The study of the use of multi-disc safety friction clutches in the working bodies of crank presses

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

Abstract. Studies of safety couplings have shown that when the friction forces in splined joints of friction disks and half couplings are taken into account, the moment of friction of the clutch increases compared to the moment calculated without taking these forces into account. Among the safety clutches, friction clutches occupy a special place in use in crank presses due to the increased accuracy of limiting the transmitted load. This allows you to increase the efficiency of protection of parts and drive units of machines, as well as reduce their weight and dimensions.

Friction disks are pressed against each other with the force created by the spring, and the moment of friction arising between them ensures the connection of the leading parts with the followers. The friction force in the slots significantly reduces the transmitted force of pressing on subsequent disks, but its influence is not taken into account in existing studies.

1. Introduction
Studies of safety couplings have shown that when the friction forces in splined joints of friction disks and half couplings are taken into account, the moment of friction of the clutch increases compared to the moment calculated without taking these forces into account.

Among the safety clutches, SFC friction clutches occupy a special place due to the increased accuracy of limiting the transmitted load. This allows you to increase the efficiency of protection of parts and drive units of machines, as well as reduce their weight and dimensions.

Friction disks are pressed against each other with the force created by the spring, and the moment of friction arising between them ensures the connection of the leading parts with the followers. The friction force in the slots significantly reduces the transmitted force of pressing on subsequent disks, but its influence is not taken into account in existing studies [1-5].

Previous studies have shown that when the friction forces in splined joints of friction disks and half couplings are taken into account, the moment of friction of the clutch increases compared to the moment calculated without taking these forces into account.
Consider the multi-disc chart SFC presented in Figure 1.

The coupling consists of coupling halves 1 and 2, a package of friction disks 3 and 4, a thrust disk 5, a pressure disk 6, a control device and a pressure spring 7. The disks 3 are connected to the drum of the coupling half 2, and the disks 4 to the hub of the pressure disk, which is mounted on the hub of the coupling half 1 with the possibility of rotation relative to it. The thrust disk 5 is rigidly mounted on the hub of the coupling half 1.

The control device consists of rolling elements 8, which are placed with a gap in the sockets (Fig. 1, section AA) of the thrust and pressure plates.

When SFC is operating, press forces act on the spring pull force 7, circumferential force and spacer (squeezing) force (Fig. 1, section AA). Force action reduces the force of pressing against each other by friction pairs 3-4. As the friction pairs move away from the support flange of the pressure plate, the action of the spacer force decreases due to the influence of friction forces , .

2. Methods
Consider two cases:
1. Friction in the elements of the connection of the disks with the leading and driven parts is not taken into account.

The moment of friction forces SFC is determined by the formula:

\[ T = zF_p R_p \frac{f}{1 + (z - 1)C_f}. \]  

2. Subject to the consideration of friction in splined joints.

Let us consider successively the transfer of the friction moment from the leading disks to the driven disks, shown in Fig. 1.

The moment of friction forces of the friction group of the clutch is equal to:

\[ T = z R_p f (F_n - F_p), \]  

3. Results
Spacer force is determined by the formula:

\[ F_p = T \tan \alpha, \]  

\[ T_1 \] – torque transmitted by friction pairs that are associated with the pressure plate 6:
Substituting expressions (1) and (2) into equation (3), we obtain:

\[ F_p = \frac{\left(\frac{z-1}{z}\right)F_Cf}{1 + \left(\frac{z-1}{z}\right)Cf} , \]

(5)

Torque \( T_1 \) can be imagined as

\[ T_1 = F_1 R_{cp} f , \]

(6)

\( F_1 \) – normal force acting on the first friction working surface from the support flange of the pressure plate:

\[ F_1 = F_n - F_{t1} , \]

(7)

\( F_{t1} \) – friction force in spline joints of the pressure plate:

\[ F_{t1} = F f = \frac{T_1}{R_H} f , \]

\( f_1 \) – coefficient of friction between the working surfaces of the spline connection; \( R_H \) – outer radius of friction disk 4; \( F \) – circumferential force transmitted by the friction disk:

\[ F = (F_n - F_p) f . \]

The moment of friction, perceived by the second disk (the first driven disk)

\[ T_2 = f_1 R_{cp} (F_1 + F_2) . \]

\( F_2 \), acting on the second pair is determined by the formula

\[ F_2 = F_1 - F_{t2} = F_1 - f \frac{T_2}{R_B} = F_1 - f_1 f \frac{R_{cp}}{R_B} F_1 , \]

(8)

\[ F_2 = F_1 \left( \frac{1 - f_1 f \frac{R_{cp}}{R_B}}{1 + f_1 f \frac{R_{cp}}{R_B}} \right) = F_1 \left( \frac{1 - f_1 f \frac{R_{cp}}{R_B}}{1 + f_1 f \frac{R_{cp}}{R_B}} \right) \]

\( R_B \) – internal radius of the friction disk.

The moment of friction transmitted by the third disk: \( T_3 = (F_2 + F_3) f R_{cp} . \)

\( F_3 \), acting on the third pair is determined by the formula [10-14]

\[ F_3 = F_2 - F_{t3} = F_2 - f_1 \frac{T_3}{R_H} = F_2 - f_1 f (F_2 + F_3) \frac{R_{cp}}{R_H} , \]
Using expressions (8) and (9), we write the spacer force for an arbitrary odd pair $z_1$:

$$ F_{z_1} = F \left[ \frac{1 - f_i f \frac{R_{np}}{R_n}}{\left( 1 + f_i f \frac{R_{np}}{R_n} \right)^2} \right] \left[ \frac{1 - f_i f \frac{R_{np}}{R_n}}{\left( 1 + f_i f \frac{R_{np}}{R_n} \right)^2} \right]. $$

(10)

Accordingly, for an arbitrary even pair $z_2$ spacer force $F_{z_2}$ defined as

$$ F_{z_2} = F \left[ \frac{1 - f_i f \frac{R_{np}}{R_n}}{\left( 1 + f_i f \frac{R_{np}}{R_n} \right)^2} \right] \left[ \frac{1 - f_i f \frac{R_{np}}{R_n}}{\left( 1 + f_i f \frac{R_{np}}{R_n} \right)^2} \right]. $$

(11)

The total moment of friction forces SFC is

$$ T_f = \sum z_T = \sum z_{R_{np}} f(F_n - F_c), $$

(12)

$T_z$ – moment of friction forces of one friction pair.

We write expression (12) in the following form:

$$ T_f = R_{np} f \left[ (F_n - F_1) + (F_n - F_2) + (F_n - F_3) + \ldots + (F_n - F_z) \right]. $$

$$ T_f = R_{np} f \left[ zF_n - (F_1 + F_2 + F_3 + \ldots + F_z) \right] = R_{np} f \left[ zF_n - \left( \sum F_1 + \sum F_z \right) \right]. $$

(13)

In view of relations (11) and (12), expression (13) can be written as

$$ T_f = R_{np} f \left[ \sum_1^{z_1} \left( \frac{1 - f_i f \frac{R_{np}}{R_n}}{\left( 1 + f_i f \frac{R_{np}}{R_n} \right)^2} \right) \right] \left( \frac{1 - f_i f \frac{R_{np}}{R_n}}{\left( 1 + f_i f \frac{R_{np}}{R_n} \right)^2} \right) + \sum_2^{z_2} \left( \frac{1 - f_i f \frac{R_{np}}{R_n}}{\left( 1 + f_i f \frac{R_{np}}{R_n} \right)^2} \right) \left( \frac{1 - f_i f \frac{R_{np}}{R_n}}{\left( 1 + f_i f \frac{R_{np}}{R_n} \right)^2} \right). $$

(14)

We introduce the following notation:
Then expression (12), taking into account formula (11), can be written as

$$T_f = R_p f \left[ z F_n - \frac{f (F_n - F_p)}{1 + ff_1 \frac{R_p}{R_H}} \left( 1 + A + AB + A^2 B + \ldots + A^{\frac{z}{2}} B^\frac{z}{2} \right) \right] =$$

$$= F_n R_p f \left[ z - \frac{f}{1 + (z-1)Cf} \left( 1 + A \right) \left[ 1 - (AB)^\frac{z}{2} \right] \right].$$

(15)

$$K_t$$ calculated by the formula:

$$K_t = \frac{T_{\text{max}}}{T_{\text{min}}}.$$

$$T_{\text{min}}, T_{\text{max}}$$  respectively, the minimum and maximum SFC torques, taking into account friction in splined joints, or, taking into account expression (15) [15-19]:

$$K_t = \frac{f_{\text{max}}}{f_{\text{min}}} \times \left[ z - \frac{f_{\text{max}}}{1 + (z-1)f_{\text{max}} \frac{R_p}{R_H}} \right.\left[ 1 - (A_{\text{max}} B_{\text{max}})^\frac{z}{2} \right] \right] \left[ 1 - A_{\text{max}} B_{\text{max}} \right].$$

(16)

To plot functions $$T_f (z), K_t (z)$$ taking into account friction in spline joints, the following initial data were adopted: $$F_n=1000 \text{ N}; f_{\text{min}}=0.1; f_{\text{max}}=0.8; f_1=0.1; \ f=0.1; \ C=1.25; \ R_p=100 \text{ mm}; \ z=10.$$

The geometric relationships of friction discs are found based on the following reasoning. The outer, middle and inner radii of the friction disk are interconnected by the ratio:

$$\nu = \frac{2(R_H - R_b)}{R_p},$$

(17)

$$\nu$$  – disk width ratio.

The second ratio, connecting the outer, middle and inner radii of the friction disk used in the calculations, has the form:

$$R_p = \frac{R_H + R_b}{2}.$$  

(18)

Solving the system of equations (1) and (4), we find:
\[ R_H = \frac{R_{c_p}(4 + \nu)}{4}, \quad (19) \]
\[ R_B = \frac{R_{c_p}(4 - \nu)}{4}. \quad (20) \]

Usually accepted \( \nu = 0.20 \), then, taking into account relations (18) and (19), we find:
\[ R_H = 0.106 \text{ m}; \quad R_B = 0.094 \text{ m}. \]

![Figure 2. Graphs of the dependence of the torque SFC on the number of friction pairs](image)

4. Discussion

Graphs are constructed using relations (3), (16) and (18). The accuracy factor SFC without taking into account friction in splined joints was calculated by the formula:
\[ K_t = \frac{f_{\text{max}}[1 + (z-1)Cf_{\text{max}}]}{f_{\text{min}}[1 + (z-1)Cf_{\text{max}}]}. \]

Dependency graphs \( T(z) \) are shown in fig. 2. Curve 1 reflects the dependence of the SFC torque excluding friction in splined joints, curve 2 taking into account friction.

![Figure 3. Graphs of the dependence of the accuracy factor SFC on the number of friction pairs](image)

In fig. 3 curve 1 shows the dependence \( K_t(z) \) excluding friction in the splines, curve 2 - taking into account friction.

Analysis of the graphs shows the following:
- SFC torque with regard to friction in splined joints at = 10 is approximately 2 times higher than without friction (Fig. 2);

Accuracies the accuracy coefficient SFC, taking into account friction in splined joints, increases with the number of friction pairs of the friction group, asymptotically approaching the value of the accuracy coefficient of the safety friction clutch without the control device (Fig. 3, curve 2);

- with a minimum number of friction pairs (z=2) indicators of load capacity and SFC response accuracy taking into account friction in spline joints and without taking into account friction are close to each other, which is explained by the minimal influence of friction forces in spline joints.

Thus, the study found that taking into account the frictional forces in the spline joints of friction disks with half couplings in multi-disc SFCs allows one to obtain refined dependences of the torque and accuracy coefficient on the number of friction pairs.

With a large number of friction pairs, the torque of multi-disc SFCs, taking into account friction in spline joints, significantly (two or more times) exceeds the torque without taking into account friction in spline joints.

Accounting for friction in splined joints significantly (up to 4 times) increases the accuracy coefficient of multi-disc SFC with a large number of friction pairs.

Acknowledgements
The study had no sponsorship.

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.

[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.-79с.

[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 Stobbeanspruchungen 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