УДК 531.8:075.8

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VIBRATIONS DECREASE IN MACHINES WITH CONTINUOUS IMPACT ACTION BY INTRODUCTION OF NON-LINEAR RESILIENT ELEMENTS

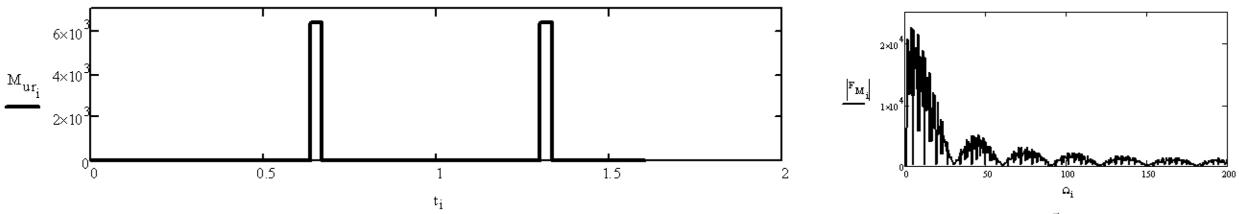
The influence of non-linearity in resilient characteristics on oscillation processes in machines drives working in continuous shock action for the purpose to improve its reliability and service life was considered. The condition of vibrations diminishing in a drive is shown.

Key words: *machine drive, non-linear oscillation, service life.*

The drive composed of electric motor, fly-wheel, reduction gear and two couplings with non-linear characteristics is considered. The drive is designed to work together with presses and casting-breakers in cyclic duty. The presence of periodic disturbances of resistance forces is the design feature of this kind of the drive. Loads in the drive are of shock character. The moment of useful resistance forces brought to incoming shaft of a working machine exceeds several times the rated drive (nominal) forces moment brought to the same shaft (for press – 2–4 times, for casting – breakers – 10–15 times, fig. 1, *a*). Disturbances spectrum is of a continuous nature in a wide range of frequencies (fig. 1, *b*).

As a consequence with continuous machine operation the possibility of resonance arises. Damping of such vibrations is usually achieved through dissipation of energy that results in its losses as well as in high wear out of resilient elements and decrease in reliability and service life of a drive. Introduction of non-linearity results in connection of natural vibration frequency and amplitude and this may influence on origination of resonance phenomena.

General view of a drive and its kinematical scheme together with working machine are shown in fig. 2. Here, J_1 *is* moment of inertia *of a* fly-wheel, J_2 *is* brought to output shaft the inertia moment of revolving parts of reduction gear, J_3 is moment of inertia of moving parts of working machine brought to the incoming shaft; c_1, c_2 are rigidity coefficients of resilient couplings; k_1, k_2 are energy dissipation coefficients in the couplings resilient elements; U_{12} is a gear ratio of a reduction gear.



a б

Fig. 1. Force characteristics of a drive loading: cyclic loading (*a*) and loading spectrum (*b*)

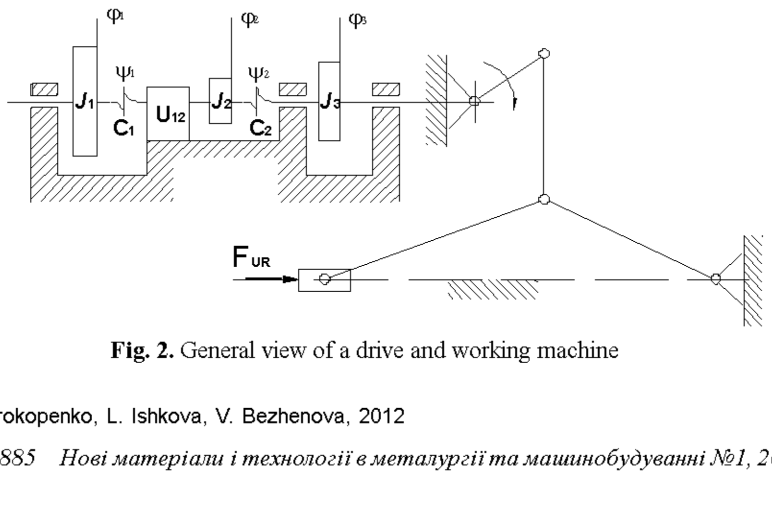


Fig. 2. General view of a drive and working machine

For non-linear elements the power relationship of the torsion moment and angular deformation is taken. Generalized coordinates are: φ_1 is the angle of rotation of motor shaft, ψ_1, ψ_2 are angles of deformation of couplings. A fly-wheel with high inertia moment on a drive shaft enables to overcome peak loadings with a low value of vibration factor. In this case it is possible to accept $d\varphi_1/dt = \omega_1 = \text{const}$ and to describe the drive vibrations with two equations

$$\ddot{\psi}_1 + a\cdot\psi_1^n + b\cdot\psi_2^n + c\cdot\dot{\psi}_1 + d\cdot\dot{\psi}_2 + e = 0;$$

$$\ddot{\psi}_2 + f\cdot\psi_1^n + g\cdot\psi_2^n + h\cdot\dot{\psi}_1 + k\cdot\dot{\psi}_2 + l(\varphi_3) = 0;$$

$$\varphi_3 = (\varphi_1 - \psi_1)/U_{12} - \psi_2, \qquad (1)$$

where e, $l(\varphi_3)$, φ_3 are disturbances from the engine, working machine and angle of incoming shaft rotation of working machine, and factors of equations are connected with system parameters like a following ratios:

$$a = c_1\cdot\frac{u_{12}^2}{J_2};\quad b = -\frac{c_2}{J_2}\cdot u_{12};\quad c = k_1\cdot\frac{u_{12}^2}{J_2};\quad d = -\frac{k_2}{J_2}\cdot u_{12};$$

$$e = -\frac{M_m}{J_1};\ f = -\frac{c_1\cdot u_{12}^2}{J_2};\quad g = c_2\left(\frac{1}{J_2}+\frac{1}{J_3}\right);$$

$$h = -\frac{1}{J_2}\cdot k_1\cdot u_{12};\quad k = k_2\cdot\left(\frac{1}{J_2}+\frac{1}{J_3}\right);\quad l = -\frac{M_{resist}}{j_3}. \qquad (2)$$

The solution of this problem on the whole is of great complexity. That is why features of machines class being examined are used. They include stationary areas of loading. First area where power disturbances of the engine act upon M_m = const and the second one where disturbances of an engine M_m = const and working machine M_{resist} = const act upon. Initial disturbances of every area are caused by the transient features between the areas. Presence of these disturbances only within the limits of every area allows to consider this problem as a piece-stationary one. On the other hand, the existent of continuous nature disturbances and relation of self-disturbances to frequency and amplitude allow to influence upon the resonance.

The solution of the first problem includes the search of stationary points in every area and estimation of their stability. The computation of stationary points of coordinates is realized with equations of motion (1) on every area with $\ddot{\psi}_1 = 0,\ \ddot{\psi}_2 = 0,\ \dot{\psi}_1 = 0,\ \dot{\psi}_2 = 0$ provision of conditions, $e \neq 0,\ l = 0$ (on the first area), and $e \neq 0,\ l \neq 0$ (on the second area).

Stability of stationary points has been estimated by the Rauth-Hurwitz criterion. Then the Hurwitz determinants for linearized system being considered by the method of a harmonic linearization have the form:

$$b_3 = c + k;\ b_2 = 0{,}75\cdot a\cdot y_0^2 + 0.75\cdot g\cdot y_2^2 - d\cdot h + c\cdot k;$$

$$b_1 = 0{,}75\cdot y_2^2(c\cdot g\cdot - b\cdot h) - 0{,}75\cdot y_0^2\cdot(d\cdot f - a\cdot k);$$

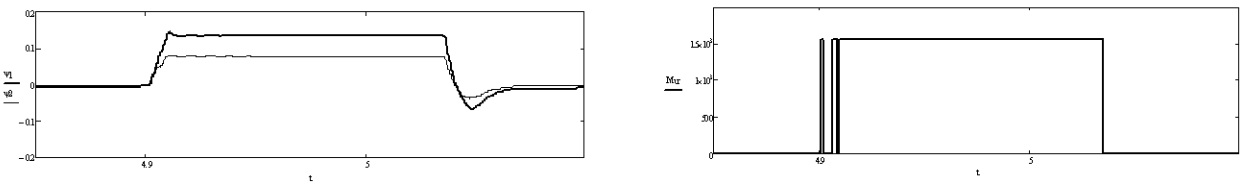
$$b_0 = 0{,}5625\cdot y_0^2\cdot y_2^2\cdot(a\cdot g - b\cdot f); R = \frac{b_1\cdot b_2\cdot b_3}{b_1^2 + b_3^2\cdot b_0}, \qquad (3)$$

where y_0, y_2 are the stationary points.

For stability of stationary points it is necessary that that Hurwitz determinants b_3, b_2, b_1, b_0 and R be more than zero.

With chosen parameters of working machine and a drive, that provide the stability of stationary points, $J_1 = \infty, J_2 = 0{,}25\ \kappa gm^2, J_3 = 0{,}1\ \kappa gm^2, c_1 = 4\cdot 10^4\ Nm/rad,$ $c_2 = 3{,}14\cdot 10^6\ Nm/rad,\quad U_{12} = 15{,}7,\ k_1 = 1\ Nms/rad,$ $k_2 = 20\ Nms/rad,\ M_M = 40\ Nm,\ M_{conp} = 3140\ Nm$, through numerical methods the solution for disturbed motion was obtained. Results of the solution, the angles of deformation in couplings and loads in a working machine are shown in fig. 3.

Computations have proved that including of non-linearity into the system and provision of stability for stationary points enables to obtain little vibrations in drives of machines being considered. Fig. 4 shows the alteration in vibrations at transition to operating loading regime (A_1) and an idle operation duty (A_2) with respect to degree (n) of polynomial of non-linearity of resilient elements. So, when changing the linear system for cubic one the non-linear vibration amplitude decreases for an order. This is related as to the shifting of vibration frequency at the increase of amplitude in areas of high frequencies where the energy of disturbance spectrum is significantly lower, as well as with progressive dissipation of vibration energy in area of high frequencies, and with positive effect of transient processes on the rate of amplitude growth at origination of the resonance.



a *б*

Fig. 3. Dynamics of a system with non-linear elements: angle of deformation in couplings (*a*) and change of forces in working machine (*b*) with present of vibrations

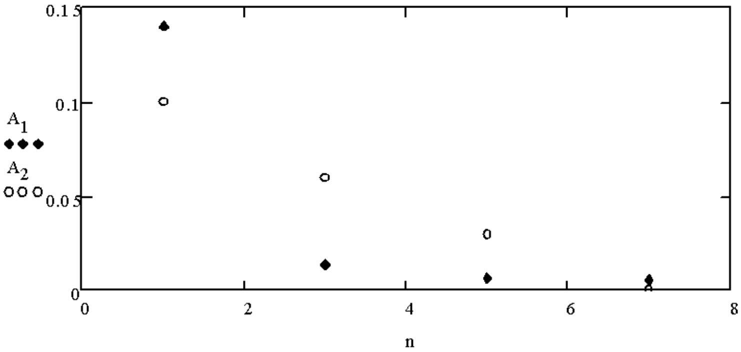


Fig. 4. Relation of amplitude vibrations to polynomial degree

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Одержано 05.07.2011

Коляда О.Ф., Сокол Т.О., Прокопенко Д.С., Ішкова Є.О., Бєженова В.С. Зменшення коливань у машинах безперервної ударної дії введенням нелінійних пружних елементів

У статті розглядається вплив пружних нелінійних характеристик на коливальні процеси в приводах машин, що працюють в ударному режимі, з метою підвищення надійності і довговічності. Показані умови зменшення коливань у приводі.

Ключові слова: *привод машини, нелінійні коливання, термін життя.*

Коляда А.Ф., Сокол Т.А., Прокопенко Д.С., Ишкова Е.А., Беженова В.С. Уменьшение колебаний в машинах непрерывного ударного действия введением нелинейных упругих элементов

В статье рассматривается влияние нелинейностей упругих характеристик на колебательные процессы в приводах машин, работающих в ударном режиме, с целью повышения его надежности и долговечности. Показаны условия уменьшения колебаний в приводе.

Ключевые слова: *привод машин, нелинейные колебания, срок службы.*

УДК 631.316.022

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МАТЕМАТИЧЕСКАЯ МОДЕЛЬ ПОВЕРХНОСТИ ЗУБЧАТОГО РАБОЧЕГО ОРГАНА ДЛЯ ОБРАБОТКИ ПОЧВЫ

Предложена математическая модель, которая при определенных условиях позволит разработать зубчатый орган с такой формой рабочей поверхности, который будет удовлетворять агротехническим, технологическим и экономическим показателям при обработке почвы. Получены уравнения, которые описывают поверхность рабочего органа на участках впадин и выступов.

Ключевые слова: *зубчатый рабочий орган, почва, выступы и впадины зубьев, логарифмическая спираль, парабола четвертого порядка, математическая модель, матрично-векторное решение.*

Плодородие черноземных почв как основного экономического ресурса Украины зависит от качества их обработки. Известно, что определяющим критерием качества обработки почвы является ее крошение. Уровень крошения почвы зависит от геометрии рабочего органа и от кинематики его движения в процессе ра-