinate; t is the time.

The boundary conditions of Equation (4) for the body's cross sections which touch rigid masses and which are not connected with the drive mechanism are determined according to the following expressions:

$$I_1 \frac{\partial^2_{g1}(0,t)}{\partial t^2} - GI_{p1}\left(\frac{\partial_{g1}}{\partial x_1} + m_1 \frac{\partial^2_{g1}}{\partial x_1 \partial t}\right)\Bigg|_{x_{1=0}} = -M_{c1} sign\left[\frac{\partial_{g1}(0,t)}{\partial t}\right]; \tag{5}$$

$$I_i \frac{\partial^2_{g_1}(0,t)}{\partial t^2} + GI_{p,i-1}\left(\frac{\partial_{g_{i-1}}}{\partial x_{i-1}} + m_{i-1} \frac{\partial^2_{g_{i-1}}}{\partial x_{i-1} \partial t}\right)\Bigg|_{x_{i-1}=l_{i-1}} -$$

$$-GI_{pi}\left(\frac{\partial_{g_i}}{\partial x_i} + m_i \frac{\partial^2_{g_i}}{\partial x_i \partial t}\right)\Bigg|_{x_i=0} = -M_{ci} sign\left[\frac{\partial_{g_i}(0,t)}{\partial t}\right], \tag{6}$$

$$g_{i-1}(l_{i-1}, t) = g_i(0,t) \quad (i = 2, 3, ..., k-1, k+1, k+2, ..., n-1);$$

$$I_n \frac{\partial^2_{g_{n-1}}(l_{n-1},t)}{\partial t^2} + GI_{p,n-1}\left(\frac{\partial_{g_{n-1}}}{\partial x_{n-1}} + m_{n-1} \frac{\partial^2_{g_{n-1}}}{\partial x_{n-1} \partial t}\right)\Bigg|_{x_{n-1}=l_{n-1}} =$$

$$= -M_{cn} sign\left[\frac{\partial_{g_{n-1}}(l_{n-1},t)}{\partial t}\right], \tag{7}$$

where I_{p1}, $I_{p2}, \ldots,$ $I_{p,n-1}$ – are the polar moments of inertia of the cross-section of the body; M_{c1}, $M_{c2}, \ldots,$ $M_{c,k-1}$, $M_{c,k+1}$, $M_{c,k+2}, \ldots,$ M_{cn} - are the antitorque moments.

With taking into account the bending deformations of the axis of rotation of the body, the errors of manufacturing and assembly of its elements, the antitorque moment is represented by the following expressions:

$$M_{cj} = M^*_{cj}(1 + x \cdot \sin g_k), \qquad (j = 1, 2, ..., k-1, k+1, k+2, ..., n),$$

where M^*_{cj} – is the moment of forces of resistance to mass rotation of the body without taking into account the errors of manufacturing and assembly, x – is the coefficient of deviation of the amplitude of the antitorque moment.

The boundary conditions for the cross-section which coincides with the place of the crown gear are

$$\left(I_k + a^2 L_m\right)\frac{\partial^2_{g_k}(0,t)}{\partial t^2} + 2a\, L_m \frac{\partial a}{\partial_{g_k}(0,t)}\left[\frac{\partial_{g_k}(0,t)}{\partial t}\right]^2 +$$

$$+GI_{p,k-1}\left(\frac{\partial_{g_{k-1}}}{\partial x_{k-1}} + m_{k-1} \frac{\partial^2_{g_{k-1}}}{\partial x_{k-1} \partial t}\right)\Bigg|_{x_{k-1}=l_{k-1}} -$$

$$-GI_{p,k}\left(\frac{\partial_{g_k}}{\partial x_k} + m_k \frac{\partial^2_{g_k}}{\partial x_k \partial t}\right)\Bigg|_{x_{k=0}} = -M_{ck} sign \frac{\partial_{g_k}(0,t)}{\partial t^2}; \tag{8}$$

$$g_{k-1}(l_{k-1}, t) = g_k(0,t),$$

where M_{ck} – is the antitorque moment in the crown wheel, a – is the gear ratio of the crown gearing.

The value of a is determined according to the expression

$$a = \left[1/U_2 + 1/U_2 \sum_{j=1}^{S_i} E_{ij} \cos(jz_2 j_m / U_2 + e_{ij}) \right]^{-1} . \qquad (9)$$

In the case when the angle gap in the crown gearing is partially open, the equation of motion of the pinion is the following:

$$L_m \frac{d^2 j_m}{dt^2} + c_{m-1}\left(j_m - j_{m-1}\right) + u_{m-1}\left(\frac{d j_m}{dt} - \frac{d j_{m-1}}{dt} \right) = 0 . \qquad (10)$$

With this, the boundary conditions for equation (10) take the form

$$I_k \frac{\partial^2_{g_k}(0,t)}{\partial t^2} + GI_{p,k-1}\left(\frac{\partial_{g_{k-1}}}{\partial x_{k-1}} + m_{k-1} \frac{\partial^2_{g_{k-1}}}{\partial x_{k-1}\partial t} \right)\Bigg|_{x_{k-1}=l_{k-1}} -$$

$$-GI_{p,k}\left(\frac{\partial_{g_k}}{\partial x_k} + m_k \frac{\partial^2_{g_k}}{\partial x_k \partial t} \right)\Bigg|_{x_{k=0}} = -M_{ck} sign \frac{\partial_{g_k}(0,t)}{\partial t^2}$$

$$g_{k-1}\left(l_{k-1},t\right) = g_k\left(0,t\right) . \qquad (11)$$

The dynamic properties of the electric motor are taken into account by means of joint integration of motion equations with the equations of electromagnetic state [9]. This mathematical model of electrical machine, which is represented in normal Cauchy form and facilitates the solving of this problem. The equations are the following:

$$pi_s = a_s\left(u_s + e_s - r_s i_s\right) + b_s\left(e_r - r_R i_R\right); \qquad (12)$$

$$pi_R = b_R\left(u_s - e_s - r_s i_s\right) + a_s\left(e_r - r_R i_R\right). \qquad (13)$$

In these equations, the indexes s and R indicate the belandind of the parameters to the winding of stater and rotor respectiveli; i_s and i_R – are the colum-matrices of the projections of the currents into x and y coordinate axes.

$$i_s = col\left(i_{sx}, i_{sy}\right); \; i_R = col\left(i_{Rx}, i_{Ry}\right);$$

e_s and e_R – are the colum-matrices of rotative forces

$$e_s = \begin{array}{|c|c|} \hline & w_0 \\ \hline -w_0 & \\ \hline \end{array} \; y_s ; \qquad e_R = \begin{array}{|c|c|} \hline & w_0 - w \\ \hline w - w_0 & \\ \hline \end{array} \; y_R ;$$

where w_0 – is the cyclic frequency of the power supply, w – is the angular velocity of the motor's rotor rotation, y_s, y_R – are the column matrices of full magneticflux linkage; a_s, a_R, b_s, b_R – are the constant coefficients

$$a_s = a_s\left(1 - \frac{a_s}{a_m + a_s + a_R}\right); \qquad a_R = a_R\left(1 - \frac{a_R}{a_m + a_s + a_R}\right);$$

$$b_s = b_s = -1 - \frac{a_s a_R}{a_m + a_s + a_R} .$$

Here a_s, a_R, and a_m – are the quantities which are inverse to the leakage inductance of windings and inverse to the working inductance of the machine; r_s, r_R – are the resistances of the windings, U_s - is the column-matrix of the current voltage

$$U_s = coL\left(U_m, \mathrm{O}\right), \qquad (14)$$

where U_m – is the amplitude of the power-supply voltage.

The column-matrix y_s and y_R are determined according to the expressions:

$$y_s = \left(\frac{1}{a_s} + \frac{1}{a_m}\right) i_s + \frac{1}{a_m} i_R ; \quad y_s = \left(\frac{1}{a_R} + \frac{1}{a_m}\right) i_R + \frac{1}{a_m} i_s .$$

An electromagnetic moment is found by the formula

$$M_e = 3 p_0 \left(i_{sy} i_{rx} - i_{sx} i_{ry} \right) / 2 a_m , \tag{15}$$

where p_0 is the number of couples of magnetic poles; i_s and i_r are the column-matrixes of projection of currents on axes X and Y, a_m is the quantity reverse to working inductance of the motor.

The initial conditions for calculating the process of start are set in the form

$$i_{sx}(0) = 0 ; \quad i_{sy}(0) = 0 ; \qquad i_{Rx}(0) = 0 ; \quad i_{Ry}(0) = 0 ; \tag{16}$$

$$j_i(0) = 0 ; \quad p j_i(0) = 0 ; \qquad \left(i = 1, 2 ..., m \right) ; \tag{17}$$

$$g_i(0) = 0 ; \quad p g_i(0) = 0 ; \qquad \left(i = 1, 2 ..., n \right) . \tag{18}$$

For illustration of the aforesaid technique, let us make a calculation of non-stationary processes in the drive of a kiln rotary furnace whose body is 4.5 m in diameter and whose length is 170 m. This calculation is reduced to solution of a cauchy problem for a system of non-linear differential equations. High order of this system, as well as the change of its structure in time causes the necessity to apply numerical methods.

The determination of loads which emerge in elements of electromechanical system of rotary furnaces in transient processes is conducted according to the following algorithm. Write the expression (3) in the form

$$g_k = l + (\Delta - 1) d , \tag{19}$$

where $l = j_m / U_2 + A_0 + \sum_{j=1}^{S_i} E_{ij} \sin(j z_2 j_m / U_2 + e_{ij})$.

The parameter Δ corresponds to different interlocation of gears of the crown gearing: $\Delta = 1$ – when the gap between teeth of the crown gearing is fully closed, $\Delta = 2$ – when the gap between teeth the crown gearing is fully open, $\Delta = 3$ – when teeth of the gears of the crown gearing are in intermediate position. The determination of parameter Δ is carried out in each step of integration depending on the ratio of the coordinates j_m and g_k and on the value of the torque moment M_F, which is transmited by the crown gearing.

In the case when $g_k = l \quad M_F > 0$, the gap in meshing of the crown gearing is fully closed $\left(\Delta = 1\right)$. The motion of the drive system is described by the equations (1), (2), (4-8), (12), (13) with taking into account the relations (3), (9), (15).

If the condition

$$g_k = l + d ; \quad M_F < 0$$

is satisfied, the gap is fully open $\left(\Delta = 2\right)$, then the mathematical model must be formed on the basis of the equations (1), (2), (12), (13) with taking into account the relation (15).

If the condition

$$l < g_k < l + d ; \quad M_F = 0$$

is satisfied, partical opening of the gap takes place in the meshing $\left(\Delta = 3\right)$. In this case, the motion of the drive mechanism of the furnace is described by the expressions (1), (2), (10) - (13) with taking into account the relation (15).

Notice that the start of the motion proceedes, mainly, under partical opening of the gap $\left(\Delta = 3\right)$.

The considered calculation model makes it possible to investigate dynamic phenomena under non-stationary and stationary operation conditions in electromechanical drives and allows us to take into consideration the influence of the value of the angle gap in maching of the open gearing, variation of the, value of the gear ratio of open large-sized gearing due to the teeth wear, elastic-dissipative properties of links, as well as that of electromagnetic processes in electric motors.

Discussion and results. As an example, some results of determination and ways of decreasing of vibroacoustics parameters of kiln rotary furnaces with AK 114-6M electric motors whose power is 320 kW and with Ax600x900x1400 main reduction gear have been considered. In our case, the body of the furnace rotates on 7 carrying rollers.

From the graph in Fig. 2, one can see that in the drive system the intensive oscillation of electromagnetic moment takes place in the start period with a frequency close to the frequency of oscillation of the supply voltage. The maximum value of electromagnetic moment exceeds nominal moment 2…3.2 times depending on the change of additional resistance of the rotor circuit. Thus, rational selection of additional resistances at rheostat starting of furnaces contributes to fluency of load of the elements of the drive, considerable decrease of vibroacoustic parameters under transient conditions. The use of the fluent thyristor drive shows better results.

For the majority of the existing designs of large-sized kiln rotary furnaces, the lower frequency of natural oscillation is less than the frequency of gears of the open crown pair, that indicates the possibility of resonance occurance.

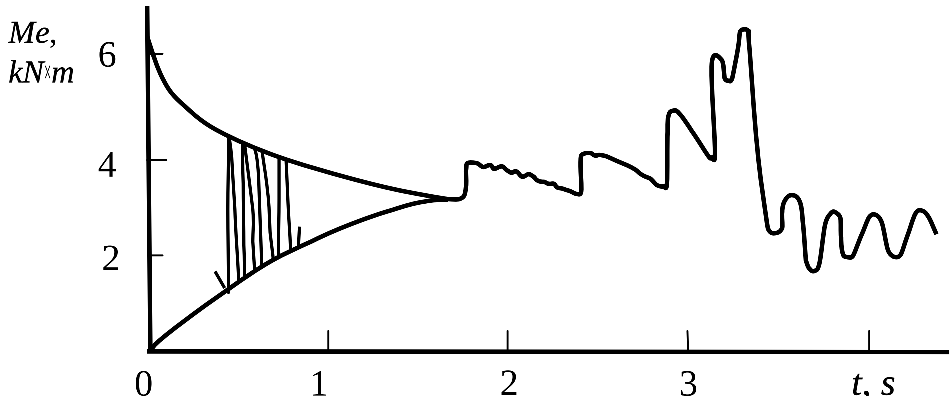

Fig. 2. Dependence of electromagnetic moment in motor

At practical well-founded values of torsion rigidity of shafting which feeds energy to the pinion gear (Fig. 3), resonance conditions of the system due to coincidence of natural frequency of oscillation with the frequency of force change in meshing of the crown gearing are possible. Resonance phenomena which occur during operation of the drive system can increase the amplitude of the oscillation of torque moment in the crown gearing in 1.4 … 1.9 times.

In the non-resonance zone, the intensity of oscillation is essentially influenced by the value of the amplitude of deflection of turning angle of the crown wheel and by the antitorque moment of the furnace. The effective method of decreasing harmful influence of oscillation phenomena is the correct selection of reasonable elastic-dissipative and inertia parameters, calculation and design of active vibro-isolating systems; for example, the set for damp rotary oscillation [10]. The purpose of this invention is to increase the effectiveness of damp of oscillation of shaft vibration. Good results can be obtained resulting from reasonable modifications of structures of the existing driving mechanisms of kiln rotary furnaces and drying drums [11].

In the process of operation of kiln rotary furnaces and drying drums it is necessary to periodically conduct diagnostics of vibroacoustics and load parameters of parts of drive systems [12, 13].

Fig. 4 shows graphic dependencies of dynamic coefficients of the change of normal force in meshing of the crown gearing at different values of angle gap of gearing and deflection of turning angle of the crown wheel. In the graphs, one can see that the increase of angle gap to d = 0.003 rad leads to the increase of dynamic coefficient by 20%, and with the same angle gap and at E = 0.0005 rad, increases by 30% and more.

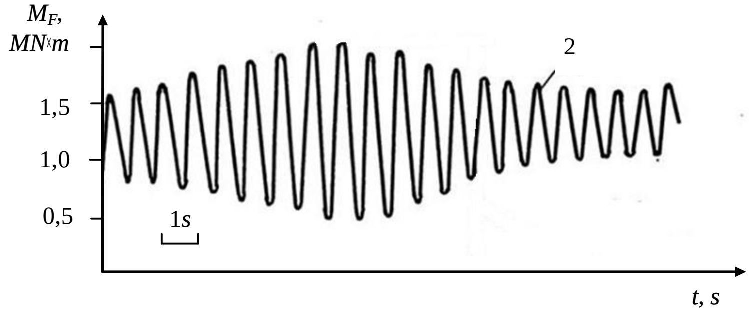

Fig. 3. Oscillation of torgue moment in crown gearing under resonans condition of operation:
1 – determined from experiment;
2 – determined by means of mathematical modeling

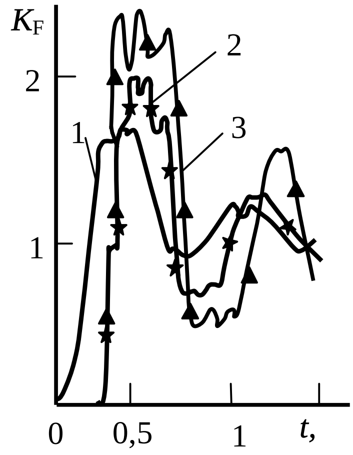

Fig. 4. Calculation dependences of dynamic magnification coefficient K_F of normal load of crown gearing on the parametres E and d:
1 - d = E = 0;
2 - d = 0.003 rad, E = 0;
3 - d = 0.003 rad, E = 0.0005 rad

Conclusions. Application of the driving mechanism with smooth selection of angle gap in of the crown gear pair permits to essentially decrease vibration activity of the machine in the starting process.

With the increase of rigidity of the elements of attachment of the crown wheel to the furnace body, the influence of the value of angle gap on intensity of oscillation phenomena in systems increases. Therefore, the use of the attachment of the crown wheel with the help of flexible contacts [14] makes it possible to decrease percussion phenomena in the drive elements during non-stationary operation conditions.

The investigation has shown that the proposed procedures allow us to considerably decrease vibroacoustic parameters of the machine aggregates under study and to improve work conditions of service personnel. The suggested programs can be included into the united system of CAD of the considered aggregates, they also can be useful in diagnostics of drives of rotary furnaces in the course of their operation and reconstruction.

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V.M. Korendiy, I.V. Kuzio, V.V. Vergeles
Lviv Polytechnic National University,
Department of Mechanics and Automation of Machine Building

ANALYSIS OF POWER REGULATION MECHANISMS OF HORIZONTAL-AXIS WIND TURBINES AND PROSPECTS OF THEIR IMPROVEMENT

Проаналізовано основні типи механізмів регулювання потужності горизонтально-осьових вітроустановок малої потужності. З метою підвищення точності регулювання потужності, ефективності й надійності функціонування вітроустановок у широкому діапазоні швидкостей вітру обґрунтовано необхідність подальшого вдосконалення існуючих механізмів регулювання. Розглянуто перспективи створення комбінованих механічних систем, у яких передбачено можливості одночасного повороту й складання лопатей, повороту (або складання) лопатей і виведення вітроколеса з-під вітру.

Main types of mechanisms of power regulation of horizontal-axis wind turbines of low power are analyzed. The necessity of further improving of existent regulation mechanisms is motivated for the purpose of increasing of accuracy of power regulation, effectiveness and reliability of wind turbines functioning in wide range of wind speeds. The prospects of creation of combined mechanical systems are considered, where the possibilities of simultaneous blades turning and folding, blades turning (or folding) and wind-wheel deflection out of wind direction are provided.

Problem stating. The operation modes of wind turbines (WTs) fall into two types: stationary and transient [1]. In stationary operation modes WTs can be operating during unlimited time spans over the whole life cycle. The shut-down mode and normal (nominal) functioning of WT with partial or full load on