SOME SPECIFIC VIBRATORY EFFECTS ARISING IN MECHANISMS WITH NONLINEAR POSITION FUNCTION

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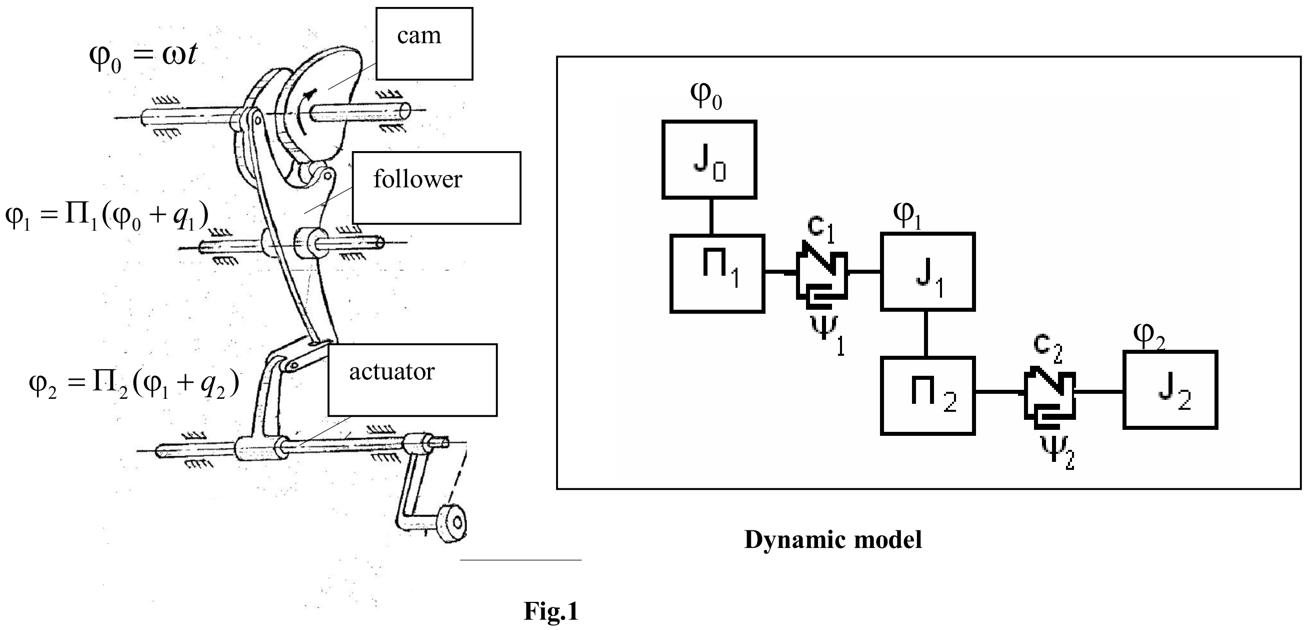
Abstract

In the paper we have presented an overview of some problems, among which we should single out the elimination of following negative dynamic effects: the effects arising from the nonlinear geometrical characteristics of mechanisms; the effect arising from the joint action of nonlinear position function and clearances; the effects of excitation vibratory regimes caused by non-stationary friction in kinematic pairs of cyclic mechanisms.

1. The effects arising from nonlinear geometrical characteristics of mechanisms

Preliminary remarks. A distinctive property of machine drives are the motion transformation and programmable displacement of actuators according to nonlinear position function $\Pi(\varphi)$, where φ is the coordinate of an input link [1,2]. The mechanisms realized programmed motion ("cyclic" mechanisms) are playing a double role in the vibratory system on the one hand being the source of vibration excitation, and on the other hand being a critical object to vibration protection.

For example, on Fig.1 is presented a typical cyclic mechanism and its dynamic model.



Dynamic model

Fig.1

Those models were created by combining the block diagrams of vibratory mechanical systems with certain kinematic analogs Π_i, which determine the kind of connections between the input and output links (Fig. 1). Here J_i, c_i, ψ_i are respectively moments of inertia, stiffnesses and energy dissipation coefficients of the corresponding links in the kinematic chain.

Any coordinate in the absolute motion φ_i is a combination of the "large" coordinate $\Pi_i(\varphi_0)$, realizing the motion of the absolutely rigid drive, and the "small" coordinate $\Delta\varphi_i$, whose ensemble corresponds to the number of vibratory system degrees of freedom *H*.

Strongly speaking, in this case the set of differential equations is nonlinear. However, with representing the position functions as truncated Taylor series the linearization in the vicinity of the current value of φ_0 is carried out. Than the position functions retained their nonlinear properties relative to the large coordinate, and only small deformations entered the corresponding expressions in a linear fashion. After the transformation to the quasinormal coordinates the original system can be described by differential equations of the form

$$\ddot{y}_r + 2n_r(t)\dot{y}_r + p_r^2(t)y = W_r(t) \qquad (r = \overline{1,H}) \ . \qquad (1)$$

Using the method of conventional (fictitious) oscillator (Vulfson, 1969) the solution y_r has the following structure [2,3]:

$$y_r = \mu_r \sum_{j=0}^{s-1} D_{jr} \exp\left[-\int_{t_j}^{t} n_r dt\right] \sqrt{\frac{\Omega_r(t_j)}{\Omega_r(t)}} \sin\Psi_{jr} + Y_{sr}, \qquad (2)$$

$$\left.\begin{aligned}
&\mu_r = \left[1 - 2\exp\left(-\vartheta_r \bar{N}_r\right)\cos 2\pi\bar{N}_r + \exp\left(-2\vartheta_r \bar{N}_r\right)\right]^{-0,5};\\
&Y_{sr} = \frac{1}{\sqrt{\Omega_r(\tau)}} \int_{t_{s-1}}^{t} \frac{W_r(u)}{\sqrt{\Omega_r(u)}} \exp\left[-\int_u^t n_r d\xi\right] \sin\left[\int_u^t \Omega_r(\xi)d\xi\right] du;\\
&\Psi_{jr} = \int_{t_j}^{t} \Omega_r(\tau)d\tau,
\end{aligned}\right\} \qquad (3)$$

where $\Omega_r(t) = \bar{p}_r \exp z_r$ is conventional "natural" frequency; $\bar{p}_r$ — an optional parameter with the dimension of a frequency; $N_r = \bar{p}_r/\omega$.

According to the method of conventional oscillator the relation between the function z_r and variable frequency $p_r(t)$ will have the form of the following differential equation, responding to the particular "conventional oscillator" with the excitation $2p_r^2(t)$ [2,3]:

$$\ddot{z}_r - 0,5\dot{z}_r^2 + 2\bar{p}_r^2 e^{2z} = 2p_r^2(t). \qquad (4)$$

By slow changes of $p_r^2(t)$ the dynamic components in the Eq:(4) is small in relation to the static one. For this case the solution (2) corresponds to the *WKBJ* approximation method $(\Omega_r \approx p_r)$..

Violating of dynamic stability conditions by slow change of parameters. According to (2) the amplitude of free and accompanying vibrations excited is changing proportional to the function

$$S_r = p_r(t)^{-0,5} \exp[-\int_0^t n_r(\xi)d\xi]. \qquad (5)$$

With changing parameters it may happen that $dS_r / dt > 0$, therefore the customary decrease of amplitudes can be disturbed. In the similar case the amplitude modulation exists, where the zone of decrease alternates of the zone of increase. Therefore, contrary to a parametric resonance we do not experience the unlimited increase of amplitudes. Under some unfavorable conditions the increase of amplitudes may become rather intensive. Using (5) the dynamic stability conditions on any time interval can be written as [2–5]

$$n_r + 0{,}5\dot{p}_r / p_r > 0. \qquad (6)$$

It is possible to show that condition (6) can also be obtained by the direct Lyapunov method and is, consequently, the sufficient condition for asymptotic stability. Compliance with this condition removes also the possibility of the build-up in the zones of the main parametric resonances.

Using the method of conventional oscillator a dynamic effect caused by a sudden temporary change in the “natural” frequency of a system (so called "parametric impulse") is considered in [2,6].

2. The effect arising from the joint action of nonlinear position function and clearances.

For cyclic mechanisms the clearance-effect leads to possibility of vital distortion of kinematical characteristics and increase the drives vibroactivity. Two cases are revealed. In the first case the clearance proves as a nonlinear element to which a possibility of generating vibratory impact modes is connected. In the second case reaction to a clearance manifests itself as an impulse in linear systems. This dynamic effect is equivalent to impact arising from disruption of a continuity of the function $d\Pi / d\varphi_0$. Some dynamic criterions that allow forecasting the excitation of vibratory impact regimes are offered [5–9].

In the linkages the clearance effect sometime softens due to the conjugate action between the contracting surfaces of hinge (Fig. 2). On many researches of this problem it is supposed, that the vibration excitation at elimination of breaks of the kinematic contact in clearances do not arise. However at parametrical pulses the arising effect is close to impact. This effect, named *pseudo-impact*, under certain conditions is transformed to the impact with disruption of contact of a kinematic circuit.

The offered model of clearance-joint (Fig. 3) is submitted as a pendulum that oscillates in a rotating power field about the elastic support [9]. The analysis of this model allows defining conditions of stability on the limited time intervals and critical values of parameters of system at which the excitation close to impact takes place.

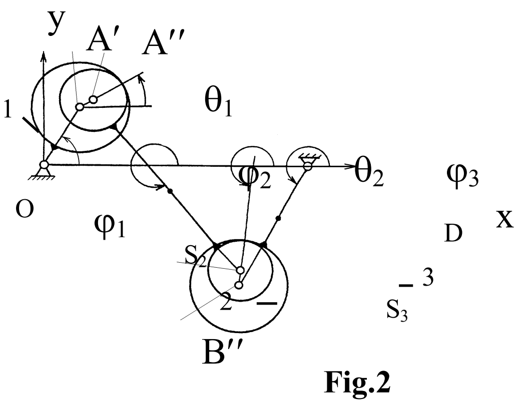


Fig.2

B′

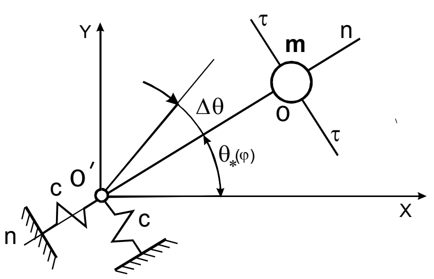


Fig.3

The relation between linearized tangential and normal stiffnesses of "links" $A'A'', B'B'' - c_i^\tau, c_i^n$ can be presented as [9]

$$c_i^\tau / c_i^n = \left|R_i^*\right| / (0,5 c_i^n s_i + \left|R_i^*\right|) \quad , \tag{7}$$

where R_i^* (i=1,2) is the reaction in the corresponding hinge calculated at the kinetostatic level; s_i is the value of clearance.

In order to obtain comparable results in varying the system parameters the quasielastic coefficients were scaled by the stiffness coefficient $c_0 = J_3 p_0^2 / l_3^2$, where p_0 is the partial frequency realized conditionally for a zero pressure angle and clearance-free elastic joint between coupler and the rocker; J_3 is the reduced moment of inertia of the rocker; $l_3 = BD$. For the closed kinematic chain an increase of natural frequency parameter $\eta_0 = p_0 / \omega_1$ results in the slow frequency variation being interrupted by intense parametric pulsations. In turn this may result in a considerable growth in the level of dynamic errors and the vibration activity of the mechanism [9].

The dynamic effect under analysis is illustrated on Fig.4 by the plots of functions φ_{2*}'', φ_{3*}'', which are proportional to the ideal angular accelerations of coupler and rocker, and φ_2'', φ_3'', calculated taking into account the clearances and elasto-dissipative properties of hinges. We see that at a rather high value of $\eta_0 = 50$ (Fig.4a) accompanying vibrations are intensively excited in the zones of parametric pulses. At $\eta_0 = 10$ (Fig.4b) the effect is considerably attenuated. An analysis shows that the dynamic stability of the system over a finite time interval is an important factor. The corresponding sufficient conditions (7), based on the method of conditional oscillator, lead to the form [2]

$$\vartheta_r > \left|\vartheta_r^0\right| = \left|\ln(\eta_r^+ / \eta_r^-)\right|, \tag{8}$$

where $\vartheta_r, \left|\vartheta_r^0\right|$ are logarithmic decrement and its critical value for the mode r,

$\eta_r = \eta_r(\varphi_1 + \Delta\varphi_r)$, $\eta_r = \eta_r(\varphi_1 - \Delta\varphi_r)$, $\Delta\varphi_r = \pi / \eta_r(\varphi_1)$.

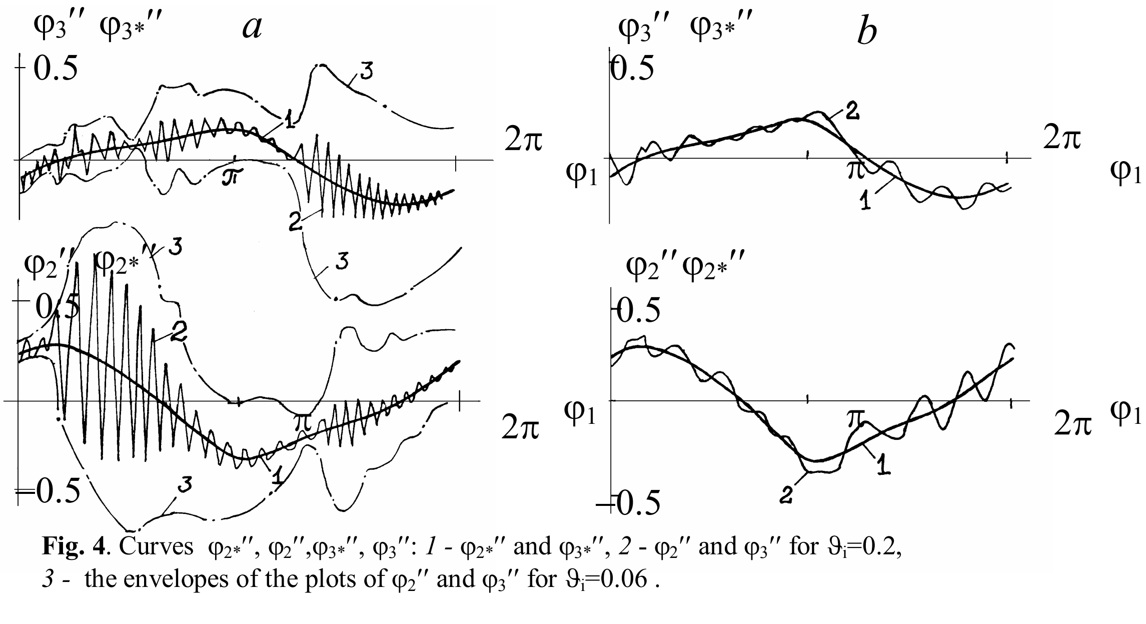


Fig. 4. Curves φ_{2*}'', φ_2'', φ_{3*}'', φ_3'': *1* - φ_{2*}'' and φ_{3*}'', *2* - φ_2'' and φ_3'' for ϑ_i=0.2,
3 - the envelopes of the plots of φ_2'' and φ_3'' for ϑ_i=0.06 .

Real damping is determined by the effective value of the logarithmic decrement $\vartheta_r^* = \vartheta_r - \left|\vartheta_r^0\right|$. The value $\max \vartheta_r^0$ can be used as an efficient criterion for characterizing the level of vibration. The envelopes shown in Fig.4a (curves 3) correspond to the logarithmic decrement decreasing from ϑ_i =0,2 to 0,06. We see that a decrease in ϑ_i intensifies the dynamic instability and vibration excitation.
For η_0=10 and ϑ_i =0,2 ,ϑ_i^* remains nearly zero for a rather long time, thus compensating the dissipative factors (Fig.4b).

The transformation the pseudo-impact to impact is evidently visible at comparison of phase trajectories (Fig.5)

Pseudo-impact Impact

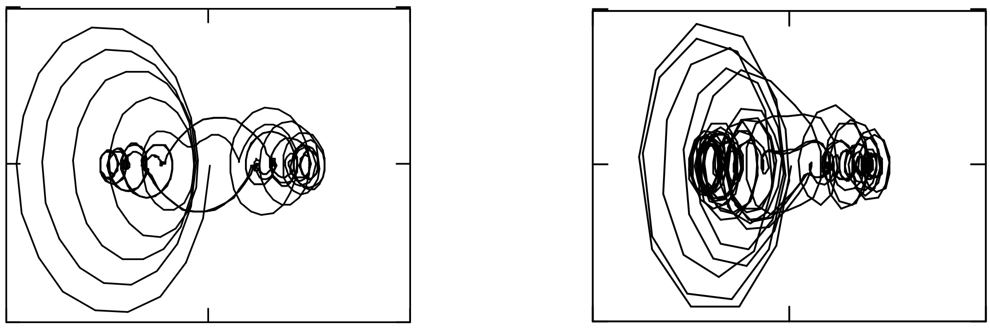


Fig. 5

3. The effect of excitation owing to fall out of synchronism in multi-sections drives with lattice structure.

The researched effect is observed in drives with regular structure, which are used in machines for realization of repeating technological and transport operations [2,5,10]. The theory of regular oscillatory systems is based on the analysis of the lattices consisting of masses and springs. For the first time this problem was considered by Born and Karman with reference to the analysis of the heat capacity of crystals. The basic directions of the further development of this theory are reflected in [11].

With reference to machines with cyclic mechanisms the theory of regular oscillatory systems requires additional development. First, dynamic models of drives have more complex internal structure of each repeating module which formed not only a simple connected chain, but also brunched and ring structured vibratory systems with nonlinearities and nonstationary dynamic connections [2,5,10,12]. Should be noted, that in some cases the conditions of regularity are realized only approximately.

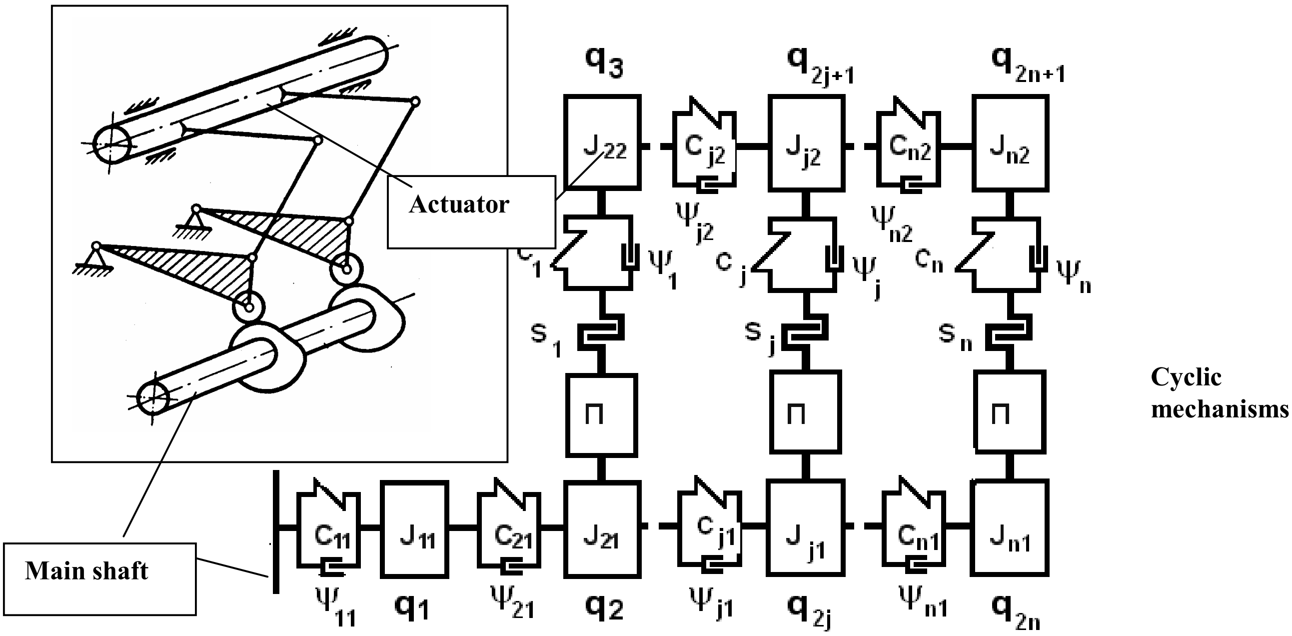


Fig. 6

The dynamic model of a drive (Fig. 6), consisting of subsystems of the main shaft ($k=1$) and an actuator ($k=2$) connected to the main shaft with n cyclic mechanisms is considered. Each of mechanisms is submitted as consecutive connection of the elements, which are taking into account elastic-dissipative, inertial, and kinematic characteristics, and clearances. The following symbols are accepted: $J_{j,k}$ - the moments of inertia; $c_{j,k}$, c_j - factors of rigidity; $\psi_{j,k}$, ψ_j - factors of dispersion; $\Pi(\varphi_{j,1})$ - function of position. It is supposed, that dynamic characteristics of the main shaft and an agency are given to sections of entrance and target parts of cyclic mechanisms. Besides angular speed ω on "input" is accepted by a constant, that usually as a first approximation corresponds to real machines at a rational choice of characteristics of the electric drive and transfer mechanisms. The considered oscillatory system has $2n+1$ degrees of freedom. As the generalized coordinates we shall accept the dynamic mistakes equal to deviations of absolute coordinates in the appropriate sections of inertial elements from coordinates of program movement.. Thus for the main shaft $q_1=\varphi_{1,1}-\varphi_0$, $q_{2(j-1)}=\varphi_{j,1}-\varphi_0$, where $\varphi_0=\omega t$, $j=\overline{1,n+1}$, and for an actuator – $q_{2j-1}=\varphi_{j,2}-\Pi(\varphi_0)$ $(j\geq 2)$. The accepted dynamic model is described by the set of nonlinear differential equations with slowly varying factors [12]. At $\Pi'=r_0\sin\varphi_0$ the computer simulation researched was carried out with variation of number of identical mechanisms n and other parameters of system [12].

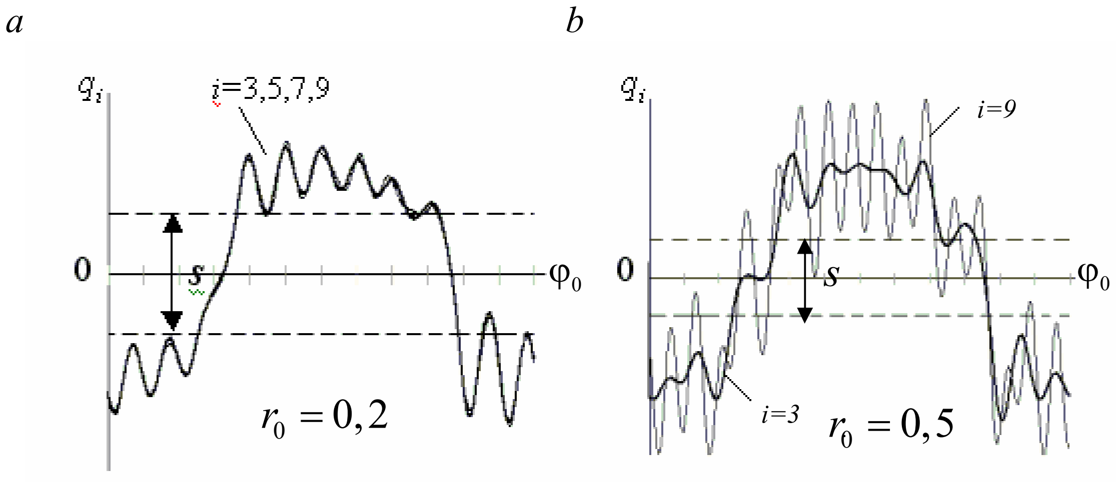


Fig. 7

Under favorable conditions the queasy-synchronous mode of actuators vibrations is realized (a mode of type 1, Fig.7a). At this form the elastic element c_{j2} is not deformed. For modes of type 1 the "natural" frequency is close to a case with $c_{j1} \to \infty$ (the rigid main shaft) and are described by dependence

$$p_m \approx p\sqrt{1 + 2\varsigma_2[1 - \cos((m-1)\pi / n)]}, \quad (m = \overline{1, n}; \ \varsigma_2 = c_{j2} / c_j) \qquad (9)$$

Synchronous moving corresponds to the lowest frequency (m=1).

The conditions at which the synchronous form of vibrations of an output link is strongly broken were received. Usually it corresponds to the most deformed site of the main shaft (a mode of type 2, Fig.7b). In this case the plots q_i for first three mechanisms $(i = 3,5,7)$ differ a little, but for the coordinate q_9 not damping vibrations and violation of synchronism are observed.

Strong additional excitation of the mechanism n is arise due a specific influence of the subsystem formed by the previous mechanisms, which energy is partially "pumped over" in the asynchronous form. Thus the violating of dynamic stability conditions on finite time intervals plays a large role (see above). At vibratory impact modes the spectrum of frequency can be change, and the level of vibration increase. For analysis of these phenomena the method harmonious linearization of force was used [12].

One of ways for the constructive decision of the problem for elimination of asynchronous modes with large amplitudes is the transition to branched-ring structured drive, where separate sections replace the long actuator with limited number of mechanisms.

4. The effects of excitation vibratory regimes caused by non-stationary friction in kinematic pairs of cyclic mechanisms.

Preliminary remarks. As it was shown above one of sources of accompanying vibration are the distortions of continuity of power or kinematic perturbations and collisions by reversals in clearances [2], [5], [8]. The other source of accompanying vibrations is connected with the forces of friction.

The study of frictional oscillations in mechanisms is limited more often to case, when the coefficient of friction in time of rest differs from the coefficient in time of motion. However modification of these coefficients is the reason of self-oscillations only with small relative velocities between rubbing surfaces. Meanwhile in mechanisms, which realize the programmed motion of operating members, the frictional self-oscillatory regimes can arise even in the case with large velocities.

Usually it happens when the drive include the self-locking mechanisms (for example, - worm- and-wheel mechanism). The similar situation came into existence in cyclic mechanisms with “nonreversible” kinematic pairs, in particular, when the line of the reaction does not coincide with direction of the force of friction in prismatic kinematic pair [13] .

Dynamic model. The elementary mechanisms, in which the dynamic effects under study are best demonstrated, are shown in Fig.8. We have considered these effects by the example of a dynamic model of an output link of the slider-crank mechanism (Fig.9). The following symbols here are accepted: c_o, ψ_0 - coefficients of stiffness and dissipation; *S*-center of the masses of slider ; $P=-m\Pi''\omega^2-P_0$, where P_0- external force; $\Pi''=d^2\Pi/d\varphi^2$; $\Pi(\varphi)$ - position function of slider; $\omega=d\varphi/dt$; φ - the coordinate of input link; h_0, h_1, h_2 - constructive sizes show on Figure; α - angle of pressure.

The forces of friction in quads is equal

$$F=F_1+F_2= - \mu_0 \, sign\, v_0 |R| \, \varsigma(h_0, h_1, h_2, \alpha) cos\alpha \; , \qquad (10)$$

where μ_0 - coefficient of friction; v_0 - velocity of the slider in rigid mechanism; R - reaction in the hinge B (in the assumption of relative small forces on a coupler).

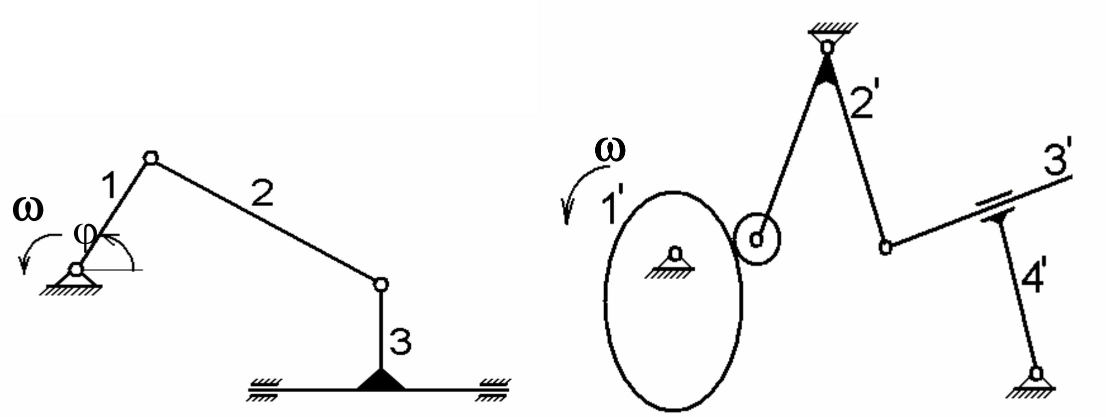


Fig. 8. Examples of the mechanisms with “nonreversible” kinematic pairs

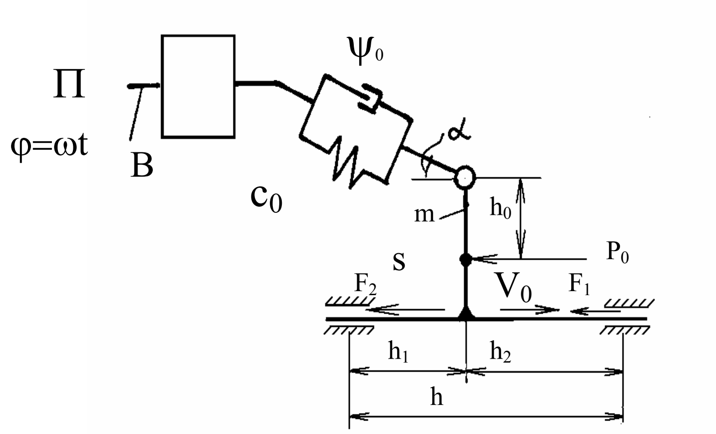


Fig. 9. **Dynamic model**

According to (10) the vibration of slider are described by the nonlinear differential equation

$$m\ddot{q}+b(\varphi)\dot{q}+c(\varphi)q=-\mu_0\zeta\cdot \mathrm{sign} v_0\big(b(\varphi)\dot{q}+c(\varphi)q\big(-m\Pi''\omega^2-P_0, \qquad (11)$$

where $q = x - \Pi(\varphi)$ — the deviation of the coordinate of slider from the programmed motion;

$c = c_0 \cos^2 \alpha$ — the reduced coefficient of stiffness; b — coefficient of equivalent linear resistance.

Let us next reduce equation (11) to the following form:

$$q'' + \Psi(q, q', \varphi) = -\Pi''(\varphi) - w_0(\varphi), \qquad (12)$$

where $(\ )' = d/d\varphi$; $\varphi = \omega t$ —"nondimensional time"; $\Psi = 2n_0 q' + k_0^2(1 + \mu \cdot \mathrm{sign}v \cdot \mathrm{sign}q)$; $2n_0 = b/(m\omega^2)$; $\mu = \zeta\mu_0$ — reduced coefficient of friction, $w_0 = P_0/m$; $v = \Pi' + q'$ (here is accept $\mathrm{sign}(b\dot{q} + cq) \approx \mathrm{sign}q$, that occurs practically with real values of parameters).

On such mechanisms the "design" parameter $\varsigma \approx 2h_0/h > 1$, therefore $\mu > \mu_0$.

Frequency and dissipative characteristics. By the approximated solution of equation (12) and by analysis the combination of methods of harmonic linearization and fictitious oscillator was used [3]. The right-hand side of the equation (12) is a slowly varying function, therefore the approximate solution we shall seek as

$$q = A_0(\varphi) + A(\varphi)\sin\theta. \qquad (13)$$

Here $A_0(\varphi), A(\varphi)$ - slowly varying functions; θ - fast varying phase, corresponding to vibrations with slowly varying "natural" frequency.

Based on the solution (13), let us present the nonlinear function Ψ as

$$\Psi \approx \Psi_0 + \Psi_C \cos\theta + \Psi_S \sin\theta, \qquad (14)$$

where Ψ_0, Ψ_C, Ψ_S - the coefficients, which are depending on "slow time"φ:

$$\Psi_0 = \frac{1}{2\pi}\int_0^{2\pi} \Psi(\varphi,\theta)d\theta; \quad \Psi_C = \frac{1}{\pi}\int_0^{2\pi} \Psi(\varphi,\theta)\cos\theta d\theta; \quad \Psi_S = \frac{1}{\pi}\int_0^{2\pi} \Psi(\varphi,\theta)\sin\theta d\theta.$$

The summarized velocity $v = \Pi' + q'$ changes sign only in a small vicinity of values φ that are corresponding to change of a sign of Π'. If the sign q does not vary, than on the given part of kinematic cycle $n = (1 \pm \mu)n_0$ and $k^2 \approx (1 \pm \mu)k_0^2$; here $2n = \Psi_c/(Ak)$, $k^2 = \Psi_s/A$; $k_0^2 = c/(m\omega^2)$; $n_0 = \psi k_0/(4\pi)$. Further let us take a look at a more complicated case, when q of the period of "fast' motion changes the sign.

Let us consider a new variable $\gamma = \arcsin(A_o/A)$. On the period $\theta \in [0, 2\pi]$ the change of sign q happens twice, namely with $\gamma = \gamma_1$ and $\gamma = \gamma_2$. Than after some simplifications by a slow modification of functions $A(\varphi)$, $A_0(\varphi)$ we get

$$A_0 = Wk_0^{-2}[1 + \frac{2}{\pi}\mu\,\mathrm{sign}v(\gamma + 0{,}5\sin 2\gamma)]^{-1}; \qquad (15)$$

$$k^2 = k_0^2[1 + \frac{2}{\pi}\mu \cdot \mathrm{sign}v(\gamma + 0.5\sin 2\gamma)]; \qquad (16)$$

$$2n = 2n_0 - k_0^2\mu \cdot \mathrm{sign}v \cdot \sin 2\gamma \cdot \sin(0.5\Delta\gamma)/\pi k, \qquad (17)$$

where γ, $\Delta\gamma = \gamma_2 - \gamma_1$ the average value and increment of this function an a period of a "fast" motion $\tau = 2\pi/k$; $W = -\Pi'' - w$; $W' = dW/d\varphi$. The requirement sign v=const during the period τ reduces to the following condition:

$$|\gamma| > \arcsin|W/(k^2\Pi')| \qquad (18)$$

In the small-sized zones, which are in a vicinity of change of sign v a force of friction exhibits itself as a strong dissipative factor.

The plots of nondimensional "natural" frequency $f_1 = k^2/k_0^2$ are listed on Fig.10.

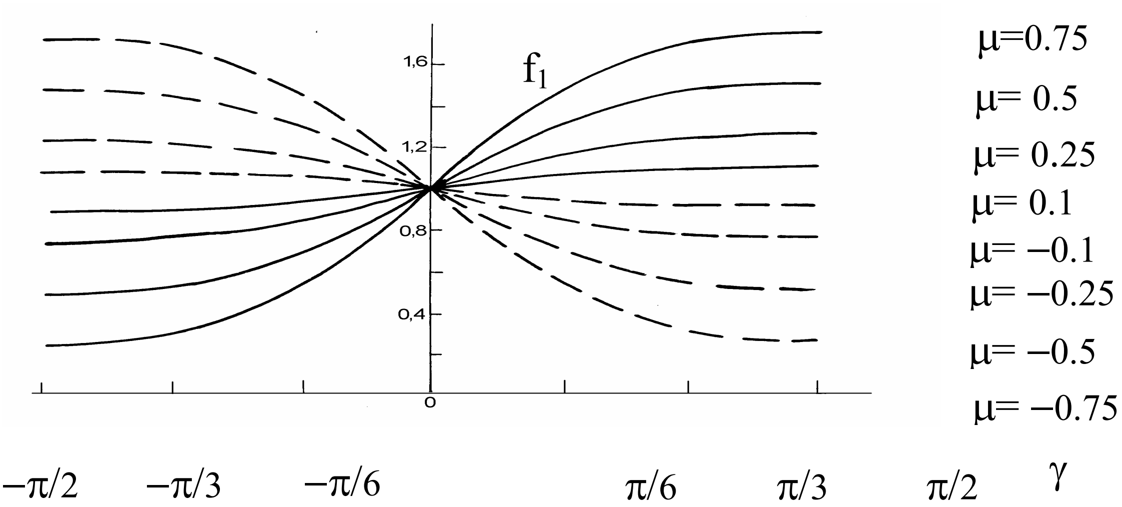


Fig. 10. Curves $f_1(\gamma,\mu)$

With purposes of convenience of the analysis the sign A_0 is taken into account directly in function γ, therefore further we shall accept $A_0=|A_0|$. Besides we shall consider $\mu>0$, when the force of friction is directed against a velocity of an output link, and $\mu<0$ - on the direction of a velocity. Both these cases are illustrated in Fig.1.

With application of (15)–(17) alongside with condition (18) the condition $|A_0|\le A$ should be also observed. This condition testifies that are in existence a critical value of amplitude $A=A_*$, depending from exterior disturbances, below which the nonlinear properties of a system practically are not exhibited.

Conditions of excitation of self-occultation regimes. Let us introduce into consideration an equivalent instantaneous value of logarithmic decrement

$$\vartheta=\vartheta_0+\vartheta_\mu+\vartheta_\nu. \qquad (19)$$

Here $\vartheta_0=0{,}5\psi_0$ - corresponds to dissipation of the drive; ϑ_μ to the dissipation at Coulombic friction; $\vartheta_\nu=-2\pi(0{,}5-\nu)k^2dk/d\varphi$, where ν=0,1,2 corresponds to vibration's motion, velocities, and accelerations [2], [3]. [5], [10]. (The parameter ϑ_ν is taking into account a modification of amplitudes owing to variability of frequency.)

The main objective of this material is to study the nonlinear vibrations connected with friction. Because of this the influence of a last addend of formula (10) that corresponds to $k_0=p_0/\omega$=const, may be disregarded. Based on the (6)-(8),(10) in such cases we have

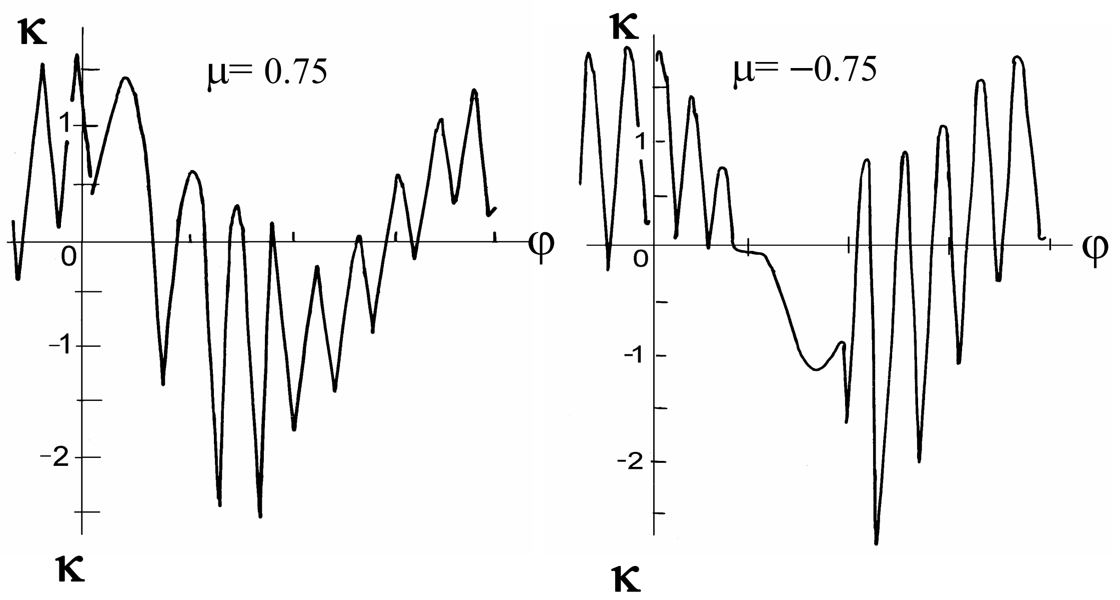
$$\vartheta=\vartheta_0-\frac{\pi f_2 W'\,\mathrm{sign}\nu}{k_0\left(W\left((f_1+2\mu\cdot\sin 2\gamma\cdot\mathrm{sign}\nu/\pi\right)\right.}-2\pi(\nu-0.5)k_0^{-2}\frac{dk_0}{d\varphi}\ , \qquad (20)$$

where $f_2=0.5\mu\cdot\sin 2\gamma[\sqrt{f_1}\sin\gamma-(\nu-0.5)/\pi]$,

As following from (20), under certain conditions there come into being a possibility of "negative damping", when $\vartheta<0$. In this case the conditions of dynamic stability an any time interval of the kinematic cycle are disrupted [2], [5]. The variability of the sign of "damping" come to the existence a specific quasiperiodic regime, which can be treated as self-oscillations supported by nonstationary friction.

For scotch-yoke mechanism the following correction should be adopted: there is a need to insert into formula (10) $\zeta=\zeta_{max}|\sin\varphi|$ instead of ζ=const.

Computer simulation of self-oscillation regimes. For detection of qualitative peculiarities of the established vibration regimes and to test the validity of obtained analytical dependencies the computer-simulation was carried out over a wide range of variation of parameters. On Fig.11 the plots of function $\kappa(\varphi)=x''/\Pi''$ for typical dynamic regimes are listed. The function $\kappa(\varphi)$ characterizes the ratio between accelerations allowed for vibration and with a given programmed motion. The following input dates were here accepted: $\Pi''=5{,}236\cdot10^{-2}\cos\varphi$, m; $k_0=p_0/\omega=10$; $\vartheta_0=0{,}2$.
With $\mu=0{,}75$ as we might expect according to the formula (7),the lowering of “natural” frequency occurs when $0<\varphi<\pi/2$; $\pi<\varphi<3\pi/2$, and with $\mu=-0{,}75$ when $\pi/2<\varphi<\pi$; $3\pi/2<\varphi<2\pi$.
According to the formula (11) distortions of the dynamic stability and the excitation of non-stationary self-oscillations is possible with $\mathrm{sign}(f_2 vW')=1$. The similar situation for the most part occurs in the first and second quadrants with $\mu>0$, and in the second and fourth - with $\mu<0$.
With $\mu=0{,}5$ it is possible to upset the self-oscillations with insignificant modifications of parameters. On this case in zone of sign change of velocity the force of friction plays a role of essential dissipative factor.
With $\mu=0{,}95$ besides highly essential distortions of programmed motion occurs well-defined relaxation self-oscillations. However, these regimes, which were close to self-locking, do not represent a practical interest.
In the case with installation the spring unloader between a slider and housing with a stiffness factor $c_u=m\omega^2$, we have $\kappa<1{,}25$. Here it is possible to realize the acceptable conditions of operation even in the case when exists strong relaxation self-oscillations with absence of a dynamic unloading.



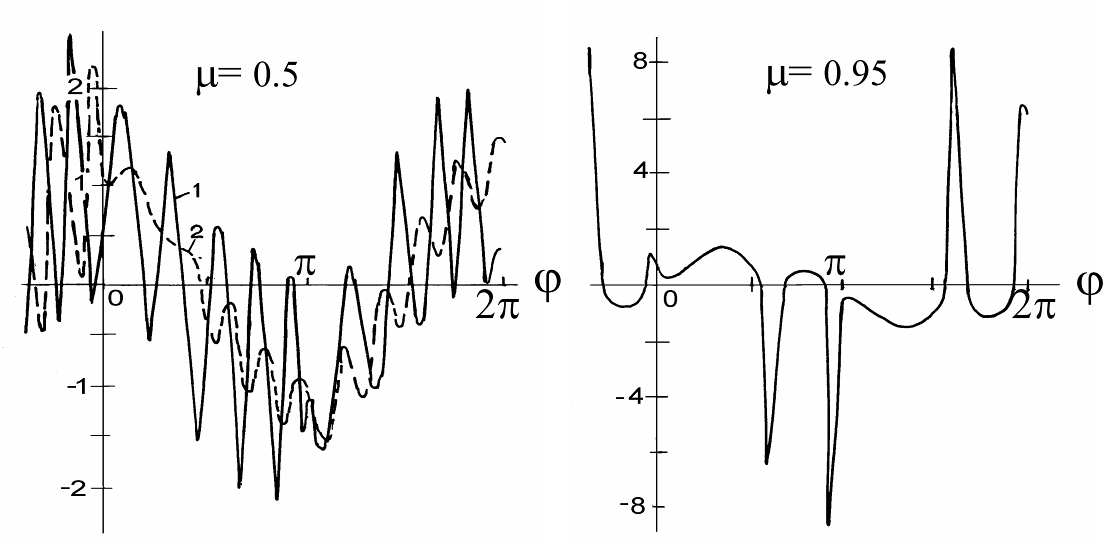


Fig. 11. Plots $\chi(\varphi)$

Hence, it is determined, that the variable force of friction on defined phases of a kinematic cycle can have not only dissipative character, but also play a destabilizing role. In such cases the conditions of a dynamic stability can be broken, and self-oscillations can arise.

REFERENCES

1.Vulfson I.I., Kolovsky M. Z., Nonlinear Dynamic Problems of Machines (in Russian). Izdat Mashinostroenie, Leningrad (1968).

2. Vulfson I.I. Vibrations of machines with cyclic mechanisms [in Russian], Mashinostroenie, Leningrad, 1990.

3. Vulfson I.I. Vibrations of system with time-dependent parameters (in Russian). Prikladnaya matematika i mechanika. 1969, №2. V.33. Pp. 331–337.

4. .Vul'fson I.I. Analytical investigation of the vibration of mechanisms caused by parametric impulses. //Mechanism and Machine Theory. Vol.10, 1973. P.p. 305–313.

5. Dresig H., Vul'fson I.I.: Dynamik der mechanismen, Springer, Wien, New York 1989.

6. Vulfson I.I. Dynamic effects and criterions of vibroactivity of cyclic mechanisms with clearances [in Russian]. VNTR. N2, 2007.

7. Senevirtane L.D., Earles.S.W. Chaotic behavior exhibited during contact loss in a clearance joint of four bar mechanism. Mech. Mach. Theory, vol. 27, N3, 1992. Pp. 307–321.

8. Vulfson I.I. Optimization of the parameters of the vibration systems of cyclic mechanisms taking clearances into account. Journal of Machinery Manufacture and Reliability, N3, 1994. Pp.9-16.

9. Vulfson I.I., Preobrazhenskaya M.V. Parametric pulses during shockless reversals in linkage clearances. Journal of Machinery Manufacture and Reliability, N5, 1995. Pp.18–23.

10.Vulfson I. Vibroactivity of branched and ring structured mechanical drives. New York, London: Hemisphere Publishing Corporation, 1989.

11. Landa P.S. Nonlinear oscillations and waves in dynamical systems. Dordrecht-London: Kluwer Academic Publ.,1996.

12. Vulfson I.I., Preobrazhenskaya M.V. Research of the oscillatory modes arising in clearances of cyclic mechanisms, connected to the common actuator. Journal of Machinery Manufacture and Reliability, N1, 2008. Pp.29–37.

13. Vulfson I.I. Nonlinear vibration in cyclic mechanisms induced by nonstationary friction forces in kinematic pairs. Journal of Machinery Manufacture and Reliability, №4, 1999. Pp.20–26.

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