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Calculation of planetary drive of mechanical press

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Abstract
Algorithm of calculation of planetary drive of mechanical presses, with the help of which there are defined the main parameters: constructive sizes of the main gear wheels, necessary moments of brakes of inclusion and switching off, energy costs of inclusion and switching off and so forth, is presented in the article.
Keywords: PLANETARY DRIVE GEAR, MECHANICAL PRESS, CONSTRUCTIVE SIZES, MOMENT

Planetary gears are used in a drive gear of mechanical presses with double aim. On the one hand, planetary drive gear with the increased transmission ratio successfully replaces intermediate cylindrical gear pair with the smaller relation, and on the other hand, planetary gears are used as a component of system of press activation, this is their main advantage. First, energy costs for activation of such gear are distributed on expenses at switching on and switching off, and the greatest expenses at switching off occur after a working stroke at the end of idle running back that provides greater power opportunities of the gear. Secondly, the total value of costs of energy of activation and switching off of the drive gear in comparison with traditional systems of activation is considerably (by 3-5 times) less. The last advantage is especially typical for those presses, where the clutch and a brake were mounted on lay shafts. Thirdly, in planetary gears there used two brakes, working conditions of which are much better, than operating conditions of a clutch that considerably increases a resource of their work [1].

Calculation of planetary gear of mechanical press lies in determination, first of all, of the constructive sizes of the main gear wheel, necessary moments of brakes of activation and switching off, energy costs for activation and switching off. But at the stage of outline design carrying out the full dynamic analysis of the system is connected with a number of difficulties.

In such cases it is possible to use results of approximate calculation of the main energydynamic parameters of processes of activation and stop, which is based on the analytical solution of the equations of mathematical model of drive gear after introduction of additional assumptions and prerequisites [2].

Settlement dependences for determination of duration of the period of activation of t_{1n}, of minimum angular speed of a guide link a during the activation period ω_{amn}, of the necessary braking torque look as follows:

$$t_{1n} = \frac{J_a j \omega_{an} p (1+z)}{M_{bp}(1+j_n)} \quad (1)$$

$$\omega_{amn} = \frac{\omega_{an}}{1+j_n} \quad (2)$$

$$M_{bp} \geq \frac{J_a j \omega_{an}^2 (1+z)^2}{\varphi_{bt}(1+j_n)(2+z)} \quad (3)$$

Similar dependencies during gear stop will be

$$t_{1o} = \frac{J_a j \omega_{ao}(1+p)(1+z)}{M_{hp}(1+j_o)} \quad (4)$$

$$\omega_{amo} = \frac{\omega_{ao}}{1+j_o} \quad (5)$$

$$M_{hp} \geq \frac{J_a j \omega_{ao}^2 (1+z)^2}{\varphi_{ht}(1+j_o)(2+z)} \quad (6)$$

Where J_a – inertia moment of drive portions; j_n, j_o, j – relative inertia moments of driven parts

$$j_ï = \frac{J_h}{J_a(1+p)^2}; \quad j_o = \frac{J_b}{J_a p^2}; \quad j = j_ï + j_î + j_ï j_î .$$

J_b, J_h – are the moments of inertia of the driven parts of a drive gear; ω_{an}, ω_{ao} – are the initial angular speed of a link a at activation and stop respectively; $p= z_b/z_a$ – is the kinematic reducer parameter; z_b, z_a – are the number of teeth of external and internal pinion gears of a reducer respectively; z - is indicator of intensity of brake activation. Average values of an indicator of z are equal to 0.2…1.2; M_{bp}, M_{hp} – are the necessary braking torques providing braking of links b and h respectively on the set angles φ_{bt} and φ_{ht}.

Rational distribution of total reduction ratio of drive gear on the basis of calculation of parameters criterion of optimality, which may be as total weight of the gear and energy consumption for activation and stop [3] is a paramount task at calculation of planetary gear.

Full weight of the gear is represented as follows [4]:

$$G_\Sigma = 12{,}2 \cdot 10^{-5} \frac{M_{гм}}{[K_0]_R} \left(\frac{C_R x_R}{i_Z} + \frac{C_Z x_Z [K_0]_R}{i_Z [K_0]_Z} \right) \qquad (7)$$

where $M_{гм}$ – is torque rating on the drive shaft; $[K_0]_R$ - is admissible power factor of material of gear wheels of planetary reducer; x_R - is the coefficient depending on the type of the planetary gear and parameters of gearings; x_Z - is the coefficient depending on a design of gear wheels [5]:

$$x_z = \frac{i_z + 1}{i_z}\left(C_1 + C_2 i_z^2\right)$$

where $[K_0]_Z$ - is admissible power factor of material of gear wheels; $i_Z = d_2 / d_1$ - is transmission ratio of toothed gearing; d_1, d_2 - are the diameter of a delitelny circle of the driving and driven wheels respectively; C_R, C_Z – are the coefficients of a design of planetary reducer and toothed gearing; C_1, C_2 - are the coefficients of a design of the driving and driven wheels respectively.

For planetary gears the coefficient x_R is determined by a formula

$$x_R = \frac{k_à + k_g n_\omega (p-1)^2 + k_b p^2}{n_\omega (p-1)}$$

where k_a, k_g, k_b - are the coefficients of a design of gear wheels of a reducer; n_ω – is the number of satellites;

Coefficients of a design k_a, k_g, k_b of gear wheels of a planetary reducer are accepted equal to unit for wheels with external teeth and 0.3 - for wheels with internal teeth [5].

Value $M_{гм}/[K_0]_R$ is a constant for each press, that is why the correlation $G_\Sigma M_{гм}/[K_0]_R$ is directly-proportional to full weight of the gear and can be accepted as the first criterion of optimality of parameters (weight minimum) R_G.

For planetary gears the coefficient of x_R is determined by formula

$$R_G = \frac{12{,}2 \cdot 10^{-5}}{i_Z}\left(C_R x_R + C_Z x_Z\right) \qquad (8)$$

In the comparative analysis coefficients C_R and Cz are accepted as equal to C_R=1,2…2,6, Cz=1,0…1,3, and C_z acquires greater values at band brakes , less - at disk brakes. Coefficients C_1 and C_2 of gear drive are equal to C_1=1,45…2,6, C_2=0,36…0,44. Coefficients C_1 and C_2 get great values for drives with rather narrow tooth gears (at $\psi 2 \leq 0.15$) [6].

In fig. 1 there shown the dependence of R_G criterion on the general transmission ratio i_o and parameter of a reducer p. To each i_o value there corresponds one combination of p and i_Z (as $i_î = i_z/(1-p)$), where G weight is the smallest. The minimum values of R_G criterion in the figure are connected by the S_G line.

Energy consumption for activation and stop can be defined according to the formula:

$$A = N_a\left[i_z p(1+p)\right]^2 \left(\sqrt[3]{\frac{1}{(p-1)i_z}}\right)^5 \qquad (9)$$

where

$$N_a = \left(\sqrt[3]{M_{гм}}\right)^5 \omega_н^2 C_в \frac{\left(\psi_b{}^4 - 1\right)\pi\gamma\psi_a\left(1+\kappa_j\right)}{314}\left(\sqrt[3]{\frac{2}{\psi_a n_\omega [K_o]_R}}\right)^5$$

$\omega_н$ – is rated angular speed of the main shaft of a press; $C_в$ -is coefficient of design of external pinion gear; γ - is firmness of wheel material; ψ_a - is relative width of a wheel a; k_j – is coefficient of ratio of inertia moments of the driven masses $\kappa_j = j_n / j_o$; ψ_b – is the relative width of a wheel b;

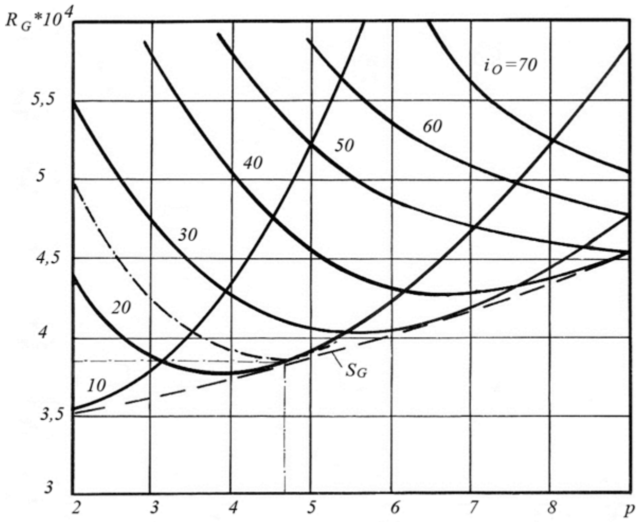

Figure 1. Dependence of R_G criterion on the relaying correlations of the gear

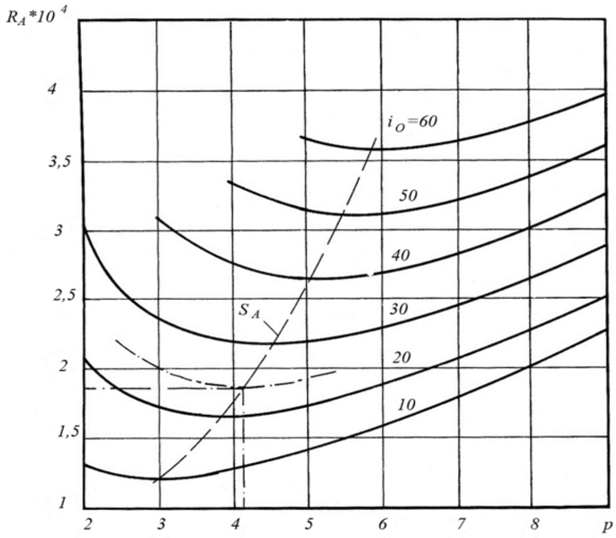

Figure 2. Dependence of R_A criterion on the relaying correlations of the gear

In the comparative analysis the constant *Na* doesn't depend on parameters of a drive gear and for each press is a constant. Therefore correlation

$$R_A = \frac{A}{N_a} \quad (10)$$

is the dependence of relative energy consumption on activation and stop of the main executive gear on parameters of a drive gear and is accepted as the second criterion of an optimality of R_A.

Fig. 2 shows the dependence of criterion of R_A on the parameter *p* at a certain general transmission ratio i_o. From the figure one may see that for each transmission ratio of i_o there is the only combination of *p* and i_Z at which the value R_A will be the smallest. Optimum values of parameter *p* may be found as the coordinate of a point of curve crossing R_A for the corresponding i_o value with the S_A line, which is the line of minimum values of R_A criterion.

After definition of the transfer relations of the gear the number of wheel teeth of a reducer is determined by the following dependences, received after the joint solution of a condition of the neighbourhood and assembly:

$$z_1 = \frac{g}{g+q} n_\omega A; \quad z_2 = \frac{q-g}{2(g+q)} n_\omega A; z_3 = \frac{q}{g+q} n_\omega A.$$

where *A* – random integer number; *q, g* – are the smallest integers, which relation is equal to transmission ratio of a reducer.

Conclusions

Thus, the article shows the algorithm of calculation of planetary gear of mechanical presses, with the help of which it is possible to define critical parameters: constructive values of the main gear wheels, necessary moments of brakes of activation and switching off, energy costs for activation and switching off and so forth.

References

1. Yavtushenko A.V., Rudenko A.V., Rybinok V.A. Sovershenstvovanie sistem vkljuchenija krivoshipnyh kuznechno-pressovyh mashin [*System development of activation of crank-type of forging and pressing machines*]. Kiev, Znanie, 1980, 24 p.
2. Yavtushenko A.V., Sereda B.P., Vasilchenko T.A.,Glebenko A.V. (2010). Priblizhennyi raschet osnovnyh jenergodinamicheskih parametrov processov vkljuchenija i ostanovki planetarnogo privoda [Approximate calculation of main energydynamic process parameters of activation and stop of planetary drive]. *Metal treatment under pressure, collection of scientific papers*. Kramatorsk, DGMA, No 3(24), p.p. 175-179.
3. Yavtushenko A.V., Sereda B.P., Vasilchenko T.A., Glebenko A.V. (2009). Optimizacija parametrov planetarnogo privoda kuznechno-pressovih mashin [Parameter optimization of planetary drive of forging and pressing machines]. *Metal treatment under pressure, collection of scientific papers*. Kramatorsk, DGMA, No 1(20), p.p. 306-311
4. Sereda B.P., Vasilchenko T.A. (2009). Minimizacija ras hoda jenergii na vkljuchenie i

ostanovku glavnogo ispolnitel'nogo mehanizma KPM [Minimization of energy consumption for activation and stop of the main actuating mechanism of forging and pressing machines]. Bulletin of the National technical university "Kharkov polytechnic institute". Collection of scientific papers, special issue – New solutions in modern technologies. Kharkiv, No 31, p.p. 22-25.

5. Planetarnye peredachi [*Planetary drives*]. Reference book. Under the editorship of V.N. Kudryavtsev and Yu.N. Kirdyashev. Lviv, Mashinostroenie, 1977, 536 p.
6. Tynyanov V. N. (1976). O proektirovanii zubchatyh privodov s naimen'shej massoj dlja krivoshipnyh pressov [About the design of gear drive with the lowest mass for mechanical press]. *Die forging*. No 9, p.p. 31-34.

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