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THE RESEARCH OF THE TORSIONAL VIBRATIONS OF THE SCREW IN TERMS OF IMPULSIVE FORCE IMPACTS

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ДОСЛІДЖЕННЯ КРУТИЛЬНИХ КОЛИВАНЬ ШНЕКА У ВИПАДКУ ДІЇ ІМПУЛЬСНИХ СИЛ

Purpose. To analyze the impulsive force impact on nonlinear torsional vibrations of a screw.

Methodology. The methodology is based on the Bubnova-Galorcina's and Van-der-Pol's methods combination, which allowed receiving the equations in a standard form in terms of the impulsive force impacts.

Findings. The mathematical model of torsional vibrations of the screw in terms of the impulsive force impact was presented. The abrupt nature of changes in the amplitude-frequency characteristics of torsional vibrations of the screw was studied. The resonance torsional vibrations of the screw were considered under the impulsive force impacts. The torsional vibrations of the screw were explored on condition that the moment of forces of resistance is proportional to the relative angular velocity of the screw motion and the moment of impulsive forces is approximated by a nonlinear function. It has been determined, that in such a case the influence of impulsive forces becomes apparent only while changing the screw vibration frequency. The amplitude frequency characteristics of the torsional vibrations of the screw in different geometric parameters were presented.

Originality. The influence of impulsive forces on resonance and on non-resonance torsional vibrations of the screw was explored. The proper amplitude frequency characteristics were established.

Practical value. It has been established that in the non-resonance terms the impulsive nature of the screw loading results in the abrupt change of the screw amplitude and the phase of vibrations when the impulsive forces are performed. Its influence rises as the time of the screw machines exploration passes and it can result in considerable amplitudes of torsional vibrations of the screw. It has been established that the screw resonance falls as the vibration frequency raises.

Keywords: *screw, torsional vibrations, impulsive forces*

Introduction. Screw conveyor transport and technological mechanisms are widely used in different branches of industry, including mining industry, for the transportation of bulk and lump materials. The efficiency of the operation of many bays, shops and the whole enterprises depends on their reliable functioning. Screw conveyors can be characterized by the simplicity of their design and, consequently, high reliability, easiness of operation and adjustment when used in automated systems and by being ecologically-friendly to the environment because of their hermeticity [1–3]. High-speed screw conveyors are used for all-purpose loading and unloading complexes, which are designed to transport load on horizontal, declining the vertical routes. The existing methods are based on a number of theoretical and experimental investigations as well as on the analysis of the statistical data on the results of their exploitation. In order to provide the reliability and the quality of the technological processes performed by conveyor mechanisms, it is necessary to take into account the dynamic vibrations, caused by outside power factors and the peculiarities of the functioning of screw conveyors.

Latest researches and publications overview. The fundamentals of the designing and the investigation of screw conveyors were laid by such scientists as Hryhorev A.M., Hevko B.M., Owen P.J. [1], Rohatynskyi R.M., Rorres C. [2], Loveikin V.S. [4], Hevko R.B. [3] and other. The development of the theory of vibrations was elaborated by V.S. Loveikin [4], L.Q. Chen [5] and others. In case of forced vibrations, in other words those vibrations, which are caused by the influence of periodical forces, the frequency of which is altering in time, the amplitude of vibrations and the dynamic stresses depend essentially on the frequency of the forcing power. When the above-mentioned frequencies are the same, or when the frequency of forcing power approximates the natural frequencies in a screw and in the case of low damping, resonance is developing [6, 7], that is the amplitude of vibrations is increasing rapidly. Such a rise in the amplitude causes an essential increase of a twist angle or deflection of a screw. With the increase of angle or linear deformations, the dynamic stress in the working bodies of a screw increases as well. In this case, dynamic stresses (resonance) depend both on inside factors (physical and mechanical parameters of a screw, its geometric sizes etc.) and on the outside ones. The outside factors include the angular rate of the rotation of a screw and the value of outside disturbing forces for bending [7] and torsional vibrations [6].

Unsolved problems of the general issue. During the work of the screw mechanisms the force impulsive impact on the working parts due to the peculiarities of lump cargo transported and the screw bending deformations can often lead to the significant torsional amplitudes of vibrations in the screw, and therefore to reducing the reliability of its work. The problem of investigating the torsional vibrations of the screw working parts that occur at various modes of its operation in non-resonant and resonant zones, and, relatively, to the reliability of screw in the transport and technological systems, is not currently paid enough attention resulting in the need for further researches.

The purpose of the research is to analyze the impulsive force impacts on the nonlinear torsional vibrations of the screw.

Research results. General results, presented in the papers [6, 7], are used for the investigation of the impulsive force impacts on the torsional vibrations of a screw. A screw rotates around the axis, making torsional and bending vibrations. In many cases, the last ones cause the short-lasting periodical influence on torsional vibrations. The question is about the contact of a screw and a casing, transportation of bulk loads of relatively large sizes and other. A mathematical model of the torsional vibrations of a screw conveyor for the above-mentioned influence of the external immediate forces is the following differential equation

$$I\frac{\partial^2\theta}{\partial t^2}-\frac{\partial}{\partial x}\left(GJ\frac{\partial\theta}{\partial x}\right)=Q\left(\theta,\frac{\partial\theta}{\partial t},\frac{\partial\theta}{\partial x},\right)+ +\sum_{i=1}^{n}Q_i\left(\theta,\frac{\partial\theta}{\partial x},\frac{\partial\theta}{\partial t}\right)\cdot\sum_{j=1}\delta\left(t-(t_i+j\tau)\right), \tag{1}$$

where $\theta(x,t)$ is a twist angle of a screw, I is a linear moment of inertia of a screw about a strainless axis, G is a shear modulus of the screw material, J is an equatorial moment of the screw cross-section, $\delta(...)$ – Dirac function, that acts periodically over a period of τ at time moments t_i, $Q_i\left(\theta,\frac{\partial\theta}{\partial x},\frac{\partial\theta}{\partial t}\right)$ – function, which characterizes the intensity of impulsive force impacts at the time moments mentioned.

If the properties of δ–function are used

$$f(t)\delta(t)=f(0)\delta(t);$$

$$\sum_j\delta(t-j\tau)=\frac{\upsilon}{\pi}\left[\frac{1}{2}+\sum_{j=1}\cos j\upsilon t\right]; \tag{2}$$

$$\int_{-\infty}^{t}\delta(t)dt=\begin{cases}1, npu & t>0\\ 0, npu & t\le 0\end{cases}, \tag{3}$$

the system of differential equations (1) after the averaging is as follows

$$\frac{da}{dt}=-\frac{\mu}{\omega_\theta}\left\{F_{k0}^{s}(a)+\frac{\upsilon}{2\pi}\sum_{i=1}F_{ik0}^{s}(0)\right\}; \tag{4}$$

$$\frac{d\varphi}{dt}=-\frac{\mu}{a\omega_\theta}\left\{F_{k0}^{c}(a)+\frac{\upsilon}{2\pi}\sum_{j=1}F_{jk0}^{c}\right\},$$

where $\upsilon=\frac{2\pi}{\tau}$, ω_θ is a frequency of the vibrations in a screw.

From the technical point of view, the above-mentioned equations can be integrated and the dynamic process of a screw conveyor can be shown as

$$\theta(x,t)=a(t)X(x)\cos(\omega_\theta t+\varphi(t)). \tag{5}$$

In (5) the amplitude of torsional vibrations $a(t)$ and its phase $\psi=\omega_\theta t+\varphi(t)$ are determined by the system (4).

The indicated solution will be the first approximation to the task stated. In order to describe a jump pattern of change for the main parameters of the torsional vibrations of a screw, it is necessary to find its first improved approximation. In order to find it, we assume, that the solution of the differential equations (4) is the functions $a=a(t)$ i $\psi=\psi(t)$. Then, the first "improved" approximation of the parameters a and ψ is represented as follows

$$\begin{aligned} a_{покр.} = a-\frac{\mu}{\omega_\theta}\Bigg\{&\frac{\upsilon}{\pi}\sum_{i=1}^{n}F_{i0}^{s}(a)\sigma(t,t_i)+\\ &+\frac{\upsilon}{2\pi}\sum_{i=1}^{n}\sum_{n}\frac{-F_{in}^{ss}(a)\cos(n\psi)+F_{in}^{sc}(a)\sin(n\psi)}{n\omega_\theta}+\\ &+\frac{\upsilon}{2\pi}\sum_{i=1}^{n}\sum_{n}\Bigg(\frac{F_{in}^{ss}(a)}{(n\omega_\theta)^2-(k\upsilon)^2}(2n\omega_\theta\sin(n\psi)\cos k\upsilon(t-t_i)-\\ &-2k\upsilon\cos(n\psi)\sin k\upsilon(t-t_i))+\\ &+\frac{F_{in}^{ss}(a)}{(n\omega_\theta)^2-(k\upsilon)^2}(-2n\omega_\theta\cos(n\psi)\cos k\upsilon(t-t_i)+\\ &+2k\upsilon\sin(n\psi)\sin k\upsilon(t-t_i)))+\\ &+\sum_{n}\frac{-F_{n}^{sc}(a)\cos(n\psi)+F_{n}^{sc}(a)\sin(n\psi)}{n\omega_\theta}\Bigg\}; \end{aligned} \tag{6}$$

$$\begin{aligned} \psi_{покр.} = \omega t-\frac{\varepsilon}{a\omega_\theta}\Bigg\{&\frac{\upsilon}{\pi}\sum_{i=1}^{n}F_{ki0}^{c}(a)\sigma(t,t_i)+\\ &+\frac{\upsilon}{2\pi}\sum_{i=1}^{n}\sum_{n}\frac{-F_{in}^{cc}(a)\cos(n\psi_k)+F_{in}^{cs}(a)\sin(n\psi_k)}{n\omega_\theta}+\\ &+\frac{\upsilon}{2\pi}\sum_{i=1}^{n}\sum_{n}\Bigg(\frac{F_{in}^{cs}(a)}{(n\omega_\theta)^2-(k\upsilon)^2}(2n\omega_\theta\sin(n\psi)\cos k\upsilon(t-t_i)-\\ &-2k\upsilon\cos(n\psi)\sin k\upsilon(t-t_i))+\\ &+\frac{F_{in}^{cs}(a)}{(n\omega_\theta)^2-(k\upsilon)^2}(2n\omega_\theta\cos(n\psi_k)\cos k\upsilon(t-t_i)+\\ &+2k\upsilon\sin(n\psi)\sin k\upsilon(t-t_i)))+\\ &+\sum_{n}\frac{-F_{n}^{ss}(a)\cos(n\psi_k)+F_{n}^{ss}(a)\sin(n\psi_k)}{n\omega_\theta}\Bigg\}, \end{aligned}$$

where $\sigma(t,t_i)$ is a periodical function, which comprises the sum of $\sum_j \frac{\sin j\upsilon(t-t_i)}{j}$.

The above-mentioned formulae show, that in terms of non-resonance the impulsive pattern of loading becomes apparent in the amplitude drop change $a_{покр}$ and phase $\psi_{покр}$ at the moment of impulsive forces action. During the use of screw machines their action increases and sometime later it can lead to considerable amplitudes of the torsional vibrations.

The resonance torsional vibrations of a screw under the impulsive force impacts should be considered. Much more important case of torsional vibrations is the one, where the frequency of natural oscillations is connected with the frequency of impulsive disturbance by the correlation $\omega_\theta \approx \frac{q}{p}\upsilon$ (p,q – reciprocals); here $\upsilon = \frac{2\pi}{\tau}$.

The above mentioned substantiates the following differential equation for impulsive force impacts

$$\frac{d^2T}{dt^2}+\left(\frac{q}{p}\upsilon\right)^2 T=\mu(\bar{F}\left(T,\frac{dT}{dt},\upsilon t\right)-\Delta T+$$
$$+\sum_{j=1}^{n}\bar{F}_j\left(T,\frac{dT}{dt},\upsilon t\right)\cdot\sum_{i=1}^{m}\delta\left(t-(t_i+j\tau)\right)); \quad (7)$$

$$\omega_\theta^2=\left(\frac{q}{p}\upsilon\right)^2+\mu\Delta, \quad (8)$$

where $\mu\Delta$ is a deregulation of frequencies $T(t)=a\cos\psi$. In this case, the usual differential equations relative to variables $a(t)$ and $\varphi(t)$ acquire the form of

$$\frac{da}{dt}=-\sin\psi\frac{p\mu}{q\upsilon}\left(\bar{F}\left(a\cos\psi,-a\upsilon\frac{q}{r}\sin\psi\right)-\Delta a\frac{q}{p}\upsilon\cos\psi_k+\right.$$
$$\left.+\sum_{i=1}^{n}\bar{F}_i\left(a\cos\psi,-a\upsilon\frac{q}{p}\sin\psi\right)\sum_j\delta\left(t-(t_i+j\tau)\right)\right);$$

$$\frac{d\varphi}{dt}=-\cos\psi\frac{p\varepsilon}{aq\upsilon}\left(\bar{F}\left(a\cos\psi,-a\upsilon\frac{q}{p}\sin\psi\right)-\Delta a\frac{q}{p}\upsilon\cos\psi+\right.$$
$$\left.+\sum_{i=1}^{n}\bar{F}_i\left(a\cos\psi,-a\upsilon\frac{q}{p}\sin\psi\right)\sum_{j=1}\delta\left(t-(t_i+j\tau)\right)\right). \quad (9)$$

Taking into consideration the assumption, that $Q(...)$ and $Q_j(...)$ – multinomial, the functions $\bar{F}(a\cos\psi,-a\omega_\theta\sin\psi)$ and $\bar{F}_i(a\cos\psi,-a\omega_\theta\sin\psi)$ are represented in the form of Fourier series. Using the information above and the properties of δ of Dirac function (2, 3), the system of the differential equations (9) after the approximation acquires the form of

$$\frac{da}{dt}=-\frac{\mu p}{2\pi q\upsilon}\sum_{i=1}^{n}(F_{i0}^{s}(a)+\sum_n(F_{in}^{sc}(a)\cos n(p\varphi+q\upsilon t_i)+$$
$$+F_{in}^{ss}(a)\sin n(p\varphi+q\upsilon t_i)))+F_0^s(a)).$$

Thus, in terms of resonance as a contrast to non-resonance, the additional terms have appeared in the approximated equations. But being similar to the non-resonance case, the values $a_{покр}$ and $\psi_{покр}$ at the moment of impulsive forces action show a jump change.

The torsional vibrations of a screw should be considered in the case, when the moment of sustaining power is proportional to relative angular velocity of a screw conveyor $\frac{\partial\theta(x,t)}{\partial t}$, and the moment of impulsive forces is approximated by the function $\lambda\theta(x,t)+\gamma\theta^3(x,t)$. The differential equation of the torsional vibrations of a screw is as follows

$$\frac{\partial^2\theta}{\partial t^2}-\frac{GJ_0}{I_0}\frac{\partial^2\theta}{\partial x^2}=\mu(\lambda\theta(x,t)+\gamma\theta^3(x,t)\times$$
$$\times\sum_{j=1}^{n}\sum_{i=1}^{m}\delta\left(t-(t_i+j\tau)\right)-\beta\frac{\partial\theta}{\partial t}). \quad (10)$$

According to the Bubnov-Halorkin method, the solution to the equation (10) is shown to be the same as in paper [6] in the form of $\theta(x,t)=X(x)T(t)$. After simple transformations, the differential equations are reduced to a simple form of differential equation

$$\frac{d^2T(t)}{dt^2}+\left(\frac{k\pi}{l}\right)^2\frac{GJ_0}{I_0}T(t)=\mu(\left(\lambda T+\gamma T^3\right)\times$$
$$\times\sum_{i=1}^{2}\sum_j\delta\left(t-(t_i+j\tau)\right)-\beta\frac{dT}{dt}). \quad (11)$$

For non-resonance vibrations of a screw conveyor, the amplitude and the vibration frequencies according to the results given in paper [6] ($t_1=0$, $t_2=\frac{\pi}{2\upsilon}$) are described by means of the differential equations

$$\frac{da}{dt}=-\mu\beta\frac{a}{2};$$

$$\frac{d\psi}{dt}=\omega_\theta-\frac{\mu}{\omega_\theta\pi}\left(\frac{\lambda\upsilon}{8}+\frac{3\gamma a^2\upsilon}{32}\right).$$

Having integrated the obtained system of differential equations, the first approximation of the solution to the equation of the torsional vibrations of a screw conveyor is found

$$T(t)=a_0e^{-\frac{\beta}{2}t}\cos\left(\omega t+\theta_0-\frac{\lambda\upsilon t}{8\pi\omega}+\frac{3\gamma\upsilon a_0^2}{32\pi\beta\omega}e^{-\beta t}\right), \quad (12)$$

where a_0 and θ_0 are determined to be starting conditions.

Thus, the influence of impulsive forces results only in change of the frequency of the vibrations of a screw.

Resonance vibrations should be considered. Let's assume, that the frequency of natural oscillations of a screw conveyor is connected with the frequency of the impulsive disturbance by the following correlation

$$\omega \approx q\frac{\upsilon}{2}. \qquad (13)$$

In this case, a first approximation of the solution of the differential equation (12) has the following form

$$T(t) = a(t)\cos\left(\frac{\upsilon}{2}qt + \varphi(t)\right),$$

where functions $a(t)$ and $\varphi(t)$ are determined from the system of differential equations.

$$\frac{da}{dt} = -\mu\Big(\frac{\beta a}{2} + \frac{a\left(2\lambda+\gamma a^2\right)}{4\pi q}\left(\sin 2\varphi + \sin\left(2\varphi + q\frac{\pi}{2}\right)\right) -$$

$$-\frac{\gamma a^3}{8\pi q}\left(1+(-1)^q\right)\sin 4\theta\Big);$$

$$\frac{d\varphi}{dt} = \frac{\omega_\theta^{\,2} - \left(\frac{q\upsilon}{2}\right)^2}{\omega_\theta} - \mu\Big(\frac{4\lambda+3\gamma a^2}{4\pi q} + \frac{\lambda+\gamma a^2}{2\pi q} \times$$

$$\times\left(\cos 2\varphi + \cos\left(2\varphi + q\frac{\pi}{2}\right)\right) + \frac{\gamma a^2}{8\pi q}\left(1+(-1)^q\right)\cos 4\varphi\Big). \qquad (14)$$

The figure shows the amplitudes of the torsional vibrations of a screw when there is a transition through the resonance at different parameter values $\omega_\theta = \frac{k\pi}{l}\sqrt{\frac{GJ_0}{I_0}}$ at l=10m; G=80GPa; I_0=3,4675kgm.

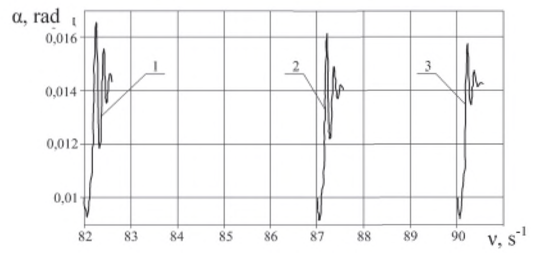

Fig. The amplitudes of resonance torsional vibrations at different values of the parameter ω_θ *at: 1 –* $J_0 = 0{,}10248\times10^{-4} m^4$; *2 –* $J_0 = 0{,}1147\times10^{-4} m^4$; *3 –* $J_0 = 0{,}1215\times10^{-4} m^4$

Conclusions and development prospects. The represented graphical dependencies and their comparison with the resonance curves in case of bending vibrations [7] make it possible to state, that resonance value of the amplitude of the vibrations of a screw takes a smaller value at larger frequencies. The obtained results give the possibility to avoid resonance torsional vibrations of a screw in case of its exploitation under the influence of impulsive forces due to the change of material transportation conditions. Based on the taken out equations, it is possible to develop the automated systems of management for the processes of material transportation using screw machines.

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Мета. Провести аналіз впливу імпульсних сил на нелінійні крутильні коливання шнека.

Методика. Методика базується на поєднанні методів Бубнова-Гальоркіна та Ван-дер-Поля. З її допомогою отримані рівняння у стандартному вигляді для випадку дії імпульсних сил.

Результати. Представлена математична модель крутильних коливань шнека у випадку дії імпульсних сил. Встановлено стрибкоподібний характер зміни амплітудно-частотних характеристик крутильних коливань шнека. Розглянуті резонансні крутильні коливання шнека під дією імпульсних сил. Досліджені крутильні коливання шнека за умови, що момент сил опору пропорційний відносній кутовій швидкості руху шнека, а момент імпульсних сил апроксимується нелінійною функцією. Встановлено, що в цьому випадку вплив імпульсних сил проявляється у зміні лише частоти коливань шнека. Представлені амплітудно-частотні характеристики крутильних коливань шнека за різних значень його геометричних параметрів.

Наукова новизна. Досліджено вплив імпульсних сил на резонансні та нерезонансні крутильні коливання шнека, встановлені відповідні амплітудно-частотні характеристики.

Практична значимість. Встановлено, що в нерезонансному випадку імпульсний характер навантаження на шнек проявляється у стрибкоподібній зміні амплітуди та фази коливань шнека в момент дії імпульсних сил. Їх дія за період експлуатації шнекових машин наростає та з часом може привести до значних амплітуд крутильних коливань шнека. Встановлено, що резонансне значення амплітуди коливань шнека зменшується при збільшенні частоти коливань.

Ключові слова: *шнек, крутильні коливання, імпульсні сили*

Цель. Провести анализ влияния импульсных сил на нелинейные крутильные колебания шнека.

Методика. Базируется на сочетании методов Бубнова-Галёркина и Ван-дер-Поля. С ее помощью получены так называемые уравнения в стандартном виде для случая действия импульсных сил.

Результаты. Представлена математическая модель крутильных колебаний шнека в случае действия импульсных сил. Установлен скачкообразный характер изменения амплитудно-частотных характеристик крутильных колебаний шнека. Рассмотрены резонансные крутильные колебания шнека под действием импульсных сил. Исследованы крутильные колебания шнека при условии, что момент сил сопротивления пропорционален относительной угловой скорости движения шнека, а момент импульсных сил аппроксимируется нелинейной функцией. Установлено, что в этом случае влияние импульсных сил проявляется в изменении лишь частоты колебаний шнека. Представлены амплитудно-частотные характеристики крутильных колебаний шнека при разных значениях его геометрических параметров.

Научная новизна. Исследовано влияние импульсных сил на резонансные и нерезонансные крутильные колебания шнека, установлены соответствующие амплитудно-частотные характеристики.

Практическая значимость. Установлено, что в нерезонансном случае импульсный характер нагрузки на шнек проявляется в скачкообразном изменении амплитуды и фазы колебаний шнека в момент действия импульсных сил. Их действие за период эксплуатации шнековых машин нарастает и со временем может привести к значительным амплитудам крутильных колебаний шнека. Установлено, что резонансное значение амплитуды колебаний шнека уменьшается при возрастании частот колебаний.

Ключевые слова: *шнек, крутильные колебания, импульсные силы*

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