Theoretical foundations of the use of single-circuit negative feedback in safety friction clutches with differentiated friction pairs installed in forging equipment

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Abstract. From studies aimed at summarizing the results of private surveys, conditions of high accuracy SFC with single-loop negative feedback are obtained. These conditions reflect the dependences of the magnitude of the coefficient of force and the spacer force generated by the feedback control device on the current coefficient of the gain value of the coefficient of friction, which theoretically ensure the limiting value of the torque transmitted by the clutch, which leads to failure-free operation of the crank presses.

1. Introduction

The safety friction clutch (SFC) installed in crank presses with single-circuit negative feedback is classified by the first generation couplings and is most widely used in practice among other types of couplings. The reason for this, despite the relatively low gain enhance the accuracy of the response, is the structural simplicity, extensive research on this type of couplings, proven tuning techniques and extensive operational experience, which allowed us to obtain the results necessary to optimize the design parameters of the couplings.

It should be noted that in relation to the SFC of this type, only private studies have been carried out relating to certain aspects of their functioning. So, the influence on the response accuracy of the so-called gain of the feedback control device, the influence of the gain value on the load capacity of the couplings is studied quite thoroughly, questions of tuning the SFC to improve the reliability of load transfer in the kinematic circuit of the drive, as well as stability issues of the movement of the drive with SFC.

From studies aimed at summarizing the results of private surveys, conditions of high accuracy SFC with single-loop negative feedback are obtained. These conditions reflect the dependences of the magnitude of the coefficient of force and the spacer force generated by the feedback control device on the current coefficient of the gain value of the coefficient of friction, which theoretically ensure the limiting value of the torque transmitted by the clutch, which leads to failure-free operation of the crank presses.

2. Methods

A schematic diagram of a first-generation SFC with single-circuit negative feedback having differentiated friction pairs of the type “leading pairs - driven pairs” is shown in Fig. 1 (the upper part of
Two kinematic half-couplings 1 and 2 coaxial with each other are connected to each other in the circumferential direction of the friction group consisting of friction disks 3 and 4. The disks 3 are connected in the circumferential direction with the hub of the pressure disk 5, the disks 4 are connected to the half-coupling drum 2.

The pressure disk 5 is devoid of kinematic connection with the hub of the coupling half 1 in the circumferential direction, with the exception of slight friction between them, which is not considered further [1-3].

The control device is made in the form of rolling bodies 6, which are placed in beveled sockets made on the end surfaces of the pressure disk facing one another and rigidly fixed on the hub of the coupling half 1 of the thrust disk 7 (Fig. 1, section A-A).

The force closure of friction pairs is created by a spring 8, the force of which is transmitted to the pressure disk through a thrust bearing 9 to reduce friction between them[4-6].

![Figure 1. Schematic diagram of the first generation PFM with differentiated friction pairs.](image)

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Given the action of the spacer force in the SFC (see Fig. 1, section AA), we write the formula for the torque in the form:

$$ T_p = zR f(F_p - F), $$

$$ F $$ – spacer force; other designations are looked above.

When the structural-layout scheme of the friction group, built according to the type of "leading friction pairs driven friction pairs", the spacer force is determined by the formula:

$$ F = \frac{(z - z_1)T_p}{zr} \tan \alpha, $$

$$ z_1 $$ – the number of leading friction pairs of the friction group.

Putting the expression (2) in the formula (1), we obtain:

$$ T_p = zF_p R \frac{f}{1 + (z - z_1)Cf}, $$

The greatest accuracy of the SFC operation will be when $T_p = \text{const}$. In this regard, the question arises of choosing the required magnitude of the torque of the coupling.

The nominal value of the torque at which the SFC is adjusted is usually taken as the required value. SFC tuning to the rated torque is carried out according to the minimum coefficient of friction, which allows minimizing the clutch response when the coefficient of friction is reduced and increasing the reliability of load transfer without interrupting the course of the process performed by the machine.
3. Results

Based on the foregoing, we write the formula for the nominal torque of the SFC, using relation (3):

\[ T_{\text{min}} = z F_p R \frac{f_{\text{min}}}{1 + (z - z_i)C f_{\text{min}}}, \]  

(4)

\( f_{\text{min}} \) – minimum coefficient of friction.

Equating to each other the right-hand sides of relations (3) and (4), we find:

\[ C_i = \frac{1}{z - z_i} \left( \frac{1}{f_{\text{min}}} - \frac{1}{f_i} \right), \]  

(5)

\( C_i \) – current gain gain value; \( f_i \) – current coefficient value of the coefficient of friction.

Expression (5) shows that for maximum accuracy of the SFC operation, the gain should be variable, and it functionally depends on the friction coefficient [7-10].

In addition, from the expression (5) it follows that for \( f_{\text{min}} = f_i \) should be \( C_{\text{min}} = 0 \).

Finally, expression (5) shows that the value of the gain should increase with increasing \( f_{\text{min}} \).

The graph of function (5) is shown in Fig. 2 Curve 1 is constructed from the following source data: \( z = 6, \ z_i = 1, \ f_{\text{min}} = 0.1 \).

Curve 1 shows that, with the indicated initial data, the absolute increase in the gain is 1.75 (for \( C_{\text{min}} = 0 \)).

Parameter Influence \( z_i \) the magnitude of the gain is shown by curve 2, constructed at \( z_i = 2 \): the absolute increase in gain in this case is 2.19.

![Figure 2. Graphs of the dependence of the gain on the coefficient of friction](image)

Therefore, with an increase in the number of driven friction pairs, higher values of the gain are required to ensure the “ideal” SFC load characteristic. Physically, this is due to the fact that when \( T_{p,\text{min}} = \text{const} \) a decrease in the number of driven friction pairs leads to a decrease in the torque transmitted by the pressure plate and rolling elements of the control device. To compensate for the decrease in spacer force, an increase in gain is required.

Second condition \( f_{\text{min}} = 0 \) theoretically impossible, since the gain, according to the physical meaning of the gain (see above), must be equal to zero at least one of the two input parameters, \( R \) или \( \alpha \).

Value \( R \) cannot be equal to zero, since the gain is lost any meaning of the SFC functioning [11-15].

To achieve the condition \( F_p = 0 \) theoretically parameter \( \alpha \) may be zero. However, from a practical point of view, this is not possible, since the gain at \( f_{\text{min}} > f_{\text{min}} \) SFC should work with negative feedback, i.e., with \( F_p \neq 0 \). Therefore, from the physico-mechanical point of view, in this case, the
transition of the rolling elements of the control device from a zero pressure angle to a non-zero angle is necessary. At $\alpha_{\text{min}} = 0$ this seems impossible.

In the SFC of the first generation with differentiated friction pairs, there is a limit from above on the magnitude of the gain, which in this case has the form:

$$C_p \leq \frac{1}{z_1 f_{\text{max}}}.$$  

(6)

$f_{\text{max}}$ – maximum coefficient of friction.

Replacing the parameter in relation (5) $f_i$ per parameter $f_{\text{max}}$ and using the right-hand sides of relations (5) and (6) in the equality $C_i = C_p$, we get:

$$z_1 = \frac{z}{m},$$  

(7)

$m$ – relative width of the range of the coefficient of friction $f_{\text{min}} \ldots f_{\text{max}}$:

$$m = f_{\text{max}} / f_{\text{min}}.$$

If accept $z_1 = 1$, then the condition for equal values of gain $C_i$, $C_p$ performed when

$$z = m.$$

(8)

Relation (7) shows that to ensure a minimum integer value $z_1 = 1$ it is necessary that the total number of SFC friction pairs be at least equal to the coefficient $m$. Usually $m = 8 \ldots 10$, which corresponds to a multi-disc version of the clutch.

The same conclusion can be drawn from the analysis of relation (8).

Therefore, the implementation of the limitation of the magnitude of the gain at the maximum value of the coefficient of friction is possible only in the multi-disc version of the SFC with differentiated pairs of friction. The number of leading friction pairs should be minimal.

We are also investigating a variant of the structural-layout scheme of the friction group in which friction pairs are not differentiated into leading and driven, but all are driven. The indicated option is shown in fig. 1 (lower part relative to the axis of rotation of the coupling) [16-19].

The difference between the considered SFC variant is the introduction of a thrust bearing 10, installed between the leftmost (in Fig. 1) friction disk 4 and the thrust disk 7. The kinematic connection between them is broken, and the entire load of the coupling is transferred from the coupling half to the coupling half 2 (or vice versa) by means of rolling 6.

Applying the same method as for the SFC variant with differentiated friction pairs, the calculation gain method, we find the expression for calculating the coupling torque:

$$T_p = z f R \frac{f}{1 + z C_f}.$$  

(9)

Relation (9) differs from relation (3), with the same parameters, by a lower value of the reduced coefficient of friction equal to:

$$f_p = \frac{f}{1 + z C_f},$$

and, accordingly, lower torque $T_p$.

This is explained by the greater magnitude of the spacer force acting on the friction pair, due to the fact that in this case the rolling elements of the control device transmit the full torque of the SFC, and not a certain part of it, as in the SFC with differentiated pairs of friction.

Using the mathematical calculations similar to the above, we find the dependence of the magnitude of the gain on the coefficient of friction:

$$C_i = \frac{1}{z} \left( \frac{1}{f_{\text{min}}} - \frac{1}{f_i} \right).$$  

(10)
The relation (10) is similar to the relation (5): it gives the same general dependencies between the function and variable arguments \( z, f_i \): The only difference is the smaller the current gain, the gain value at equal current gains, the values of the friction coefficient.

Since the gain, as noted above, at \( f_{\min} \) should be \( C_{\min} = 0 \), introduce a new parameter \( C'_{\min} \neq 0 \). Using this parameter in the initial relation (4) and having performed the mathematical action described above, we find:

for the first version of the SFC layout scheme:

\[
C' = \frac{1}{z - z_i} \left( \frac{1}{f_{\min}} - \frac{1}{f_i} \right) + C'_{\min},
\]

(11)

for the second version of the SFC layout scheme:

\[
C' = \frac{1}{z - z_i} \left( \frac{1}{f_{\min}} - \frac{1}{f_i} \right) + C'_{\min}.
\]

(12)

Formulas (11) and (12) differ from similar formulas (5) and (10) by the presence of terms \( C'_{\min} \) \( C''_{\min} \) respectively. In this case, when \( f_i = f_{\min} \) will be \( C_{\min} = C'_{\min} \) \( C''_{\min} \). This means that with a minimum value of the coefficient of friction in the SFC, negative feedback will act.

The minimum and, at the same time, tuning (nominal) torque in this case should be calculated using relation (4) with the replacement of the parameter in it \( C \) per parameter \( C'_{\min} \) (for the first version of the SFC).

For the second variant of SFC, formula (4) can also be used with the above replacement \( C \) and without taking into account the parameter \( z_1 \).

Consider the issue of parameter values \( C'_{\min} \) \( C''_{\min} \). We will solve this problem on the basis of the same nominal load capacity of the compared SFC options.

Using relations (3) and (9) with the replacement of parameters in each of them \( f, C \) respectively on the parameters \( f_{\min} \), \( C_{\min} \), \( (C'_{\min} \) \( C''_{\min} \) ), which corresponds to the nominal torque of the SFC, we find:

\[
C'_{\min} = \frac{1}{z - z_i} \left( \frac{zFR}{T_p} - \frac{1}{f_{\min}} \right),
\]

(13)

\[
C''_{\min} = \frac{1}{z} \left( \frac{zFR}{T_p} - \frac{1}{f_{\min}} \right),
\]

(14)

An analysis of relations (13) and (14) shows that \( C'_{\min} > C''_{\min} \). Therefore, given this result, we can conclude that, taking into account relations (11) and (12), \( C_i' > C''_i \), moreover, the difference between these values increases in comparison with the values calculated by the relations (5) and (10).

The difference in the values of the gain for the first and second versions of the SFC, however, affects their nominal load capacity. Indeed, substituting relations (13) and (14) into expressions (3) and (9), respectively, leads to different results – \( T'_{\min} > T''_{\min} \) (respectively, the nominal torques of the first and second variants of the SFC).

Moments \( T'_{\min}, T''_{\min} \) obtained from expressions (3) and (9) by replacing the parameter in them \( f \) per parameter \( f_{\min} \).

4. Discussion

For the first SFC variant, it is important to check the maximum value of the gain coefficient for its restriction due to the limitation of the gain above, for SFCs having differentiated friction pairs to reduce overload in crank presses. For SFC with all driven friction pairs of such a restriction, the gain
in the stationary loading mode does not exist.

Using relation (6) for \( z_1 = 1 \) and expression (11), taking into account formula (13), in the equality
\[
C'_{\max} = C'_n,
\]
we find:
\[
C'_{\min} = \frac{1 + \frac{1}{m} - \frac{1}{(z - 1)(1 - \frac{1}{m})}}{1 + (z - 1)f'_{\min}}.
\] (15)

When deriving relation (15), expression (4) was used with a replacement in the last parameter \( C \) per parameter \( C'_{\min} \).

Relation (15) allows us to abstract from such SFC parameters as \( T_p, F_p, R \), use when calculating \( C'_{\min} \) minimum amount of source data.

As follows from relation (15), the value \( C'_{\min} \) inversely proportional to the coefficient of friction \( f'_{\min} \), if we consider the value of the coefficient of friction \( f'_{\max} \) variable, and \( m = \text{const} \). Therefore, to minimize \( C'_{\min} \) friction materials with friction coefficients as high as possible should be assigned \( f'_{\min} \).

From relation (15) it also follows that the value \( C'_{\min} \) decreases with increasing coefficient \( m \). In this regard, it is advisable from the point of view of minimization \( C'_{\min} \), use friction materials of friction pairs with the greatest dispersion of the coefficient of friction.

![Figure 3](image.png)

**Figure 3.** Graphs of the dependence of the minimum value of the gain on the number of friction pairs

Nature of function \( C'_{\min}(z) \) can be traced by the graph gain, constructed in accordance with relation (15). The graph is built at \( f'_{\min} = 0,1 \) and \( m = 8 \), which corresponds to real data.

The graph (Fig. 3) shows that \( C'_{\min} \) with increasing increases in the range of 2 ... 4, then, having reached a maximum, decreases. With this minimum value \( z \) corresponds to \( C'_{\min} \), much smaller than with \( z > 4 \).

Therefore, with the minimum value of the parameter \( z \) better use is achieved from the point of view of minimizing the spacer force with a minimum value of the coefficient of friction and, thus, increasing the nominal load capacity of SFC.

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References

[1] Zhivov L.I., Ovchinnikov A.G., Skladchikov E.H. Forging and stamping equipment: Textbook for universities / Ed. L.I. Zhivova. - M.: Publishing House of MSTU. N.E. Bauman, 2015. -- 560 f : ill.

[2] Zhivov L.I., Chumakov B.N., Drozdov N.G. Features of the dynamics of the hot stamping crank press for stamping low forgings. “University News. Mechanical Engineering ”, 2012, No. 1, p. 155-159

[3] Zalessky V.I. Equipment forging shops. Ed. 2nd, overwork, and add. Textbook for high schools. M., "Higher School", 2009. 632 p.

[4] Crank forging machines / Ed. Vlasova V.I. - M.: Mechanical Engineering, 2012. -- 424 p.

[5] Noskov G.P., Rodov G.M., Vyatkin V.P. An experimental study of the loads in the crank press drive during a technological operation. Forging and stamping production, 2016, No. 5, p. 30 - 32

[6] Svistunov V.E. Forging and stamping equipment. Crank presses: Study Guide. - M.: MGIU, 2016. -- 704 p.

[7] Svistunov V.E. The results of mathematical modeling of crank presses with compact actuators. Forging and stamping production, 2015, No. 10, 24 - 27

[8] Sokov V.I. Experimental determination of the friction moment when the clutch is engaged and braking of the hot-stamping crank presses. Forging and stamping production. Metal forming. 2015, No.10, 29-35

[9] Truskovsky V.I. The dynamics of forging machines. - Chelyabinsk: Publishing house of SUSU, 2015.-79c.

[10] Fedorkevich V.F. About the rigidity of modern crank hot stamping presses / Forging and stamping. Metal forming. 2013, No. 5, p. 23 - 25

[11] Hoopfer P. Dynamic loads in crank presses. “Forging and stamping production”, 2011, No. 2, p. 28 - 31

[12] Chubukov V.A., Gartvig A.A. Investigation of the influence of the design and parameters of hot-stamping crank presses on the nomenclature and accuracy of stamped products. Collection of scientific reports of the VI International Conference "The participation of young scientists, engineers and educators in the development and implementation of innovative technologies." - M.: MGIU, 2016, p. 134-138

[13] Scheglov V.F., Maksimov L.Yu., Linz V.P. Forging machines. - 2nd ed., Revised. and add. - M.: Mechanical Engineering, 2010. -- 304 e., Ill.

[14] Patent RU 2427466. A method for protecting crank presses from overloads by force on a slider. Svistunov V.E., Chubukov V.A., Matveev A.G. Publ. 08/27/2011

[15] Schumann K. Methode zur rechnerischen Untersuchung der technologischen Stobbeanspruchung mechanismischer Pressen. Umformtechnik, 24 (2012), No. 2, p. 29 - 35

[16] Hiraishi Kenji, Kagawa Toshiaki. Sumitomo jukikaigiho. Sumitomo Heavy Ind. Techn. Rev. 2010, No. 164, p. 9

[17] Schnellaufer-Press produziert Platinen. Maschinenmarkt. 2009, No. 41, p. 36

[18] Stanzzoder Umformautomat: Application 1867469 EPO, IPC B 30 V 15/04 (2006.01), B30 B 15/00 (2006.01). Haulick + Roos GmbH, Siegel Andreas (Hoeger, Stellrecht & Partner Patentanwälte Uhlandstrasse 14 with 70182 Stuttgart): No. 06012074.8; Claim 06/12/2008; Publ. 12/19/2009.

[19] Stanzautomat selbst erganzen. Blesh InForm. 2009, No. 4, p. 87