

COMPLEX APPROACH TO PROVIDING OF CHAIN DRIVES QUALITY

КОМПЛЕКСНЫЙ ПОДХОД К ОБЕСПЕЧЕНИЮ КАЧЕСТВА ЦЕПНЫХ ПЕРЕДАЧ

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Abstract: Complex approach to design and realization of chain drives is presented, including in itself account of the real dynamic processes in time their operation, application of integrated parts from polymeric composites and computer-aided design.

KEYWORDS: CHAIN DRIVES, DYNAMICS, INTEGRATED PARTS, COMPUTER-AIDED DESIGN

1. Introduction

A multimass chain drive is the system, which consists of interactive and intercaused indivisible elements, has many possible realization in the process of functioning and that is why behaves to the complicated systems.

The new approach to calculation and designing of chain drives must be based on the account of the real dynamic processes which take place during their work, use of polymeric composites for making of sprockets and chains and change-over to the automated optimal design, which will enable to choose the aggregate of values of their parameters, at which yet on the stage of design high dynamic quality of transmissions and drives will be provided.

The aim of the article consists in complex illumination of questions which touch upgrading of chain drives quality.

2. Previous publications

In [1] scientific bases of synthesis of chain-drives are systematic expounded on the base of new conception of their design which consists, first of all, that a chain-drive is considered as an oscillating system. Work of chain-drive is described by differential equations of torsion, transversal, parametric and longitudinal vibrations, the solution of which on PC by numerical methods gives self-resonant frequencies and forms of vibrations of transmission with practically any amount of the masses (sprockets).

In further works [2-15] found the illumination of questions of the practical making of chains from polymeric composites, research of them stress-strained state, development of theory of chain transmissions in direction of optimization of them structural, parametric and operating specifications.

Last achievements, which touch the questions of the high-quality design, making and operating of chain transmissions from the point of view of the developed theory and application of the newest materials and resource-saving technologies expounded in monograph [16]. Others developments and results of the already done works, touching the complex providing of quality of chain-drives are further presented.

3. Results of research of the dynamic system of chain-drive by frequency methods

Mathematical model of work of n -mass of chain-drive can be written as follows:

$$I_i \ddot{\varphi}_i + \eta(\dot{\varphi}_i R_i - \dot{\varphi}_2 R_2) R_i + c_{1,2}(\varphi_1 R_1 - \varphi_2 R_2) R_i + \eta(\dot{\varphi}_1 R_1 - \dot{\varphi}_n R_n) R_i + c_{1,n}(\varphi_1 R_1 - \varphi_n R_n) R_i = -c_{1,2}(S'_{x1} - S'_{x2}) R_i - c_{1,n}(S'_{x1} - S'_{xn}) R_i + M_D; \quad (1)$$

$$I_i \ddot{\varphi}_i + \eta(\dot{\varphi}_i R_i - \dot{\varphi}_{i-1} R_{i-1}) R_i + c_{i-1,i}(\varphi_{i-1} R_{i-1} - \varphi_i R_i) R_i + \eta(\dot{\varphi}_i R_i - \dot{\varphi}_{i+1} R_{i+1}) R_i + c_{i,i+1}(\varphi_i R_i - \varphi_{i+1} R_{i+1}) R_i = -c_{i-1,i}(S'_{xi} - S'_{xi-1}) R_i - c_{i,i+1}(S'_{xi} - S'_{xi+1}) R_i;$$

$$I_n \ddot{\varphi}_n + \eta(\dot{\varphi}_n R_n - \dot{\varphi}_{n-1} R_{n-1}) R_n + c_{n-1,n}(\varphi_{n-1} R_{n-1} - \varphi_n R_n) R_n + \eta(\dot{\varphi}_n R_n - \dot{\varphi}_1 R_1) R_n + c_{n,1}(\varphi_n R_n - \varphi_1 R_1) R_n = -c_{n-1,n}(S'_{xn} - S'_{xn-1}) R_n - c_{n,1}(S'_{xn} - S'_{x1}) R_n - M_R,$$

where $i = 2, 3, 4, \dots, n-1$; $I_1, \dots, I_n$ – the reduced moments of inertia of the rotating masses; $\varphi_1, \dots, \varphi_n$ – current angles of rotation of the masses (sprockets); $c_{1,2}, \dots, c_{i,i+1}, \dots, c_{n,1}$ – rigidity of the conforming branches of a chain contour; $R_1, \dots, R_n$ – radiuses of arrangement of joints of a chain contour on sprockets;

$S_{x1}, \dots, S_{xi}, \dots, S_{xn}, S'_{x1}, \dots, S'_{xi}, \dots, S'_{xn}$ – functions of a disturbance of terminal joints of branches; η – damping ratio; M_D, M_R – driving moment and moment of resistance accordingly.

The solution of system of differential equations (1) without right members gives own frequencies of system:

$$p_\varphi^2 = [M]^{-1} \cdot [C] \quad (2)$$

where $[M]$ – diagonal matrix of rotated masses;

$[C]$ – symmetrical matrix of rigidities of branches of a chain contour.

On fig.1 for an example is presented frequency diagram, where all own frequencies are shown for 2 - mass chain drive (horizontal lines: transversal vibrations f_u , torsion vibrations f_φ , longitudinal vibrations f_x , frequencies of disturbances (from the polygonal effect of sprockets f_z , accumulated error of length of branch f_w , $2f_w$, eccentricities of sprockets f_d) and corresponding resonance numbers of frequencies of rotation (on axis of abscissas)

$n_z^u, n_z^\varphi, n_d^u, n_w^u, n_z^x$ for a chain-drive, equipped by the chain of ПП-15,875-2270-2 ГОСТ13568-75, tension of leading branch is $S_f=1$ kN, her length $l=nt=33\,15,875=523,87$ mm, number of links in the contour of $W=65$, gear-ratio of $u=1,8$, moments of inertia, $I_1=0,13$ kgm², $I_2=0,17$ kgm².

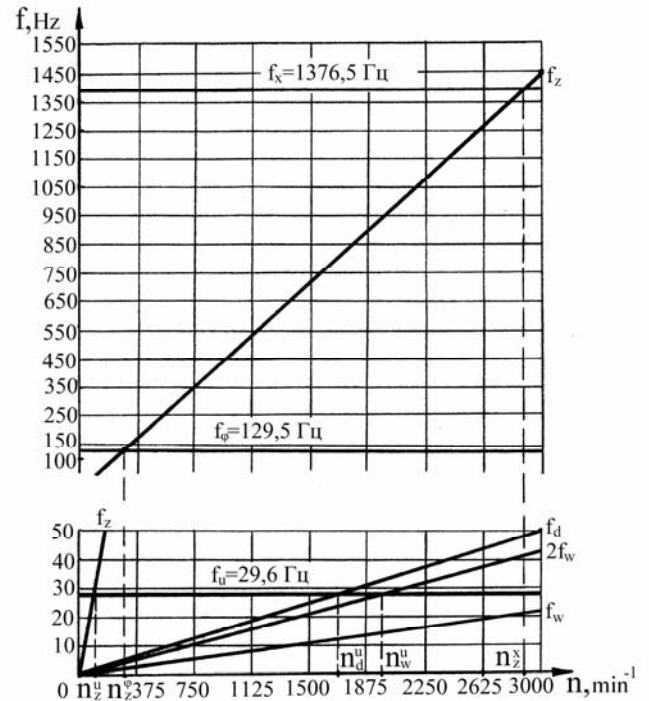


Fig.1. Frequency diagram

As evidently from fig.1, resonance frequencies of rotation from the eccentricities of sprockets on the transversal vibrations n_d^u and from the accumulated error of length of branch on the transversal vibrations n_w^u lie in the range of 1500...2250 min⁻¹. Resonance frequencies of rotation from polygonal effect on longitudinal vibrations arrive at almost 3000 min⁻¹. As for resonance frequencies of rotation from eccentricities and accumulated error of lengths of

branch on torsion vibrations, then they lie so highly, that not fall even in the range of speeds, practically accessible for roller chain-drives up-to-date development of technique.

On the other hand, resonance frequencies of rotation for transversal n_z^u and torsion n_z^p vibrations from the polygonal effect of sprockets lie in the bottom range of frequencies of rotation (to 375 min⁻¹). As for resonance frequencies of rotation from the accumulated error of length of branch, then their values will lie in area of frequencies of rotation (8...15) of 10³ min⁻¹, id est in an unattainable for the real chain-drives range.

Practically most possible is an inflow of the system in resonance with basic tone of own frequencies. Therefore there will be a decline of the dynamic loading at any after resonance speeds, that and confirmed experimentally. However, most possible speed is dictated by shock strength of rollers, promoting which is possible application of polymeric toothed crowns of sprockets and chains (or even them internal links) from polymeric composites.

4. Basic kinematic and dynamic advantages of chain-drives, equipped by polymeric parts

The detailed solution of the equations (1) for 2-mass chain drives resulted in [14]. As results of solution this equations were fined the dependences for determination of angular speeds (and accelerations) of rotary masses. For example,

$$\begin{aligned} \phi_1 = & -c_1 \cdot \frac{R_1}{p_1 J_1} (\eta p_1 + k_1 + k_2) \cdot e^{p_1 \tau} - c_2 \cdot \frac{R_1}{p_2 J_1} \cdot (\eta p_2 + \\ & + k_1 + k_2) \cdot e^{p_2 \tau} + \left(\frac{M_d}{J_1} - \frac{(k_1 + k_2) R_1}{J_1} \cdot \xi_0 \right) \cdot \tau + c_3 + \\ & + \sum_{n=1}^5 \frac{-R_1}{J_1 n z_1 \omega_1} [((k_1 + k_2) \alpha_n - (k_1 + k_2) M_n^{x_1} + \\ & + \eta n z_1 \omega_1 N_n^{x_1}) \cdot (-\cos(n z_1 \omega_1 \tau)) + \\ & + ((k_1 + k_2) N_n^{x_1} - \eta n z_1 \omega_1 M_n^{x_1}) \cdot \sin(n z_1 \omega_1 \tau)] + \\ & + \sum_{n=1}^5 \frac{-R_1}{J_1 n z_2 \omega_2} [(-(k_1 + k_2) \beta_n - (k_1 + k_2) M_n^{x_2} + \\ & + \eta n z_2 \omega_2 N_n^{x_2}) \cdot (-\cos(n z_2 \omega_2 \tau - 2\pi j)) + \\ & + (-(k_1 + k_2) N_n^{x_2} - \eta n z_2 \omega_2 M_n^{x_2}) \cdot \sin(n z_2 \omega_2 \tau - 2\pi j)] + \\ & + \sum_{n=1}^5 \frac{-R_1}{J_1 n z_1 \omega_1} [((k_1 + k_2) \alpha_n - (k_1 + k_2) M_n^{x_1} + \\ & + \eta n z_1 \omega_1 N_n^{x_1}) \cdot (-\cos(n z_1 \omega_1 \tau - \xi_1)) + \\ & + ((k_1 + k_2) N_n^{x_1} - \eta n z_1 \omega_1 M_n^{x_1}) \cdot \sin(n z_1 \omega_1 \tau - \xi_1)] + \\ & + \sum_{n=1}^5 \frac{-R_1}{J_1 n z_2 \omega_2} [(-(k_1 + k_2) \beta_n - (k_1 + k_2) M_n^{x_2} + \\ & + \eta n z_2 \omega_2 N_n^{x_2}) \cdot (-\cos(n z_2 \omega_2 \tau - \xi_2 - 2\pi j)) + \\ & + (-(k_1 + k_2) N_n^{x_2} - \eta n z_2 \omega_2 M_n^{x_2}) \cdot \sin(n z_2 \omega_2 \tau - \xi_2 - 2\pi j)]; \quad (3) \end{aligned}$$

(Solving the system of equations (1) in relation to angular velocity

ω_1, ω_2 and angular accelerations ε_1 and ε_2 (first and second derivatives from ϕ_1, ϕ_2) leading and conducted the rotating masses of 2-mass chain-drive, it is possible to get them graphic dependences. On fig.2,3 the represented changes of the indicated data during starting only for conducted rotating mass, as it is considered that leading mass is revolved evenly (that does not answer reality, although leading mass has a less unevenness of rotation, than conducted)).

The type of the curves of angle speeds and accelerations (fig.2,3) testifies that motion of the rotating masses of chain drive is not even. Value of angle speeds and accelerations in chain drive, equipped by a polymeric chain, less comparatively with the same values of chain drive, equipped by a standard metallic chain. Beginning from 0,08 s after starting, motion of conducted rotating mass becomes practically almost even in case of application of

polymeric chain, in that time as at application standard metallic - motion continues to be uneven.

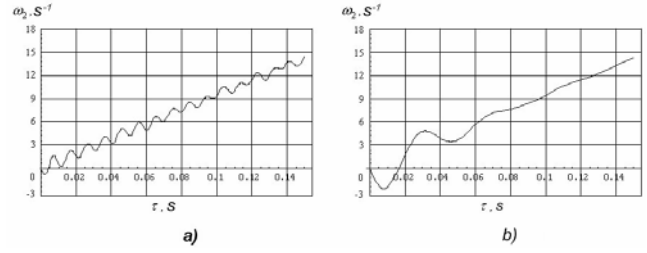


Fig.2. Angular speed ω_2 the second rotating mass with a metal a) and polymeric b) chains

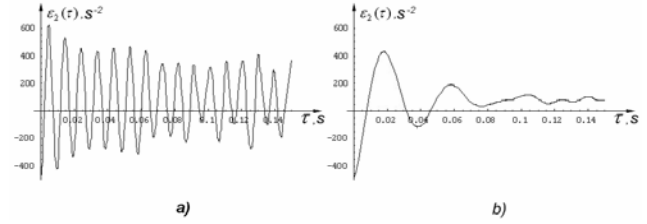


Fig.3. Angular acceleration ε_2 the second rotary mass with a metal a) and polymeric b) chain

The type of the curves of angle speeds and accelerations (fig.2,3) testifies that motion of the rotating masses of chain drive is not even. Value of angle speeds and accelerations in chain drive, equipped by a polymeric chain, less comparatively with the same values of chain drive, equipped by a standard metallic chain. Beginning from 0,08 s after starting, motion of conducted rotating mass becomes practically almost even in case of application of polymeric chain, in that time as at application standard metallic - motion continues to be uneven.

Solving the system of equations (1) in relation to deformation of leading of branch of chain $\phi_1 R_1 - \phi_2 R_2$, will find the dynamic loading in leading and conducted branches (for 2-mass chain-drive), predefined by the polygonal effect of sprockets (fig. 4 a, b) :

$$\begin{aligned} F_{dun1}(\tau) = & c_1 \cdot (\phi_1 R_1 - \phi_2 R_2 + \frac{t}{\pi} \sum_{k=1}^5 (-1)^k \frac{\sin(k z_1 \omega_1 \tau)}{k(1 - k^2 z_1^2)} - \\ & - \frac{t}{\pi} \sum_{k=1}^5 (-1)^k \frac{\sin(k z_2 \omega_2 \tau - 2\pi j)}{k(1 - k^2 z_2^2)}); \\ F_{dun2}(\tau) = & c_2 \cdot (\phi_1 R_1 - \phi_2 R_2 - \frac{t}{\pi} \sum_{k=1}^5 (-1)^k \frac{\sin[k(z_1 \omega_1 \tau - \xi_1)]}{k(1 - k^2 z_1^2)} + \\ & + \frac{t}{\pi} \sum_{k=1}^5 (-1)^k \frac{\sin[k(z_2 \omega_2 \tau - \xi_2 - 2\pi j)]}{k(1 - k^2 z_2^2)}), \quad (4) \end{aligned}$$

where t – step of a chain; k – number of harmonics; $j = \frac{\varepsilon \cdot z_i}{2\pi}$ –

factor of number of links; ε – phase lag between disturbances of terminal joints of branches, belongs to adjacent sprockets;

$\xi_i = \alpha_i - p_i \frac{2\pi}{z_i}$ – phases lag between motion of terminal joints of

branches, belongs to one sprocket; α_i – angles of contact by a chain of a sprocket; p_i – number of joints of a chain on contact arc.

The inertia loading is from the unevenness of rotation of the masses

$$F_{\partial i} = m_i \ddot{x}_i = F_{i,i+1} - F_{i-1,i}, \quad (5)$$

where m_i – the rotating masses; $\ddot{x}_i$ – their acceleration

$$(x_i = \varphi_i R_i).$$

Dynamic unevenness of rotation of the masses (sprockets)

$$\delta_i = 2\varphi_i z_i \cdot 100\%, \quad (6)$$

where z_i – numbers of teeth of sprockets are.

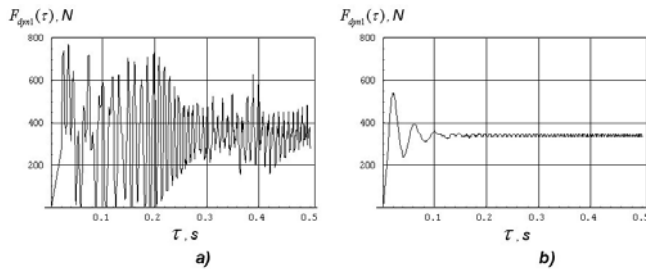


Fig. 4. Dynamic loads at start-up in a leading branch conditioned by polygonal effect of sprockets with metal a) and polymeric b) chains

From fig. 4a,b see that a steady-state behavior of operations of chain drive, equipped by a polymeric chain, comes considerably before (about 0,15 s), than drive, equipped by a metal chain (about 0,45 s). Amplitude of vibrations of the dynamic loading in leading and conducted branches of chain drive, equipped by a metal chain more than in drive, equipped by a polymeric chain. Amount of vibrations of the dynamic loading in a chain drive with a metal chain considerably anymore, than in drive with a polymeric chain, as a link of metal chain consists of 6 elements, that results in the different dimension steps of chain, which does not characterize the parameters of chain from polymeric compos, which has very small difference in pitch (as links of chain are made by pressure-die casting from the same press-form on thermoplast automatic machines).

Apparently on fig. 5, dynamic loadings in a leading branch of chain drive and shock loading at an input joint of a chain in gearing with a sprocket tooth essentially decrease.

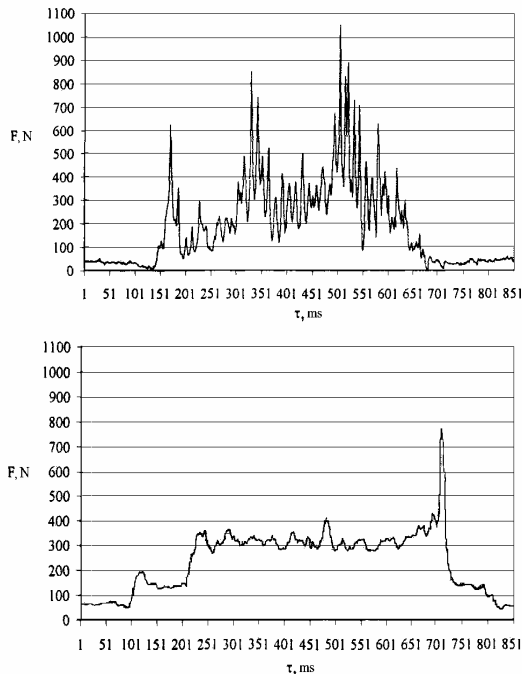


Fig.5. Oscillograms of dynamic loadings at the steady-state movement, received experimentally, in a leading branch of the chain drive equipped standard metallic (above) and polymeric (below) elements

5. Reduction of contact stresses by application of polymeric composites

One of reasons of death drive roller chains there is cleaving of rollers as a result of high contact stresses which arise up in the

moment of engagement of roller of chain with the tooth of sprocket. Even the insignificant angularity result in the considerable increase of value of contact stresses. One of the most effective ways of reduction of value of contact stresses is application of polymeric composites as to material for making of sprockets. Will consider on a concrete example the results of such application and will compare them to traditional. Initial data: chain ИП-25,4-5760 ГОСТ 13568-75. Power $N=6,6$ kW. Number of teeth of sprocket $z=18$. For making of sprocket the glass-filled polyamide ПА-66-KC ($E_2=3,53 \cdot 10^5$ МПа, $\mu_2=0,38$). is applied. For steel

$$E_1=2,06 \cdot 10^5 \text{ МПа}, \mu_1=0,3.$$

In a table.1 contact stresses, calculated on the known formula of Hertz, are brought.

Table 1. Contact stresses are depending on the profile of teeth of sprockets and materials of their making

Profile of teeth	σ_H max, MPa	
	metal sprocket	polymeric sprocket
ГОСТ 591–69	771,5	136,2
ГОСТ 592–75	982,1	173,3

As see, the use of polymeric sprocket diminishes the value of contact stresses approximately in 5,7 times. Obviously, such correlation will be just for the different types of chain, and also for the different initial of data, because it is determined only by properties of materials. However, all brought calculations over are just only for a "ideal" case, id est, when a contact takes place along the formative cylindrical surfaces of contacting bodies, and a corner between their axes equals a zero.

In practice, in connection with the features of making of drive roller chains, between axes any two nearby rollers there are angularity the value of which can arrive at a few ten of minutes. As a result of loading distributed very unevenly on length of roller, or even a contact takes place not on all his length.

As a result, even the insignificant corners of angularity result in the considerable increase of value of contact stresses. Yes, in the already mentioned example, at in corner of angularity $5'$ contact stress it is grown in 3 times, and at $40'$ — more, than in 9 times (a variant is with a metal sprocket).

Character of dependence of value of contact stresses σ_γ from the angularity γ is represented on fig. 6, where curves 1 and 2 answer dependences for a metal sprocket with the profiles of tooth accordingly for ГОСТ 592-75 and ГОСТ 591-69, and curves 3 and 4 — for a polymeric sprocket with the same profiles.

On a graph clearly noticeable two areas: first — at $\gamma < \gamma_{kp}$, second

— at $\gamma > \gamma_{kp}$. It is possible to note, that angularity from $15'$ to $45'$, that mostly met in practice, for metal sprocket completely are fall on the second section (that answer to the most unacceptable case of contact), in the same time for polymeric — mainly on first section.

A very evident picture we will get, comparing dependence of coefficient K_γ that is numeral equal σ_γ / σ_H , from the corner of angularity γ . On fig. 7 graphs of these dependences (1 — for the contact of metal chain with a metal sprocket, 2 — for the contact of the same chain with a polymeric sprocket) are brought. As evidently from graphs, influence of corner of angularity on the increase of contact stresses is far more substantial in case with metal sprocket. Yes, critical corner of angularity γ_{kp} , at what $K_\gamma = 2$, in case of application of metal sprocket folds just $1'50''$, at that time as for a polymeric sprocket he is evened $44'40''$.

Thus, at angularity $15' \dots 45'$ which mostly meet in practice, in transmissions, equipped by polymeric sprockets, a contact takes place on all length of roller, thus contact stresses grow no more than twice in comparing to the "ideal case", and in transmissions, equipped by metal sprockets, length of line of contact folds from

28,4 % to 14,7 % accordingly, here contact stresses grow in 5,2 ... 9,7 times.

Certain interest presents that, how contact stresses will change at presence of defects for the chains of different pitch. To that end will conduct the comparing analysis of value of contact stresses for the same initial of data, that and in the examined example. That terms remained the same, id est the pitch circle of sprocket did not change, it is necessary for the different values of pitch of chain to accept the different number of teeth of sprocket.

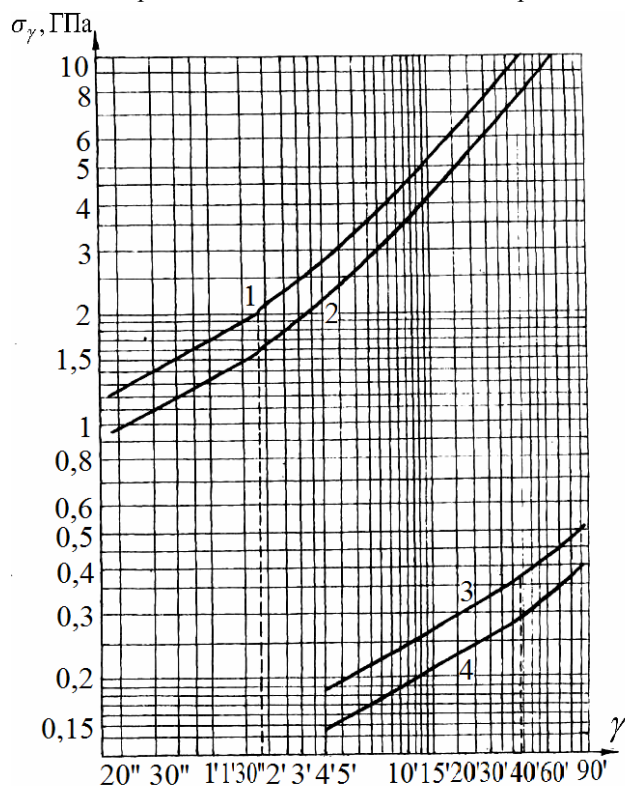


Fig. 6. Dependences of value of contact stresses from the angularity of roller: 1,3 - metal and polymeric sprockets accordingly for ГOCT 592-75; 2,4 - metal and polymeric sprockets accordingly for ГOCT 591-69

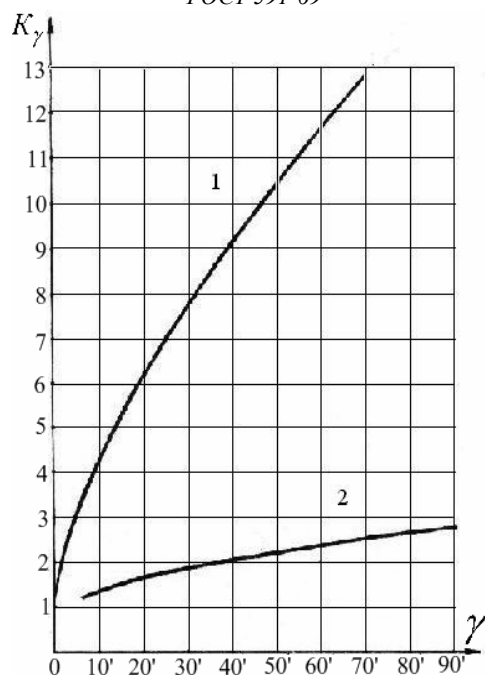


Fig. 7. Dependence of coefficient of K_γ from a corner of the angularity: 1 – a contact of metal chain with a metal sprocket; 2 – a contact of metal chain with a polymeric sprocket

What is why if at $t = 25,4$ mm is accepted $z = 18$, then for the values t , that 19,05 mm, 15,875 mm and 12,7 mm of value z , is evened will present according to 24, 29 and 36 (at such values of z deflection on the deviating diameter of sprockets does not exceed 0,4 %).

Increase of pitch, and also increase width of chain at invariable pitch diminishes the critical value of corner of angularity γ

substantially, although correlation $\gamma_{\kappa p}^{n o n} / \gamma_{\kappa p}^{m e m}$ does not depend from pitch and presents about 24,3.

Other relative description, $\sigma_\gamma^{m e m} / \sigma_\gamma^{n o n}$ which, at bottom, and is the index of effectiveness of substitution metal sprockets on polymeric, grows at to the increase of pitch and width of chain, and if pitch is identical — at increase of corner of angularity. On condition $\gamma = 0$ correlation $\sigma_\gamma^{m e m} / \sigma_\gamma^{n o n}$ is not change and at any pitch and width of chain equal 5,67, id est it is determined only by properties of materials contacting bodies.

All presented calculations and conclusions are correct also at other of initial conditions (power, peripheral velocity and others like that). Will notice, however, that at growth of peripheral effort (or power) in n times also in n times increase critical value of corner of angularity, and a value of contact stresses increase in $\sqrt{n}$ times.

Thus, the greatest effectiveness on the reduction of contact stresses at change of metal sprocket on polymeric is arrived at the most values of pitch and width of chain.

6. Integrated parts of chain-drives

Production of parts of chain-drives from the polymeric composites is low-wasted and not very much power-consuming. From them it more easily to make the parts of difficult form, they are so technological, that allow to create the so-called integrated parts production of which from a metal far more expensive or it is impossible in general. As specific gravity of polymeric composition materials in several times below specific gravity of steel, the moments of inertia of parts and consumable energy diminish during acceleration and braking, that allows raising frequency of rotation. Thus they are not exacting to lubrication, work practically without noise and own relative high durability.

Except requirements, presenting to the constructions of parts from polymeric materials which are produced by injection casting, during creation of new constructions of elements of chain-drives possibility of implementation of parts was foreseen by integrated, id est one monolithic part which is made for one technological operation can include for itself a maximal number structurally functional elements (for example, 6 elements of link of standard metallic roller chain).

On fig. 8 elastic monolithic link from a polymeric composite and chains, made from such elastic monolithic links are shown. It is important to mark that from such links it is possible to assemble two-, three- and multirow chains. Industrial prototypes of such chains presented on fig. 8. On the same fig. is shown also a metal-polymeric chain (from below) the internal links of which are executed from a polymeric composite as one integrated monolithic part (fig. 9), what consists of two cylindrical elements with holes, connected by lamellar elements, and external links which standard metallic connecting links can serve in a role of, id est such chain is fully demountable.

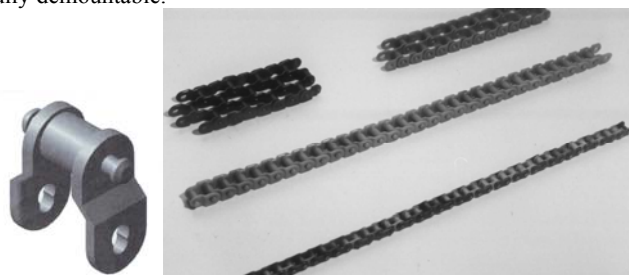


Fig. 8. Elastic monolithic link (on the left) and metal-polymeric single-row (from below), polymeric one, two- and three-row chains

It should be noted that in case of change of metallic chain on polymeric or metal-polymeric, in which internal links is made from polymer composite, dimension exactness of chains, made from grass-filled brands of polyamides, arrives at exactness of the best standards of foreign firms, which apply the selective assemblage of metallic elements of links of roller chains (English firm Renold and Japanese - Tsubaki).



Fig. 9. Metal-polymeric chain with integrated internal link from a polymeric composite

On fig.10 is shown sprocket from a polymeric composite, integrated with a drum and fastening lid.



Fig. 10. Integrated sprocket (with a drum and fastening lid)

7. Designing of n -mass of chain-drive

Designing of chain-drive with any amount of the masses is considered as a two basic job mix: to the choice of structure (structural synthesis) and choice of numerical values of parameters of the got structure (parametrical synthesis). Part of parameters which are manage in sense of dynamic quality of the system, distinguished as varied. The choice of values of parameters, which are varied, folds the article of parametrical synthesis.

Will distinguish the system, functional, structural and technological levels of designing. At system level the analysis of work of the dynamic system of chain-drive comes true on the whole and her decoupling on structural principle as independent subsystems (of two-mass transmissions – of the partial systems) for the next stages of designing.

At functional level of designing is formed mathematical model as a system of differential equations which describe a structure and functioning of the dynamic system of multimass chain-drive. On such model it is possible to carry out imitation experiments, related to the choice of parameters in the conversational or automatic modes.

At structural level a basic task is a synthesis of parameters: co-ordinates of location and orientation of sprockets (topologies), interaxial distances, angles of contact, type, pitch and parameters of chain and sprockets. The task of technological level of designing is a synthesis of technological process as a sequence of productive operations of treatment of surfaces of parts (for metallic sprockets

and chains), or by volume making of parts for one technological cycle from polymeric composites, assembly and control of quality.

A structural synthesis realizes on the basis of construction of optimal structural scheme which consists of sum of 2-mass modules (two sprockets, connected by a branch of the chain contour), id est partial systems which topologically form practically any great number of chain-drives, with any amount of the masses, located by different character in the Cartesian system of co-ordinates. Thus are taken into account the different cases of location of sprockets in a chain contour and conditions at which a structural scheme of multimass chain-drive is geometrically compatible and her structural realization - possible.

In general case in a polygon, formed by the lines of interaxial distances, values of corners of their crossing it is possible how pleasingly to change, keeping geometrical parameters. Thus the centers of joints of chain will always remain in points the touch of step-type lines with the pitch circle of sprockets. It enables to the designer to provide the optimal structural scheme of multimass chain-drive at the chosen finally basic geometrical parameters: interaxial distances, lengths of conjugating branches of chain contour with the integer of links and angles of phase synchronism. At any amount of the masses (sprockets) in a chain contour a calculation realizes for every sprocket and two nearby with her, here calculation dependences embrace all cases of their location and function (drive, tightening, deflecting). Otherwise speaking, a multimass chain-drive is built on module principle, according to which a few simple module elements (in this case sprockets, embraced by a chain) topologically form practically any great number of chain-drives, located by any character in the Cartesian system of co-ordinates. Thus the place of every module element is exactly set in a structure. A specific consists in the choice of initial modules and in order of their location, id est in the combinatorial completing of the limited number of simple module elements (two-mass chain-drives), topologically oriented according to list of determinate dependences.

Parametrical optimization of such module chain-drive is executed on the volume (to mass) occupied by her and parametrical function, which contains maximal durability, efficiency and resistance to the wear (at maintenance of minimum volume). As an criterion function during optimization of chain-drive with any the amount of the masses on the criterion of vibroactivity the accepted amplitudes of the dynamic loading and most integral description are a coefficient of dynamism of chain-drive, dependent, in particular, from the absolute root-mean-squares of the maximal dynamic loading in branches of chain contour.

The worked out complex of the automated optimal designing of chain-drives POSCD, built on block principle of the module designing as application packages of geometrical, power and dynamic calculations, enables to the designer, working in the conversational mode with PC, to build a chain contour which contains from 2 to 24 masses (sprockets), that satisfies to both the requirements specification and requirements of the optimal designing : minimum to the cost, material capacity, occupied volume, mass and high dynamic quality.

A complex enables to carry out the simulation of work of chain-drives on PC which provides the rapid and comprehensive analysis of influence of different parameters on quality of transmission: co-ordinates of location of centers of sprockets, frequencies of their rotation, operational conditions, type and row of chain, method of lubrication, power, tenure of employment, values of the rotating masses and applied materials.

Existent methods of assembling and adjustment of chain-drives, in particular, determination of location of tightening element (sprocket, roller or chock), does not provide minimization of the dynamic loading, as under adjustment of chain-drive for some reason understand adjusting of tightening of branches of immobile transmission, id est in statics. In the real conditions of the dynamic loading during exploitation of chain-drives the task of optimization of location of tightening element becomes actual with the aim of minimization of amplitudes of vibrations and dynamic loading. The put task gets untied by means of autonomous block of construction

of maps of dynamic load capacity and adjustment of chain-drive SET, which works in the conversational mode.

The lines of possible positions of center of tightening element are built for the different levels of the dynamic loading. Analysis of series of these lines (maps of dynamic load capacity) enable to choose position of tightening element which provides the system low dynamic loading. The same way there can be built maps of distribution of amplitudes of vibrations, dynamic loading in the branches of chain, inertia, shock loading and dynamic unevenness of motion of elements of transmission depending on that, which from the enumerated characteristics is taken for the criterion of optimization.

As a result get the maps of distribution of dynamic characteristics for every branch or mass on boundary of area which enables to the designer to see that dynamic characteristics change at varying the site of tightening element, and to choose such his position at which all dynamic characteristics or part from them (for example, only dynamic loading in branches of chain contour which depends on select of criterion optimization), minimized.

Decrease of labour-intensiveness and reduction of terms of designing of both polymeric parts and press-forms for their pressure-die casting on thermoplast automatic machines during series production, arrived at by automation of designer works, and also standardization and normalization of parts of forms, that allowed to lead the task of designer to designing of forming profile of matrix taking into account the different types of shrinkages and technological admittances.

Conclusion

Presented complex approach to designing and realization of chain transmissions and drives, that is based on the account of the real dynamic processes which accompany their operation, application of polymeric composite materials, resource-saving technologies and equipment for their realization, automated optimal designing, as a result of what a designer will be able to choose the scientifically well-founded aggregate of values of parameters, at which yet on the stage of designing high dynamic quality of chain transmissions and drives is provided.

References

1. Пилипенко О. И. Научные основы и синтез цепных передач. Дисс. докт. техн. наук., ХПГУ, Харьков, 1996, 467с.

2. Пилипенко О.И., Степенко А.П., Козар І.Ф. Вплив похибок виготовлення ланцюгів на контактну взаємодію елементів ланцюгових передач з різних матеріалів. Вісник Чернігівського технологічного інституту №3. Машинобудування. Електроніка. Чернігів, 1997. – С.75-84.

3.Пилипенко О.І., Козар І.Ф., Максименко В.А. Дослідження ланцюгового зацеплення в передачах з полімерних композитів. Вісник Чернігівського технологічного інституту №4. Машинобудування. Чернігів, 1997.– С.125-133.

4.Пилипенко О.І., Борщов О.О., Степенко А.П., Козар І.Ф. Оптимізація ланцюгової передачі по параметричній функції, що містить максимальні довговічність, ККД і опір спрацюванню. Вісник Чернігівського технологічного інституту №3. Машинобудування. Електроніка. Чернігів, 1997. – С.85-94.

5.Пилипенко О.І., Козар І.Ф. Дослідження розмірної точності приводних ланцюгів з полімерних композитів. Вісник Чернігівського державного технологічного університету №9, серія Технічні науки. Чернігів, 1999. – С.41-46.

6.Пилипенко О.И., Кришук Н.Г., Козар И.Ф., Максименко В.А. Компьютерное моделирование напряжённо-деформированного состояния деталей передач зацеплением. Международный сборник научных трудов «Прогрессивные технологии и системы машиностроения». Выпуск 13. Донецк-Севастополь, 11-17.09. 2000. – С.80-85.

7. Пилипенко О.І., Максименко В.А., Козар І.Ф. Дослідження динаміки перехідного та усталеного режиму роботи привода, оснащеного металевими і полімерними ланцюгами // Вісник Чернігівського держ. техн. ун-ту, №10. Серія технічні науки. Чернігів, 2000. – С.61-69.

8. Пилипенко О.І., Максименко В.А. Динаміка руху привода, оснащеного ланцюговою передачею з врахуванням характеристики двигуна // Вісник Чернігівського держ. техн. ун-ту, №12. Серія технічні науки. Чернігів, 2001. – С.25-28.

9. Пилипенко О.І., Максименко В.А. Аналіз пуску ланцюгового привода при застосуванні двигунів різного типу // Вісник Чернігівського держ. техн. ун-ту, №13. Серія технічні науки. Чернігів, 2001. – С.35-39.

10.Пилипенко О.І., Максименко В.А., Ільїн Д.О. Методи зниження динамічних навантажень і вибору основних параметрів ланцюгових передач. Вісник ЧДТУ №18, серія Технічні науки. Чернігів, 2002. – С.12-21.

11.Пилипенко О.І. Динамічна настройка роликів ланцюгових передач. Вісник ЧДТУ №15, серія Технічні науки. Чернігів, 2002. – С.35-38.

12. Пилипенко О.І., Ільїн Д.О. Моделювання динаміки ланцюгового привода, оснащеного деталями з полімерних композитів. Автоматизація виробничих процесів у машинобудуванні та приладобудуванні, №40, Львів, 2006. – С. 183-189.

13.Пилипенко О. Динаміка та розрахунок ланцюгового приводу з полімерних композитів. Машинознавство №2, 2004, Львів. – С.19-24.

14. Oleg Pilipenko, Dmitrij Ilyin. Research and Development of Chain Drives Equipped by Parts of Polymeric Composites. Proceedings of the 6-th International Conference “Research and Development in Mechanical Industry” RaDMI 2006, Budva, Montenegro, 2006. – P.1-8.

15. Пилипенко О.І., Ільїн Д.О. Дослідження динаміки ланцюгового привода чисельними методами. Вісник ЧДТУ №28, серія Технічні науки. Чернігів, 2007. – С.22-28.

16. Pilipenko O.I. Synthesis of Chain Drives Based on Dynamic Methods, New Materials and Technologies. Machine Design, Monograph. Novi Sad, Serbia, 2007. – P.307-314.