MANUFACTURING ENGINEERING AND AUTOMATED PROCESSES

МАШИНОБУДУВАННЯ, АВТОМАТИЗАЦІЯ ВИРОБНИЦТВА ТА ПРОЦЕСИ МЕХАНІЧНОЇ ОБРОБКИ

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DEFINING PARAMETERS OF ELASTIC-SAFETY CLUTCHES FOR SCREW CONVEYERS

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Summary. *The designs of elastic-safety clutches of screw conveyors are offered allowing, except safety conditions, due to damping and elastic properties to ensure smooth start of conveyor screw while in drive run-up and reduce the sharpness of changes in load on it during operation and overloading. The analytical dependence of the determination of impact loads with axial and torque shocks in the investigated clutch at the boundary moment of its actuation is given. The graphical dependences of the change in the torque value depending on the movement of the movable half-clutch in the mode of protection device actuation at different angle of the hitch inclination are shown.*

Key words: *screw conveyers, drives, safety clutches.*

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Problem setting. Creation of new and improvement of the existing designs of transport-technological mechanisms works for to mechanization and automation of production processes and increase in labour productivity. During transport-technological processes by screw conveyers (SC) performance the overloadings caused by technological processes operation as well as by stochastic phenomena result in significant deformation and breakdown of the screw operating devices and other mechanism parts. It is possible to prevent overloadings of technological character providing efficient loading of SC but it is difficult to predict stochastic overloadings and they can be prevented using safety clutches (SC) with elastic and protecting properties in SC drives constructions.

Analysis of previous investigations. The problem of the improvement of SC is investigated by Zenkov R.L. [1], Hryhoriev A.M. [2], Hevko B.M. [3], Rohatynskyi R.M. [4, 5] and other scientists. The researches carried out by Holybentsev A.N. [6], Komarov M.S. [7], Leveikin V.S. [8] and other scientists deal with dynamic processes in machines. Machine drives are investigated by Poliakov V.S. [9], Tepynkycheiev V.K. [10], Malashchenko V.S. [11], Nahorniak S.H., Lutsiv I.V. [12] and others. At the same time the problems of dynamic behaviour of machine drives with safety mechanisms still remain unsearched hence scientific-practical problem is vital.

The aim of the investigation is to research and analyze impact loadings in screw conveyer drives with safety clutches and to decrease their size.

Realization of the investigation. Safety friction, cam, ball-type and combined elastic-

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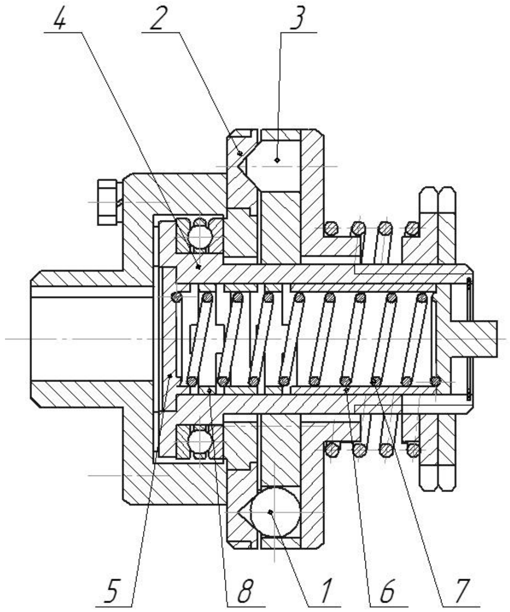
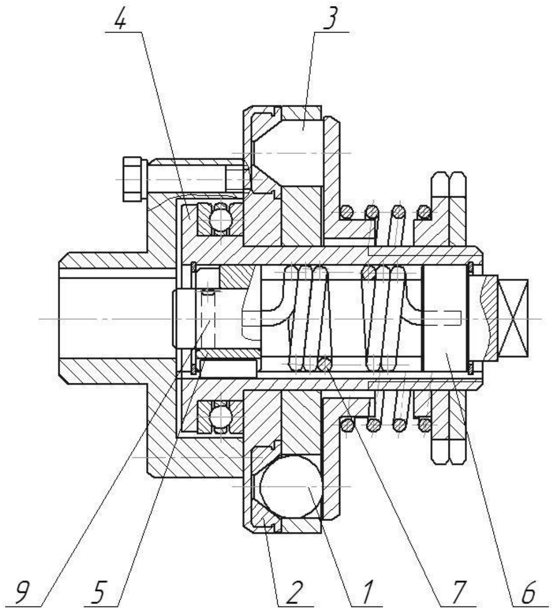
safety clutches (SC) [2, 8, 12] are widely used in SC structures. However not all SC structures completely meet the requirements concerning effective disconnection of kinematic chain in case when overloading occurs in SC. Thus for SC drives protection from overloading cam and ball-type safety clutches at low speeds, small rotational torques and rotating masses of connectors are used since at high speeds and large mass such clutches produce frequent overloadings at the moment of repeated connections. Above mentioned loadings cause quick wearing of cams and balls surfaces and instability of the rotational torque. Low frequency safety clutches are used but their disadvantage is the absence of damping properties while in operation and ability of the SC screw smooth start at drive run-up.

Therefore to avoid these disadvantages a number of elastic-safety clutches are offered which due to their safety as well as damping and elastic properties provide smooth start of conveyer screw at drive run-up and decrease abrupt changes of loads on it during operation and overloading (Fig. 1). Ball-hitch (Fig. 1: couples 1 – 2) and conical pin-hitch (Fig.1: couples 3 – 2) couples perform the functions of gearing elements between half-couplings in such clutches. When these gearing elements are properly made of materials with damping properties (balls or conical pins) they can function as dampers when half-couplings interact between themselves. To provide elastic properties in these clutch structures the use of torsional spring 7 inside journals 4 between fixed and movable in radial direction 6 flanges and cam gearing elements 8 (for clutch shown in Fig. 1a) and movable pin joint 9 (for clutch shown in Fig. 1b) is introduced. At drive start the torsional spring 7 provides screw smooth start due to torsional twisting and cam elements 8 phase-in gearing (Fig. 2a) or rotation of movable pin joint 9 (Fig. 2b) about defined angle. In case when overloading occurs on the screw, the release of half-couplings by means of gearing elements (balls 1 and conical pins 2) recess action from hitches 2 takes place. During the following couplings of the half-couplings in the skidding mode the gearing elements made of damping materials act as damper when half-coupling interact with each other.

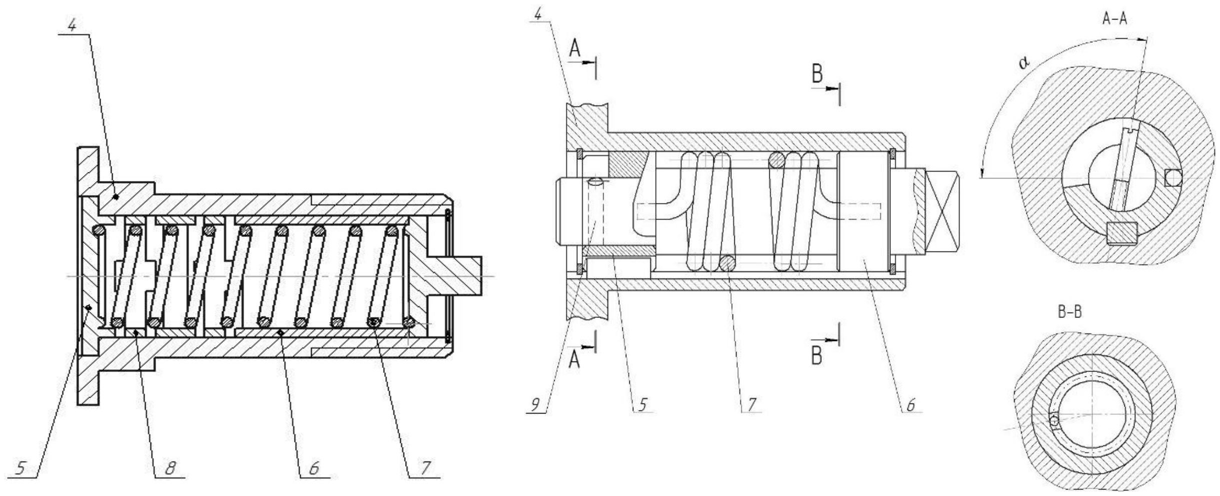


a)



b)

Figure 1. Elastic-conical safety clutches: a) Patent of Ukraine # 115032 b) self-designed project



a) b)

Figure 2. Elastic elements of the clutch a) cam elements of gearing b) pin movable joint

Let us consider the state of ball and damper rolling elements when half-couplings interact by gearing elements ball-damper-hitch with conical half-coupling surfaces (Fig. 3, Fig. 4). In rolling elements contact areas the response forces P_b, F_b, N_b with hitches surfaces and respectively frictional forces F_{fb} at half-coupling relative rotation angle occur.

Working conical surface of damper sector is effected by normal component N_d which depends on the force value of serpentine ring spring. Thus when half-couplings barring frictional force F_{fd} occurs on the damper surface.

Radial displacement of sectors (x) is described by the following equation:

$$x / \cos\alpha = h / \sin\alpha, \tag{1}$$

where a, h – is the ascent angle of sector conical surface and hitch depth.

Such sector displacement can be provided in case when overloading occurs. Here ball axial movement is within certain limits and depends on hitch depth h, that in its turn is connected with half-coupling angle displacement by the following dependence:

$$\varphi_2 - \varphi_3 = h \cdot ctg\beta / P_b,$$

2)

where β, R_b – is ascent angle of hitch profile and ball location radius, φ_2, φ_3 – is angle dimension relatively to half-coupling rotation.

The scheme of determination of rolling elements and damper operating conditions is shown in Figure 3.

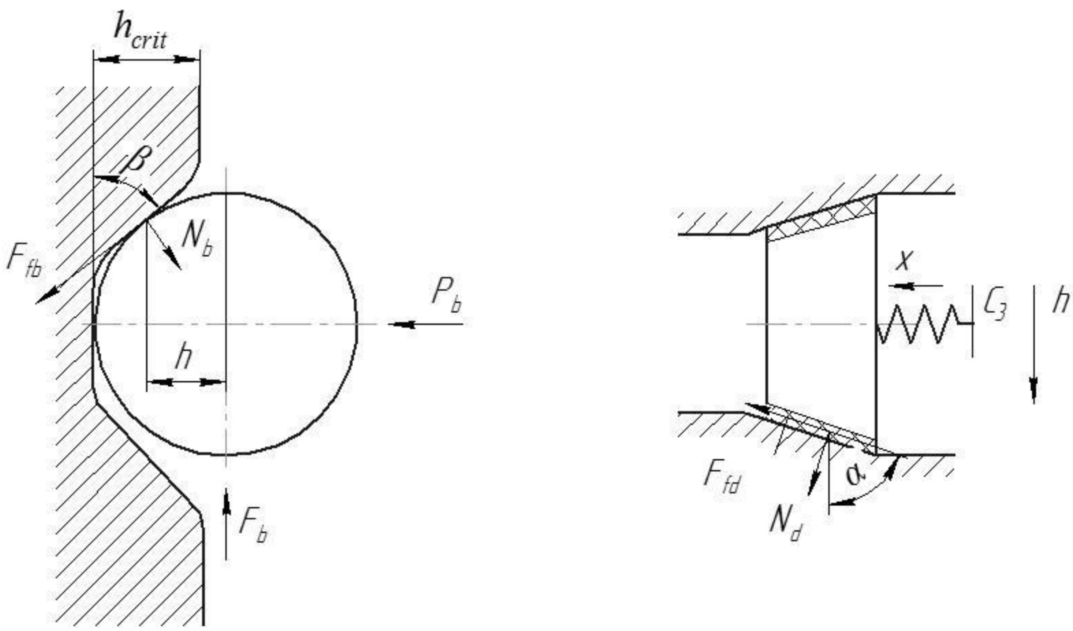


Figure 3. Calculating scheme of rolling element and damper operation

When the ball interacts with the hitch, equilibrium conditions can be represented by the following equations:

$$\begin{cases} P_b - N_b \cos\beta + F_{fb} \sin\beta = 0 \\ F_b - N_b \sin\beta - F_{fb} \cos\beta = 0 \\ F_{fb} = N_b \cdot f_{fb}. \end{cases} \qquad (3)$$

When half-couplings barring, friction forces occur in splines and damper working surface and are determined by the following dependence:

$$\begin{cases} F_s = (1/R_s)C_2(\varphi_2 - \varphi_4)/f_{fs} + \dot{x}\, tg\alpha \cdot \mu_s \\ F_{fd} = \dfrac{\dot{x}}{\cos\alpha}\mu_{fd} + N_d f_{fd}, \end{cases} \qquad (4)$$

where C_2, f_{fs} – is the stiffness of driven system and the friction ratio in splines resulted to axial force;

μ_s, μ_{fd}, R_s – respectively the coefficients of viscous friction in splines and damper on the wedge and the mean diameter of splines location;

f_{fd} – the coefficient of dry friction in damper;

F_s, φ_4 – respectively the friction force in splines and the torsion angle of the driven system.

Considering the freedom degree of such mechanism we can come to the conclusion that at relative half-coupling displacement the links have axial and radial movement. The coefficients of viscous friction effecting the joining elements of the half-couplings and their response time have significant influence on the friction process.

Determination of the rotational torque dependence on movable clutch axial displacement is the problem of the force calculation.

The calculation scheme of the safety clutch deformation mechanism is shown in

Fig. 4.

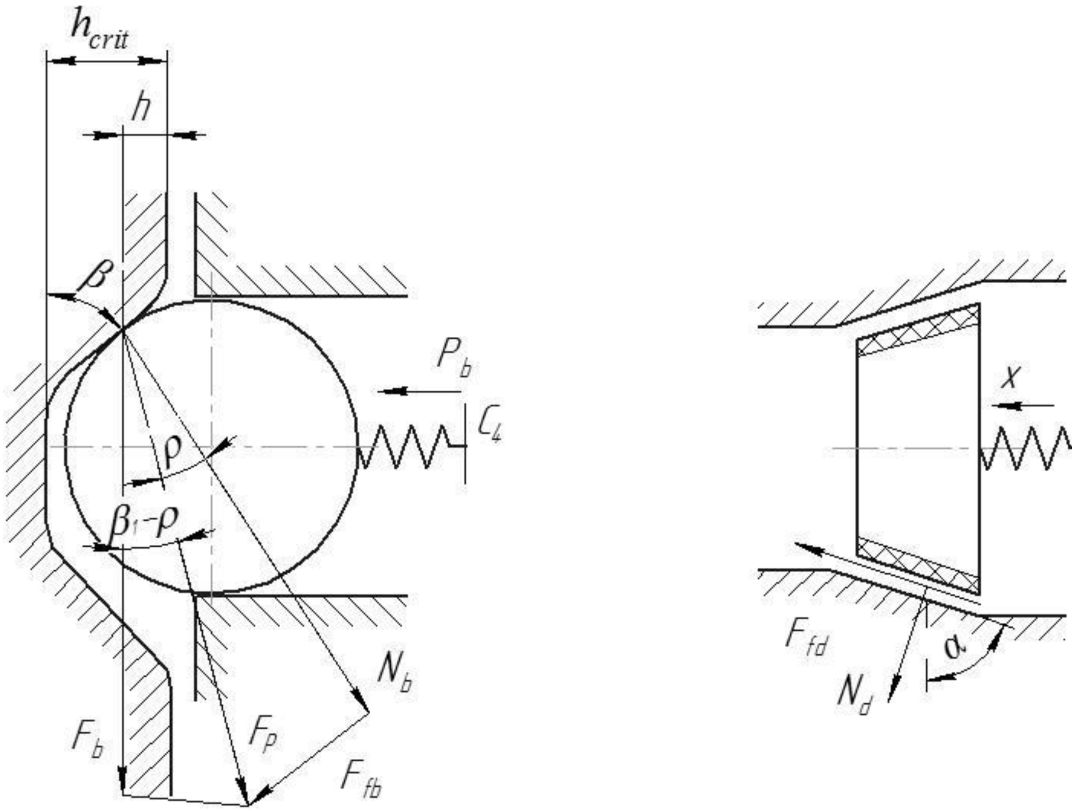


Figure 4. The calculation scheme of the safety clutch deformation mechanism

When half-couplings interact the normal response N_b and the friction force F_{fb} directed towards the side opposite to driven link movement occur in the contact area of the ball and the hitch. The resultant force F_ρ is displaced towards the ring force F_b on friction angle value ρ.

If overloaded the clutch complete release results from the movable half-coupling displacement in axial direction on value h_{crit} taking into account h and the half-coupling rounded radius value.

Taking into account the profile of the ball gearing we can consider the torque value changes on the following stages of the clutch actuation: on the first stage the displacement of the movable half-coupling passes on value h' from 0 to $(\text{h}'_{crit} - r(1-\cos\beta))$, on the second stage – up to the complete recess action of the balls.

Considering the forces and taking into account that $\beta_1 = (90^\circ - \beta)$ we determine dependence between ring force F_b and spring force P_b applying the known equation:

$$P_b = F_b(tg(\beta_1 - \rho) - \frac{R_b f_{fs} \mu_s}{R_s}, \tag{5}$$

where β_1 – hmitch inclination angle; R_s – mean diameter of splines location.

In the second case the spring force value P_b is expressed as $P_b = C_4\lambda_0 + C_4 h'$, taking into account the spring stiffness C_4 and its previous compression λ_0.

Substituting the values P_b in formula (5) we determine the ring force F_b depending on the location radius of the balls, splines and previous spring tension and stiffness.

$$F_b = (C_4\lambda_0 + C_4 h') / (tg(\beta_1 - \rho) - R_b f_{fs}\mu_{fs} / R_s). \tag{6}$$

Taking into account the value of ring force F_b in equation (6) and radius of the balls location on the orbit we determine the clutch rotational torque when the movable half-coupling is displaced in axial direction on value h'.

$$T = R_b(C_4\lambda_0 + C_4\,\mathrm{h}') / (tg(\beta_1 - \rho) - R_b f_{fs}\mu_s / R_s). \tag{7}$$

The displacement of the normal response N_b direction relatively to the working surface results from further ball movement along rounded area. Thus it is necessary to determine the dependence of direction changes on the movable half-coupling displacement.

The dependence of current inclination angle β_1' value of normal response on further displacement $(h'_{crit} - h')$ of the movable half-coupling is expressed as follows: $\sin\beta_1' = (r - (h'_{crit} - h'))/r,$ where $(h'_{crit} - h') = y'$ – is current value of gearing between the ball and the hitch.

The value y' is also expressed through the movable half-coupling displacement $y' = y_{\max} - y,$ where $y_{\max} = (h'_{\max\ \mathrm{crit}} - h'_{\max})$ is the maximum ball gearing value at which the angle β_1' of the normal response N_b equals the hitch inclination angle.

Making corresponding substitutions we define the dependence of the inclination angle β_1' on the half-coupling displacement *y*.

$$\sin\beta_1' = \frac{r - y_{\max} + y}{r}. \tag{8}$$

$$\beta_1' = \arcsin\frac{r - y_{\max} + y}{r}. \tag{9}$$

Let us symbolize the value $(r - y_{\max})$ through *z* and making corresponding substitutions we get $\beta_1' = (\arcsin z + y)/r.$ The value magnitude of this angle taking into account $tg\,\beta_1$, is expressed as follows:

$$tg\beta_1 = \frac{\sin(\arcsin\frac{z+y}{r})}{\left(1 - \sin^2(\arcsin\frac{z+y}{r})\right)^{1/2}} = \frac{z+y}{\left(r^2 - (z+y)^2\right)^{1/2}}. \tag{10}$$

Relying on equation (10) we solve the tangent equation in the following way:

$$tg(\beta_1 - \rho) = \frac{\dfrac{(z+y)}{(r^2-(z+y)^2)^{1/2}} - tg\rho(r^2-(z+y)^2)^{1/2}}{1 + \dfrac{tg\rho(z+y)}{(r^2-(z+y)^2)^{1/2}}},$$

$$tg(\beta_1 - \rho) = \frac{(z+y) - tg\rho(r^2-(z+y)^2)^{1/2}}{(r^2-(z+y)^2)^{1/2} + tg\rho(z+y)}. \tag{11}$$

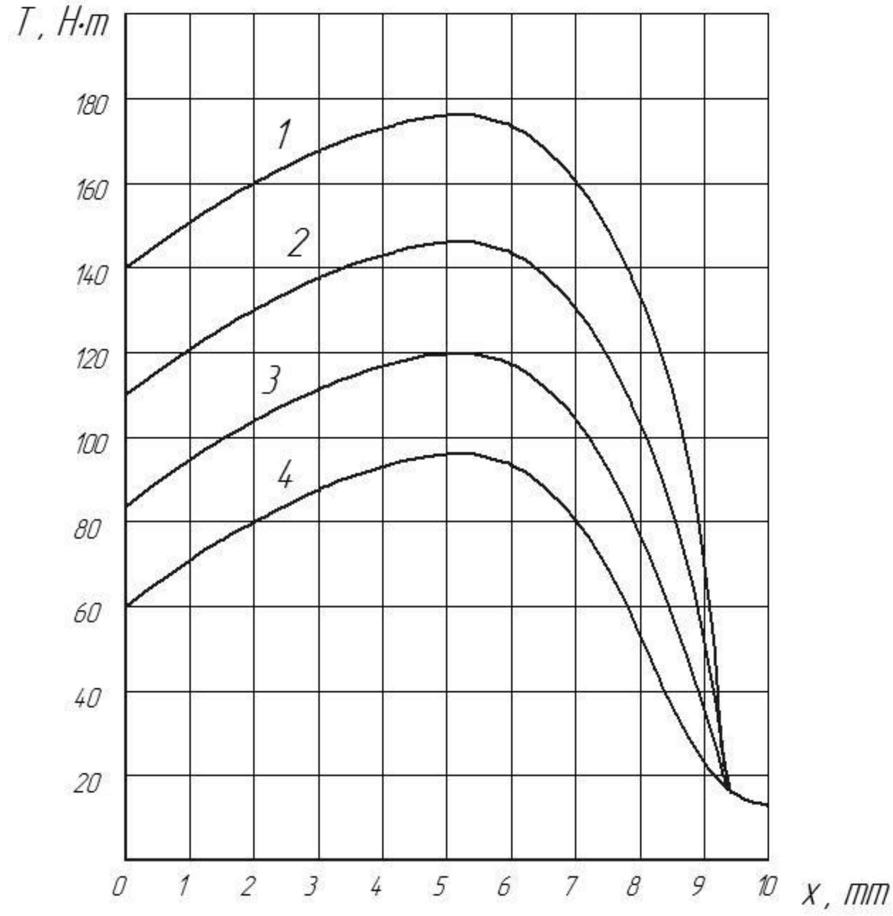
Plugging this equation in dependence (5) we determine the value of the ring force depending on the spring compression force

$$P_b = F_b \frac{(z+y) - tg\rho(r^2(z+y)^2)^{1/2}}{(r^2-(z+y)^2)^{1/2} + tg\rho(z+y)} - \frac{R_b f_{fs} \mu_s}{R_s}. \tag{12}$$

The transfer moment value on the second stage of the clutch actuation is:

$$T = \frac{F_b[C_4\lambda_0 + C_4(h'_{crit} - y')]}{\dfrac{(r-y) - tg(2r\cdot y' - (y')^2)^{1/2}}{(2r\cdot y' - (y')^2)^{1/2} + tg\rho(r-y')} - \dfrac{R_b f_{fs}\mu_s}{R_s}}. \tag{13}$$

The dependence graph of rotational torque value T on the movable half-coupling axial displacement at various inclination angles β_1 is shown in Fig. 5.



1 – β_1= 30°, 2 – β_1= 45°, 3 – β_1= 55°, 4 – β_1= 60°
Figure 5. Dependence Graph of torque T on different values of inclination angel β_1

On the basis of the previous calculations we can come to the conclusion that at torque transmission in the range of 0 to $h_{crit} - r(1-\sin\beta_1)$ its value is determined from the formula (7), and when the displacement value changes within the limits from $h'_{crit} - r(1-\sin\beta_1)$ to h_{crit} the equation (13) is used.

Conclusion. Having calculated by numerical method the moment equation using formulas (7) and (13) when R_b= 80 mm; C_4 = 80 N/mm; f_{fs}= 0.15; λ_0= 15 mm, R_s= 16 mm, h = (4–10) mm, the graphs of the moment value changes depending on the movable half-coupling displacement in the mode of safety device actuation at various hitch inclination angles are shown in Fig. 5.

While investigating these dependences we came to the conclusion that the boundary moment of the safety clutch operation is determined from formula (7) at maximum movable half-coupling displacement on value h_{crit}. The hitch inclination angle changes within the limits 60° – 30° result in the actuation torque increase by 1.8 times.

As the result of carried out researches the suggested safety clutch increases reliability and durability of the screw conveyer drives.

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ВИЗНАЧЕННЯ ПАРАМЕТРІВ ПРУЖНО-ЗАПОБІЖНИХ МУФТ ГВИНТОВИХ КОНВЕЄРІВ

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Резюме. *Запропоновано конструкції пружно-запобіжних муфт гвинтових конвеєрів, які, окрім запобіжних, дозволяють за рахунок демпфувальних і пружних властивостей забезпечувати плавний запуск шнека гвинтового конвеєра під час пуску привода й зниження різкості зміни навантажень на нього під час експлуатації та перевантажень. Наведено аналітичну залежність визначення ударних навантажень при осьовому та крутильних ударах у досліджуваній муфті в граничний момент її спрацьовування. Наведено графічні залежності зміни величини моменту залежно від переміщення рухомої півмуфти в режимі спрацьовування захисного пристрою за різних кутів нахилу лунок.*

Ключові слова: *гвинтові конвеєри, приводи, запобіжні муфти.*

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