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Increasing stable operation of the working body in crank presses [*]

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Повышение стабильности функционирования рабочего органа в кривошипных прессах[***]

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Introduction. Static and dynamic loading systems of the safety friction clutch (SFC) are investigated. A schematic diagram of the frictional contact of solids in the forging and stamping machines is synthesized. From the point of view of increasing the operational stability of the working body in crank presses, the following factors are considered: response time, current friction factor and a change in torque under static and dynamic loading of the safety clutch.
Materials and Methods. The response time of the SFC with differentiated friction pairs is determined. The sought indicator corresponds to the period of the uptime in which the load is amplified (between values of the rated torque and the response time). The parameters of a dual-mass system correspond to the parameters of an equivalent system that includes a clutch and key parts of the drive. The system elements include mass of inertia; mass of inertia including the engine rotor and the main (driving) part of the SFC; elastic connection with the specified value of the reduced angular stiffness.
Research Results. Values of the load arising in elastic bonds not conditioned by the working body operation are specified. Formulas that should be used to determine the values of the driving moment and generalized coordinates are presented. Start conditions with an increase in the load value from the initial indicators are described.
Discussion and Conclusions. The dependence is found for calculating the minimum number of friction pairs of the basic friction group. It is shown that at this minimum, the gain used to implement an “ideal” SFC load characteristic, does not exceed the maximum permissible value, even if the value of the friction coefficient is maximum. A fundamental SFC model is presented, in which, with a minimum value of the friction coefficient, negative feedback does not work. In the functional

Введение. Исследованы статическая и динамическая системы нагружения предохранительной фрикционной муфты (ПФМ). Синтезирована принципиальная схема фрикционного контакта твердых тел в кузнечно-штамповочных машинах. С точки зрения исследования процесса повышения стабильного функционирования рабочего органа в кривошипных прессах рассмотрены следующие факторы: время срабатывания, текущий коэффициент трения и изменение вращающего момента при статическом и динамическом нагружении предохранительной муфты.
Материалы и методы. Определено время срабатывания ПФМ, имеющей дифференцированные пары трения. Искомый показатель соответствует участку рабочего времени, на котором усиливается нагружение (между значениями номинального вращающего момента и момента срабатывания). Параметры системы, состоящей из двух масс, соответствуют параметрам эквивалентной системы, включающей муфту и ключевые части привода. Элементы системы: масса инерции; масса инерции, включающая ротор двигателя и основную (ведущую) часть ПФМ; упругая связь с указанным значением приведенной угловой жесткости.
Результаты исследования. Определены значения нагрузки, возникающей в упругих связях, не обусловленных функционированием рабочих органов. Представлены формулы, которые следует использовать для определения значений движущего момента и обобщенных координат. Описаны условия старта при увеличении значения нагрузки от начальных показателей.
Обсуждение и заключения. Найдена зависимость для вычисления минимального числа пар трения основной фрикционной группы. Показано, что при этом минимуме коэффициент усиления, используемый для реализации «идеальной» нагрузочной характеристики ПФМ, не превышает предельно допустимое значение, даже если величина коэффициента трения максимальна. Представлена принципиальная модель ПФМ, в которой при минимальном значении коэффициента трения отрицательная обратная связь не действует. В принципиальной схеме модернизации

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diagram of the basic SFC version modernization, there is no negative feedback with a minimum value of the friction coefficient in order to increase the accuracy of operation and the rated load capacity.

базового варианта ПФМ для повышения точности срабатывания и номинальной нагрузочной способности отсутствует отрицательная обратная связь при минимальном значении коэффициента трения.

Keywords: crank press, friction coefficient, working mechanism, gain factor, overload, accuracy

Ключевые слова: кривошипный пресс, коэффициент трения, рабочий механизм, коэффициент усиление, перегрузка, точность.

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Introduction. At the present stage of production development, to increase reliability and to extend the lifetime of manufacturing facilities are critical tasks. In particular, it is of interest to study the static and dynamic loading systems of a safety friction clutch (SFC). In the framework of this research, the static and dynamic loading systems of the SFC are considered. A schematic diagram of the frictional contact of solids in forging and stamping machines is synthesized. From the viewpoint of investigating the process of increasing the stable operation of the working body in crank presses, the following factors are considered: response time, current friction coefficient and change in torque under static and dynamic loading of the safety clutch.

Materials and Methods. Determine the response time of the SFC with differentiated friction pairs. The required indicator corresponds to the part of the working time on which the load is intensified - between the values $T_н$ (rated torque) and T_i (response time). The parameters of a dual-mass system correspond to the parameters of an equivalent system including a clutch and key parts of the drive (Fig. 1).

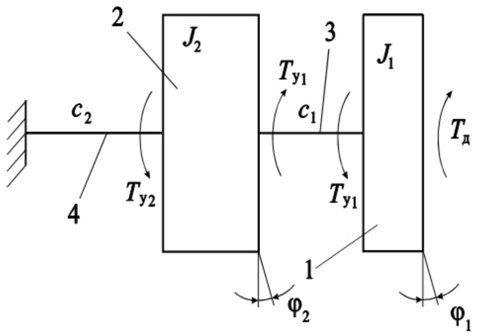

Fig. 1. Dynamic design diagram 1

The system components are as follows:

- mass of inertia (the equivalent system is linked up with the main (drive) shaft of the SFC (position 1 in Fig. 1);
- mass of inertia including the engine rotor and the main (drive) part of the SFC (position 2 in Fig. 1);
- elastic connection with the specified value of the reduced angular stiffness c_1 and c_2, respectively (positions 3 and 4 in Fig. 1

Establish that the damping value in this system and in the SFC is low and should not be used. The value of the reduced moment of resistance forces is equal $T_н$. Given these factors, we obtain the equations of motion [1]:

$$J_1\ddot{\phi}_1 + c_1(\phi_1 - \phi_2) = T_д\,, \qquad (1)$$

$$J_2\ddot{\phi}_2 + c_2\phi_2 = c_1(\phi_1 - \phi_2)\,, \qquad (2)$$

where J_1, J_2 are the values of the given moments of inertia of the SFC and the non-essential (driven) part of the drive, respectively; $T_д$ is the value of the driving torque; ϕ_1, ϕ_2 are the values of the generalized coordinates of the system motion (the values of the rotation angles of the mass of inertia 1 and 2, respectively).

The equations (1) and (2) show that the engine has the required resource of power level and $\phi_1 = \omega t$ (ω is the value of the angular velocity of inertia masses 1, $\omega = \text{const}$, t is the time value) [2–5].

Let us introduce these conditions into the equation (2):

$$\ddot{\phi}_2 + \frac{c_1 + c_2}{J_2}\phi_2 = \frac{c_1}{J_2}\omega t .$$

As a result, we write:

$$\phi_2 = A\sin\sqrt{\frac{c_1 + c_2}{J_2}}t + B\cos\sqrt{\frac{c_1 + c_2}{J_2}}t + \frac{c_1}{c_1 + c_2}\omega t.$$

Values of the integration constants A and B should be calculated when specifying the initial values: at t =0 $\phi_2 = T_{\text{н}} / c_2$, $\dot{\phi}_2 = \omega$. Then

$$B = \frac{T_{\text{н}}}{c_2};\ A = \frac{c_2\omega}{c_1 + c_2}\sqrt{\frac{J_2}{c_1 + c_2}} .$$

Hence

$$\phi_2 = \frac{c_2\omega}{c_1 + c_2}\sqrt{\frac{J_2}{c_1 + c_2}}\sin\sqrt{\frac{c_1 + c_2}{J_2}}t + \frac{T_{\text{н}}}{c_2}\cos\sqrt{\frac{c_1 + c_2}{J_2}}t + \frac{c_1}{c_1 + c_2}\omega t .$$

Values of the loads, that the elastic bonds 3 and 4 receive, are equal:

$$T_1 = c_1(\phi_1 - \phi_2) = \frac{c_1c_2\omega t}{c_1 + c_2} - \frac{c_1c_2\omega}{c_1 + c_2}\sqrt{\frac{J_2}{c_1 + c_2}}\sin\sqrt{\frac{c_1 + c_2}{J_2}}t - \frac{c_1}{c_2}T_{\text{н}}\cos\sqrt{\frac{c_1 + c_2}{J_2}}t , \tag{3}$$

$$T_2 = c_2\phi_2 = \frac{c_1c_2\omega t}{c_1 + c_2} + \frac{c^2{}_2\omega}{c_1 + c_2}\sqrt{\frac{J_2}{c_1 + c_2}}\sin\sqrt{\frac{c_1 + c_2}{J_2}}t + T_{\text{н}}\cos\sqrt{\frac{c_1 + c_2}{J_2}}t , \tag{4}$$

$$\sin\sqrt{\frac{c_1 + c_2}{J_2}}t = 0,\ \cos\sqrt{\frac{c_1 + c_2}{J_2}}t = 1 \text{ or } \sin\sqrt{\frac{c_1 + c_2}{J_2}}t = 1,\ \cos\sqrt{\frac{c_1 + c_2}{J_2}}t = 0 . \tag{5}$$

Consider the equation:

$$\frac{c_1c_2\omega}{c_1 + c_2}\sqrt{\frac{J_2}{c_1 + c_2}} = \frac{c_1}{c_2}T_{\text{н}} . \tag{6}$$

The only real value of the solution to the equation (6):

$$c_1 = c_2\left(\sqrt[3]{\frac{c_2J_2\omega^2}{T^2{}_{\text{н}}}} - 1\right). \tag{7}$$

The cubic equation analysis

$$c^3{}_1 + 3c^2{}_1c_2 + 3c_1c^2{}_2 + c^3{}_2 - \frac{c^4{}_2J_2\omega^2}{T^2{}_{\text{н}}} = 0$$

allows us to make the following statement: for values of the variables c_1 smaller than the value of the variable calculated according to (6), the value of the left-hand side of (7) is much larger than of the right [6–8]. At the trigonometric functions specified in (3), with a possible decrease in c_1, the value of the sinusoidal vibrational amplitude increases $\left(c_1c_2\omega\sqrt{J_2 / (c_1 + c_2)} / (c_1 + c_2)\right)$ and the value of the cosinusoidal vibrational amplitude decreases $\left(c_1T_{\text{н}} / c_2\right)$.

$$c_1 < c_2\left(\sqrt[3]{\frac{c_2J_2\omega^2}{T^2{}_{\text{н}}}} - 1\right). \tag{8}$$

The time interval when the value of the elastic coupling moment 3 is identical $T_{\text{п}}$:

$$t_{\text{c}} = \frac{c_1 + c_2}{c_1c_2\omega}\left(T_{\text{п}} + \frac{c_1}{c_2}T_{\text{н}}\right). \tag{9}$$

Then $\sin\sqrt{(c_1 + c_2)t_{\text{c}} / J_2} = 0$; $\cos\sqrt{(c_1 + c_2)t_{\text{c}} / J_2} = 1$, resulting in:

$$\sqrt{(c_1 + c_2)t_{\text{c}} / J_2} = 2\pi n ,$$

where n = 0, 1, 2,…., n.

Considering (9), we write:

$$(c_1+c_2)\sqrt{c_1+c_2}\left(T_{п}+\frac{c_1}{c_2}T_{н}\right)=2\pi n J_2 c_1 c_2 \omega\,. \tag{10}$$

For rigid SFC $\left(c_1>c_2\left(\sqrt[3]{c_2J_2\omega^2/T^2_{н}}-1\right)\right)$, we use the expression $\sin\sqrt{(c_1+c_2)t_{c}/J_2}=1$, $\cos\sqrt{(c_1+c_2)t_{c}/J_2}=0$. At that,

$$t_{c}=\frac{c_1+c_2}{c_1c_2\omega}\left(T_{п}+\frac{c_1c_2\omega}{c_1+c_2}\sqrt{\frac{J_2}{c_1+c_2}}\right). \tag{11}$$

In this equation, angular stiffness of the SFC:

$$\sqrt{\frac{c_1+c_2}{J_2}}t_{c}=\frac{\pi}{2}+2\pi n\,.$$

The value can be obtained in the form of an analytical result:

$$c_1`=\sqrt[3]{c^2_2(3c_2-G)-\left(c_2-\frac{G}{3}\right)^3+\left\{\left[c^2_2-\left(\frac{c_2}{3}-\frac{G}{9}\right)\right]^3+\left[\left(c_2-\frac{G}{3}\right)^3-c^2_2\left(3c_2-G\right)\right]^2\right\}^{\frac{1}{2}}}+$$

$$+\sqrt[3]{c^2_2(3c_2-G)-\left(c_2-\frac{G}{3}\right)^3-\left\{\left[c^2_2-\left(\frac{c_2}{3}-\frac{G}{9}\right)\right]^3+\left[\left(c_2-\frac{G}{3}\right)^3-c^2_2\left(3c_2-G\right)\right]^2\right\}^{\frac{1}{2}}}-c_2+G.$$

As a result, $G=J_2c^2_2\omega^2\left(\pi/2+2\pi n-1\right)^2/T^2_{п1}$. From knowing c_1 and t_{c}, we find the values of the torques acting on the elastic coupling 4. For this, we use the dependences presented below [6–9].

— Safety friction clutch (elastic):

$$T'_2=T_{п1}+\frac{c_1+c_2}{c_2}T_{н}\,. \tag{12}$$

— Safety friction clutch (rigid):

$$T''_2=T_{п1}+c_2\omega\sqrt{\frac{J_2}{c_1+c_2}}\,. \tag{13}$$

Based on the results of the calculations, it is required to make a number of comments. With an increase in c_2, the value of the moment arising on the elastic coupling 4 (elastic clutch) decreases. If c_2 grows, so does the torque (rigid clutch).

Consider the types of drive loads. It is important to note that prior to the SFC operation, overloading is not the reason to stop the machine working bodies (Fig. 2).

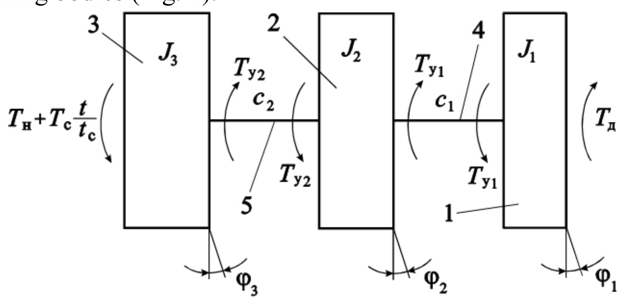

Fig. 2. Dynamic design diagram 2

The equations for this system are:

$$J_1\ddot{\phi}_1+c_1(\phi_1-\phi_2)=T_{д}\,, \tag{14}$$

$$J_2\ddot{\phi}_2-c_1(\phi_1-\phi_2)+c_2(\phi_2-\phi_3)=0\,, \tag{15}$$

$$J_3\ddot{\phi}_3 - c_2(\phi_2 - \phi_3) = -T_{\text{н}} - T_{\text{c}}\frac{t}{t_{\text{c}}}. \tag{16}$$

Here, ϕ_1, ϕ_2, ϕ_3 are generalized angular coordinates of inertia masses 1, 2, and 3; T_{c} is the value of the possible increase in torque in the working body device; t_{c} is time (associated with the value of the overload growth rate and T_{c}). The torque value T_{c} (depends on the type of production machine and its applicability) is written in the form: $(1,2...4)T_{\text{н}}$ [10–13]. The working body has a power reserve, and the value of the angular velocity of inertia masses 1 is assumed unchanged, i.e. $\phi_1 = \omega t$. In this case, considering (16), we obtain [13]:

$$c_1(\phi_1 - \phi_2) = T_{\text{д}}.$$

Let us do the summation:

$$J_2\ddot{\phi}_2 + (c_1 + c_2)\phi_2 - c_1\omega t - c_2\phi_3 = 0. \tag{17}$$

We sum (15) and (16):

$$J_2\ddot{\phi}_2 - c_1\omega t + c_1\phi_2 + J_3\ddot{\phi}_3 = -T_{\text{н}} - T_{\text{c}}\frac{t}{t_{\text{c}}}. \tag{18}$$

We differentiate twice (16):

$$J_2\frac{d^4\phi_2}{dt^4} + (c_1 + c_2)\frac{d^2\phi_2}{dt^2} - c_2\frac{d^2\phi_3}{dt^2} = 0. \tag{19}$$

Using (18), we obtain:

$$\frac{d^2\phi_3}{dt^2} = \frac{1}{J_3}\left(c_1\omega t - c_1\phi_2 - T_{\text{н}} - T_{\text{c}}\frac{t}{t_{\text{c}}}\right) - \frac{J_2}{J_3}\frac{d^2\phi_2}{dt^2}.$$

Substitute the last expression in (16):

$$\frac{d^4\phi_2}{dt^4} + \frac{c_2J_2 + (c_1 + c_2)J_3}{J_2J_3}\frac{d^2\phi_2}{dt^2} + \frac{c_1c_2}{J_2J_3}\phi_2 = \frac{c_2}{J_2J_3}\left(c_1\omega t - T_{\text{н}} - T_{\text{c}}\frac{t}{t_{\text{c}}}\right). \tag{20}$$

From (20), we obtain:

$$\phi_2 = \frac{1}{c_2}\left(J_3\frac{d^2\phi_3}{dt^2} + c_2\phi_3 + T_{\text{н}} + T_{\text{c}}\frac{t}{t_{\text{c}}}\right). \tag{21}$$

We differentiate twice (21) and get:

$$\frac{d^2\phi_2}{dt^2} = \frac{1}{c_2}\left(J_3\frac{d^4\phi_3}{dt^4} + c_2\frac{d^2\phi_3}{dt^2}\right). \tag{22}$$

Substitute (22) and (21) in (18):

$$\frac{d^4\phi_3}{dt^4} + \frac{c_2J_2 + (c_1 + c_2)J_3}{J_2J_3}\frac{d^2\phi_3}{dt^2} + \frac{c_1c_2}{J_2J_3}\phi_3 = \frac{c_2}{J_2J_3}\left[c_1\omega t - \frac{c_1 + c_2}{c_2}\left(T_{\text{н}} + T_{\text{c}}\frac{t}{t_{\text{c}}}\right)\right]. \tag{23}$$

We write the general solutions of the equations (22) and (23):

$$\phi_2 = A_1\sin k_1t + B_1\cos k_1t + C_1\sin k_2t + D_1\cos k_2t + \frac{1}{c_1}\left(c_1\omega t - T_{\text{н}} - T_{\text{c}}\frac{t}{t_{\text{c}}}\right), \tag{24}$$

$$\varphi_3 = A_2\sin k_1t + B_2\cos k_1t + C_2\sin k_2t + D_2\cos k_2t + \frac{1}{c_1 + c_2}\left[c_1\omega t - \frac{c_1 + c_2}{c_2}\left(T_{\text{н}} + T_{\text{c}}\frac{t}{t_{\text{c}}}\right)\right]. \tag{25}$$

Here,

$$k_{1,2} = \sqrt{\frac{c_2J_2 + (c_1 + c_2)J_3}{2J_2J_3} \pm \sqrt{\left[\frac{c_2J_2 + (c_1 + c_2)J_3}{2J_2J_3}\right]^2 - \frac{c_1c_2}{J_2J_3}}}.$$

Research Results. Using the expressions (22) and (23), we find the values of the load arising in elastic couplings not represented in (18). In this case, we use the values of the torque $T_{\text{д}}$ equal to $(\phi_1 - \phi_2)(t)$. The presence of the moment $T_{\text{н}}$ provides recording the generalized coordinates: $\phi_1 - \phi_2 = T_{\text{н}} / c_1$, $\phi_2 - \phi_3 = T_{\text{н}} / c_2$, $\phi_1 = \omega t$. We describe the conditions for starting at the increasing load (from the initial indicators): at $t = 0$ $\phi_2 = -T_{\text{н}} / c_1$,

$d\phi_2 / dt = \omega$; $\phi_3 = -(c_1 + c_2)T_{\text{н}} / c_1c_2$, $d\phi_3 / dt = \omega$. Given (17) and (18) and the starting conditions, we obtain (at $t = 0$):

$$\frac{d^2\phi_2}{dt^2} = -\frac{c_2}{J_2c_1}T_{\text{н}};\ \frac{d^3\phi_2}{dt^3} = 0;\ \frac{d^2\phi_3}{dt^2} = \frac{c_2 - 2c_1}{J_3c_1}T_{\text{н}};\ \frac{d^3\phi_3}{dt^3} = 0.$$

We use the starting conditions, the obtained indicators and the main expressions (22) and (23). We obtain constant values of the integration process:

$$A_1 = -\frac{k^2{}_2T_{\text{c}}}{k_1(k^2{}_1 - k^2{}_2)c_1t_{\text{c}}};\ B_1 = \frac{T_{\text{н}}c_2}{c_1J_2(k^2{}_1 - k^2{}_2)};$$

$$C_1 = \frac{k^2{}_1T_{\text{c}}}{k_2(k^2{}_1 - k^2{}_2)c_1t_{\text{c}}};\ D_1 = -\frac{T_{\text{н}}c_2}{c_1J_2(k^2{}_1 - k^2{}_2)};$$

$$A_2 = -\frac{k^2{}_2}{k_1(k^2{}_1 - k^2{}_2)} \times \frac{(c_1 + c_2)T_{\text{c}}}{c_1c_2t_{\text{c}}};\ B_2 = -\frac{T_{\text{н}}(c_2 - 2c_1)}{c_1J_3(k^2{}_1 - k^2{}_2)};$$

$$C_2 = \frac{k^2{}_1}{k_2(k^2{}_1 - k^2{}_2)} \times \frac{(c_1 + c_2)T_{\text{c}}}{c_1c_2t_{\text{c}}};\ D_2 = \frac{T_{\text{н}}(c_2 - 2c_1)}{c_1J_3(k^2{}_1 - k^2{}_2)}.$$

We substitute the values obtained after integration into (22) and (23):

$$\phi_2 = \frac{1}{(k^2{}_1 - k^2{}_2)c_1}\left[\frac{T_{\text{c}}}{t_{\text{c}}}\left(\frac{k^2{}_1}{k_2}\sin k_2t - \frac{k^2{}_2}{k_1}\sin k_1t\right) + \right.$$
$$\left. + \frac{T_{\text{н}}c_2}{J_2}(\cos k_1t - \cos k_2t)\right] + \frac{1}{c_1}\left(c_1\omega t - T_{\text{н}} - T_{\text{c}}\frac{t}{t_{\text{c}}}\right);$$

$$\varphi_3 = \frac{1}{(k^2{}_1 - k^2{}_2)c_1}\left[\frac{(c_1 + c_2)T_{\text{c}}}{c_2t_{\text{c}}}\left(\frac{k^2{}_1}{k_2}\sin k_2t - \frac{k^2{}_2}{k_1}\sin k_1t\right) + \right.$$
$$\left. + \frac{(c_2 - 2c_1)T_{\text{н}}}{J_3}(\cos k_2t - \cos k_1t)\right] + \frac{1}{c_1 + c_2}\left[c_1\omega t - \frac{c_1 + c_2}{c_2}\left(T_{\text{н}} + T_{\text{c}}\frac{t}{t_{\text{c}}}\right)\right].$$

Torque values perceived by elastic couplings 4 and 5:

$$T_1 = (\phi_1 - \phi_2)c_1 = \frac{1}{k^2{}_1 - k^2{}_2}\left[\frac{T_{\text{c}}}{t_{\text{c}}}\left(\frac{k^2{}_2}{k_1}\sin k_1t - \frac{k^2{}_1}{k_2}\sin k_2t\right) - \right.$$
$$\left. - \frac{T_{\text{н}}c_2}{J_2}(\cos k_1t - \cos k_2t)\right] + T_{\text{н}} + T_{\text{c}}\frac{t}{t_{\text{c}}}; \tag{26}$$

$$T_2 = (\phi_2 - \phi_3)c_2 = \frac{c_2}{(k^2{}_1 - k^2{}_2)c_1}\left[T_{\text{н}}\left(\frac{c_2}{J_2} + \frac{c_2 - 2c_1}{J_3}\right)(\cos k_1t - \cos k_2t) - \right.$$
$$\left. - \frac{T_{\text{c}}c_1}{c_2t_{\text{c}}}\left(\frac{k^2{}_1}{k_2}\sin k_2t - \frac{k^2{}_2}{k_1}\sin k_1t\right)\right] + \frac{c^2{}_2}{c_1 + c_2}\omega t + \frac{c_1 - c_2}{c_1}\left(T_{\text{н}} + T_{\text{c}}\frac{t}{t_{\text{c}}}\right). \tag{27}$$

The values k_1t and k_2t are not related to each other. The values $\sin k_1t$, $\sin k_2t$, $\cos k_1t$, $\cos k_2t$ can be positive or negative: $\sin k_1t = 1$ and $\sin k_2t = -1$ or $\cos k_1t = -1$ and $\cos k_2t = 1$ [10–12]. The values of these time intervals are from (27):

$$t_1 = (T_{\text{п}} - T_{\text{н}})\frac{t_{\text{c}}}{T_{\text{c}}} - \frac{k^3{}_1 + k^3{}_2}{k_1k_2(k^2{}_1 - k^2{}_2)}, \tag{28}$$

$$t_2 = \frac{t_{\text{c}}}{T_{\text{c}}}\left\{T_{\text{п}} - \left[\frac{2c_2}{(k^2{}_1 - k^2{}_2)J_2} + 1\right]T_{\text{н}}\right\} \tag{29}$$

Substituting (27) and (28) into (29), we obtain the values of interconnected torques in the elastic coupling 5 [13]:

$$T'_2 = \frac{(k^3{}_1 + k^3{}_2)c_2}{k_1k_2(k^2{}_1 - k^2{}_2)}\left(\frac{T_{\text{c}}}{c_1t_{\text{c}}} - \frac{c_2\omega}{c_1 + c_2}\right) + (T_{\text{п}} - T_{\text{н}})\frac{t_{\text{c}}}{T_{\text{c}}}\left[\frac{c^2{}_2\omega}{c_1 + c_2} + \frac{(c_1 - c_2)T_{\text{c}}}{c_1t_{\text{c}}}\right] + \frac{c_1 - c_2}{c_1}T_{\text{н}}, \tag{30}$$

$$T''_2=\frac{t_\text{c}}{T_\text{c}}\left\{T_\text{п}-\left[1+\frac{2c_2}{(k^2{}_1-k^2{}_2)J_2}\right]T_\text{н}\right\}\left[\frac{c^2{}_2\omega}{c_1+c_2}+\frac{(c_1-c_2)T_\text{c}}{c_1t_\text{c}}\right]+$$
$$+T_\text{н}\left[\frac{c_1-c_2}{c_1}-\frac{2c_2}{(k^2{}_1-k^2{}_2)c_1}\left(\frac{c_2}{J_2}+\frac{c_2-2c_1}{J_3}\right)\right]. \quad (31)$$

We use the expression for calculating the expansion force [13] to obtain the peak torque transmitted by the friction clutch when the external moment increases to T_i :

$$F_\text{p}=\frac{T_i-T'_2}{r}\text{tg}\alpha .$$

We use this expression to find the torque T'_2 :

$$T'_2=R_\text{cp}f_i\left(F_\text{п}-\frac{T_i-T'_2}{r}\text{tg}\alpha\right).$$

We consider the value for the full torque $T_\text{п}$ of the SFC and obtain $T_\text{п}=T_i$. Thus, by the end of the 2nd time interval $t_1\dots\ t_2$, when the load distribution in the safety friction clutch is completed, the values of the friction moment and the external torque are the same.

Discussion and Conclusions. The dependence is found for calculating the minimum number of friction pairs of the main friction group. It is shown that with this minimum, the gain factor used to implement the “ideal” load characteristic of the SFC does not exceed the maximum permissible value, even if the value of the friction coefficient is maximum.

A fundamental model of SFC is presented, in which negative feedback does not function at the minimum value of the friction factor. In the basic scheme of modernization of the basic version of SFC, at a minimum value of the friction factor, the negative feedback is absent to increase the operation accuracy and the rated load capacity.

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