Aspects of using single-circuit negative feedback in a safety friction clutch with all leading friction pairs, taking into account installation in crank presses

K O Kobzev, E S Bozhko, A V Mozgovoi, Y E Chertov, I A Zanina
Don State Technical University, Gagarin's square 1, Rostov-on-Don 344000, Russia
E-mail: 5976765@mail.ru

Abstract. A feature of the considered option is that, as an analog, a safety friction clutch (SFC) of the usual accuracy of operation in crank presses, in which there is no feedback control device, is adopted. Moreover, this case is applied only with a minimum value of the coefficient of friction, i.e., a constructive version of such a SFC should provide for the feedback control device to be turned off from operation under the indicated condition. Taking into account the structure of the ratio for calculating the magnitude of the gain, as well as the fact that for the “ideal” SFC, this value should be variable, it should be noted, taking into account the previous experience in calculating and constructing SFC of this type, that it is most convenient to vary the magnitude of the gain by changes in the angle (pressure angle of the rolling element control device).

1. Introduction
A feature of the considered option is that, as an analogue, a safety friction clutch (SFC) of the usual accuracy of operation in crank presses, in which there is no feedback control device, and therefore, $C = 0$. Moreover, this case is applied only with a minimum value of the coefficient of friction, i.e., a constructive version of such a SFC should provide for the feedback control device to be turned off from operation under the indicated condition. Taking into account the structure of the ratio for calculating the magnitude of the gain, as well as the fact that for the “ideal” SFC, this value should be variable, it should be noted, taking into account the previous experience in calculating and constructing SFC of this type, that it is most convenient to vary the magnitude of the gain by changes in the angle (pressure angle of the rolling element control device) [1-4]. Obviously, in this case, when $f_{\min}, C_{\min} = 0, \alpha_{\min} = 0 (\alpha_{\min} - \text{minimum pressure angle}).$

2. Methods
It is technically impossible to fulfill this condition, since there will be no spacer force at zero angle of inclination of the side walls of the nests $F_p$, and at any additional external load, the rolling elements will not transition to the profile of the side wall of the socket with a non-zero angle $\alpha$. Consequently, the principle of automatic SFC regulation will be lost.
The indicated principle can be implemented, for example, as shown in Fig. 1. Here is a schematic diagram of a SFC with a control device that conditionally implements a zero angle $\alpha$, $f_{\text{min}}$. This is achieved due to the fact that the connection between the rolling bodies 6 and the pressure plate 5 in the axial direction is not directly carried out, as in the basic version of the SFC (Fig. 1), but by means of the support sleeve 11, which is installed in the central hole of the pressure plate.

The support sleeve is spring-loaded with respect to the pressure disk 5 by the main spring 12, and with respect to the stop 13 rigidly fixed on the coupling half 1 - with the help of an additional spring 14 (Fig. 1). To transmit torque between the pressure disk 5 and the supporting sleeve 11, a key 15 is installed. The seats for the rolling elements 6 made in the thrust disk 7 and in the supporting sleeve 11 are profiled by curved lines convex in the direction of the rolling bodies to reduce the angle $\alpha$ from the bottom of the nest to its periphery [5-8].

According to the condition of functioning of the control device and the task, the tension force of the spring 12 should be zero at a minimum value of the coefficient of friction. This is due to the lack of a spacer force acting on the pressure plate 5 under the specified condition. The tension force of the spring 14 is not equal to zero, since the gain is not a zero angle. Therefore, the tension force of the spring 14 should compensate for the spacer force, i.e.

$$F_{\text{p, min}} = F_{14},$$

$F_{\text{p, min}}$ - minimum spacing; $F_{14}$ - spring tension force 14 at $f_{\text{min}}$. $F_{\text{p, min}}$ can be calculated using the previously obtained ratio.

3. Results
Using the result contained in the right-hand side of equality (1) in expression (2), we can find the minimum spring tension $F_{\text{p,14}}$.

Equality (2) contains the term $C_{\text{min}}$, to determine which it is necessary to fulfill a condition that excludes self-braking when moving to the right (in Fig. 1) of the support sleeve 11 relative to the guide key 15.

The ability to carry out the movement of the support sleeve 11 to the right as a result of the influence of a spacer force on it is determined by the ratio of the values of the friction forces between the support sleeve and the guide key and the spacer force feedback control device. This ratio is expressed by the following equilibrium of the support sleeve

$$F_{\text{p, min}} - F_{\text{r}} - F_{\text{p,14}} = 0,$$

$F_r$ – friction force between the support sleeve and the key.

The thrust force is calculated by the formula:

$$F_{\text{p, min}} = \frac{T_p (z - z_i)}{z r} \tan \alpha_{\text{min}},$$

$\alpha_{\text{min}}$ – the minimum angle between the tangent at the contact point of the rolling body 6 with the socket of the support sleeve 11 and the axial line of the socket [9-13].

Given the previously obtained expression, we can write expression (4) in the form:

$$F_{\text{p, min}} = \frac{(z - z_i) F_p R_{\text{vr}} f_{\text{min}} \tan \alpha_{\text{min}}}{r},$$

The friction force between the support sleeve and the guide key is calculated by the formula:

$$F_r = \frac{2(z - z_i) T_p f_1}{z d_0},$$

$f_1$ – coefficient of friction between the support sleeve and the key 15; $d_0$ – the diameter of the Central hole of the support sleeve 11.

Angle $\alpha_{\text{min}}$, as shown above depends on the coefficient of friction $f_1$. To establish the relationship between them, we turn to the circuit shown in Fig. 2. The diagram shows a fragment of the nest of the support sleeve 11 and the rolling body 6 located in the nest. Minimum angle $\alpha_{\text{min}}$ between the tangent A-A to the profile of the wall of the nest at the point of contact between it and the rolling body and the axial line o-o nests corresponds to the position of the rolling body with a minimum value of the coefficient of friction. Normal $\vec{N} - \vec{N}$ held to the guide key 15 and forms normal pressure forces with the vector $\vec{F}_p$ angle equal to angle $\alpha_{\text{min}}$. The friction angle between the support sleeve and the guide key is indicated as $\rho_{\text{min}}$.

The condition for the motion of the bodies of the frictional contact “body 11 – body 15” is written as:

$$\alpha_{\text{min}} \geq \rho_{\text{min}}.$$

After substituting the right-hand sides of expressions (6) and (7) into equation (5), taking into account condition (3) (in the form of equality, as a limiting case of the relative motion of bodies 11 and 15), we obtain:

$$F_{\text{p,14}} = (z - z_i) F_p R_{\text{vr}} f_{\text{min}} f_1 \left(1 - \frac{2}{d_0}\right).$$

In the expression (8), the replacement $\tan \alpha_{\text{min}} = f_1$. The second case relates to the operation of the SFC in adaptive mode with a minimum value of the coefficient of friction. Obviously, in this case, the
gain should not be zero. Then the nominal torque of the coupling will be equal, in accordance with the expression:

\[ T_p = zF_p R \frac{f_{\text{min}}}{1 + (z - z_1)C_{\text{min}} f_{\text{min}}} \]  

(9)

Equating to each other the right parts of the previously found expression and relation (9) and replacing in the first \( f_i, f_i^f, C_i, C_i \), we get the dependence \( C_i, f_i \) based on the solution of the equation:

\[ C_i = \frac{1}{z - z_1} \left[ \frac{1}{f_i^{\text{min}}} - \frac{1}{f_i^f} + (z - z_1)C_{\text{min}} f_{\text{min}} \right] . \]  

(10)

Comparing the previously obtained dependence and dependence (10), we see that the current coefficient is the gain value, the gain coefficient \( C_i \) in the second case, as in the first case. In addition, as follows from expression (10), the nominal SFC torque is less than \( 1 + (z - z_1)C_{\text{min}} f_{\text{min}} \) times than the SFC torque with differentiated friction pairs[14-19].

Using the previously obtained relation, we find the restrictions on the values for the parameters \( z, z_1 \). Solving quadratic inequality with respect to the unknown \( z_1 \):

\[ z_1 \in \left[ -\infty; \frac{f_{\text{max}} (1 + zC_{\text{min}} f_{\text{min}}) - \sqrt{\left( f_{\text{max}} (1 + zC_{\text{min}} f_{\text{min}}) - 4zC_{\text{min}} f_{\text{min}} f_{\text{max}} \right)^2}}{2C_{\text{min}} f_{\text{min}} f_{\text{max}}}, +\infty \right] \cup \left[ \frac{f_{\text{max}} (1 + zC_{\text{min}} f_{\text{min}}) + \sqrt{\left( f_{\text{max}} (1 + zC_{\text{min}} f_{\text{min}}) - 4zC_{\text{min}} f_{\text{min}} f_{\text{max}} \right)^2}}{2C_{\text{min}} f_{\text{min}} f_{\text{max}}}, +\infty \right] . \]  

(11)

Many solutions of (11) were obtained taking into account the positive term \( C_{\text{min}} f_{\text{min}} f_{\text{max}} \) with an unknown second degree in the square inequality, as well as the fact that \( D > 0 \) (\( D \) - discriminant of square inequality). Really, \( D = f_{\text{max}}^2 + 2zC_{\text{min}} f_{\text{min}} f_{\text{max}}^2 + z^2C_{\text{min}}^2 f_{\text{min}}^2 f_{\text{max}}^2 - 4zC_{\text{min}} f_{\text{min}} f_{\text{max}}. \)  

(12)

In expression (12) we compare the second and fourth terms

\[ \Delta = 2zC_{\text{min}} f_{\text{min}} f_{\text{max}} (f_{\text{max}} - 2f_{\text{min}}) . \]  

(13)

From relation (13) it follows that \( \Delta > 0 \), since the gain is the difference in brackets is positive (for known combinations of modern friction materials used in SFC).

4. Discussion
Graphically, the set of solutions (13) is shown in Fig. 3. Line 1 reflects the right boundary of the left half-region of solutions (13), line 2 - the left border of the right half-region. Parameter Values \( z_1 \) satisfying the necessary constraint \( C \) above (see above), are located in the regions of solutions located respectively below line 1 and above line 2 (see Fig. 3).
The graphs shown in Fig. 3, are constructed according to the following initial data: \( f_{\text{max}} = 0.8 \), \( f_{\text{min}} = 0.1 \), \( C_{\min} = 0.3 \), \( z = 8 \). It follows from solutions (13) that the parameter \( z_1 \) increases with increasing \( z \): slower for the right boundary of the left half-region of solutions, faster for the left border of the right half-region of solutions (lines 3 and 4, respectively, constructed with the same initial data, except for the parameter \( z = 16 \)).

At the same time, it follows from Fig. 3 that the left boundary of the right half-region of solutions (13) reflects the values of the parameter \( z_1 \), which exceed the value of the parameter \( z \), taken as the initial one when plotting charts. This contradicts the physical picture, according to which the parameters \( z, z_1 (z > z_1) \). Therefore, the right half-domain of solutions (13) cannot be taken into account, since the gain leads to results that really cannot be put into practice.

In addition, the graph in Fig. 3 (line 1) shows that the value of the parameter \( z_1 \), gain necessary to achieve a sufficient upper limit of the value \( C_{\text{max}} \), does not meet the set condition, since the gain is less than unity. Parameter increase only \( z \) up to sixteen allows you to implement the option SFC with parameter value \( z_1 = 1 \) (line 3).

Thus, if all the SFC friction pairs are leading, then the current gain is the feedback gain less than when the SFC is made with differentiated friction pairs.

A schematic diagram of a feedback control device with switching off the action of the latter with a minimum value of the coefficient of friction is proposed and calculated, which allows the construction of a SFC with increased load capacity for crank presses.

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