The process of increasing the stable operation of the working body in crank presses

Kirill Kobzev^{1*}, *Yuliya Goncharova*¹, and *Nataliya Kobzeva*²

¹ Don State Technical University, pl. Gagarina, 1, Rostov-on-Don, 344002, Russia

² Rostov State Medical University, 29 Nakhichevan lane, Rostov-on-Don, 344022, Russia

Abstract. The article considers the study of the synthesized concept of frictional contact of solids in crank presses. As a result of the analysis the possibility of obtaining the maximum load characteristic of the friction contact within the interval of variation of the friction coefficient has been established. The possibility of equality of the values of the friction force of the friction contact at the boundaries of the indicated interval in the presence of a maximum and the achievement of their greatest stability under these conditions is also revealed. As result the upgraded concept of frictional contact allows theoretically to obtain a very high stability of the friction force, however, due to the relatively large value of the angle and force parameter, it is used inefficiently. A necessary condition for the absence of zero reversal of the output parameter of the main friction group of the friction contact in the interval of variation of the friction coefficient and the presence of the maximum function of the load capacity of the friction contact is the transfer of its full load by the sensitive elements of the additional friction group. An additional condition for the existence of a maximum is the need for the sensitive elements of the main friction group to transfer part of its full load at an equal number of friction pairs of both friction groups, and for the sensitive elements to transfer the full load of the main friction group at a lower number of friction pairs than in the additional friction group.

1 Introduction

This article investigated the static and dynamic loading system of the safety friction clutch and synthesized the schematic diagram of the friction contact of solids in forging and stamping machines. As a result of the analysis the possibility of obtaining a maximum load characteristic within the interval of change in the coefficient of friction is established. The possibility of equality of the friction force values at the boundaries of the specified interval in the presence of a maximum and achieving their greatest stability under these conditions is also revealed. The results show a high stability of the friction force during slippage of bodies, although at large values of the pressure angle of the sensitive elements of the sensor-transducer, the maximum friction force can be briefly proportional to the current value of the friction coefficient [1-15].

*Corresponding author: 5976765@mail.ru

2 Materials and methods

There is a need to conduct research to find time of operation of a safety clutch (PFM) that has a differential pair of friction which are corresponding to the area of working time, which in the loading process increases from the value of the nominal torque T_n to the value of the actuation torque T_i . The system that are consisting of two masses corresponds to the equivalent system in its parameters, which has a clutch and key parts of the drive indicated in Fig.1.

Fig.1. Design dynamic scheme 1

Components of the system are the mass of inertia (equivalent system is brought to the main (driving) shaft PFM- (Position 1, Fig.1.); the mass of inertia (including the rotor of the engine and the main (leading) part of the PFM – (Position 2, in Fig.1.); elastic bond (with the specified value of the reduced angular stiffness and respectively - (Positions 3 and 4, in Fig.1). Similarly, the elastic PFM system is shown by positions 3 and 4 in Fig.1.

We establish that the damping value in the current system and in the PFM is low and it should not be used, and the value of the reduced moment of the resistance forces is equal T_H , we obtain the equations of motion [1]:

$$J_1 \ddot{\varphi}_1 + c_1(\varphi_1 - \varphi_2) = T_n, \quad (1)$$

$$J_2 \ddot{\varphi}_2 + c_2\varphi_2 = c_1(\varphi_1 - \varphi_2), \quad (2)$$

where J_1, J_2 – is the values of the given moments of inertia, respectively PFM and non-core (slave) part actuator; T_n – is the value of the driving torque; φ_1, φ_2 – is the values of generalized coordinates of motion of the system (the values of the rotation angles, respectively, the masses of inertia 1 and 2).

As a result of the study of the above equations (1) and (2), we establish that this engine has the necessary resource of the power value and $\varphi_1 = \omega t$ (ω – is the value of the angular velocity of the masses of inertia $\omega = \text{const}$, t – is the value of time) [2-5].

We introduced conditions data into equation (2):

$$\ddot{\varphi}_2 + \frac{c_1 + c_2}{J_2} \varphi_2 = \frac{c_1}{J_2} \omega t$$

There is final result:

$$\varphi_2 = A \sin \sqrt{\frac{c_1 + c_2}{J_2}} t + B \cos \sqrt{\frac{c_1 + c_2}{J_2}} t + \frac{c_1}{c_1 + c_2} \omega t.$$

The values of the integration constants A and B necessary are calculated when specifying initial values: when $t=0$ $\varphi_2 = T_H / c_2$, $\dot{\varphi}_2 = \omega$. Then

$$B = \frac{T_H}{c_2}; \quad A = \frac{c_2 \omega}{c_1 + c_2} \sqrt{\frac{J_2}{c_1 + c_2}}.$$

As a result of the study of this level we got:

$$\varphi_2 = \frac{c_2 \omega}{c_1 + c_2} \sqrt{\frac{J_2}{c_1 + c_2}} \sin \sqrt{\frac{c_1 + c_2}{J_2}} t + \frac{T_H}{c_2} \cos \sqrt{\frac{c_1 + c_2}{J_2}} t + \frac{c_1}{c_1 + c_2} \omega t.$$

The values of the loads that receive the elastic bonds 3 and 4 are equal to:

$$T_1 = c_1(\varphi_1 - \varphi_2) = \frac{c_1 c_2 \omega t}{c_1 + c_2} - \frac{c_1 c_2 \omega}{c_1 + c_2} \sqrt{\frac{J_2}{c_1 + c_2}} \sin \sqrt{\frac{c_1 + c_2}{J_2}} t - \frac{c_1}{c_2} T_H \cos \sqrt{\frac{c_1 + c_2}{J_2}} t \quad (3)$$

$$T_2 = c_2 \varphi_2 = \frac{c_1 c_2 \omega t}{c_1 + c_2} + \frac{c^2 \omega}{c_1 + c_2} \sqrt{\frac{J_2}{c_1 + c_2}} \sin \sqrt{\frac{c_1 + c_2}{J_2}} t + T_H \cos \sqrt{\frac{c_1 + c_2}{J_2}} t \quad (4)$$

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

We conducted the study of the following equation:

$$\frac{c_1 c_2 \omega}{c_1 + c_2} \sqrt{\frac{J_2}{c_1 + c_2}} = \frac{c_1}{c_2} T_H. \quad (6)$$

As the result we obtain the only real value of the solution of equation (6):

$$c_1 = c_2 \left(3 \sqrt{\frac{c_2 J_2 \omega^2}{T_H^2}} - 1 \right). \quad (7)$$

We conducted the analysis of the cubic equation of type

$$c^3_1 + 3c^2_1 c_2 + 3c_1 c^2_2 + c^3_2 - \frac{c^4_2 J_2 \omega^2}{T_H^2} = 0$$

which believes that by values of variables c_1 are smaller than the value of the variable calculated by (6), the value of the left part in (7) is much larger than the right part [6-8]. As a result of the search coefficient values for these trigonometric functions in equation (3), with a possible decrease in S1 the amplitude of the oscillations increases, that we study by sinusoid $(c_1 c_2 \omega \sqrt{J_2 / (c_1 + c_2)}) / (c_1 + c_2)$, and the value of the amplitude of oscillations

decreases, which are committed the cosinusoid ($c_1 T_n / c_2$).

$$c_1 < c_2 \left(\sqrt[3]{\frac{c_2 J_2 \omega^2}{T_H^2}} - 1 \right) \quad (8)$$

The length of time for which the value of the elastic coupling moment 3 is identical T_n :

$$t_c = \frac{c_1 + c_2}{c_1 c_2 \omega} \left(T_n + \frac{c_1}{c_2} T_H \right) \quad (9)$$

Then $\sin \sqrt{(c_1 + c_2) t_c / J_2} = 0$; $\cos \sqrt{(c_1 + c_2) t_c / J_2} = 1$, in consequence of what we found:
 $\sqrt{(c_1 + c_2) t_c / J_2} = 2\pi n$,

Where $n=0, 1, 2, \dots, n$. At the value of expression (9), the last presented equation changes into an expression of type:

$$(c_1 + c_2) \sqrt{c_1 + c_2} \left(T_n + \frac{c_1}{c_2} T_H \right) = 2\pi n J_2 c_1 c_2 \omega \quad (10)$$

For hard of PFM $\left(c_1 > c_2 \left(\sqrt[3]{c_2 J_2 \omega^2 / T_H^2} - 1 \right) \right)$ is necessary to approve expression $\sin \sqrt{(c_1 + c_2) t_c / J_2} = 1$, $\cos \sqrt{(c_1 + c_2) t_c / J_2} = 0$, herewith

$$t_c = \frac{c_1 + c_2}{c_1 c_2 \omega} \left(T_n + \frac{c_1 c_2 \omega}{c_1 + c_2} \sqrt{\frac{J_2}{c_1 + c_2}} \right) \quad (11)$$

Here the value of angular stiffness PFM is equal to:

$$\sqrt{\frac{c_1 + c_2}{J_2}} t_c = \frac{\pi}{2} + 2\pi n$$

The value of the found expression is obtained in the form of an analytical total:

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

Result of this is $G = J_2 c^2 2 \omega^2 (\pi/2 + 2\pi n - 1)^2 / T_n^2$. The values of c_1 and t_c were obtained, these are the values of the torques that are acting on the elastic bond 4, are found in the following dependencies [6-9]:

- safety friction clutch-elastic type

$$T'_2 = T_{n1} + \frac{c_1 + c_2}{c_2} T_H \quad (12)$$

- safety friction clutch-hard type

$$T''_2 = T_{n1} + c_2 \omega \sqrt{\frac{J_2}{c_1 + c_2}} \quad (13)$$

As a result of the conducted researches it is necessary to specify: as the value increases c_2 the value of the moment which are occurring at the elastic of 4 (indicated by flexible coupling) reduces; with increasing c_2 the torque T''_2 (indicated the hard clutch) increases. We performed a study of the types of drive load, where before actuation of the safety clutch overload value is not the cause of stopping the working bodies of the machine. In this regard, the scheme is presented in Fig.2.

Fig. 2. Calculated dynamic scheme 2

We presented the equations for this system in the following form

$$J_1 \ddot{\varphi}_1 + c_1(\varphi_1 - \varphi_2) = T_d, \quad (14)$$

$$J_2 \ddot{\varphi}_2 - c_1(\varphi_1 - \varphi_2) + c_2(\varphi_2 - \varphi_3) = 0, \quad (15)$$

$$J_3 \ddot{\varphi}_3 - c_2(\varphi_2 - \varphi_3) = -T_H - T_c \frac{t}{t_c}, \quad (16)$$

where $\varphi_1, \varphi_2, \varphi_3$ is values of generalized angular coordinates of masses of inertia 1, 2 and 3; T_c – it's the value of the possible increase in torque in the device of the working body; t_c – it's the time value (is related to the value of the overload growth rate and T_c). The value of the torque size T_c depends on the type of process machine and the possibilities of its application, and is indicated as (1,2...4) T_H [10-13]. We established that the working body has a power reserve, and the value of the angular velocity of the masses of inertia 1 is assumed to be unchanged, i.e. $\varphi_1 = \omega t$.

As a result (16) we got [13]:

$$c_1(\varphi_1 - \varphi_2) = T_d.$$

We had to perform the summation operation as:

$$J_2\ddot{\varphi}_2 + (c_1 + c_2)\varphi_2 - c_1\omega t - c_2\varphi_3 = 0 \quad (17)$$

Result of the summation operation of equations (15) and (16):

$$J_2\ddot{\varphi}_2 - c_1\omega t + c_1\varphi_2 + J_3\ddot{\varphi}_3 = -T_H - T_c \frac{t}{t_c} \quad (18)$$

We need to differentiate this twice (16):

$$J_2 \frac{d^4\varphi_2}{dt^4} + (c_1 + c_2) \frac{d^2\varphi_2}{dt^2} - c_2 \frac{d^2\varphi_3}{dt^2} = 0 \quad (19)$$

Using equation (18) we obtain:

$$\frac{d^2\varphi_3}{dt^2} = \frac{1}{J_3} \left(c_1\omega t - c_1\varphi_2 - T_H - T_c \frac{t}{t_c} \right) - \frac{J_2}{J_3} \frac{d^2\varphi_2}{dt^2}$$

If we substitute the last expression into equation (16), we will obtain:

$$\frac{d^4\varphi_2}{dt^4} + \frac{c_2J_2 + (c_1 + c_2)J_3}{J_2J_3} \frac{d^2\varphi_2}{dt^2} + \frac{c_1c_2}{J_2J_3} \varphi_2 = \frac{c_2}{J_2J_3} \left(c_1\omega t - T_H - T_c \frac{t}{t_c} \right) \quad (20)$$

From (20) we will find:

$$\varphi_2 = \frac{1}{c_2} \left(J_3 \frac{d^2\varphi_3}{dt^2} + c_2\varphi_3 + T_H + T_c \frac{t}{t_c} \right) \quad (21)$$

If we differentiate the expression (21) twice, we will obtain:

$$\frac{d^2\varphi_2}{dt^2} = \frac{1}{c_2} \left(J_3 \frac{d^4\varphi_3}{dt^4} + c_2 \frac{d^2\varphi_3}{dt^2} \right) \quad (22)$$

After we had substituted expressions (22) and (21) into equation (18), we got:

$$\frac{d^4\varphi_3}{dt^4} + \frac{c_2J_2 + (c_1 + c_2)J_3}{J_2J_3} \frac{d^2\varphi_3}{dt^2} + \frac{c_1c_2}{J_2J_3} \varphi_3 = \frac{c_2}{J_2J_3} \left[c_1\omega t - \frac{c_1 + c_2}{c_2} \left(T_H + T_c \frac{t}{t_c} \right) \right] \quad (23)$$

The General solutions of equations (22) and (23) are written as follows:

$$\varphi_2 = A_1 \sin k_1 t + B_1 \cos k_1 t + C_1 \sin k_2 t + D_1 \cos k_2 t + \frac{1}{c_1} \left(c_1\omega t - T_H - T_c \frac{t}{t_c} \right) \quad (24)$$

$$\begin{aligned} \varphi_3 = & A_2 \sin k_1 t + B_2 \cos k_1 t + C_2 \sin k_2 t + D_2 \cos k_2 t + \\ & + \frac{1}{c_1 + c_2} \left[c_1\omega t - \frac{c_1 + c_2}{c_2} \left(T_H + T_c \frac{t}{t_c} \right) \right], \end{aligned} \quad (25)$$

Where

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

3 Research result

It's necessary to search for the value of the load system with usage of the expressions (22) and (23). It enters in the elastic bonds that are unbound in expression (18), as a result it appears the use of the value of the driving moment T_{Π} that are equal to the expression $(\varphi_1 - \varphi_2)(t)$. When the moment value appears T_H , the values of the generalized coordinates will be $\varphi_1 - \varphi_2 = T_H / c_1$, $\varphi_2 - \varphi_3 = T_H / c_2$, $\varphi_1 = \omega t$. If we have the increase in the load value from the initial indicators, the start conditions will be equal: at $t=0$, $\varphi_2 = -T_H / c_1$, $d\varphi_2 / dt = \omega$; $\varphi_3 = -(c_1 + c_2)T_H / c_1 c_2$, $d\varphi_3 / dt = \omega$. We have used (17) and (18) and the starting conditions, then we got (if $t = 0$):

$$\frac{d^2\varphi_2}{dt^2} = -\frac{c_2}{J_2 c_1} T_H; \quad \frac{d^3\varphi_2}{dt^3} = 0; \quad \frac{d^2\varphi_3}{dt^2} = \frac{c_2 - 2c_1}{J_3 c_1} T_H; \quad \frac{d^3\varphi_3}{dt^3} = 0;$$

We have used the values of the starting conditions and the obtained readings in the main expressions (22) and (23), we obtain the values of the constants of the integration process:

$$\begin{aligned} A_1 &= -\frac{k^2_2 T_c}{k_1(k^2_1 - k^2_2)c_1 t_c}; & B_1 &= \frac{T_H c_2}{c_1 J_2(k^2_1 - k^2_2)}; \\ C_1 &= \frac{k^2_1 T_c}{k_2(k^2_1 - k^2_2)c_1 t_c}; & D_1 &= -\frac{T_H c_2}{c_1 J_2(k^2_1 - k^2_2)}; \\ A_2 &= -\frac{k^2_2}{k_1(k^2_1 - k^2_2)} \times \frac{(c_1 + c_2)T_c}{c_1 c_2 t_c}; & B_2 &= -\frac{T_H(c_2 - 2c_1)}{c_1 J_3(k^2_1 - k^2_2)}; \\ C_2 &= \frac{k^2_1}{k_2(k^2_1 - k^2_2)} \times \frac{(c_1 + c_2)T_c}{c_1 c_2 t_c}; & D_2 &= \frac{T_H(c_2 - 2c_1)}{c_1 J_3(k^2_1 - k^2_2)}. \end{aligned}$$

The result of substitution of values that was obtained after the integration process in (22) and (23):

$$\begin{aligned} \varphi_2 &= \frac{1}{(k^2_1 - k^2_2)c_1} \left[\frac{T_c}{t_c} \left(\frac{k^2_1}{k_2} \sin k_2 t - \frac{k^2_2}{k_1} \sin k_1 t \right) + \right. \\ &\quad \left. + \frac{T_H c_2}{J_2} (\cos k_1 t - \cos k_2 t) \right] + \frac{1}{c_1} \left(c_1 \omega t - T_H - T_c \frac{t}{t_c} \right); \\ \varphi_3 &= \frac{1}{(k^2_1 - k^2_2)c_1} \left[\frac{(c_1 + c_2)T_c}{c_2 t_c} \left(\frac{k^2_1}{k_2} \sin k_2 t - \frac{k^2_2}{k_1} \sin k_1 t \right) + \right. \\ &\quad \left. + \frac{(c_2 - 2c_1)T_H}{J_3} (\cos k_2 t - \cos k_1 t) \right] + \frac{1}{c_1 + c_2} \left[c_1 \omega t - \frac{c_1 + c_2}{c_2} \left(T_H + T_c \frac{t}{t_c} \right) \right]. \end{aligned}$$

We obtained the values of the torques that have been perceived by the elastic bonds 4 and 5:

$$\begin{aligned} T_1 = (\varphi_1 - \varphi_2)c_1 &= \frac{1}{k^2_1 - k^2_2} \left[\frac{T_c}{t_c} \left(\frac{k^2_2}{k_1} \sin k_1 t - \frac{k^2_1}{k_2} \sin k_2 t \right) - \right. \\ &\quad \left. - \frac{T_H c_2}{J_2} (\cos k_1 t - \cos k_2 t) \right] + T_H + T_c \frac{t}{t_c}; \end{aligned} \quad (26)$$

$$T_2 = (\varphi_2 - \varphi_3)c_2 = \frac{c_2}{(k^2_1 - k^2_2)c_1} \left[T_H \left(\frac{c_2}{J_2} + \frac{c_2 - 2c_1}{J_3} \right) (\cos k_1 t - \cos k_2 t) - \right. \\ \left. - \frac{T_c c_1}{c_2 t_c} \left(\frac{k^2_1}{k_2} \sin k_2 t - \frac{k^2_2}{k_1} \sin k_1 t \right) \right] + \frac{c^2_2}{c_1 + c_2} \omega t + \frac{c_1 - c_2}{c_1} \left(T_H + T_c \frac{t}{t_c} \right). \quad (27)$$

We established that $k_1 t$ and $k_2 t$ find the values of expressions unrelated to each other, as well as the values of the quantities $\sin k_1 t$ and $\sin k_2 t$, $\cos k_1 t$ and $\cos k_2 t$ have signs of the same or different character. The main condition for determining the value of the positive maximum of this equation is the values of simultaneous expressions $\sin k_1 t = 1$ and $\sin k_2 t = -1$ or $\cos k_1 t = -1$ and $\cos k_2 t = 1$ [10-12]. The period values of the specified time intervals were taken from (27):

$$t_1 = (T_\Pi - T_H) \frac{t_c}{T_c} - \frac{k^3_1 + k^3_2}{k_1 k_2 (k^2_1 - k^2_2)} \quad (28)$$

$$t_2 = \frac{t_c}{T_c} \left\{ T_\Pi - \left[\frac{2c_2}{(k^2_1 - k^2_2)J_2} + 1 \right] T_H \right\} \quad (29)$$

If we substitute (27) and (28) in (29), we will obtain the values of the interrelated numbers of torques that are the part of the elastic bond 5 [13]:

$$T'_2 = \frac{(k^3_1 + k^3_2)c_2}{k_1 k_2 (k^2_1 - k^2_2)} \left(\frac{T_c}{c_1 t_c} - \frac{c_2 \omega}{c_1 + c_2} \right) + (T_\Pi - T_H) \frac{t_c}{T_c} \left[\frac{c^2_2 \omega}{c_1 + c_2} + \frac{(c_1 - c_2)T_c}{c_1 t_c} \right] + \\ + \frac{c_1 - c_2}{c_1} T_H; \quad (30)$$

$$T''_2 = \frac{t_c}{T_c} \left\{ T_\Pi - \left[1 + \frac{2c_2}{(k^2_1 - k^2_2)J_2} \right] T_H \right\} \left[\frac{c^2_2 \omega}{c_1 + c_2} + \frac{(c_1 - c_2)T_c}{c_1 t_c} \right] + \\ + T_H \left[\frac{c_1 - c_2}{c_1} - \frac{2c_2}{(k^2_1 - k^2_2)c_1} \left(\frac{c_2}{J_2} + \frac{c_2 - 2c_1}{J_3} \right) \right]. \quad (31)$$

If we want to derive the maximum value of the torque that are transmitted by the safety friction clutch when the external torque increases to T_i , we'll have to apply the

expression to find the value of the spacer force [13]: $F_p = \frac{T_i - T'_2}{r} \operatorname{tg} \alpha$. With the

application of the current expression the torque value T'_2 is found as

$$T'_2 = R_{cp} f_i \left(F_\Pi - \frac{T_i - T'_2}{r} \operatorname{tg} \alpha \right),$$

whence we obtain that $T_\Pi = T_i$. In this connection, 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 value of the external torque is the same as the value of the friction torque.

4 Discussion and conclusions

As you can see, the upgraded circuit diagram theoretically allows to obtain a very high stability of the friction force, but due to the relatively large value of the parameter $\operatorname{tg}\alpha$, the force F_{Π} is used inefficiently. A necessary condition for the absence of zero reversal of the output parameter in the interval of change of the coefficient of friction and the presence of a maximum of the load capacity function is the transfer of its full load by the sensing elements. An additional condition for the existence of the maximum is the need to transfer the sensing elements of its full load at an equal number of friction pairs of both friction groups, and transfer the sensing elements of the full load of the main friction group at a lower number of friction pairs than in the additional friction group.

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