Determination of the practical accuracy of the operation of the safety friction clutch with a variable gain installed in the forging equipment ensuring the absence of breakdowns

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Abstract. The created version of SFC with a variable value of the feedback gain theoretically has the accuracy of operation of the “ideal” SFC when it is installed in forging and stamping equipment. A variable value of the gain is obtained by changing the pressure angle of the rolling element control device depending on the transmitted load.

To obtain a variable value, the gain in the SFC was changed in the design of the control device.

The experimental studies of the full-scale SFC sample with a variable gain, however, showed that the coupling does not have the accuracy of operation inherent in the "ideal" SFC. Since the gain was experimentally studied in the static loading mode on the installation with a shortened kinematic chain, i.e., with a smooth increase in the load (the time of the full overload, which leads to the operation of the SFC, is longer than the period of natural vibrations of the system), the effect of unaccounted dynamic loads is practically excluded.

1. Introduction

The created version of the first-generation SFC with a variable value feedback gain theoretically has the accuracy of operation of the "ideal" SFC. A variable value of the gain is obtained by changing the pressure angle of the rolling element control device depending on the transmitted load.

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The experimental studies of the full-scale SFC sample with a variable gain, however, showed that the coupling does not have the accuracy of operation inherent in the "ideal" SFC. Since the gain was studied experimentally in a static loading mode on a setup with a shortened kinematic chain, i.e., with a smooth increase in load (the time of full overload, which leads to SFC operation, is longer than the period of natural vibrations of the system), the effect of unaccounted dynamic loads was practically eliminated.

2. Methods

The equation of axial equilibrium of the support sleeve 6, according to the diagram of forces depicted in Fig. 1 has the form:

\[ F_p - F_n - F_f = 0, \]

\[ F_p \] current coefficient reinforcing spacer force between rolling elements 4 and supporting sleeve 6; \[ F_n \] coefficient of reinforcing tension force spring 9; \[ F_f \] friction force between the pressure plate...
7 and the guide key 8 [1-3].

Unlike the first three members of equation (1), the force $F_{\tau}$ adopted linearly as a function of torque SFC. The value of the coefficient of friction between the pressure plate and the guide key, according to the work, is considered constant.

In equation (1), the terms can be represented as

$$F_{\tau} = \frac{T_{\text{nom}}}{r} \cdot \tan \alpha_i; \quad F_{\nu} = \frac{2T_{\text{nom}}}{d} \cdot f_i$$

$T_{\text{nom}}$ – rated torque SFC; $\alpha_i$ – current coefficient reinforcing pressure angle between the rolling body 4 and the socket wall; $r$ – radius of the circle on which the rolling bodies are located 4; $c_0$ – axial spring stiffness 9; $x_i$ – current coefficient reinforcing abscissa of the contact point between the rolling body 4 and the wall of the socket; $f_i$ – coefficient of friction between the pressure plate 7 and the guide key 8; $d$ – diameter of the central hole of the pressure plate 7.

Substituting the right sides of formulas (2) into equation (1), we obtain:

$$\frac{\tan \alpha_i}{r} + \frac{2rf_i}{d} = \frac{2T_{\text{nom}}}{d} \cdot f_i.$$

Integrating function (3), we find:

$$y_i = \frac{c_0r}{2T_{\text{nom}}} x_i^2 + \frac{2rf_i}{d} x_i + C.$$

$C$ – integration constant determined from the initial conditions: $y = y_0, \ x = 0$. $C = y_0, y_0$ – the initial ordinate of the generatrix of the nest, which is determined according to the scheme depicted in Fig. 2. According to the scheme we get:

$$y_0 = \frac{D}{2} \cos \alpha_{\text{min}} = \frac{D}{2\sqrt{1 + \tan^2 \alpha_{\text{min}}}}.$$

Then relation (5) can be written in the following form:

$$y_i = \frac{c_0r}{2T_{\text{nom}}} x_i^2 + \frac{2rf_i}{d} x_i + \frac{D}{2\sqrt{1 + \tan^2 \alpha_{\text{min}}}}.$$

The rated torque SFC is calculated, according to the work, with a minimum value of the coefficient of friction, according to the following formula:

$$T_{\text{nom}} = zF \cdot R \cdot \frac{f_{\text{min}}}{1 + zC_{\text{min}}f_{\text{min}}}.$$
To establish dependence $C(f_i)$ we use the condition according to which the current coefficient is the amplifying value of the torque SFC at any value $f_i$ remains constant and must be equal to the nominal torque of the coupling [4-9].

Equating to each other the right parts of the relation and relations (7), we obtain an equation whose solution has the following form:

$$C_i = \frac{f_i - f_{\min} + zC_{\min}f_i f_{\min}}{zf_{\min} f_i}.$$  
(8)

On the other hand $C_i = \frac{R_p}{r^2} \tan \alpha_i$, therefore, from expression (8) we obtain:

$$\tan \alpha_i = \frac{r(f_i - f_{\min} + zC_{\min}f_i f_{\min})}{R_p zf_{\min} f_i}.$$  
(9)

From relations (2) and (8) we find:

$$x_i = \frac{T_{\min}}{c_0} \left[ \frac{f_i - f_{\min} + zC_{\min}f_i f_{\min}}{R_p zf_{\min} f_i} - \frac{2f_i}{d} \right].$$  
(10)

Substituting (10) in (8), we obtain:

$$y_i = \frac{rT_{\min}}{2c_0} \left[ \frac{f_i - f_{\min} + zC_{\min} f_i f_{\min}}{R_p zf_{\min} f_i} - \frac{2f_i}{d} \right] + \frac{2rT_{\min} f_i}{dc_0} \left[ \frac{f_i - f_{\min} + zC_{\min}f_i f_{\min}}{R_p zf_{\min} f_i} - \frac{2f_i}{d} \right] + \frac{D}{2\sqrt{1 + \tan^2 \alpha_{\min}}}.$$  
(11)

3. Results

Consider the circuit shown in Fig. 3. The diagram shows the process of moving the rolling body 4 with automatic control [10-15].

Position I, II of the rolling body corresponds to the operation of the SFC with the minimum and maximum values of the coefficient of friction. Abscissa axis $x$ passes through the center of the rolling body, the ordinate axis $y$ through the point of amplification of the contact of the side wall of the socket and the rolling body.
In the indicated position 1, the initial ordinate of the contact point is \( y_{\min} \), and position 2 of the rolling body corresponds to the ordinate of the contact point \( y_{\max} \):

\[
\Delta y = y_{\max} - y_{\min}.
\]  

Substituting relations (10) and (9) into equality (12), we obtain:

\[
\begin{align*}
\Delta y &= \frac{r T_{\text{nom}}}{2c_0} \left[ \frac{f_{\max} - f_{\min} + zC_{\min}f_{\max}f_{\min}}{R_{zp}zf_{\min}f_{\max}} - \frac{2f_1}{d} \right] + \\
&+ \frac{2r T_{\text{nom}} f_1}{dc_0} \left[ \frac{f_{\max} - f_{\min} + zC_{\min}f_{\max}f_{\min}}{R_{zp}zf_{\min}f_{\max}} - \frac{2f_1}{d} \right].
\end{align*}
\]

Consider the circuit shown in Fig. 4. The diagram corresponds to the view shown by the arrow in fig. 3. Positions I and II in Fig. 2.11 correspond to similar positions of the rolling body in Fig. 3.

Moving the rolling body from position I to position II (Fig. 3) leads to relative twisting of the SFC half-coupling by an angle \( \varphi_{\max} \) (fig. 4). By producing geometric constructions in fig. 4, we find

\[
\varphi_{\max} = 2 \arcsin \frac{\Delta y}{2r},
\]

and

\[
\Rightarrow \varphi_{\max} = 2 \arcsin \left( \frac{T_{\text{nom}}}{4c_0} \left[ \frac{f_{\max} - f_{\min} + zC_{\min}f_{\max}f_{\min}}{R_{zp}zf_{\min}f_{\max}} - \frac{2f_1}{d} \right] + \frac{T_{\text{nom}} f_1}{dc_0} \left[ \frac{f_{\max} - f_{\min} + zC_{\min}f_{\max}f_{\min}}{R_{zp}zf_{\min}f_{\max}} - \frac{2f_1}{d} \right] \right).
\]

Using the main idea of the energy balance method, we write:
\[ A_i = P_{np} + A_p. \]

\[ A_p = \frac{\Delta T}{2} \phi_{\text{max}}, \]

\[ A_{tr} = 2F_{\text{np}} x_{\text{max}} = \frac{2\Delta T}{d} f_{\text{t}}, x_{\text{max}}, \]

\[ \Delta T \] – magnitude of overload, i.e. maximum increase in external load [16-19].

Using the relations, we find the expression for calculating the overload \( T_{\Delta} \), associated with an increase in the coefficient of friction and the onset of general overload in the drive of the machine:

\[ \Delta T = \frac{4d_{c_0} x_{\text{max}}^2}{d \phi_{\text{max}} - 4 f_{\text{t}} x_{\text{max}}}. \]

The practical accuracy factor SFC can be calculated by the formula:

\[ K_i = \frac{T_{\text{max}}}{T_{\text{min}}} + \frac{\Delta T}{T_{\text{max}}} = 1 + \frac{\Delta T}{T_{\text{max}}}. \]

Obviously, the quantity \( K_i \) will approach unity if the torque decreases \( \Delta T \).

4. Discussion

Since the gain is the magnitude of the torque \( T_{\text{nom}} \) during the calculation and design, the SFC is the initial one and is defined as a function of various operational factors (the rated drive power of the machine, the location of the SFC in the kinematic circuit of the drive, etc.), it is considered constant.

Consider the possibility of decreasing \( \Delta T \) by changing the value \( x_{\text{max}} \). We turn to relation (16).

Decrease value \( x_{\text{t}}, x_{\text{max}} \) possibly due to an increase in axial stiffness \( c_0 \). In relation (19) we obtain that indicates a decrease in torque \( \Delta T \) with increasing \( c_0 \).

\[ \text{Figure 5. Areas of solution.} \]

Dependency Function Charts \( K_i (f), \phi_i (f) \) are shown in fig. 5. When calculating \( \phi_{\text{max}} \) relation (16) is used, in which, in turn, the quantity \( y_{\text{max}} \) was calculated by relation (15), without taking into account the last term in it. In this case, the following initial data are accepted: \( r = 0,03 \text{m}, \ D = 0,016 \text{m}, \ z = 6, \ R_{\text{cp}} = 0,1 \text{m}, \ f_{\text{min}} = 0,2 ,\ f_{\text{max}} = 0,8, \ T_{\text{nom}} =16 \text{Nm} \ u \ F_{\text{n}} =366 \text{N} \).

The graph in fig. 5, the curve is constructed using relation (18). Curve \( \phi_i (f) \) constructed using
relation (18). Angle change $\Phi_i$ directly proportional to the change $f_i$.

Thus, a kinematic scheme for the occurrence of overloads during the operation of the first generation SFC with a variable gain is developed.

The cause of overloads during operation of the SFC with a variable gain is the additional energy costs necessary to move the rolling bodies relative to the side walls of the sockets of the control device.

With an increase in the coefficient of friction $f_i$ (0.2 to 0.8) overload value $\Delta T$ rises from 0 to the maximum value.

The maximum overload during operation of the SFC occurs at the maximum value of the coefficient of friction, if before the increase in the coefficient of friction, the coupling worked at the minimum value of the coefficient of friction.

The SFC accuracy factor increases from 1 to 1.105 as the coefficient of friction increases from minimum to maximum.

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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 mechanischer 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] Stanzoder 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 ergänzen. Blesh InForm. 2009, No. 4, p. 87