IDENTIFICATION OF ANALYTICAL DEPENDENCIES
OF THE OPERATIONAL CHARACTERISTICS OF THE WORKPIECE CLAMPING MECHANISMS
WITH THE ROTARY MOVEMENT OF THE INPUT LINK

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Abstract: The research is devoted to the problem of determining the efficiency of the workpiece fixing mechanism operation. Improving characteristics of workpiece fixing is one of the required conditions to increase the cutting modes, which may help to enhance the machining productivity. The study investigates the main characteristics and general features of a new structure of clamping mechanisms with electromechanical actuators for fixation of rotation bodies. The main advantages of using electromechanical clamping actuators with self-braking gear are presented. Two simplified dynamical models for the description of different stages of the clamping process are developed. The calculation scheme was formulated to find out how the mass-geometric parameters of mechanism elements should influence the main characteristics of the clamping mechanisms of this type.

Keywords: the mechanism for clamping of rotation bodies, clamping actuator, self-braking screw gear, kinematic characteristics,
workpiece clamping efficiency

1. INTRODUCTION

One of the main trends in the metalworking machine tool industry, which contributes to improving the processing efficiency, is to increase the cutting modes: the cutting speed (related to a spindle speed) and the thickness of the layer of material being removed in one pass of the tool (related to feed). As the possibility to increase the maximum processing modes to a large extent depends on the quality of fixture of a workpiece, it is necessary to provide the required value and stability of the clamping force over a wide range of spindle speeds. The process of clamping workpieces in a metalworking machine tool is carried out by using the clamping mechanism (CM). Since the CM is mounted on the spindle of the machine, it affects not only the efficiency of the workpiece clamping process but also the performance characteristics of the spindle assembly (spindle unit), in general. Any CM contains two main subsystems: "clamping actuator" (CA) and "clamping chuck". CA can be defined as an intermediate device for converting and transmitting energy in the form of a power flux from an energy source (ES) to the clamping chuck with functional dependence on it. Characteristics of the CA have a large influence on the main power characteristics of a CM. Thus, the magnitude of the clamping force, stability of the clamping force under the influence of various perturbations such as the influence of centrifugal forces, deviation size of a clamping surface, heterogeneity of physical-mechanical characteristics of the material of the workpiece, and change of power supply characteristics are essential. Characteristics such as response time during mechanism operation, reliability of maintaining a tense system state while holding a workpiece, possibility of effective management and regulation of work of a CM are also important. A typical CA uses different methods to maintain the workpiece clamping force (tense system state) during processing. The need to use a large number of energy converters to create clamping forces by using of typical CA designs leads to additional energy losses, reduced reliability and

efficiency of control of these mechanisms, increased sensitivity to external perturbations, complications of design, manufacturing and operation, etc. Not only does the efficiency of the energy converters work very often decrease with the increase of a spindle rotational speed, but their design also deteriorates the dynamic characteristic of a spindle assembly. The main researches for improving the productivity of the spindle unit in machine tools are dedicated to the detection of influence of bearings, optimal interaction of nodes, temperature influence, increasing vibration resistance and functional rigidity. Increasing the efficiency of creation clamping efforts can help to improve the processing efficiency through better utilization of the potential abilities of machine tools of the existing designs in particular. This explains the expediency of creating new designs of CA based on new structures that meet the above requirements.

2. THE MAIN DISADVANTAGES OF TRADITIONAL DESIGNS OF ACTUATORS OF THE CLAMPING MECHANISM

As a result of the analysis, the common problems of the main types of CA were identified and formulated. In the case when mechanical energy is supplied to the CA as an external axial force, there is an additional axial load on the spindle and its bearings. This can also lead to uncontrolled movement of the elements of the spindle assembly due to elimination gaps or elastic deformations. This reduces the accuracy of processing. Centrifugal inertia forces appear on the moving elements of CA converters (lever, wedge, etc) and constantly try to move them. It can cause uncontrolled negative changes in power characteristics of this mechanism during high-speed rotation of the spindle. The value of clamping effort created by a CA with the mechanisms of geometric closure depends on the variations in the size of the workpiece from its nominal values. The problem of a CA actuated by hydraulic pressure is that there is an occurrence of losses working fluid

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when bringing it to the rotating spindle. Also, the inertia forces acting on the liquid at high speeds counteract its displacement, which reduces the efficiency of the mechanism. Both types of CA require using large external mechanisms for preparing and supplying input energy (mechanical or hydraulical).

On the basis of the obtained information, ways of improving structure for the development of new CA designs are revealed:

- Characteristics of the energy converters should not depend on the influence of the centrifugal forces of inertia.
- Keeping the system's tense state while holding the workpiece must be ensured without supplying energy from the outside.
- The number of different types of energy converters should be minimal to simplify construction and reduce energy loss.
- Input energy must be supplied without external force interaction that can cause appearance of uncontrolled radial displacement of spindle units or offset of the rotation axis.
- Units that may complicate balancing of the spindle assembly or disturb it over a short period should be avoided.
- Change and adjustment of the characteristics of the clamping process must be available without disassembling the mechanism.

The safest and most economical CAs are those whose contour of force interaction is closed by braking inside the mechanism. CA of this kind can be disconnected from the power source after reaching the required clamping force and provide support even in the case of a power failure of a machine tool. Several basic types of self-braking gears can be used in the CA: screw, wedge, helix and cam. Self-braking mechanisms in the form of screw gearing have become most widespread as they are more economical in manufacturing and operation due to their simpler construction.

3. DEVELOPMENT OF A TYPICAL STRUCTURE OF A NEW CLAMPING MECHANISM

Following the identified requirements, a new CA structure was developed, with predictably better characteristics and where the input energy is electric. The widespread electrical actuator is an electromotor with rotation motion. The output force of a CA must be linear for work as a clamping chuck. To transform rotation motion in linear, it's reasonable to use a screw gear with the ability to self-brake (self-lock caused due to friction forces). It can keep a tensioned state of the system making it possible to hold a workpiece during processing without having to supply energy from an outer source. In such a way, the number of energy converters is reduced to two: one electromagnetic and one mechanical. Working characteristics of both converters do not depend on of centrifugal inertial forces and deviations of clamp surface size from nominal values. One variant of this structure is embodied in the patented CA design [Patent UA95323]. A feature of this design is the specific usage of an electromotor when a stator is attached to the body of the machine tool and the rotor is installed on a spindle. To create an effective CA based on this structure, it is necessary to identify the typical mathematical dependencies for this structure between the mass-geometric parameters of the elements of the mechanism, and the kinematic and power characteristics. That is why, it is necessary to establish the basic dependencies that describe the characteristics of displacement of the CA output link (value of displacement, velocity) and the amount of energy (value of output effort) spent to clamp the workpiece in the CM with CA of the proposed design.

The block diagram of a typical collet CM in which the proposed CA is used can be represented by the following set of structural elements (Fig. 1): the energy source ES is usually an industrial electrical network with voltage U; the switching device (SD), which controls the operation of the drive motor D, consisting of a stator and rotor (on the shaft of which rotation moment Ma_0 occurs); the drive self-braking mechanism (DSBM) changes the movement characteristics of the output link of the drive motor M and captures the tense state of the CM after it is switched off; a clamping chuck (CC), which directly fixes the object of the clamping (OC) with Fr_2 radial force.

Several types of control of engine D can be implemented in this structure of CM to automatically adjust the parameters of the CM, for which different backward linkages I, II and III (Fig. 1) are commonly used. The control system of electrical CA, the core element of which is the SD switching device, must ensure that in addition to Fr_2 clamping force control, there is also mutual locking with the main drive of the machine, feed and auxiliary movements.
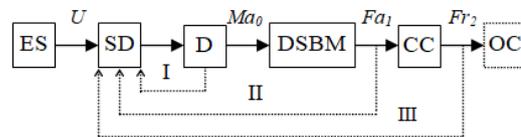

Fig. 1. Block diagram of a collet CM with the electrical CA and self- braking mechanism

The effective control of the operation of the CM with the electrical CA can be performed using a switching device (SD) by changing the stator current parameters that determine the rotor motion characteristics. The control signal may include characteristic operational values (displacement and force interaction) of the various CM subsystems and is transmitted to the SD through separate backward linkages I, II and III (Fig. 1). This improves the reliability and efficiency of the clamping process control. The influence of centrifugal inertia forces of unbalanced moving CM elements on changing the value of clamping force Fr_2 can be determined from a function where the actual spindle speed is used as a variable.

Obtaining a predetermined value of clamping force Fr_2 in CM with electrical CA can be carried out based on several principles of its control system organization. The simplest is the principle of control without deliberately limiting engine running time. In this case, the electrical CA drive motor operates in stopper mode operation until the rotor is stopped under the influence of resistance forces (overturning mode), which corresponds to the maximum value of the tense system state for the given power supply characteristics. This is reflected in the characteristics of the motor current of the motor D and gives a signal for the switching device to switch it off. The system remains in the tense state which is supported by the self-braking mechanism (DSBM). Overloading of the drive motor during stopping is acceptable since it has a small duration. Other clamping process control principles use signals from sensors giving information about the value of displacement and force interaction in the CM subsystems to limit the clamping force and provide controlled stopping of the electrical CA engine. In these cases, the response time of the control equipment (relay and sensors) must be taken into account. The possibility of using different control schemes of electrical CA increases the efficiency of its usage in the CM for different technological equipment and different operation conditions.

In the proposed design of an electric CA (Fig. 2), the rotor of

the induction motor D is located on the threaded surface of the spindle with the possibility of their mutual movement, and the stator of electric CA is rigidly connected to the housing of the spindle assembly. Energy supply to the rotating spindle is carried out without contact – due to the influence of the stator electromagnetic torque. At the initial moment of operation of this CM, the rotor of the motor D electrical CA has an angular velocity relative to the stator, which is equal to the speed of rotation of the spindle. Rotation acceleration or deceleration of the rotor of motor D and its movement in relation to the spindle occurs under the action of the electromagnetic moment of stator due to the activation of motor D. This also results in the axial movement of the rotor on the spindle threaded surface with a force Fa1, which is the output force of electric CA.
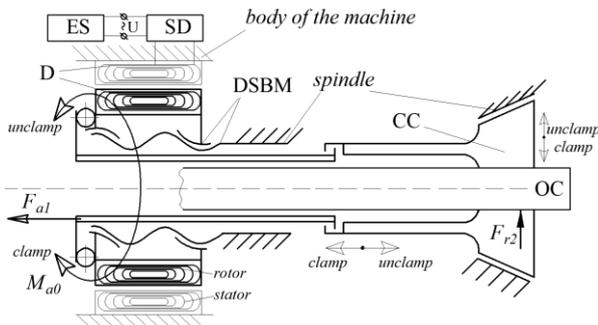

Fig. 2. Schematic diagram of electric CA according to the patent of Ukraine № 95323

4. IDENTIFICATION OF ANALYTICAL DEPENDENCIES

The mathematical description of a multi-mass dynamic model, which reflects the working process of the CM with the proposed electric CA, can be carried out in accordance with the calculation scheme (Fig. 3a). Detailed calculations of this type are related to the difficulties in computation and their volume. Therefore, for practical calculations and further analytical studies, the multi-mass system is reduced to a single-mass system (Fig. 3b, c) with the equivalent moment of inertia J_Π and the equivalent torque $M_\Pi = M_{a0}$. On the scheme (Fig. 3b, c), the influence of forces of potential character (forces of elasticity) is reflected by the action of elastic elements with the corresponding stiffness coefficient $c_{\Pi\Pi}$, $c_{\Pi p}$, c_κ.

4.1. Development of calculation schemes

It is reasonable to consider CM as a system in two cases: during the clearance Δ adjustment, when mainly active forces of dissipative character are acting in the system, and during the creation of tension of the elastic system of CM (Fig. 3 b) when the forces of potential character dominate. The analysis of the clamping process in the CM with the electric CA (Fig. 3 b, c) should also take into account the forces of elasticity of the collet petals with the rigidity c_{nn}, stiffness of electric CA – c_{np}, and contact stiffness in the joints of the collet chuck cartridge c_κ. The study of the CA in the SM (Fig. 3) requires consideration of the kinematic characteristics of two interacting mechanical energy converters: a screw self-braking mechanism (DSBM) and a collet (CC).

In a screw-driven self-braking mechanism, the rotation rotor of D (which works like a nut in a screw mechanism) with an angular velocity ω_∂ ensures its translational motion with the speed V_m transmitted to the output link of CA – "clamp pipe". The relation-

ship between these kinematic parameters can be established based on the equality between the rotation angle of the motor rotor φ_∂ and the axial movement of the "clamp pipe" x_m in one complete turn of the rotor $\phi_\partial/x_m = 2 \cdot \pi/h$ or

$$x_m = \frac{\phi_\partial \cdot h}{2\pi}, \qquad (1)$$
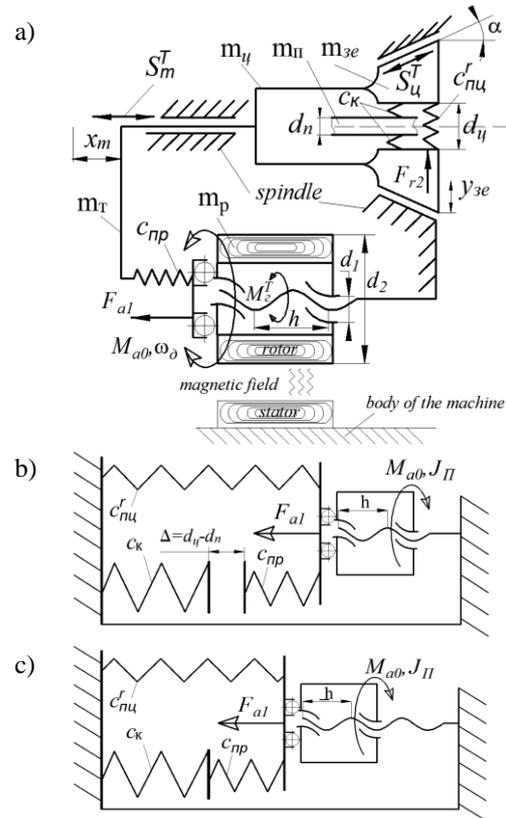
where h is the pitch of the thread.

Fig. 3. Scheme of the CM: a) calculation scheme; b) simplified dynamic models during elimination gaps; c) simplified dynamic models when creating tension in the system

After differentiating the left and right parts of equations (1) in time, we obtain $\partial x_m/\partial t = (\partial \phi_\partial/\partial t) \cdot h/2\pi$,

$$V_m = \omega_\partial \cdot \frac{h}{2\pi} = 0.5\omega_\partial \frac{h}{\pi} = \omega_\partial \cdot 0.5 d_1 \cdot tg\psi, \qquad (2)$$

where d_1 is the diameter of the thread, ψ is the lead angle of the screw and ω_∂ is the angular velocity of the rotor of the engine.

Express the value of $i_s = 0.5 d_1 \cdot tg\psi$ and substitute it in the equation (2). In that way, we get $V_m = \omega_\partial \cdot i_s$ or:

$$V_m = \dot{\phi}_\partial \cdot i_z. \qquad (3)$$

The movement of the collet chuck in the axial direction with velocity V_T provides radial movement of the clamping elements of the collet with velocity V_{3E}. The relationship between these kinematic parameters can be established based on the equality of relation of axial displacement x_m of the "clamp pipe" together with the body of the collet to the radial displacement y_{3e} of the collet surfaces intended to contact the object of fixation OC, $x_m/y_{3e} = ctg\alpha$ from this it derives that:

$$y_{ze} = \frac{x_m}{ctg\alpha}, \qquad (4)$$

$\partial x_m/\partial t = (\partial t_{3e}/\partial t) \cdot ctg\alpha$, $V_m = V_{3e} \cdot ctg\alpha$ where α is half of cone angle of collet. $i_y = ctg\alpha$, therefore,

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$$V_{se} = \frac{V_m}{i_u} . \tag{5}$$

After adding (3) to (5), we obtain

$$V_{se} = \frac{\dot{\phi}_o \cdot i_e}{i_u} = \dot{\phi}_o \frac{i_e}{i_u} . \tag{6}$$

Analysis of the energy balance of the CM with proposed electrical CA (Fig. 3, a) during the workpiece clamping can be done by comparing the energy E_3 consumed by the CM during a clamping process in general to the effective clamping energy $E_{e\Phi}$ consumed in the contact zone of collet surfaces with the object of fixation OC to create contact deformations. It should be noted that, unlike with pneumatic and hydraulic actuators, there is no component of energy lost for maintaining the system's tense state. Therefore, the calculation of the total energy required for operation of the CM can be carried out by the formula:

$$E_3 = E_{nM} + A_{mp} + E_{e\phi}, \tag{7}$$

where $E_{\eta M}$ is the energy consumed for the movement of the masses of a unit CM, A_{mp} is the work (energy equivalent) performed to overcome friction forces, and $E_{e\Phi}$ is the effective clamping energy consumed in the contact zone of collet surfaces with the object of fixation OC to create contact deformations.

4.2. Detection of dependencies between geometrical-mass parameters and power and kinematic characteristics of the mechanism

The efficiency coefficient of the mechanical part of the studied CM (Fig. 2) can be determined by the formula:

$$\eta_{3M} = \frac{E_{e\phi}}{E_{nM} + A_{mp} + E_{e\phi}} . \tag{8}$$

In this case, E_{nM} is the sum of the kinetic energy of mechanism links, which are rotating or have reciprocating motion:

$$E_{nM} = \sum_{i=1}^{i=n} \frac{m_i V_i^2}{2} + \sum_{i=1}^{i=n} \frac{I_i \omega_i^2}{2},$$

$$E_{nM} = \frac{m_{oc}}{2} V_{oc}^2 + \frac{m_{p\partial}}{2} V_{p\partial}^2 + J_\partial \frac{\omega_\partial^2}{2}, \tag{9}$$

$m_{oc} = m_p + m_m + m_u$, where V_{oc}, m_{oc} are, respectively, linear velocity (equal to "clamp pipe" velocity V_m) and total mass of CM elements moving in the axial direction: motor rotor, "clamp pipe", collet of the clamping mechanism; $m_{p\partial}$, $V_{p\partial}$ are, respectively, the mass and linear velocity of the CM elements moving in the radial direction, in this case $m_{p\partial} = m_{3e}$, where m_{3e} is the mass of clamping elements of the collet and $V_{p\partial} = V_{3e}$, where V_{3e} is the radial speed of movement of clamping elements of the collet; J_∂, ω_∂ are the moment of inertia of the rotor and its angular velocity.

Analytical dependencies (6) do not take into account the mass of the workpiece that moves with the collet, since it can change significantly in each specific cycle of fixation, which distorts the value of the efficiency as an inherent characteristic of the CM of the investigated structure.

Moment of inertia of the engine rotor $J_\partial = m_p(d_1^2 + d_2^2)/8$. According to (2), (5) between ω_∂, V_{3e} and V_m, there are certain kinematic dependencies that are determined according to the calculation scheme: $V_{oc} = V_m = \omega_\partial \cdot 0.5 d_1 \cdot tg\psi$, $V_m = V_{3e} \cdot ctg\alpha$, $V_{p\sigma} = V_{3e} = \omega_\partial \cdot 0.5 d_1 \cdot tg\psi \cdot tg\alpha$. Then (9) can be written in the following form:

$$E_{nM} = \frac{1}{8} \left(\omega_o d_1 tg\psi \right)^2 \left(m_{oc} + m_{po} tg^2\alpha + 0.5 \frac{m_p \left(d_1^2 + d_2^2 \right)}{\left(d_1 tg\psi \right)^2} \right) \tag{10}$$

The work A_{mp} (energy equivalent) performed to overcome the friction forces in CM consists of the following works: friction forces S_m^T of the "clamp pipe" in contact with the spindle during their mutual movement on the value of the "clamp pipe" stroke length x_m; the friction forces S_u^T in contact with the collet cone with the spindle during their mutual movement on the value of the collet stroke length $x_u = x_m/cos\alpha$; given the proximity of the value $cos\alpha$ from 0.97 to 1, the calculations may be simplified to approximately $x_u \approx x_m$; friction in the screw gear of the DSBM, which create the torque of friction forces M_s^T when turning with the angle ϕ_∂.

$$A_{mp} = S_m^T \cdot x_m + S_u^T \cdot x_m + M_e^T \cdot \phi_\partial . \tag{11}$$

The moment created by the forces of friction in the screw gear M_s^T can be defined as the action of the tangential component of the friction forces F_t with the shoulder equal to half the average diameter of the screw thread $M_s^T = 0.5 d_1 F_t$. The torque generated by the friction in the connection bearings of the rotor to the "clamp pipe" is extremely small compared to M_s^T, so we can neglect its magnitude. The tangential component F_t of the friction force depends on the axial load Fa_1 arising at the output of the gear. $F_t = Fa_1 \cdot tg\phi^T$, where ϕ^T is the angle of friction in the threaded connection. Thus, the moment created by the friction forces in the screw transmission under load Fa_1 can be determined from the following dependency:

$$M_e^T = 0.5 \cdot d_1 \cdot Fa_1 \cdot tg\phi^T . \tag{12}$$

The friction force between the surfaces of the collet cone and the spindle cone with the angle 2α is determined by the formula:

$$S_u^T = \frac{z \times Fa_1}{\cos\alpha + \left(\sin\alpha \right)/f_u} , \tag{13}$$

where z is the number of the collet segments and f_μ is the coefficient of friction in the conical coupling of the collet with the spindle.

The force S_m^T of friction "clamp pipe" with a spindle:

$$S_m^T = g \cdot m_2 \cdot f_m , \tag{14}$$

where g is the free fall acceleration, $m_2 = m_m + m_n$ is the mass equal to the sum of the mass of the "clamp pipe" and the workpiece (as a rod), and f_m is the coefficient of friction in the coupling of the "clamp pipe" with the spindle.

Using (12), (13) and (14), we write (11) as follows:

$$A_{mp} = Fa_1 \frac{\phi_o h}{2\pi} \left(\frac{g m_2 f_m}{Fa_1} + \frac{z}{\cos\alpha + \left(\sin\alpha \right)/f_u} + tg\phi^T \cdot ctg\psi \right) . \tag{15}$$

Effective clamping energy $E_{e\Phi} = z \int_0^{h_k} Fr_2(h_k) dh_k$, where h_k is the convergence in contact points between collet and spindle under the action of a normal clamping force, and the formula for determining it was obtained experimentally $h_k = f(S^k, Fr_2, \Delta d)$, where S^k is the area of contact of the respective ribbed surfaces of the collet with the object of fixation. According to literary sources:

$$E_{e\phi} = \frac{m \cdot S^\kappa \cdot z \cdot h_k}{A_M^{(1/m)} \cdot \left(1 + m \right)} , \tag{16}$$

where A_M, m are coefficients selected according to the conditions

of the clamping process. For clamping conditions with a full fit of the clamping elements and object of fixation, *m=1, A_M=0.13* and dependency (16) takes shape:

$$E_{e\phi} \approx 3.82 \cdot S^\kappa \cdot z \cdot h_k . \qquad (17)$$

Compute of the value of the work efficiency coefficient of clamping mechanisms with the electromechanical actuator derives from putting dependencies (10), (15), (17) into (8). Choosing appropriate characteristics of the elements of the new structure conduces to elimination of the problems related to increasing efficiency of fixation of a workpiece for high-performance machining. The obtained analytical dependencies (10), (15), (17), (18) make it possible to identify values that describe the efficiency of the CM and depend on some geometric and mass parameters of the elements of the mechanism:

$$\eta_{3M} = 3.82 S^\kappa z h_k \left/ \frac{\frac{1}{8}\left(\omega_o d_1 tg\psi\right)^2 \times}{\times\left(m_{oc} + m_{po} tg^2\alpha + 0.5\frac{m_p\left(d_1^2 + d_2^2\right)}{\left(d_1 tg\psi\right)^2}\right)}\right.$$

$$+Fa_1\frac{\phi_o h}{2\pi}\left(\frac{gm_2 f_m}{Fa_1} + \frac{z}{\cos\alpha + (\sin\alpha)/f_u} + tg\phi^T \cdot ctg\psi\right) \qquad (18)$$

$$+3.82 \cdot S^\kappa \cdot z \cdot h_k .$$

To illustrate the example of using the dependence (18), the efficiency of the prototype mechanism [16] is calculated. The separated components (10), (15), (17) of the expression (18) are calculated. In accordance with the dependence (10), the value of E_{nM} is determined at the parameters of the mechanism: $\omega_\partial = 314$ rad/s, $d_1 = 0.09$m, $tg\psi = 0.01$, $m_{oc} = 3$kg, $m_{p\partial} = 0.4$ kg, $tg\alpha = 0.268$, $m_p = 2.4$kg, $d_2 = 0.14$m, $E_{nM} = \frac{1}{8}(314 \cdot 0.09 \cdot 0.01)^2 \times$

$\left(3 + 0.4 \cdot 0.268^2 + 0.5\frac{2.4(0.09^2 + 0.14^2)}{(0.09 \cdot 0.01)^2}\right) = 409.7$ Nm.

In accordance with the dependence (15), the value of A_{mp} is determined at the parameters of the mechanism: $Fa_1 = 1200$ N, $\phi_\partial = 44$ rad, $h = 2 \cdot 10^{-3}$m, $g = 9.81$ m/s², $m_2 = 3.2$ kg, $f_m = 0.3$, $z = 3$, $\alpha = 15°$, $tg\phi = 0.2$, $ctg\psi = 5$, $f_\mu = 0.1$, $A_{mp} = 1200\frac{44 \cdot 2 \cdot 10^{-3}}{2\pi} \times \left(\frac{9.81 \cdot 3.2 \cdot 0.3}{1200} + \frac{3}{0.966 + 0.259/0.1} + 0.2 \cdot 5\right) = 31.1$ Nm.

In accordance with the dependence (17), the value of $E_{e\Phi}$ is determined at the parameters: $S^k = 95 \cdot 10^{-5}$m², $z = 3$, $h_k = 100$ kN/m, $E_{e\Phi} = 3.82 \cdot 10^{-5} \cdot 3 \cdot 100$ kNm.

In accordance with the dependence (8), the efficiency coefficient η_{3M} of the CM [16] is:

$$\eta_{3M} = \frac{1080}{409.7 + 31.1 + 1080} = 0.71 .$$

The obtained dependences (18) allow to carry out the analysis of influence of the CM parameters on its work efficiency. It enhances the efficiency of design work allowing to find more appropriate parameters easier. Thus, (18) demonstrates that one of the promising ways to increase the efficiency of the CM is to reduce the value of moving masses m_{oc}, m_p, m_2 and other. Reduction of the mass m_p of the motor rotor can be achieved through the use of modern high-torque motors with a rotor that has a much smaller mass, outer diameter d_2 and moment of inertia. The weight of the elements of the CM can be reduced through the use of composite and polymeric materials which weigh three up to four times less. For example, the rod connecting the clamp drive and the clamping chuck which is made in the form of a pipe can be

made of carbon fibre. Using the formula (18) gives enhanced opportunities for the designer to make informed decisions about the feasibility of using new structural elements and materials, improving the quality of surfaces machining to reduce the coefficients of friction f_m, f_μ. That is, (18) makes it possible to evaluate the technical solutions in terms of comparing the costs of their implementation and the magnitude of productivity gains. Determining the appropriate periods between maintenance is dependent on the allowable amount of loss of the mechanism efficiency due to its normal wear. The level of degradation of the friction surfaces and the change of the friction coefficient allows to calculate the level of reduction of the mechanism productivity by means of (18) and to make more accurate decisions about the need for maintenance or overhaul.

The proposition would change the parameters m_{oc} and m_{oc} by making the parts of the mechanism from titanium with the purpose of improving. The magnitude of the change in the part's mass when changing its material while keeping its strength can be determined by comparing the coefficients of specific strength of each material. The specific strength is a material's strength divided by its density. In the manufacture of parts of titanium, their weight will be twice less than that made of alloy steel and two and a half times less compared to high-quality carbon steel with providing the same strength. The masses $m'_{oc} = 1.5$ kg and $m'_p = 1.2$ kg of the parts which are made of new material are reduced by 50% comparatively with the previous variant. The diameter of the rotor $d'_2 = 0.13$m can be reduced by 7% by using more expensive special manufacturing technologies. The parameters m_{oc}, m_p and d_2 have an influence on the value E_{nM}, and consequently on the efficiency η_{3M}. The value of E'_{nM} determined at the new parameters of the mechanism is:

$$E'_{nM} = \frac{1}{8}\left(314 \cdot 0.09 \cdot 0.01\right)^2 \times$$

$$\left(1,5 + 0.4 \cdot 0.268^2 + 0.5\frac{1.2\left(0.09^2 + 0.13^2\right)}{\left(0.09 \cdot 0.01\right)^2}\right) = 184.88\, Nm.$$

The value of the mechanism's efficiency with new parameters η'_{3M} can be calculated with (18): $\eta'_{3M} = \frac{1080}{184.88 + 31.1 + 1080} = 0.833$

The proposed changes of the masses and geometrical parameters resulted in the improvement of the efficiency coefficient $\eta'_{3M} = 0.833$ comparatively with the previous $\eta_{3M} = 0.71$ by 17%.

5. CONCLUSION

As a result of the evaluation of traditional CM design function, the main disadvantages, and factors limiting clamping process efficiency and in particular at high speeds of rotation, have been identified. Based on the obtained information, the characteristics of the elements and the structure of the CM with improved characteristics are revealed and described. Following the example of the patented mechanism design of this type, function is considered, and the calculation scheme and simplified dynamic models describing different stages of its work are made. It is revealed that for the mechanisms of this type, the first stage of operation which is related to elimination of the gaps is distinguished and displayed in the relatively enlarged rotation angle of the input link (rotor) in the free-running mode. Therefore, the special feature of the mechanism of the proposed structure is in the possibility of using the work of inertia forces of the input link, the magnitude of which increases over its acceleration period, can be adjusted and de-

pends on the magnitude of the rotation angle of the rotor during free-running. Also, by changing the geometrical-mass characteristics (moment of inertia) of the rotating input link, the force characteristics of the clamping process can be corrected.

The calculation scheme was formulated to determine mass-geometric parameters of mechanism elements should influence the main characteristics of the clamping mechanisms of this type. Based on the revealed information, dependencies for determining the specific kinematic and dynamic characteristics and the value of the work efficiency coefficient of the clamping mechanism of this type were found. The obtained dependencies are among the necessary prerequisites for creating an automatic system for calculating the work characteristics of CM of this type. Usage of obtained analytical dependence enhances the efficiency of design work and operation of the CM due to the possibility of selecting the mass and geometrical parameters of the elements.

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