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APPROXIMATE CALCULATION OF THE BASIC PARAMETERS OF ENERGODINAMIC PROCESSES STARTING AND STOPPING PLANETARY DRIVE

The paper presents a simplified engineering method, which at the stage of conceptual design allows you to define the parameters of the preliminary processes starting and stopping of the planetary drive. Introduction of the proposed simplifications, gives an opportunity to solve a differential equation of motion of the inertial elements of the dynamic model of the drive.
Key words: Planetary drive, press-forging machine, mathematical model of a drive, power dynamic parameters.

В статье представлена упрощенная инженерная методика, которая на стадии эскизного проектирования позволяет определять предварительные параметры процессов включения и остановки планетарного привода. Введение предложенных упрощений дает возможность решить дифференциальное уравнение движения инерционных элементов динамической модели привода.
Ключевые слова: Планетарный привод, кузнечно-прессовая машина, математическая модель привода, энергодинамические параметры.

У статті представлена спрощена інженерна методика, яка на стадії ескізного проектування дозволяє визначать попередні параметри процесів включення та зупинки планетарного приводу. Введення запропонованих спрощень дає змогу розв’язати диференційне рівняння руху інерційних елементів динамічної моделі приводу
Ключові слова: Планетарний привід, ковальсько-пресова машина, математична модель приводу, енергодинамічні параметри.

Starting systems that include various schemes planetary reducers have been developed recently. However, theoretical foundations of design and analysis of such a drive almost completely lack in the literature.

Works [1, 2] developed a dynamic and mathematical models of the planetary gear, which determine the basic laws of the drive units motion when started and stopped. However, at the stage of conceptual design, a full analysis of the dynamic system is irrational and often impossible, due to lack of exact original data values. The need for such indicators also arises when evaluating the effectiveness of the chosen gear scheme and its parameters. In such cases, you can use the results of an approximate calculation of the basic enerdgodynamic parameters of the starting and stopping processes.

The aim is to develop an approximate engineering method of calculating the drive with a simplified design.

An approximate calculation is based upon the drive mathematical model analytical solutions, additional assumptions and prerequisites being assumed. The

validity of such assumptions is defined by means of the results of the dynamic system theoretical analysis and experimental studies of various planetary gear drive mechanisms.

Works [1] presented a mathematical model of the planetary drive:

$$\left.\begin{aligned} \varphi_a'' &= a_a(M_d - M_{ca}) + a_b(M_{cb} + M_{tb}) + a_h(M_{ch} + M_{th}); \\ \varphi_b'' &= b_a(M_d - M_{ca}) + b_b(M_{cb} + M_{tb}) + b_h(M_{ch} + M_{th}); \\ \varphi_h'' &= h_a(M_d - M_{ca}) + h_b(M_{cb} + M_{tb}) + h_h(M_{ch} + M_{th)}), \end{aligned}\right\} \quad (1)$$

where φ_a, φ_b, φ_h – rotary angles to their own axes;

M_{ca}, M_{cb}, M_{ch}-moments of resistance to rotation of the corresponding links;

M_{tb}, M_{th} - braking torque, which provides *b* or *h* link stopping;

M_d - the motor torque.

All the quantities a_i, b_i are constants and their determination is given in [1].

The inertial elements motion equations of the planetary drive dynamic model is a second order differential equation (with respect to angular displacement φ_i) with a complex nonlinear right side. Analytic solution of these equations is possible only with a special method for nonlinear equations solving, or after the introduction of additional assumptions.

It was established that during the starting or stopping period, continuing up to the complete stop of one of the gear links due to its small length, the elastic slip of the V-belt transmission and electrodynamic inertia of the induction motor, with no change of its torque. Kinetic energy needed to accelerate the driven masses is transmitted by the energy of the flywheel and the retarded masses. Consequently, during the starting and stopping process with an error less than 3..6% suppose *Md* = *Mc*a. Such an assumption is accepted and the calculation process for integrating the drive with a clutch (before the speed leveling of driven and driving masses). [3]

Experimental and theoretical studies show that for most drives with tape and disc brakes time t_1, the mass movement to a full stop is less than the time t_b of the full brake [3]. The actual behavior of the brake torque while braking can be approximated by a power law

$$M_{ti} = M_{i1}\left(\frac{t}{t_1}\right)^z, \quad (2)$$

where M_{i1} - braking torque developed by a brake at the end of the braked mass stop,

$$M_{i1} = M_{ip}\left(\frac{t_1}{t_b}\right)^z; z < 1$$

M_{ir} - calculated braking torque;

z - index of the braking intensity, determined by formulas and graphs in [3]. Mean values of the exponent *z* equal to 0,2..1,2.

Change in the braking torque when you turn off the brake can be neglected, ie we believe that the brake is turned off instantly.

Taking the above assumptions, the system of equations (1) can be represented as admitting its analytical solution

$$\left.\begin{aligned}\varphi_a'' &= a_b\left(M_{cb}+M_{tb}\right)+a_h\left(M_{ch}+M_{th}\right);\\ \varphi_b'' &= b_b\left(M_{cb}+M_{tb}\right)+b_h\left(M_{ch}+M_{th}\right);\\ \varphi_h'' &= h_b\left(M_{cb}+M_{tb}\right)+h_h\left(M_{ch}+M_{th}\right).\end{aligned}\right\} \tag{3}$$

Thus, while the braking moment M_{tb} is determined by the formula (2), and $M_{th} = 0$, and vice versa, when you turn off $M_{tb} = 0$, and the M_{th} - according to the formula (2).

As already mentioned in [3], to determine the nature of the motion units it is sufficient to solve two of the three system equations (3), because angular velocity and displacement of the third link can be determined by the equation of the kinematic connection, knowing the speed and movement of the other two

$$\varphi_a = \varphi_b i_{ab}^h + \varphi_h i_{ah}^b, \tag{4}$$

where i_{ab}^h, i_{ah}^b - gear ratio between the two links listed in the subscript at stopped third link from the top index.

When turning on the initial conditions are indicated in [1], the moments of $M_{th} = 0$, M_{tb} is are determined by the formula (2). The activation period ends at $t=t_{1n}$, when $\omega_b = 0$, $\omega_a = \omega_{amn}$, $\omega_h = \omega_{hmn}$, $\varphi_b = \varphi_{bm}$.

Integrating the first two equations (3), we obtain

$$\left.\begin{aligned}\omega_a-\omega_{an} &= \left(a_bM_{cb}+a_hM_{ch}\right)t+\frac{a_bM_{bp}t^{z+1}}{\left[(1+z)t_{1n}^z\right]}\\ \omega_b-\omega_{bn} &= \left(b_bM_{cb}+b_hM_{ch}\right)t+\frac{b_bM_{bp}t^{z+1}}{\left[(1+z)t_{1n}^z\right]}\end{aligned}\right\} \tag{5}$$

Because at $t = t_{1n}$, $\omega_b = 0$, $\omega_a = \omega_{amn}$, from the second equation (5)

$$t_{1n} = \frac{\omega_{bn}}{b_bM_{cb}+b_hM_{ch}+\dfrac{b_bM_{bp}}{1+z}} \tag{6}$$

Substituting the value t_{1n} in the first equation (5), we find the minimum angular velocity a of the driving link during the starting period

$$\omega_{amn} = \omega_{an}\frac{\left(\psi_{1n}+\dfrac{1}{1+z}\right)\left(b_bp-a_b\right)+\psi_{2n}\left(b_hp-a_h\right)}{p\left[b_b\left(\psi_{1n}+\dfrac{1}{1+z}\right)+b_h\psi_{2n}\right]} \tag{7}$$

where $\psi_{1n} = \dfrac{M_{cb}}{M_{bp}}; \psi_{2n} = \dfrac{M_{ch}}{M_{bp}}$.

Exposing the value of the ratios a_i and b_i for dependencies [1], after transformations we obtain the final dependence for determining the duration of the starting period t_{1n} and minimum angular velocity of the driving link during a starting period ω_{amn}

$$t_{1n} = \frac{J_a\gamma\omega_{an}}{M_{bp}Q_n}\ ; \tag{8}$$

$$\omega_{amn} = \omega_{an} \frac{\left(\frac{1}{1+z} + \psi_{1n}\right)\frac{1}{p} - (1+\gamma_0)\psi_{2n}\frac{1}{1+p}}{Q_n} \quad . \tag{9}$$

When turning on the drive dependencies are defined t_{10} and ω_{amo} in a similar way

$$t_{1n} = \frac{J_a \gamma \omega_{a0}}{M_{hp} Q_0}; \tag{10}$$

$$\omega_{amo} = \omega_{ao} \frac{\left(\frac{1}{1+z} + \psi_{2n}\right)\frac{1}{1+p} - (1+\gamma_n)\frac{\psi_{10}}{p}}{Q_o} \tag{11}$$

where $\psi_{10} = \frac{M_{cb}}{M_{hp}}$, $\psi_{20} = \frac{M_{ch}}{M_{hp}}$.

In formulas (8)..(11) indicated:

$$Q_n = \frac{1+\gamma_n}{p}\left(\frac{1}{1+z} + \psi_n\right) - \frac{\psi_{2n}}{1+p}; \; Q_0 = \frac{1+\gamma_0}{1+p}\left(\frac{1}{1+z} + \psi_{20}\right) - \frac{\psi_{10}}{p}.$$

The dependencies for defining the required braking torques M_{bp} and M_{hp}, providing lockup of units b and h, respectively, at given angles φ_{bt} and φ_{ht} can be set from the system of equations (5).

Integrating the second equation, we obtain

$$\varphi_b = \omega_{bnt} + (b_b M_{cb} + b_h M_{ch})\frac{t^2}{2} + \frac{b_b M_{bp} t_{1n}^{2+z}}{t_{1n}^z (1+z)(2+z)}.$$

At $t = t_{1n}$, $\varphi_b = \varphi_{bt}$ so

$$\varphi_{bt} = \left(\omega_{bn} + (b_b M_{cb} + b_h M_{ch})\frac{t_{1n}}{2} + \frac{b_b M_{bp} t_{1n}}{(1+z)(2+z)}\right) t_{1n}.$$

Substituting the values t_{1n} of formula (6), we find

$$\varphi_{bt} = -\frac{\omega_{bn}^2}{2M_{bp}} \cdot \frac{b_b\psi_{1n} + b_h\psi_{2n} + \frac{2b_b}{2+z}}{\left(b_b\psi_{1n} + b_h\psi_{2n} + \frac{b_b}{1+z}\right)^2},$$

which, after conversion of the following

$$M_{bp} \geq \frac{J_a \gamma \omega_{an}^2}{2p\varphi_{bt}} \cdot \frac{\frac{1+\gamma_n}{p}\left(\psi_{1n} + \frac{2}{2+z}\right) - \frac{\psi_{2n}}{1+p}}{Q_n^2} \quad . \tag{12}$$

The brake moment M_{hp} value is determined by a formula derivable in a similar way

$$M_{hp} \geq \frac{J_a \gamma \omega_{ao}^2}{2(1+p)\varphi_{ht}} \cdot \frac{\frac{1+\gamma_n}{1+p}\left(\psi_{2o} + \frac{2}{2+z}\right) - \frac{\psi_{10}}{p}}{Q_0^2} \quad . \tag{13}$$

During the working stroke of the CPM loads are applied to the links of the planetary mechanism due to technological strain of the main actuator. In particular the link *b* is acted by the torque M_{bt}, tending to rotate in the direction of the carrier rotation. As link *b* must remain stationary, the magnitude of braking torque M_{bp} must be greater than the moment of M_{bt}, determined by the formula

$$M_{bt} = \frac{M_{гм} p}{(1+p) iz}. \tag{14}$$

Typically, the value of M_{bt} is more than torque M_{bp} determined by the formula (12), so the brake is designed to include the estimated time equal to M_{bt}. As a result brake application is much faster, corner braking φ_{bt} decreases (M_{bt}/M_{bp}) times, and dynamic loads, shock and wear increases. To reduce the negative consequences of this circumstance, we recommend the use of a combined braking system. The starting process must occur at a reduced braking torque that can be achieved by decreasing air pressure or the discharge coefficient of pipe filling. By the beginning of the stroke brake must engage fully and to provide braking torque equal M_{bt} (formula (14)).

If the friction in the bearings parts of the planetary mechanism and main actuator are small (ψ_1 and ψ_2 less than 0.02 ... 0,004), or can be ignored, then calculated dependencies are simplified to the form

$$t_{1n} = \frac{J_a \gamma \omega_{an} p(1+z)}{M_{bp}(1+\gamma_n)}; \tag{15}$$

$$\omega_{amn} + \frac{\omega_{an}}{1+\gamma_n}; \tag{16}$$

$$M_{bp} \geq \frac{J_a \gamma \omega_{an}^2 (1+z)^2}{\varphi_{bt}(1+\gamma_n)(2+z)}; \tag{17}$$

$$t_{1o} = \frac{J_a \gamma \omega_{ao} 1+(p)(1+z)}{M_{hp}(1+\gamma_o)}; \tag{18}$$

$$\omega_{amo} = \frac{\omega_{ao}}{1+\gamma_o}; \tag{19}$$

$$M_{hp} \geq \frac{J_a \gamma \omega_{a0}^2 (1+z)^2}{\varphi_{ht}(1+\gamma_o)(2+z)}. \tag{20}$$

Formulae (15)...(20) show that the intensity of brakes actuation (decrease of *z* value) does not affect the reduction of the angular velocity of the driving member by the end of a period on or stop. Starting or stopping period, as well as the necessary braking torque with increasing braking intensity decreases.

CONCLUSIONS

1. The paper presents the developed simplified engineering method which allows to determine the preliminary parameters of the planetary drive.

2. All the obtained dependences for calculating the parameters of the starting and stopping process are similar formulae for the drive with a friction clutch, which shows the validity of the assumptions made and the identity of the physical and mechanical phenomena that occur when started and stopped drives with these systems included.

References: 1.Явтушенко А.В., Середа Б.П., Васильченко Т.А., Глебенко А.В. Математическая модель планетарного привода кривошипных прессов. Вісник національного технічного університету України « Київський політехнічний інститут».-Київ,2010.-Вип.№60. С.200-203. **2**.Явтушенко А.В., Глебенко А.В., Васильченко Т.А., Видмич С.С. Динамическая модель планетарного привода кривошипных прессов. Вісник Кременчуцького державного технічного університету ім.. М.Остроградського. – Кременчук, КДТУ, 2009.- Вип.6/2009(59, частина 1). С.44-48. **3**.Власов В.И. Системы включения кривошипных прессов. - М.: Машиностроение.- 1969-272с.

ТЕОРЕТИЧЕСКИЙ АНАЛИЗ И МОДЕЛИРОВАНИЕ ПРОЦЕССОВ КУЗНЕЧНО-ШТАМПОВОЧНОГО ПРОИЗВОДСТВА

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УЧЕТ ТРЕНИЯ ПРИ МОДЕЛИРОВАНИИ МЕТОДАМИ ВЕРХНЕЙ ОЦЕНКИ ПРОЦЕССОВ ОБРАБОТКИ ДАВЛЕНИЕМ С НИЗКИМ КОЭФФИЦИЕНТОМ ПОДПОРА

Проведено дослідження по встановленню закономірностей пластичного тертя. Запропонований підхід врахування тертя в математичних моделях верхньої оцінки для схем штампування із низьким коефіцієнтом підпору.

Investigations to determine the regularities of plastic friction are made. Proposed an approach taking to account the friction in the mathematical models developed by the upper bound methods of the schemes forming with the low pressure.

Проведены исследования по определению закономерностей пластического трения. Предложен подход учета трения в математических моделях разрабатываемых методом верхней оценки для схем штамповки с низким коэффициентом подпора.

Ни один из процессов обработки металлов давлением не протекает без влияния на него сил трения. Трение возникает на поверхности контакта инструмента и деформируемого тела (заготовки). Одной из характерных особенностей трения при пластической деформации является наличие больших нормальных давлений на контактной поверхности, которые во многих случаях значительно превышают величину предела текучести обрабатываемого металла [1-4].

Трение в процессе пластической деформации присутствует в виде контактных напряжений, находящихся на поверхности контакта инструмента и заготовки. Контактные напряжения играют как отрицательную [5] (повышение усилия деформирования, снижение стойкости инструмента, повышение неоднородности деформаций в заготовке) так и положительную (является управляющим фактором в формоизменении некоторых процессов выдавливания)