# INVESTIGATION OF DYNAMICAL IMPACT LOADS IN SCREW CONVEYER DRIVES WITH SAFETY CLUTCHES
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# *ДОСЛІДЖЕННЯ ДИНАМІЧНИХ НАВАНТАЖЕНЬ В ПРИВОДАХ ГВИНТОВИХ КОНВЕЄРІВ З ЗАПОБІЖНИМИ МУФТАМИ*

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## ABSTRACT

*Elastic-conical SCl design for decreasing of impact loadings in half-couplings of screw conveyer safety clutches at the moment of their wearing is offered. Analytical dependence of impact loadings determination at direct and torque shocks in the investigated clutch with gap S of spring installation and certain safety clutch without gap is given. Graphic dependences of these relations on gap S and spring tension C and graphs of axial and rotational velocities dependences on impacts from inclination angle of nonworking limit β are shown.*

## РЕЗЮМЕ

*Запропонована конструкція пружно-конусної ЗМ для зменшення ударних навантажень в півмуфтах ЗМ ГК в момент їх спрацьовування. Приведені аналітична залежність визначення ударних навантажень при осьовому та крутильних ударах в досліджуваній муфті з зазорами S установки пружини і певною запобіжною муфтою без зазору. Наведені графічні залежності цих відносин від зазору S і жорсткості пружини C і графічні залежності осьової і окружної швидкості при ударі від кута нахилу неробочої межі β.*

## INTRODUCTION

The creation of new and improvement of existing designs of transport and technological mechanisms are favourable for further production development and increase in labour productivity. Screw mechanisms (SM) are integrated parts of production complex mechanization and automation. During transport and technological processes operation with screw conveyers (SC) overloadings caused by technological processes of operations performance as well as stochastic phenomena often occur in their drives. These overloadings result in significant deformations and breakages of screw working parts. It is possible to prevent overloadings of technological character providing efficient filling of SC screw interturn space by transporting material or improving chargers (bins, heads). It is difficult to predict stochastic overloadings but they can be prevented using specially designed safety clutches (SCl) in SC drives constructions.

The problem of the improvement of SM and their safety devices design is investigated by (*Zenkov, 1987; Ivashkov, 1987; Kolobov, 1987; Hryhoryev, 1972; Hevko B.M. et al., 1993*; *Rohatynskiy et al., 2014; Nahornyak et al., 1992*).

The researches of some authors *(Holybentsev, 1959; Komarov, 1979; Loveykin*, *2002; Nesterov, 2002 and other scientists)*, deal with dynamic processes in machines.

Machine drives are investigated by *Malashchenko V.S. (2006)*; *Polyakov V.S. and others (1974); Tepynkycheyev V.K*. *(1964); Hevko R.B. (2014)*. At the same time the problem of dynamic behaviour of machine drives with safety mechanisms still remains unsearched hence scientific-practical problem is vital.

## MATERIAL AND METHODS

Safety friction, jaw, ball-type clutches, combined elastic-safety clutches (SCl) (*Hryhoryev, 1972; Nahornyak, 1992; Malashchenko, 2006; Polyakov et al., 1974),* are widely used in SC structures. However not all SCl structures meet the requirements concerning effective kinematic chain interruption in a case when overloading occur in SC. Thus for SC drives protection from overloading the spring-jaw and spring-ball type safety clutches are used at low speeds, small torque moment values and rotating masses of fittings. In a case of high speeds and large masses such clutches form frequent overloadings at the moment of chain reclosing. Mentioned above loadings cause quick wearing of cams and balls surfaces and unstability of rotational torque.

To decrease dynamical impact loadings in half-couplings of screw conveyer safety clutches at the moment of their actuation and increase of their operating life, elastic-conical SCl design represented in Fig. 1 is proposed.

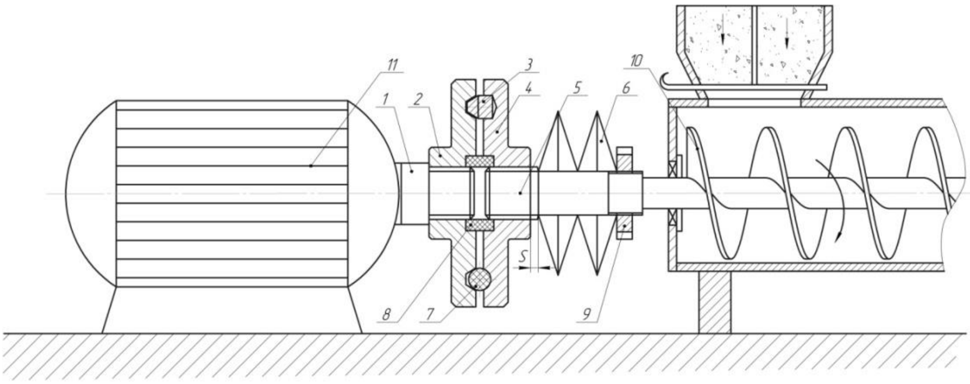


**Fig. 1 - Screw conveyer with elastic-conical safety clutch**

*1 – drive shaft; 2, 4 – driving and driven half couplings; 3 – coupling engagement components (conical pins); 5 – driven shaft; 6 – spring; 7 – holes with balls made of elastic material; 8 – elastic ring; 9 – nut; 10 – screw; 11 – electric motor*

Torque moment from electric motor *11* through the driving half coupling *2* and coupling engagement components *3* (pins) of conical shape is transferred to the driven half coupling *4* moving in axial direction and from it to the screw *10* of the screw conveyer. Half couplings are joined by compression spring 6 one end of which rests upon nut end *9* and another upon driven shaft shoulder *5*. Thus if no external force is applied to the clutch, the S-sized gap is formed between the driven clutch face and the spring. In such a case at the starting impact moment during actuation time the spring elastic force *6* does not act on the driven half coupling. Besides balls *7* made of elastic materials are located in the grooves on half coupling faces *2* and *4* and elastic ring *8* is mounted in the circular grooves. They cushion axial and radial impacts after actuation.

In a case of the screw overloading and its emergency shutdown the driven shaft *5* stopping takes place. This provides conical pins *3* and balls *7* recession with the holes of driving half coupling *2* and axial movement of driven half coupling *4* with spring *6* compression. SCl restore its engagement when the loading is to decrease to the specified. The computation model of the screw conveyer with elastic-conical SCl is given in Fig.2. Since the equation of connection between driving and driven half couplings in elastic-conical SCl can be considered like in jaw one with identical jaw profiles then the computation model of elastic-conical SCl is reduced to the jaw SCl model.

The clutch dynamic model is the mechanical system consisting of two torque masses *1* and *2* with half couplings at the ends. The given inertia mass moments are equal to $I_1$ and $I_2$ correspondingly. Half coupling is freely installed on the shaft of the driven torque mass and is spring-loaded in axial direction by spring of $C$ rigidity with previous compression. Under unloaded conditions acting upon the given mechanical system the gap $S$ is formed between the driven half coupling and thee spring.

The moment produced by the motor $T_{dr}$ and the moment transmitted by the clutch $T_{act}$ upon the driving clutch rigidly mounted with torque mass 1. The motion resistant moment produced by the screw $T_{sh}$ and the moment transmitted by the clutch $T$ affect the driven mass *2*. The axial force $F_{ax}$ from the driven half coupling side and the spring elastic force $F_{el}$ act upon the moving half coupling.

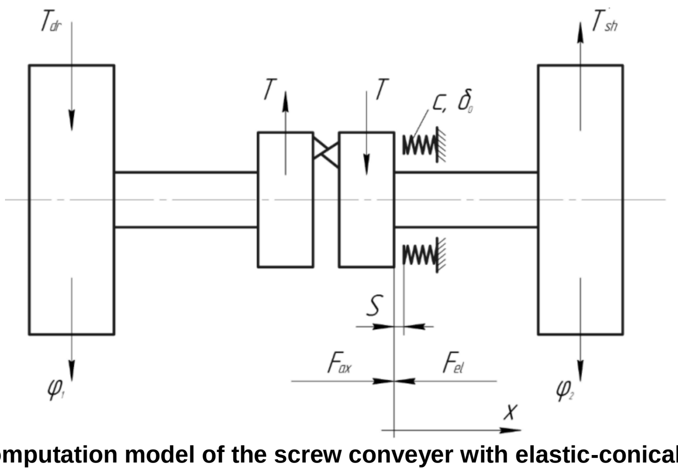


**Fig. 2 - The computation model of the screw conveyer with elastic-conical safety clutch**

In common case the torque moment produced by the driving mechanism $T_{dr}$ is the function of the rotary speed $\omega_1$ of the driving mass $T_{dr} = f(\omega_1)$. In the given problem definition it is considered to be constant in time: $T_{dr} = const$.

The torque $T_{sh}^{'}$ can be represented as the one consisting of two parts: constant and variable.

$$T_{sh}^{'} = T_{sh} + T_T \tag{1}$$

where: $T_T = f(t - t_1)$ − is its variable component as time function;

$t$ – time;

$t_1$ − certain time value, starting from action upon the system which changes in time of resistant moment component produced by the driven mechanism.

Longitudinal movement of movable half coupling up to half coupling release equals:

$$x = \frac{D}{2 \cdot tg\alpha}\psi, \tag{2}$$

where:

$D$ − is the mean diameter of conical pins;

$\alpha$ − is the inclination of holes active faces;

$\psi$ − is the turning angle of rotating mass *1* relatively to rotating mass *2*.

The axial force acting from the driving half coupling side on the driven one under the constant contact of conical pins surfaces with holes in half couplings *(Polyakov V. S., and others, 1974)* is:

$$F_\alpha = \frac{2T}{D}\left[tg(\alpha - \rho_1) - \frac{D}{d}f_1\right] \tag{3}$$

where:

$\rho_1$ − is friction angle between the conical pins and the holes;

$d$ − is shaft diameter with mounted movable half coupling;

$f_1$ − is coefficient of friction in spline joint.

Spring elastic force affecting the driven movable half coupling equals:

$$F_{el} = c[\delta_0 + x - S] \tag{4}$$

Let us assume that the clutch transmits definite moment. Hence the driven half coupling contacts with the spring one. Thus the movement of the clutch mechanical system is performed providing that $S \le x \le h_K$ and is described by the following equation system. (Presented equation system does not consider the vibration damping and rotating mass a *1* and *2* torsional rigidity).

$$\begin{gathered} I_1\ddot{\varphi}_1 = T_{dr} - T \\ I_2\ddot{\varphi}_2 = T - T_{sh} - T_T \\ m\ddot{x} = -c(\delta_0 + x - S) + \frac{2T}{D}\left[tg(\alpha - \rho_1) - \frac{D}{d}f_1\right] \\ x = \frac{D}{2 \cdot tg\alpha}(\varphi_1 - \varphi_2);\ \psi = \varphi_1 - \varphi_2 \end{gathered} \tag{5}$$

where:

$h_K$ − is the maximum driven clutch move (conical pins height);

$\varphi_1$ − is the rotation angle of torsion mass *1*;

$\varphi_2$ − is the rotation angle of torsion mass *2*.

Let us assume that at the time starting moment only the constant resistant moment of motion acts upon the driving rotating mass, i.e. motion takes place at the time moment *t* and corresponds to inequation $0 \le t \le t_1$ thus $T_T = 0$. Reducing the given above equation system we get the following equation:

$$\ddot{\psi} + \frac{chh_1}{I_{el}}\psi = \frac{T_{dr}}{I_{el}}i_2 + \frac{T_{sh}}{I_{el}}i_1 - \frac{ch}{I_{el}}(\delta_0 - S) \tag{6}$$

Here

$$I_{el}=\frac{I_1\cdot I_2}{I_1+I_2}+mhh_1;\quad i_1=\frac{I_1}{I_1+I_2};\quad i_2=\frac{I_2}{I_1+I_2};$$

$$h_1=\frac{D}{2\cdot tg\alpha};\quad h=\frac{D}{2\left[tg(\alpha-\rho)-\frac{D}{d}f_1\right]}$$

Sumbolizing:

$$\frac{chh_1}{I_{el}}=\omega^2;\ \frac{T_{dr}}{I_{el}}i_2+\frac{T_{sh}}{I_{el}}i_1=\frac{ch(\delta_0-S)}{I_{el}}=a_1$$

and solving differential equation under initial conditions:

$$\psi\big|_{t=0}=\frac{a_1}{\omega^2},\ \dot{\psi}\big|_{t=0}=0$$

we obtain:

$$\psi=\frac{a_1}{\omega^2}$$

or:

$$\psi=\frac{1}{chh_1}\left[T_{dr}i_2+T_{sh}i_1-ch(\delta_0-S)\right]$$

Correspondingly:

$$x=\frac{1}{ch}\left[T_{dr}i_2+T_{sh}i_1-ch(\delta_o-s)\right] \tag{7}$$

At a certain time moment $t_1$ the motion resistant moment $T_T$ changing in time is applied to the driven part. Let us assume that this moment depends on time linearly, i.e. $T_T=h(t-t_1)$.

Then at $t\geq t_1$ and $x\leq h_K$ the system movement is described by equation

$$\ddot{\psi}+\frac{chh_1}{I_{el}}\psi=\frac{ki_1}{I_{el}}t+\frac{T_{dr}i_2+T_{sh}i_1-ch(\delta_0-S)-hi_1t_1}{I_{el}}. \tag{8}$$

Accepting previous symbols and assuming that $\frac{ki_1}{I_{el}}=a_2$ under initial conditions $\psi\big|_{t=t_1}=\frac{a_1}{\omega^2}$, $\dot{\psi}\big|_{t=t_1}=0$, we solve the equation in the form of:

$$\psi=-\frac{a_2}{\omega^3}sin\left[\omega(t-t_1)\right]+\frac{a_2}{\omega^2}(t-t_1)+\frac{a_1}{\omega^2} \tag{9}$$

The movement according to the defined regulation is tking place until conical pins recession. Dynamic system behaviour after conical pins recession is fescribed by the following equation system:

$$\begin{cases} I_1\ddot{\varphi}_1=T_{dr}; \\ I_2\ddot{\varphi}_2=-T_{sh}-k(t-t_1); \\ m\ddot{x}=-c(x+\delta_0-S). \end{cases} \tag{10}$$

The solution of the given (10) equation system is:

$$\begin{cases} \psi=\left(\frac{T_{dr}}{I_1}+\frac{T_{sh}}{I_2}-\frac{kt_1}{I_3}\right)\frac{t^2}{2}+\frac{k}{I_2}\cdot\frac{t^3}{6}+C_{31}t+C_{32} \\ x=C_3\cdot sin\left(\sqrt{\frac{C}{m}}t+\gamma_3\right)-(\delta_0-S) \end{cases} \tag{11}$$

Constants $C_{31}$, $C_{32}$, $C_3$, $\gamma_3$ are derived from the initial conditions determined by equation (10) at $x=h_K$.

After conical pins recession they contact again (now by their nonworking surfaces) and their mutual sliding occurs up to the moment of their working surfaces impact.

Conical pins contact following the recession takes place when:

$$x = \frac{2 \cdot h_1 h_K \left(tg\alpha + tg\beta\right)}{2 \cdot h_1 tg\beta + D} \tag{12}$$

where: $\beta$ – inclination angle towards the vertical of conical pins nonworking surfaces.

The value of the axial force acting from the driving half coupling side upon the driven one after contact recovery under conical pins mutual sliding along the holes by nonworking surfaces is equal:

$$F_{ax} = \frac{2T}{D}\left[tg\left(\beta + \rho_1\right) - \frac{D}{d} f_1\right] \tag{13}$$

and the axial displacement of the driven half coupling is:

$$x = h_K\left(1 + \frac{h_2}{h_1}\right) - h_2\psi$$

where: $h_2 = \dfrac{D}{2 \cdot tg\beta}$

The equation system describing the process of conical pins sliding by their nonworking surfaces along the holes takes the form:

$$\begin{cases} I_1\ddot{\varphi}_1 = T_{dr} + T; \\ I_2\ddot{\varphi}_2 = -T - T_{sh} - T_T; \\ m\ddot{x} = F_{ax} - c\left(x + \delta_0 - S\right). \end{cases} \tag{14}$$

The initial conditions are determined at the moment of contact. Mathematical description of this process is approximate as it is unknown how speeds are distributed after the contact and if the mutual sliding of conical pins along the holes without failure actually occurs.

Within limits of the given problem set we consider that the contact of conical pins nonworking surfaces along the holes goes on “smoothly” without rebound and the impact is nonelastic i.e. relative speed of conical pins at the moment after the impact is equal to the projection of this speed before the impact on the axes coinciding with the direction of the conical pins nonworking surface (fig.3), that is:

$$\nu_{ai} = -\dot{x}_i \cos\beta + R\dot{\psi}_i \cdot \sin\beta \tag{15}$$

Consequently (15)

$$\dot{x}_{ai} = \dot{x}_i \cos^2\beta - R\dot{\psi}_i \cdot \sin\beta \cdot \cos\beta; \qquad \dot{\psi}_{ai} = -\frac{\dot{x}_i}{R}\sin\beta \cdot \cos\beta + \dot{\psi}_i \sin^2\beta \tag{16}$$

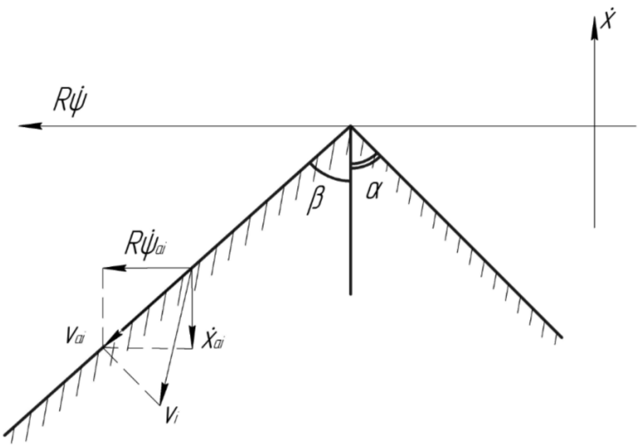


**Fig. 3 – The calculation scheme of the conical pins nonworking surfaces contact along the holes**

Here indexes *i* impact and *ai* after impact define the speeds at the moment of time that directly precedes the holes contact and moment of time that occurs directly after the moment of contact between nonworking surfaces of half coupling holes.

Equation system (14) can be reduced to the equation:

$$\ddot{\psi} + \frac{ch_2h_3}{I_{el}^*}\psi = T_{dr}\frac{i_2}{I_{el}^*} + \frac{T_{sh}}{I_{el}^*}i_1 + \frac{ki_1}{I_{el}^*}(t-t_1) + \frac{ch_3}{I_{el}^*}\left[\delta_0 - S + h_K\left(1-\frac{h_2}{h_1}\right)\right] \tag{17}$$

where: $I_{el}^* = \dfrac{I_1 \cdot I_2}{I_1 + I_2} + mh_2h_3;\;\; h_3 = \dfrac{D}{2\left[tg(\beta+\rho_1) - \dfrac{D}{d}f_1\right]}$

Introducing symbols $\omega_2^2 = \dfrac{ch_2h_3}{I_{el}^*};\;\; b_3 = \dfrac{k}{I_{el}^*}\cdot i_1;\; a_3 = \dfrac{1}{I_{el}^*}\left\{T_{dr}i_2 + T_{sh}i_1 + ch_3\left[\delta_0 - S + h_K\left(1+\dfrac{h_2}{h_1}\right)\right]\right\}$,

and solving differential equation (17) we obtain:

$$\psi = C_4\cdot\sin(\omega_2 t + \gamma_4) + \frac{b_3}{\omega_3^2}(t-t_1) + \frac{a_3}{\omega_2^2} \tag{18}$$

Constants $C_4$ and $\gamma_4$ are determined from the initial conditions.

The equation (18) describes the process until the moment when spring elastic force $F_{el}$ affects the driven half coupling. The spring interrupts its action at $x < S$.

Equation describing the system motion after interruption of the spring action is:

$$\ddot{\psi} = \frac{1}{I_{el}^*}\left[T_{dr}i_2 + T_{sh}i_1 + k(t-t_1)i_1\right] \tag{19}$$

Taking into account the introduced symbol $a_4 = \dfrac{1}{I_{el}^*}(T_{dr}i_2 + T_{sh}i_1)$, the solution of this differential equation (19) takes the following form:

$$\psi = a_4\frac{t^2}{2} + b_3\frac{(t-t_1)^3}{6} + C_{41}t + C_{42};\;\; x = h_K\left(1+\frac{h_2}{h_1}\right) - h_2\psi \tag{20}$$

Constants $C_{41}$ and $C_{42}$ are determined from the initial conditions at the moment of time when the spring interrupts its action that is $x = S$.

The impact of conical pins by their working surfaces occurs when $x = 0$. Dynamic loadings originated from conical pins of half couples impact by working surfaces result in increased wear of the impact surfaces and short clutch life (service time decrease). Dynamic loadings $F_d$ on impact are determined from static ones $F_{st}$ multiplying them by dynamic coefficient $K_d$:

$$F_d = F_{st}\cdot K_d \tag{21}$$

In the given case both axial impact of half couplings and torque impact by driving and driven rotating masses occur. That is why we should distinguish the axial impact loading $F_d$ of axial impact and dynamic impact moment $T_i$ of torque impact.

Thus dynamic coefficients $K_{Fd}$ and $K_{Td}$ are different. Let us consider the case of impact loadings effect on axial and torque impacts in the investigated clutch with spring installation gaps S and certain safety clutch without gap. All values related to the investigated clutch are indicated by index 1 and those related to the given clutch by index 2.

At the impact moment the static torque moments in both clutches are equal to:

$$T_{st1} = T_{dr}i_2 + T_{sh}i_1 + ki_1(t-t_1);$$

$$T_{st2} = T_{dr}i_2 + T_{sh}i_2 + ki_1(t-t_1) + ch_3\delta, \tag{22}$$

where: $\delta$ - preload adjustment of the spring regulating the clutch transfer moment in the known clutch.

Static axial loadings are equal to:

$$F_{st1} = \frac{T_{st1}}{h_3}; \; F_{st2} = \frac{T_{st2}}{h_3}. \tag{23}$$

To compare dynamic impact loadings in the investigated and given clutches we should find the ratio of these loadings in the above mentioned clutches. As it is difficult to determine the values of dynamic coefficients because of complexity of occurred processes let us assume dynamic coefficients to be approximately proportional to the speed modules at the impact moment:

$$K_{Fd} = |\dot{x}_i| \cdot K_{F_0} \; K_{Td} = |\dot{x}_i| \cdot K_{T_0} \tag{24}$$

where: $K_{F_0}$, $K_{T_0}$ - are proportionality coefficients correspondently.

Then in a case of torque impact we have

$$\frac{T_{d1}}{T_{d2}} = \frac{\dot{\psi}_{i1}}{\dot{\psi}_{i2}} \cdot \frac{T_{st1}}{T_{st2}}, \tag{25}$$

or $$\frac{T_{d1}}{T_{d2}} = \frac{\dot{\psi}_{i1}}{\dot{\psi}_{i2}} \cdot \frac{T_{st1}}{T_{st1} + ch_3\delta}$$

On axial impact (since $\dot{x}_1 = -h_2\dot{\psi}_i$ and $F_{st} = \frac{T_{st}}{h_3}$): $$\frac{F_{d1}}{F_{d2}} = \frac{T_{d1}}{T_{d2}} = \frac{\dot{\psi}_{i1}}{\dot{\psi}_{i2}} \cdot \frac{T_{st1}}{T_{st} + ch_3\delta}.$$

## RESULTS

Computer investigation of dynamics of proposed and given safety clutches enables to determine axial and angular mass velocities at the moment of conical pins impact by their working surfaces and relative dynamic impact loadings.

The graph dependences on gap *S* and spring rigidity *C* and graph dependences of axial and rotational velocities on impacts from inclination angle of nonworking limit *β* are shown in Fig. 5 and Fig. 4.

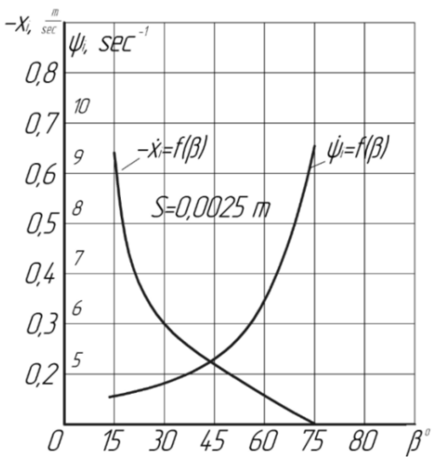


**Fig. 4 – Dependences of axial and rotational velocities of half couplings on inclination angle of conical pins active faces (engagement elements)**

It is evident from the graphs that the proposed safety clutch with the spring installed with the gap makes it possible to reduce impact loadings by 10−15%. in comparison with the given one. It is advantageous to use conical pins with active faces 45° inclination angle as in this case the axial and rotational velocities of half couplings on impacts are minimal (Fig. 4).

When the gap *S* increases, the dynamical impact loadings slightly decrease but the clutch loses its sensitivity and is unable to activating after the overload removal. That is why it is reasonable to make effective use of clutch designs with gaps $S = 1 \div 2$ mm.

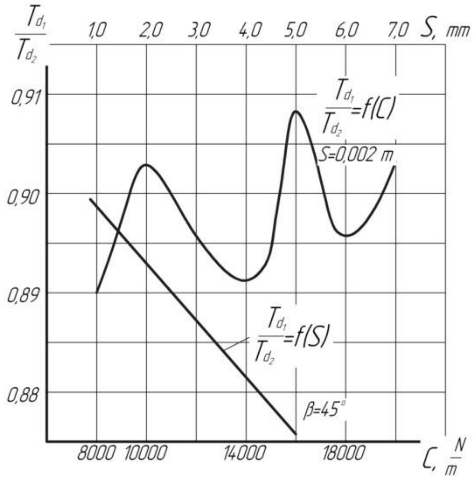


**Fig. 5 – Dependences of axial and rotational velocities of half couplings on inclination angle of conical pins nonworking faces (engagement elements)**

With the spring rigidity increase the dynamic impact loadings occurrence in compared clutches have zigzag-shaped changes but within small limits (Fig. 5). That is why it is advisable to use springs with 14000 *N/m* rigidity.

## CONCLUSIONS

Carried out experiments proved that the offered elastic-conical safety clutch of screw conveyer provides significant reduction of dynamic impact loadings under screw overloadings, increases accuracy of automatic half couplings reconnection after loading removal and at the same time considerably increases reliability and service life of screw conveyer driving mechanisms.

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