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A. K. Tuleshov, A. A. Jomartov

Institute of Mechanics and Mechanical Engineering named after U. A. Joldasbekov, Almaty, Kazakhstan.
E-mail: aman_58@mail.ru; legsert@mail.ru

DETERMINATION OF DISSIPATIVE PARAMETERS OF CRANK PRESS

Abstract. The paper considers the determination of the dissipation coefficients of the crank press's assemblies (units). Slide-crank mechanism of the press performs stamping of various parts. In such a case, there are significant dynamic loads in the assemblies and connections of the press. Work features of the crank press are connected with shock cyclic loads. Dynamic research of crank presses, at present, is a critical task. Dynamic model of the crank press is represented as an oscillatory system with many degrees of freedom. The dynamic model of the crank press consists of lumped masses connected by elastic-dissipative elements. To calculate the dynamic model of the crank press, it is required to determine dissipation coefficients of individual sections of the drive shaft. Determination of the dissipation coefficients is of great complexity. In the case of vibrations of elastic systems, energy is dissipated in machine parts and mechanical energy is transferred into thermal. Losses of mechanical energy are caused by forces of inelastic resistance, called dissipative forces. The amplitudes of the resonance oscillations, the conditions for the excitation of parametric oscillations and self-oscillations depend on the dissipation coefficients. The determination of dissipative coefficients is based on available information on the dissipation coefficients of individual elastic-dissipative elements of the system. An example is given for determining the coefficients of dissipation of the crank press. The dynamic model of the crank press was compiled on the program complex SimulationX. The obtained results show that the dynamics of oscillations of the drive shaft of the crank press changes significantly after adjustment of the coefficients of dissipation.

Key words: dynamics, crank press, hardness (toughness), dissipation, рассеяние, oscillations, SimulationX.

Introduction. Crank press is a machine with a slide-crank mechanism, designed for stamping of various parts [1-3]. When the crank press operates, significant dynamic loads occur in machine components. These dynamic loads are associated with operation aspect of the crank press, which is in shock cyclic loads. The research of the dynamics of crank presses is of great interest. Figure 1 shows the structure chart of the press [1].

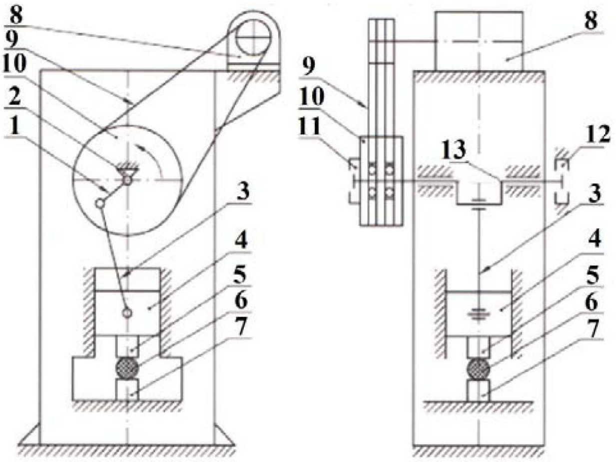


Figure 1 – Structure chart of the crank press

Principle of operation of the crank press (see figure 1): the crank 1 rotates about the axis 2 and drives the slider 4 with the punch 5 through the crank rod 3. The piece 6 is processed by moving 6 and fixed 7 parts of the punch. The press drive consists of an electric motor 8, a V-belt drive 9 and a flywheel 10. The muff of the press 11 is located between the flywheel 10 and the crank 13. Brake 12 serves to stop the press.

The crank press contains moving details and components, the mass of which is from hundred kilos to several tons. These details and components are cycled at high speeds and they are subject to large dynamic loads. To study the dynamics of the crank press, different dynamic models are worked out [4-9]. Figure 2 shows the dynamic model of the crank press. The dynamic model consists of lumped masses connected by elastic-dissipative elements.

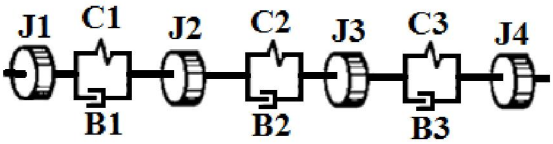


Figure 2 – Dynamic model of the crank press

The following identifications are introduced in the dynamic model (Figure 2): J1 - reduced moment of inertia of the engine rotor and pulley wheel, J2 - reduced moment of inertia of the flywheel and the brake, J3- reduced moment of inertia of the crank and executing mechanism, J4 reduced moment of inertia of the muff; C1 - V-belt drive stiffness coefficient, C2 - stiffness coefficient of the shaft section between the flywheel and the crank, C3 - stiffness coefficient of the shaft section between the crank and the muff, B1 - V-belt drive dissipation coefficient, B2 - dissipation coefficient of the shaft section between the flywheel and the crank, B3 - dissipation coefficient of the shaft section between the crank and the muff.

Inertial and stiffness parameters for the dynamic model are easily determined by known methods [4-9]. Determination of the dissipation coefficients is of great complexity.

Determination of dissipative coefficients of multi-degree-of-freedom oscillatory system. Dissipative forces have a great influence on machinery motion. Key reasons, causing power dissipates in the machinery joints are: forces of friction in kinematic couples and external friction between links of mechanism and surrounding environment; viscosity forces, arising in places of contact of elements of the fixed joints. At oscillations of elastic mechanical systems there is a dissipative forces in the joints of machine parts, in elastic elements, there is a transition of a mechanical power to thermal power. Dissipative forces, are forces of inelastic resistance due to which the mechanical power is lost [8-19].

Amplitudes of resonance oscillations, conditions of initiation of parametrical, subharmonic oscillations and self-oscillations depend on magnitude of dissipative forces. [8]

Dissipative forces have significant effect on oscillating processes with the frequencies close to natural frequencies [8-11].

For mechanical systems with one degree of freedom the resistance force, arising at oscillations, is defined as follows [8]:

$$R = -|R(q,\dot{q})| sign\, \dot{q}\,, \tag{1}$$

where q - is the generalized coordinate, describing oscillating process. Sometimes the module of resistance force does not depend on the generalized coordinate q. At viscous resistance, arising at small vibration speeds in liquid or gas $R = -b\,\dot{q}$, where b - is the dissipative coefficient. Usually for problems of dynamics of machinery, the module of resisting force depends on the generalized coordinate and depends a little on vibrospeed, in such a case resisting force is called positional $R = -|R(q)| sign\, \dot{q}$. If $|R| = P = const$, the force of dry (or Coulomb) friction takes place. Also positional - viscous friction is available, at which $R = f(q)\,\dot{q}$.

Division of the elastic dissipative force on elastic and dissipative components is conditional, and generally physically impossible. For engineering calculations efficient approximate methods are applied to assess an influence of dissipation on oscillating processes. The engineer has only restricted initial information in the form of some integral characteristics, such as a dissipation coefficient ψ or logarithmic decrement λ, which are received experimentally at single-frequency modes.

In figure 3 the graph of restoring force with account of dissipative properties is shown. The graph has two edges, top corresponds to loading, and lower - unloading. The square of the figure, restricted by the edge of loading and X axis, corresponds to the work spent at deformation, and the square of the figure, bounded above by the second edge - to the work made by elastic element when unloading. At the same time the shaded area which contour is called a hysteresis curve is proportional to the work spent in one cycle for passing the forces of inelastic resistance. The ratio of this dissipated power to the work spent at deformation is called a dissipation coefficient ψ [8].

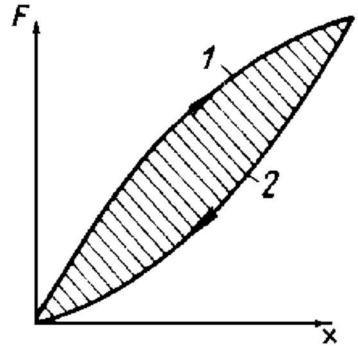


Figure 3 – The graph of the restoring force with account of dissipative properties:
1 - loading curve; 2 - unloading curve

The large number of dissipative factors, complexity and variety of the processes accompanying the oscillating phenomena lead to the fact that at the solution of engineering tasks it is necessary to resort to the dissipation parameters received from an experiment [16, 17]. In some cases an experiment determines dissipation coefficients of separate construction elements or joints, in other - some given values peculiar to the whole mechanism, knot, etc. Dissipation parameters usually are defined at single-frequency oscillations in the mode of damped free oscillations or in the resonance mode at forced oscillations. In the first case we have extinction process (figure 4) for which the dissipation coefficient can be defined as [8]

$$\psi = 1 - \left(\frac{A_2}{A_1}\right)^2,$$

where A_1 and A_2, two successive value of oscillation's amplitude divided by one period.

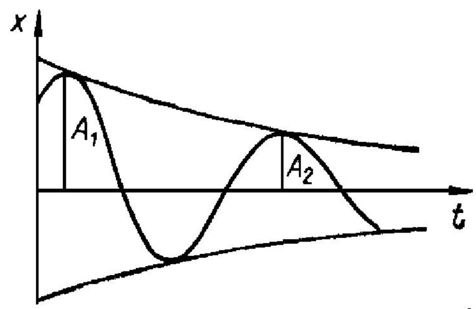


Figure 4 – Damped oscillations graph

Logarithmic decrement is a parameter $\lambda = ln\left(\frac{A_1}{A_2}\right)$. From here

$$\psi = 1 - e^{-2\lambda}$$

At small values λ we have $\psi \approx 2\lambda$. In the most general case parameters ψ and λ are not constants, and can depend on amplitude and oscillation frequency. From the analysis of the experimental data it is known that dependence of parameters of dissipation on oscillation frequency is shown very poorly.

The range of change of parameters of dissipation depending on various conditions is rather wide. So, average value of coefficient ψ for roller bearings fluctuates from 0,2 to 0,6; for dry cylindrical and conic joints from 0,03 to 0,15, and for well-greased surfaces coefficient ψ reaches value 1.

Reference data for many mechanisms of textile and polygraph machineries, machines, light industry machineries, etc., show that given dissipation factor ψ , stays within the range of values $0.4 \leq \psi \leq 0.65$ [8]. These values can be used at scoping calculation when there is no possibility of obtaining more precise information from an experiment.

Oscillatory system with one degree of freedom. Let's consider the equivalent linearization of dissipative forces in oscillating system with one degree of freedom. Let's comprise differential equation of the system with one degree of freedom [8]:

$$m\ddot{q} + cq = F(t) - |R(q,\dot{q})| sign\,\dot{q}\,, \tag{2}$$

where m, c - the reduced mass and reduced stiffness coefficient; $F(t)$ – is a driving force.

As dissipative force is non-linear a differential equation (2) is also non-linear. Because dissipative forces very slightly influence the frequency of free oscillations, and only defines a level of oscillation's amplitude at a resonance. In view of small influence of non-linear dissipative force on oscillating process, this oscillatory system is called quasi-linear. For the non-linear force $-R(q,\dot{q})$ we can find energetically equivalent linear force $R = -b\,\dot{q}$. Lets $F(t) = F_0 + sin\omega t$, then under the stationary forced oscillations $q = Asin(\omega t - \gamma)$. At the same time the energy dissipated in one period [8]

$$\Delta E = A\int_0^{2\pi} |R(Asin\varphi, \omega Acos\varphi)cos\varphi|\, d\varphi = \psi c A^2/2$$

On the other hand, $\Delta E = \pi b A^2 \omega$. Hence

$$b = \Delta E/(\pi A^2 \omega) = \psi c/(2\pi\omega) \tag{3}$$

In the formula (3) frequency of a driving force ω can be replaced by natural frequency k, dissipative forces influence strongly the oscillating process near this frequency [8].

The value ΔE corresponds to the area of hysteresis curve. Thus, from (3) follows that dissipative properties at the mono-harmonic mode are defined by the area of hysteresis curve and do not depend on the form of this loop. Dissipation coefficient ψ does not depend on amplitude if size ΔE is proportional to amplitude square [8]. It occurs at the linear resistance force or resistance force proportional to the first degree of amplitude.

From the analysis of the experimental data it is obtained that dissipation factor ψ, and logarithmic decrement λ poorly depend on an oscillation frequency [9] . Then, according to the formula (3) value of coefficient b in inverse proportion to an oscillation frequency.

Oscillatory System with multi degrees of freedom. Let's consider the equivalent linearization of dissipative forces for oscillating system with multi degrees of freedom. The dynamic model of oscillatory process is quasilinear because, non-linear dissipative forces have negligible influence on natural frequencies and forms of oscillations [8,9].

System of differential equations of model with H degrees of freedom looks as follows:

$$\boldsymbol{a}\ddot{\boldsymbol{q}} + \boldsymbol{c}\boldsymbol{q} = \boldsymbol{Q} \tag{4}$$

where $\boldsymbol{a}, \boldsymbol{c}$ – are square matrixes of inertial and quasi-elastic coefficients; $\boldsymbol{q}$ – is the vector-matrix (column) of the generalized coordinates; $\boldsymbol{Q}$ – is the vector matrix of non-conservative forces.

In the vector of generalized forces dissipative component $\boldsymbol{R}(q,\dot{q})$, we present as

$$\boldsymbol{R} = -\boldsymbol{b}\,\dot{\boldsymbol{q}}$$

where $\boldsymbol{b}$ – is the square matrix of equivalent dissipative coefficients; $\boldsymbol{q}$ – is the vector of generalized velocities.

The task is to find value of dissipative coefficients b_{jv}, the determination of which would base on the available information about the dissipation coefficients (or logarithmic decrements) of the individual elastic and dissipative elements of the system.

For purely viscous friction, where the force of resistance is proportional to the first order of velocity, for describing the dissipative properties, we usually use the dissipation function of Rayleigh Φ_R, which characterizes the intensity of the change in the total energy of the system E.

$$\Phi_R = -\frac{1}{2}\frac{dE}{dt} = \frac{1}{2}\sum_{i=1}^{H}\sum_{v=1}^{H} b_{jv}\,\dot{q}_j\dot{q}_v$$

However, for frequency-independent dissipation the coefficients b_{jv}, are unknown, and as the task is to change non-linear system by linear equivalent, here approximate approach is used.

Let us introduce the normal (main) coordinates $\theta_r, r = \overline{1,H}$

$$q_j = \sum_{r=1}^{H} \beta_{jr}\,\theta_r$$

where β_{jr} - are the shape coordinates, determined without taking into account the dissipative forces.

Then the system of differential equations (4) can be written as follows

$$\boldsymbol{a}^*\ddot{\boldsymbol{\theta}} + \boldsymbol{c}^*\boldsymbol{\theta} = \boldsymbol{Q}^* \qquad (5)$$

where $\boldsymbol{a}^* = \boldsymbol{\beta}^T\boldsymbol{a}\boldsymbol{\beta}$, $\boldsymbol{c}^* = \boldsymbol{\beta}^T\boldsymbol{c}\boldsymbol{\beta}$ – are the diagonal matrixes of inertial and quasi-elastic coefficients after the transition to the normal coordinates; $\boldsymbol{q}, \boldsymbol{Q}^*$ - are the vector-matrixes of the normal coordinates and non-conservative generalized forces; $\boldsymbol{\beta}$ – is the matrix of the shape modes coefficients.

When drawing up equation (5) the assumption of the absence of dissipative connections, between different oscillation modes was used [8].

Let's introduce in every equation of the system (5) equivalent dissipative force $R_r = -b_r^*\dot{\theta}_r$. Then

$$a_r^*\ddot{\theta}_r + b_r^*\dot{\theta}_r + c_r^*\theta_r = F_r^*(t),$$

where

$$F_r^* = \sum_{j=1}^{H} Q_j\,\beta_{jr}$$

Dissipation coefficient ψ_r^*, corresponding to r mode, is determined as

$$\psi_r^* = \textstyle\sum_{j=1}^{H}\psi_j\, c_j\beta_{jr}^2 / \sum_{j=1}^{H}\beta_{jr}^2.$$

Hence with (3) r – dissipative coefficient is determined by formula

$$b_r^* = \frac{\psi_r^* c_r}{2\pi k_r}, r = \overline{1,H}, \qquad (6)$$

where

$$k_r = \sqrt{c_r/a_r}, r = \overline{1,H}$$, - r - natural frequencies of the system.

Example. Let's consider the definition of the dissipative coefficients of the crank press (figure 1). Dynamic model of the crank press (figure 2), we will count on the software package SimulationX [20]. Figure 5 shows the dynamic model of the crank press on the software package SimulationX [21-24]. The drive shaft of the press rotates at a speed of $\omega = 80\ rad/c$.

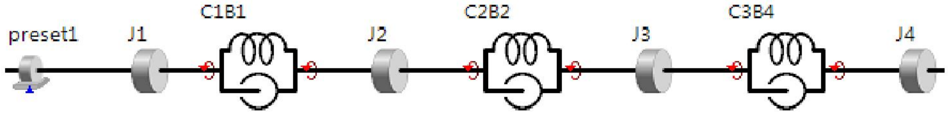


Figure 5 – The crank press model on SimulationX

Initial data:
moments of inertia of disks

$$J_1 = 0.5\ kg \cdot m^2, J_2 = 10\ kg \cdot m^2, J_3 = 5\ kg \cdot m^2, J_4 = 0.6\ kg \cdot m^2,$$

stiffness coefficients

$$c_1 = 0.16 \cdot 10^4 N \cdot m/rad, c_2 = 3.5 \cdot 10^5 N \cdot m/rad, c_3 = 3.5 \cdot 10^5 N \cdot m/rad$$

Let's take as the assumed initial values of the dissipation coefficients

$$b_1 = 50\ N \cdot m \cdot c/rad, b_2 = 60\ N \cdot m \cdot c/rad, b_3 = 60\ N \cdot m \cdot c/rad. \quad (7)$$

Determine the natural frequencies of the system (figure 6).

Natural Frequencies and Modes\Natural Frequencies
Eigenvalues

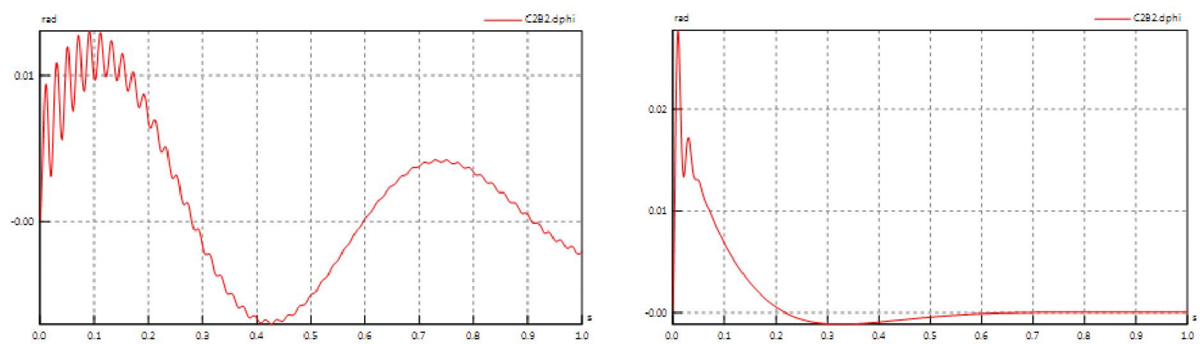
No.	Value	f [Hz] (undamped)	f [Hz] (damped)	D [-]	Time Constant [s]
f1	-0.35855±10.118 i	1.6113	1.6103	0.035414	2.789
f2	-1.0132±310.26 i	49.38	49.379	0.0032658	0.98694
f3	-5.5907±813.69 i	129.51	129.5	0.0068706	0.17887

Figure 6 – The natural frequencies of the system

Take the values of dissipation coefficients from work [9], $\psi_1 = \psi_2 = \psi_3 = 0.5$.
Determine revised values of the dissipation coefficients by formula (6),

$$b_1 = 237\ N \cdot m \cdot c/rad., b_2 = 557\ N \cdot m \cdot c/rad, b_3 = 214 N \cdot m \cdot c/rad. \quad (8)$$

Revised values of dissipation coefficients (8) substantially differ from the initial values (7).
Figure 7 shows oscillation charts of the 4th disk (of the crank press actuator) (figure 7a) with the initial and revised values of the dissipation coefficients (figure 7b).



(a) (б)

Figure 7 – Oscillation of the 4th disk (of the crank press actuator): a) with initial and б) revised data of dissipation coefficients

From the graphic charts figure 7 a, б it can be seen that the dynamics of the executive mechanism (actuator) of the crank press differ greatly because of the correction of the dissipation coefficients.

Conclusions. The method of determination of the dissipation coefficients of oscillatory system with many degrees of freedom was worked out.

The method is based on the assumption that nonlinear dissipative forces have negligible effect on the natural frequencies and oscillation modes.

Certainty of determination of dissipation coefficients depends on values of dissipation coefficients only, which must be determined experimentally, taking into account the design of the assembly units of mechanism.

Method of determination of dissipation coefficients of oscillatory system with many degrees of freedom was applied to refine the coefficients of dissipation of the crank press.

Dynamics of oscillations of the drive shaft of the crank press differs after the refinement of the dissipation coefficients.

This method of determination of dissipation coefficients is suitable for various machines with cyclic mechanisms.

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А. Қ. Тулешов, А. А. Джомартов

Ө. А. Джолдасбеков атындағы механика және машинатану институты, Алматы, Қазақстан

ҚОСИІНДІ БАСПАҚТЫҢ ДИССИПАЦИЯЛЫҚ ПАРАМЕТРЛЕРІН АНЫҚТАУ

Аннотация. Жұмыста қосиндік баспақ түйінінің диссипаттық коэффиценттерінің анықталуы қарастырылады. Баспақтың қосинді-сырғақты механизмі әртүрлі бөлшектерді қалыптайды. Осы кезде баспақ қосылыстарында және түйіндерінде шамалы динамикалық күш пайда болады. Қосинді баспақтың ерекшелігі циклді соққылы күшке байланысты. Қосинді баспақтың динамикалық зерттелуі қазіргі уақытта маңызды мәселе болып отыр. Қосинді баспақтың динамикалық үлгісі көптеген еркіндік дәрежелері бар тербеліс жүйесі сияқты болып келеді. Қосинді баспақтың динамикалық үлгісі серпімді-диссипативті элементтермен жалғанған шоғырланған массалардан тұрады. Қосинді баспақтың динамикалық үлгісі есептеу үшін жетек білігінің жеке бөліктерінің диссипация коэффициенттерін анықтау керек. Диссипация коэффициенттерін анықтау үлкен қиындық туғызады. Серпімді жүйе тербелісінде машина түйіндерінде энергия шашырауы және механикалық энергияның жылу энергиясына айналуы болады. Механикалық энергияның жоғалуы серпімсіз кедергі күштердің әсерінен болады, олар диссипативті күштер деп аталады. Резонансты тербелістер амплитудасы, автотербілістер және параметрикалық тербелістерді қоздыру жағдайы диссипация коэффициентеріне байланысты болады. Диссипация коэффициенттерінің анықталуы – жүйенің жеке серпімді диссипативті элементтерінің шашырау коэффициенттері туралы қол жетімді ақпараттарға негізделген. Қосинді баспақтың диссипация коэффициенттерін анықтау бойынша мысал келтірілген. Қосинді баспақтың динамикалық үлгісі SimulationX бағдарламалық кешенінде құрылды. Алынған нәтижелерден диссипация коэффициенттері нақтыланғаннан кейін, қосинді баспақтың жетек білігінің тербеліс динамикасы түбегейлі өзгеретінін көруге болады.

Түйін сөздер: динамика, қосинді баспақ, қатаңдық, диссипация, шашырау, тербеліс, SimulationX

А. К. Тулешов, А. А. Джомартов

Институт механики имашиноведения им. У. А. Джолдасбекова, Алматы, Казахстан

ОПРЕДЕЛЕНИЕ ДИССИПАТИВНЫХ ПАРАМЕТРОВ КРИВОШИПНОГО ПРЕССА

Аннотация. В работе рассматривается определение коэффициентов диссипаций узлов кривошипного пресса. Кривошипно-ползунный механизм пресса выполняет штамповки различных деталей. При этом возникаю значительные динамические нагрузки в узлах и соединениях пресса. Особенностью работы кривошипного пресса связаны с ударными циклическими нагрузками. Динамическое исследование кривошипных прессов, в настоящее время, является актуальной задачей. Динамическая модель кривошипного пресса представляется, как колебательная система со многими степенями свободы. Динамическая модель кривошипного пресса состоит из сосредоточенных масс, соединенных упруго-диссипативными элементами. Для расчета динамической модели кривошипного пресса, требуется определение коэффициентов диссипаций отдельных участков приводного вала. Определение коэффициентов диссипаций представляет большую сложность. При колебаниях упругих систем происходит рассеяние энергии в узлах машин и происходит переход механической энергии в тепловую. Потери механической энергии вызываются силами неупругого сопротивления, называемыми диссипативными силами. От коэффициентов диссипации зависят амплитуды резонансных колебаний, условия возбуждения параметрических колебаний и автоколебаний. Определение диссипативных коэффициентов базируется на доступной информации о коэффициентах рассеяния отдельных упругодиссипативных элементов системы. Приведен пример по определению коэффициентов диссипаций кривошипного пресса. Динамическая модель кривошипного пресса была составлена на программном комплексе SimulationX. Полученные результаты показывают, что динамика колебаний приводного вала кривошипного пресса существенно изменяется после уточнения коэффициентов диссипаций.

Ключевые слова: динамика, кривошипный пресс, жесткость, диссипация, рассеяние, колебания, SimulationX.

Information about authors:

Tuleshov A. K. – Institute of Mechanics and Mechanical Engineering named after U. A. Joldasbekov, Almaty, Kazakhstan; aman_58@mail.ru; https://orcid.org/0000-0001-9775-3049

Jomartov A. A. – Institute of Mechanics and Mechanical Engineering named after U. A. Joldasbekov, Almaty, Kazakhstan; legsert@mail.ru; https://orcid.org/0000-0002-7262-5267