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THE INFLUENCE OF CONFIGURATION PARAMETERS OF THE PLANETARY DRIVE OF CRANK PRESSES ON ITS DYNAMIC AND ENERGY-CONSUMPTION CHARACTERISTICS

Introduction. *The planetary drive has been widely found in the activation systems of crank presses as it improves working conditions of friction units and reduces press metal consumption and energy consumption.*

Problem Statement. *For studying the planetary motion of its main parts, it is advisable to consider its operation as four periods characterized by different patterns of changes in external loads and, consequently, different patterns of motion of core parts. The periods between switching the brakes when one link has not stopped, while other one starts moving is of the greatest interest. The problem of dynamic and energy-consumption parameters of the drive in transient conditions still remain a little open.*

Purpose. *The purpose of this research is to qualitatively assess the influence of configuration parameters on the dynamics of the processes of switching and stopping the planetary drive.*

Material and Methods. *The parameters of the experimental pilot plant, a model planetary drive with a nominal load of 400 kN, have been calculated. The minimum speed of the central gear has been estimated depending on the moments of inertia of the leading mass and the outer gear. The obtained dependences have been processed with the use of a standard mathematical apparatus.*

Results. *It has been found that the energy-consumption and dynamic parameters of the drive are significantly influenced by such characteristics as moments of inertia and braking moments, while decreasing angular velocity of the drive flywheel does not depend on the moments of inertia of the stopped links. The rate of braking processes is determined by the elastic properties of the friction materials and the parameters of the brake springs, not by the parameters of compressed air outflow.*

Conclusions. *The obtained analytical and graphical dependences have allowed not only qualitative, but also quantitative assessment of the influence of the parameters on the dynamics of processes. These recommendations can be used at the design stage, to select the optimal parameters of the drive with the predicted properties.*

Keywords: planetary drive, transients, angular velocity, moments of inertia, and braking points.

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The planetary drive has been increasingly used in crank press activation systems. The use of planetary mechanisms in the drives of the press allows for an unchanged number of strokes of the press to reduce several times the energy consumption for activation [1], to increase the number of revolutions of the flywheel, which leads to a significant reduction in its moment of inertia and, consequently, in size and weight [2]. Since the friction units of all planetary gearboxes act as brakes, the working conditions of which are more favorable to the friction clutches, which leads to an increase in the service life of friction elements, the cycle between repairs of the press and, subsequently, improves the press reliability and durability [3].

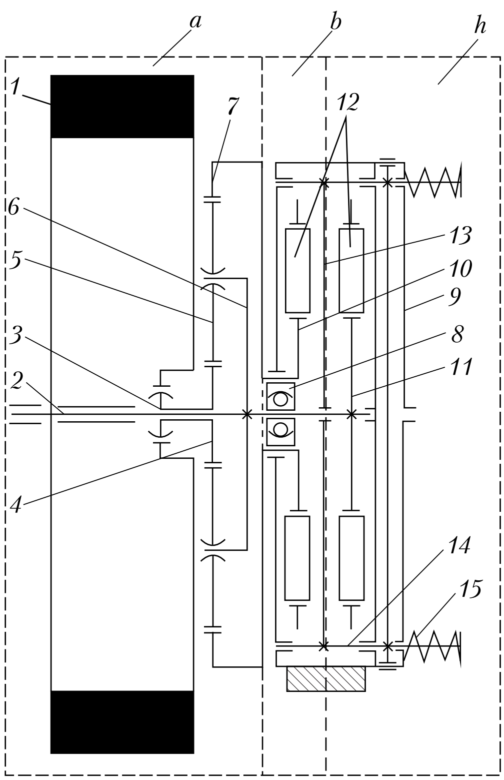

Fig. 1. Scheme of the planetary drive of the crank press: *1* — flywheel; *2* — main shaft; *3* — gear clutch; *4* — central wheel; *5* — satellites; *6* — carrier; *7* — outer wheel; *8* — support bearings; *9* — pneumatic cylinder; *10* — activation brake; *11* — stop brake; *12* — friction tabs; *13* — pressure plate; *14* — guide pin; *15* — springs

So far, the problem of changing the energy-consumption and dynamic parameters of the drive in transition modes that is, between switching the brakes, remains understudied.

The purpose of this research is to make a qualitative assessment of the influence of configuration parameters on the dynamics of the processes of activating and stopping the planetary drive.

For determining the energy parameters of activating and stopping processes, it is advisable to consider the planetary drive as a mechanical system with three concentrated masses [4]: the leading part with the flywheel and the leading central gear — link *a* (Fig. 1); the outer gear with connected brake parts — link *b* (Fig. 1); and the carrier with attached driven parts of the drive and stop brake — link *h* (Fig. 1).

The operation of a crank press with a planetary drive can be divided into four characteristic periods that differ by the nature of the main links motion (Fig. 2). The first period is a pause between two successive actuations of the drive, during which the carrier stops, while the outer and central gears are idle. The theoretical speed of the pinion gear is ω_{ao}. If the duration of the first period is sufficiently large, the angular velocity of the leading masses is restored to the initial angular velocity of the flywheel ω_{ao}.

During the second period, the permission brake is on, while the stopping brake is off, and the drive start process begins. As a result of braking, the outer gear stops, while the carrier accelerates. There is always a time interval t_{pv} between activating one brake and stopping another one. The angular velocity of the pinion gear decreases from the initial ω_{ao} to ω_{avo}. The braking of the outer gear and the simultaneous acceleration of the carrier begin through time interval t_{bv}, determined by the ratio of the driving and the braking moments acting on the outer gear. The second period ends when the outer gear stops completely.

The third period is a direct idle, stroke, and reverse idle of the main actuator, with the planetary gear engaged. The period begins at the moment the outer gear stops and continues until the start

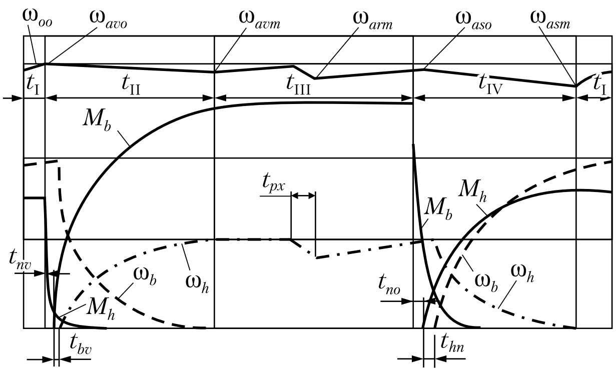

Fig. 2. Periods of motion of the planetary drive

of the brake shift. The angular velocity of the pinion gear, which by the end of the previous period decreases to ω_{avm} due to the power consumption for activation, may not fully recover by the beginning of the working stroke. During the period of working load, the angular velocity of link *a* decreases to the minimum value of ω_{arm} and, with idle reverse motion, is partially restored to ω_{aso}.

During the fourth period, the stop brake is enabled, the braking of the carrier and the simultaneous acceleration of the outer gear begin. Between the moments of switching the brakes there is also pre-start time t_{no}. The start of the carrier's motion is determined by the driving forces and the braking forces from the stopping brake. The fourth period ends when the carrier stops. The angular velocity of link *a* at the end of the period is ω_{asm}.

The simplified dependencies for the main energy-consumption and dynamic parameters of the drive are presented in [5]. The duration of the switching periods t_{1v} is determined by the formula:

$$t_{1v} = \frac{\omega_{avo} J_a \, jp}{M_{br}} \cdot \frac{(1+z)}{(1+j_v)}. \qquad (1)$$

The angular velocity of leading link *a* at the moment of stopping link *b*:

$$\omega_{avm} = \frac{\omega_{avo}}{1+j_v}. \qquad (2)$$

The angle of the brake of link *b*:

$$\varphi_{bt} = \frac{\omega^2_{avo} J_a j}{M_{br}} \cdot \frac{(1+z)^2}{(1+j_v)(2+z)}. \qquad (3)$$

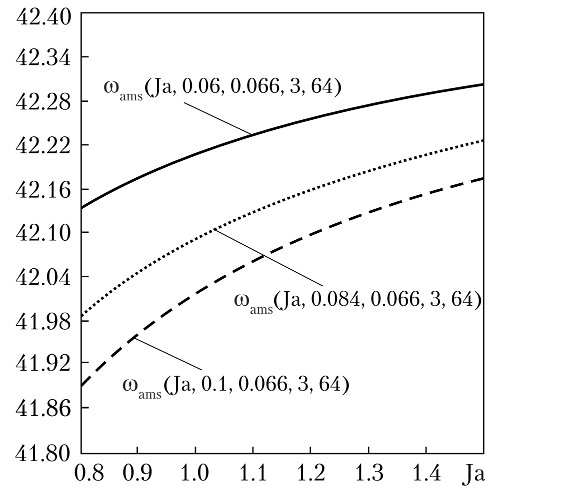

Fig. 3. Dependence of the minimum speed of the leading flywheel on the moments of inertia of the driven links

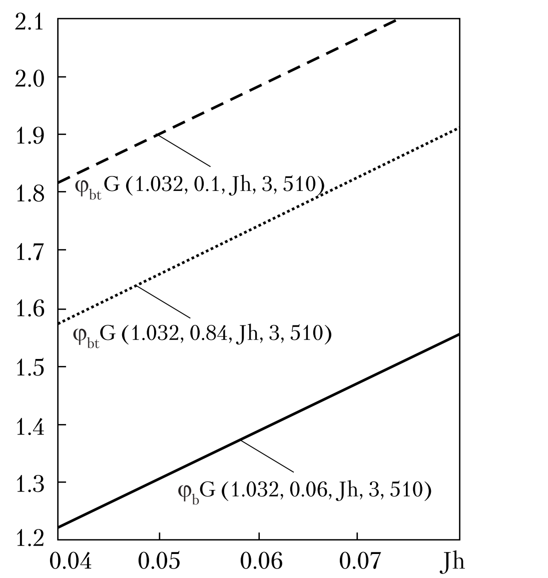

Fig. 4. Dependence of the stop angle of the outer gear on the moments of inertia of gear *b* and carrier *h*

Braking torque M_{br} required to stop the outer gear at a given angle of rotation φ_{bt}:

$$M_{br} = \frac{\omega^2_{avo} J_a j}{\varphi_{bt}} \cdot \frac{(1+z)^2}{(1+j_v)(2+z)}. \qquad (4)$$

Time t_{1s} to a complete stop of carrier *h*:

$$t_{1s} = \frac{\omega_{aso} J_a j}{M_{hr}} \cdot \frac{(1+p)(1+z)}{(1+j_s)}. \qquad (5)$$

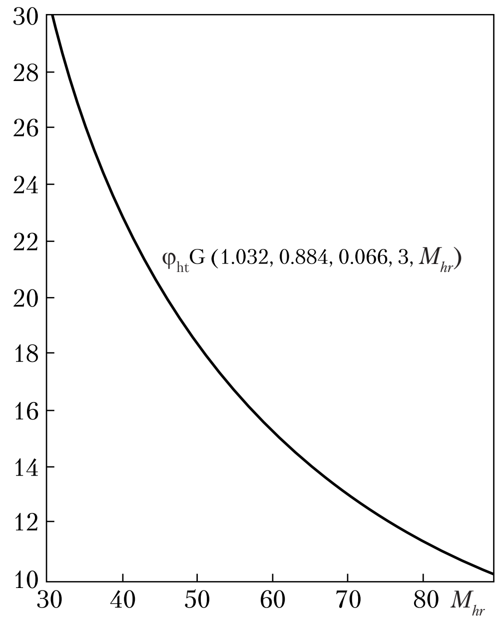

Fig. 5. Dependence of the angle of the carrier stop on maximum braking torque M_{hr}

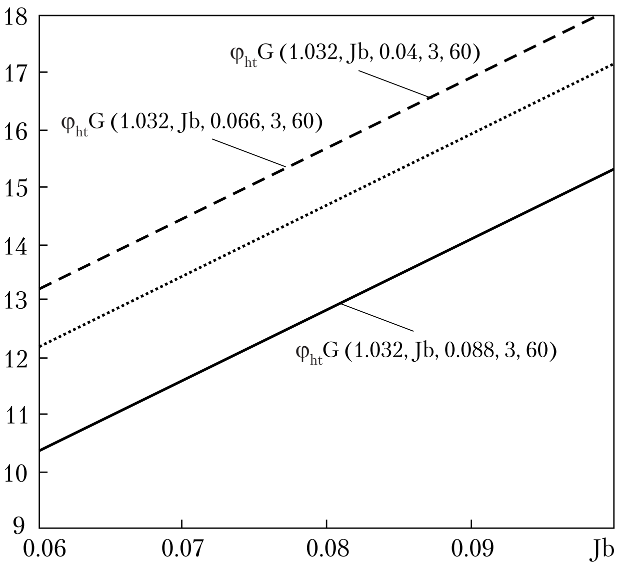

Fig. 6. Dependence of the angle of the carrier stop on the moments of inertia of gears *b* and carrier *h*

The angular velocity of leading link *a* at the moment of stopping carrier *h*:

$$\omega_{asm} = \frac{\omega_{aso}}{1 + j_s}. \qquad (6)$$

Braking angle φ_{ht} of carrier *h* for a given braking torque M_{hr}:

$$\varphi_{ht} = \frac{\omega^2_{aso} J_a j}{M_{hr}} \cdot \frac{(1+z)^2}{(1+j_s)(2+z)}. \qquad (7)$$

Calculated braking torque M_{hr} for stopping carrier *h* at a given braking angle φ:

$$M_{hr} = \frac{\omega^2_{aho} J_a j}{\varphi_{ht}} \cdot \frac{(1+z)^2}{(1+j_s)(2+z)}. \qquad (8)$$

In the formulas, *z* is the rate of braking, as determined by the formulas and graphs in [6]. The average values of the rate range 0.2—1.2; *j* is the generalized relative moment of inertia of the driven masses [7]; J_a is the moment of inertia of the leading parts of the drive; *jv* is the relative moment of inertia of the driven masses when the drive is activated; *js* is the relative moment of inertia of the driven masses when the drive stops; *p* is the kinematic parameter of planetary mechanism *A*.

It has been established that the decrease in the angular velocity of the drive flywheel does not depend on the inertia of the stopped links, and the magnitude of the engine slip does not exceed 0.5—0.9% that is much less than the nominal slip [8].

Thus, it can be seen from the above dependencies that such characteristics as inertia moments and braking moments have a priority effect on the energy-consumption and dynamic parameters of the drive; therefore, they are considered in more detail below.

For the experimental plant, which is a model planetary press drive with a nominal force of about 400 kN [9], the minimum speed of the central gear has been calculated depending on the moment of inertia of the drive mass and the moment of inertia of the outer gear (Fig. 3).

When the drive is engaged, the braking rate of the outer gear is determined by the rate of pressure increase on the friction contact of the engagement brake [10], which, in turn, depends on the rate of filling the brake cylinder cavity with compressed air and inertial parameters [11].

The angle and, accordingly, the braking time of outer gear *b* when the drive is turned on depend, first of all, on the maximum braking moment M_{br}

of the brake and the moments of inertia of gear b and carrier h (Fig. 4). As it has been already mentioned, the drive is activated under the braking torque M_{br}, determined by the conditions of load, therefore, the duration of the process is small and the angle of braking of the gear does not exceed 3—5°.

Figures 5 and 6 show the dependency of the braking angle of the carrier on the maximum braking moment M_{hr} on the moments of inertia of the driven masses.

When the drive stops, the value of the braking angle φ_{hs} of the carrier is determined by safety conditions and should not exceed 15—200. The braking rate depends little on the parameters of the outflow of compressed air, but is determined by the elastic properties of the friction material and the parameters of the brake springs. With the use of the analytical dependencies (1)—(8), it is possible to determine the influence of other structural parameters of the drive on the dynamics of transients.

The analytical dependencies of the main energy-consumption and dynamic characteristics of the drive on its configuration parameters have been obtained. The graphical interpretation of the obtained dependencies allows us to give not only a qualitative, but also a quantitative assessment of the influence of these parameters. It has been established that the main parameters determining the decrease in angular velocity are the moments of inertia of the stopped masses. The angle and, accordingly, the braking time of the outer gear when the drive is turned on depend, first of all, on the maximum braking torque of the on brake and the moments of inertia of the central gear and carrier.

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ВПЛИВ КОНСТРУКТИВНИХ ПАРАМЕТРІВ ПЛАНЕТАРНОГО ПРИВОДУ КРИВОШИПНИХ ПРЕСІВ НА ЙОГО ДИНАМІЧНІ ТА ЕНЕРГЕТИЧНІ ХАРАКТЕРИСТИКИ

Вступ. Планетарний привод знаходить все ширше впровадження у системах вмикання кривошипних пресів за рахунок поліпшення умов роботи фрикційних вузлів, зниження металоємності пресу та зменшення витрат енергії на роботу останнього.

Проблематика. При дослідженні руху основних ланок планетарного приводу його роботу доцільно подати у вигляді чотирьох періодів, які характеризуються різними закономірностями зміни зовнішніх навантажень та, як наслідок, різним характером руху основних ланок. Найбільший інтерес викликають періоди поміж черговим переключенням гальм, коли одна ланка ще не зупинилася, а інша вже починає свій рух. Питання зміни енергодинамічних параметрів привода у перехідних режимах досі лишаються мало розкритими.

Мета. Проведення якісної оцінки впливу конструктивних параметрів на динаміку процесів вмикання та зупинки планетарного приводу.

Матеріали й методи. Розрахунки виконували для експериментальної дослідної установки — макету планетарного приводу преса з номінальним зусиллям 400 кН. Прораховано значення мінімальної швидкості центральної шестерні залежно від моменту інерції провідної маси та моменту інерції зовнішньої шестерні. Обробку отриманих залежностей здійснено з використанням стандартного математичного апарату.

Результати. Встановлено, що значний вплив на енергодинамічні параметри привода чинили моменти інерції та гальмівні моменти, тоді як зниження кутової швидкості провідного маховика не залежить від моментів інерції ланок, що зупиняються, а величина ковзання двигуна значно нижче номінального параметра. Інтенсивність процесів гальмування визначається не параметрами витоку стислого повітря, а пружними властивостями фрикційних матеріалів та параметрами гальмівних пружин.

Висновки. Отримані аналітичні й графічні залежності дозволяють провести як якісну, так і кількісну оцінку впливу параметрів на динаміку процесів. Рекомендації можуть бути використані на стадії проєктування для підбору оптимальних параметрів приводу з прогнозованими властивостями.

Ключові слова: планетарний привод, перехідні процеси, кутова швидкість, моменти інерції, гальмівні моменти.