Principles of improving the smoothness of the working mechanism in forging and stamping machines

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Abstract. The analysis of safety clutches shows that, despite the high rates of scientific and technical progress in the field of mechanical engineering, the development of safety clutch designs is noticeably lagging behind, and their technical and operational characteristics in general do not fully comply with modern requirements for the protection of machine drives. Friction clutches have a high response accuracy, but at present they do not provide the necessary level of protection against overloads. For a certain period of time, these performance characteristics met the necessary requirements for accuracy of operation. However, due to the steady increase in the technical characteristics of machines and mechanisms at present, some technological equipment requires the implementation of technological processes to preserve existing power factors with minimal (up to 7...10%) deviation of the nominal values. Such equipment may include, for example, forging and stamping equipment. To ensure a high degree of constancy of the working forces and moments in the drives of the specified equipment, the use of safety couplings with a significantly higher accuracy of operation within the above limits is required.

1. Introduction

In modern machines and mechanisms, speeds and workloads, all the characteristics associated with the movement and transmission of motion are constantly increasing. Currently, the focus is on the requirements of protection elements and components of the drives from overload. Safety devices are the basis for maintaining the reliability of drive systems. The protective properties of safety devices depend on the reliability of their mechanical systems, determined by the complexity and composition of the structure.

We add that the safety friction clutches are characterized by a significant drawback consisting in the instability of the torque. The main reason for this is the inconstancy of the magnitude of the friction coefficient between the friction pairs.

The effectiveness of the use of safety clutches in the composition of the drives of the machines mainly depends on the installation location of the clutch in the kinematic drive chain, as shown in the studies. The place of installation of the coupling is justified by the protected and unprotected parts of the coupling, which boils down to the special requirements of determining the location.

The safety clutch is characterized by the presence of feedback devices in their composition - negative single-circuit, negative double-circuit, negative-zero, positive-negative or positive. A feedback device comprising a safety clutch is designed to compensate for the increase in the transmitted load with a
random increase in the friction coefficient by reducing the force with which the friction pairs are closed.

Distinctive features safety clutch:

- the ability to calculate and design a safety clutch with a predetermined accuracy of operation;
- the possibility of designing the drive of the machine, taking into account the predetermined rational installation location of the safety coupling;
- minimization of overall dimensions and the total mass of the drive in combination with a high level of protection of parts and drive assemblies from overloads.

The second-generation safety clutch occupies a special place in the classification of safety friction clutches. The second-generation safety clutch has a higher response accuracy due to the inclusion of a non-adaptive friction group. Recently, the safety clutch of the second generation have been developed in the form of various modifications.

Introduction to the design of the safety clutch in the design of the coupling of the feedback leads to a deterioration of the load capacity, which begins to fall sharply with increasing KU feedback.

As protection against overloads of torque, safety adaptive friction clutches are widely used; they perform smooth coupling at various speeds, which are successfully used in mechanical engineering in the construction of automobile coupling. Moreover, the friction clutch cannot transfer a larger moment than the frictional moment, since sliding of the contacting friction elements begins. Therefore, this property allows friction clutches to be effective fuses to protect the machine from dynamic overloads.

These couplings are compact, have the differences of smooth operation during their operation, do not require specialized devices to turn on after operation.

Classifications of safety clutch are based on their differences in the form of friction surfaces, on the design features of the control device and on the degree of angular rigidity.

Classification by the form of friction surfaces:

- disk;
- conical;
- tape.

Classification by design features of the control device:

- spacers - balls;
- spacers - rollers;
- spacers - screw pairs;
- spacers - cams;
  With combined spacer elements;
  With hydraulic feedback.

Classification according to the degree of angular rigidity:

- tough;
- elastic.

According to the classification of M.P. Shishkarev in determining the structural-functional scheme of the adaptive friction clutch and the accessories of this clutch to any section, the main distinguishing feature should be the feedback of the clutch. According to this fact, the safety clutch is divided into 3 generations.

The scheme of the first generation clutch is based on a single friction clutch with negative single-loop feedback, created using a control device.

The limited stability of the torque, the safety clutch of the first generation and the impossibility of raising it without significant modernization led to the emergence of a second-generation adaptive friction clutch, a distinctive and constructive change in which is the introduction of an additional friction group, which significantly improved the accuracy of operation and the load capacity of the couplings. A feature of the second generation safety clutch is the possibility of obtaining equality of the spacer force arising from the SU, and the strength of the closing spring. This equality distinguishes between the modes of operation of the coupling - the mode with negative feedback before, and the mode with positive feedback.
This problem is associated with the inconstancy of the magnitude of the coefficient of friction between the friction pairs.

Studies show domestic and foreign scientists Kragelsky I.V., Akhmatova A.S., Palamarenko A.Z., Afanasyev M.K., Zaporozhchenko R.M., Nagornyak S.G., Khrisanova M.I., Kostetsky B.I., Bowden F.P. and Teybor D., Volkova D.P., Mikhina N.M. and other researchers, the size of the coefficient of friction at rest is interrelated with many variables, the main of which are: the magnitude of the pressure on the surfaces of the friction contact; quality and composition of friction surfaces; the growth of the load in time preceding the actuation of the coupling; friction surface temperature; ambient humidity, etc.

Coupling industry is also widely developed in Europe, in particular, in Germany, England, France.

M.K. Afanasyev considered and proved that the change in the rate of load application changes the value of the friction coefficient of the “Steel-friction material NSF-3” pair (friction coefficient 0.3 in the interval 0.1 ... 1.0). Zaporozhchenko R.M. considers the possibility of nonnegative deviation of the size of the friction coefficient from the average value equal to 300%. To a lesser extent, the size of the friction coefficient depends on the pressure on the friction surfaces: for the steel – friction material NSF-5 friction pair: the friction coefficient increases by 1.21 times with the growth of the contour pressure by 5 times. The effect of temperature on the size of the friction coefficient is heavier: for different pairs, the friction coefficient surely decreases as the temperature rises, the coefficient of friction of other pairs passes the minimum point at temperatures of 200 ... 240°C. In changing the size of the friction coefficient, the temperature index in the clutches does not affect, however, when slipping, the friction coefficient can vary considerably.

Varying the size of the friction coefficient entails a random character and finds the size of the torque scattering field. The wear of various friction surfaces reduces the tension force of the closing spring and reduces the size of the torque, which is determined by another reason for its instability. The error associated with the systematics in this dissertation is not considered.

The deviation of the average radius of the friction surfaces from the calculated value as a result of their geometrical errors and uneven pressure against each other, the inaccuracy of the tension force of the closing spring, the geometric errors that appear during the manufacture of coupling parts, and other factors shift the tuning value of the torque and the field of operation relative to the calculated value, therefore, they can be compensated for during tuning and are not taken into account in further studies.

2. Materials and methods

There are 3 ways to increase the stability of torque in friction clutches.

Method number 1. Selection of material friction pairs

Due to the correct selection of material, the change of the friction coefficient is minimized. The friction pair “STEEL-TEXTOLIT” is insensitive to negative effects on the size of the friction coefficient of such types as pressure, load buildup rate, time of fixed contact of friction surfaces before firing, and the “setting” phenomenon. To reduce the impact on the growth of the friction coefficient of the state of friction surfaces, the quality of their treatment should minimize the effect of the molecular component of the friction force. At the same time, a pair of "STEEL-METAL-CERAMICS" has a dispersion of the magnitude of the coefficient of friction.

Method number 2. The geometric shape of the friction surfaces

M.P. Shishkareva, M.M. Saverina, L.I. Malykh and V.K. Tepinkchieva, disk clutches have a higher accuracy of operation compared with the cone clutches.

Method number 3. Enter negative feedback

Compensation for the scattering of the size of the friction coefficient by a change in the force that excites the friction torque of the coupling by introducing negative feedback. There are two options for implementing this method:

1) Due to the torque part of the safety clutch;
2) Due to the full torque of the safety clutch.

Determine the practical value of the overload in the safety clutch, after the dependency forms and
are found, which theoretically ensures full stabilization of the torque, regardless of the change in the value of the friction coefficient [2,3].

To date, the issues of methodological support for calculating and designing couplings are very limited in the literature, which does not allow us to fully consider them sufficient and necessary for full-fledged use in creating new designs and developing specific design schemes for existing models.

In the works [1-5,7-12,19-24] the influence of the operational performance of safety clutches is shown, on the weight and dimensions of the drives of the machines in which the clutches are installed. In particular, it was shown in [6, 13-18] that the accuracy of the coupling and the place of their installation in the kinematic chains of the drives influence the weight and dimensions of the drives of the machines.

This connection is due to the feature of the coupling, which lies in the fact that their weight and dimensions depend on the smoothness of the course. With increasing smoothness of the mechanism, the overall dimensions and mass of the drive increase. In addition, the overall dimensions and the mass of the coupling depend on the place of their installation in the kinematic chain of the drive [15-21]. The reason for this is the magnitude of the torque acting in the place of installation of the coupling in the drive of the machine. If the drive contains mechanical gears that change the gear ratio at the output compared to the gear input, the torque value will be different, which will affect the overall dimensions and weight of the clutch.

The question of choosing a rational installation location of the coupling in the kinematic chain of the drive is solved taking into account the complexity of the kinematic chain and the tasks assigned to the coupling to protect individual sections of the drive.

It is known that a safety clutch, including the clutch, protects that part of the drive of the machine, which is located between the source of mechanical energy (engine) and the clutch, from overloads. This principle is valid only in the case when the rate of increase of the load on the working body of the machine is greater than or equal to the speed of transmission of the shock wave along the kinematic chain in the direction from the source of the shock wave to the clutch.

At low speeds of increase in the overload on the working body of the machine, which are not comparable in magnitude with the speed of propagation of the shock wave through the kinematic chain of the drive, the safety clutch will protect almost the entire drive of the machine from overload.

In accordance with this, the dynamics of changes in the total mass of the drive elements will be different. Therefore, the first source data for the calculation and design of the coupling is the nature of the overloads that occur on the working body of the machine.

Depending on the nature of the overloads (smooth or quick-impact), the drive area of the machine protected by the coupling and, accordingly, the reduction of the total mass of its constituent elements as a result of increasing the smoothness of the trolley travel mechanism is determined. At the same time, it is possible to determine the relative increase in the mass of the coupling as a result of an increase in the accuracy of its operation.

After finding the optimum response accuracy of the clutch and the corresponding gain value (according to the mathematical model of the clutch), minimizing the mass and size characteristics of the drive, it is necessary to determine the amount of torque transmitted by the clutch. Note that the torque transmitted by the coupling, with a fixed installation location in the drive does not depend on changes in the mass and dimensions of the drive as a result of varying the accuracy of operation.

The determination of the magnitude of the torque transmitted by the clutch is the second source data for the calculation and design.

The third source data is the determination of the optimal value of the gain, at which the minimum weight and dimensions of the drive are provided. The solution to this problem seems to be one of the most labor-intensive stages of calculation, since it requires the availability of information about the initial mass data of all drive elements, as well as about the dependencies between the gain value and the masses of the drive elements.

The magnitude of the gain is calculated by the following general relationship:
\[ C = \frac{R_{av}}{r} \cdot \tan \alpha. \quad (1) \]

Where \( R_{av} \) – average radius of friction surfaces of pairs of friction group (s); \( \alpha \) – pressure angle of sensing elements; \( r \) – radius of the circle on which the sensitive elements are located.

The relation (1) contains the parameter \( R_{av} \), which determines the radial dimensions of the coupling.

Most common attitude \( R_{av}/r \) in clutches is 2.8 ... 3.3, and the final value of the gain is determined by the angle \( \alpha \), which can vary widely (from 15 to 75 ... 80) [10-16].

In addition, the parameter \( R_{av} \), among other parameters, determines the rated load capacity. Therefore, using the formula for calculating the nominal torque of the coupling for the selected model, it is possible, by designating the value of the closing spring force (group of closing springs), to find the value of the parameter \( R_{av} \).

This should be taken into account:
- surface area of friction of one pair of friction group, which is calculated by the ratio:
  \[ S = \frac{F_{sp}}{[q]}, \quad (2) \]
  where \( F_{sp} \) – force of the closing spring (group of closing springs), assigned based on the estimated axial and radial dimensions of the coupling; \([q]\) – permissible pressure on the friction surfaces for the selected combination of materials of the friction pair;
  Estimated number of pairs of friction surfaces, based on the radial dimensions of the coupling.

Based on the result of the calculation by the ratio (2) and the accepted value \( R_{av} \) You can find the basic parameters of friction pairs: \( R_o \) – outer radius of the friction disc, \( R_i \) – inner radius of the friction disk.

To do this, we write the ratio between the parameters \( R_{av} \), \( R_o \), \( R_i \), used in calculations:

\[ R_{av} = \frac{R_o + R_i}{2}. \quad (3) \]

For the specified parameters in the formula (3) there is also the following relationship:

\[ \psi = \frac{b}{2R_{av}}, \quad (4) \]

where \( \psi \) – the ratio of the width of the working surface of the friction disk; \( b \) – the width of the working surface of the friction disk.

The width of the working surface of the friction disk is calculated by the formula:

\[ b = R_o - R_i. \quad (5) \]

Having solved the system of equations (3) and (4), taking into account equality (5), we find:

\[ R_o = (1 + \psi)R_{av}, \quad (6) \]

\[ R_p = (1 - \psi)R_{cp}, \quad (7) \]
Considering that, according to reference data, it is usually accepted $\psi = 0.25$, relations (6) and (7) allow us to determine the radial dimensions of the friction disks and the radial overall dimensions of the coupling as a whole.

Knowing the number of friction pairs of the friction group, the value $R_{\text{av}}$, gain and friction coefficient value $f_{\text{min}}$ (minimum coefficient of friction), as well as the nominal torque, can be determined by the formula corresponding to the model of the coupling to determine the force of the closing spring (group of closing springs).

The second version of the method for calculating and designing the coupling is associated with a structural layout solution for determining the installation location of the coupling in the kinematic chain of the machine drive. The essence of this solution is to find such an installation location of the coupling in the kinematic chain of the drive, which minimizes the total mass of the drive components and parts, including the mass of the coupling.

The peculiarity of the design-layout solution and the calculation of the coupling is that in the presence of mechanical drive-down (gearboxes) or upshift (multipliers) gears in the machine drive, moving the coupling installation site inevitably leads to a change in the torque value acting at the installation site.

This means that the magnitude of the coupling gain is kept constant. In this case, the maximum gain value accepted for the selected clutch model is assumed, ensuring the highest clutch response accuracy and the maximum reduction in the total mass of parts and components that make up the protected part of the drive.

The possibility of varying the gain value in this case should be evaluated based on the study of the influence of the gain value on the dynamics of changes in the total mass of parts and components of the protected part of the drive, as well as the drive of the machine as a whole.

In this regard, the method of calculation and design should include:
- calculation and design of the control device as a whole and its individual elements;
- calculation and design of the main and additional friction groups;
- calculation and design of the pressure unit.

The basis of the calculation is the design of the profile of the side walls of the nests on the basis of the obtained curve equation that outlines the profile. Since the equation includes the values of the axial stiffness of the springs, it is necessary to match them with the diameter of the rolling elements, since these parameters determine the maximum axial movement of the support sleeve. This should take into account the conclusion on the effect on the practical accuracy of the coupling operation on the values of the axial stiffness of the springs.

Minimum pressure angle $\alpha_{\text{mn}}$ must be matched with the initial tension force of the spring, the nominal torque of the coupling and the magnitude of the friction angle in the movable joint “pressure disk - guide key” (to prevent the joint from getting stuck) and be limited from below.

The diameter of the rolling body is selected by the condition of contact strength, taking into account the local load transferred to them.

The dimensions of the nest under the rolling body must strictly correspond to the diameter of the rolling body, both in depth and in minimum cross section.

3. Calculation and design of the control device
Development of elements of the method for calculating and designing a nest should include:
- determination of the depth of the nest - common and working;
- determination of the size of the nest in consecutive cross sections;
- “binding” of the socket in the axial direction on the support sleeve.

Determining the depth of the nest
The basis for determining the depth of the nest is the profile of the side walls. When calculating, it is necessary to take into account two depths of the nest - total and working. Total depth $H_t$ necessary for
the development of nest processing technology, working depth $H_w$ – to “bind” the socket in the axial direction on the support sleeve and determine its total depth. The specified position is characterized by contact in cross-section along the circumference (theoretically) between the nest and the rolling body, with maximum rolling body immersion in the nest.

The dependence of the safety factor of parts and drive units on the accuracy of the safety clutches is established. It is shown that the actual value of the safety factor is inversely proportional to the accuracy coefficient, and the moment of resistance of the weakest link of the machine is proportional to the accuracy coefficient of the safety clutch.

Couplings with variable feedback KU are investigated. It is shown that in static loading mode, the AFM operates at a torque equal to its nominal torque, and couplings with a variable KU feedback require an increase in external load to operate in relation to the nominal torque.

Application of the principle of adaptability in the work of lifting mechanisms, and in the work - the use of feedback in the crane mechanisms. It is shown that the application of the above principles can improve the safety and durability of the lifting mechanisms.

The dynamic processes in drives of machine units with AFM are investigated. As a result, it was found that with a lowering characteristic of the friction force, the system is unstable and, after a minor perturbation, oscillations are self-excited with increasing amplitude oscillations, limited over time due to an increase in the sliding velocity of the bodies. It was concluded that to study the dynamic processes occurring in drives of machine units with AFM, it is advisable to use the method of synchronization of mechanical vibrations and the reduced van der Pol formula.

4. Results
The studies of the operation of the AFM in the drive of the machine, given in [66,71,72], show that the AFM of the second generation have a more severe dynamic limitation of the KU value, which is explained by the presence of a friction group in them, unused feedback, which in the “negative” mode friction “has more than a friction group covered by feedback, a destabilizing effect on the process of mechanical oscillations.

The studies of the basic version of the second-generation AFM showed that their performance could hardly be improved without a significant change in the design, and the possibilities for a significant increase in the accuracy of response were almost exhausted.

The studies have also shown that the pattern of change in the thrust force in real AFM structures of the second generation is far from the required pattern, and therefore it is necessary to create AFM structures with the implementation of the required pattern of change in the QF value.

5. Discussion and conclusion
The main direction of improvement of the second generation AFM with the aim of significantly improving the accuracy of operation is to change the design of the control device to implement the pattern of change in the thrust force depending on the variable magnitude KU and the magnitude of the friction coefficient.

The curve that delineates the side wall of the seat for the rolling body of the control device (CU) provides at each point of contact with the rolling element a pressure angle at which the torque of the safety clutch is theoretically equal to the moment. However, in order for the rolling element to move along the forming wall of the socket, the rolling body must also move along the axis of the support sleeve, which should mean compression of the springs in the coupling.

The increase in the potential energy of the springs, as well as the need to perform a certain work against the friction forces between the pressure disk and the guide key, requires an additional external load to be attached to the safety clutch. This means the occurrence of an overload, despite the theoretical characteristic of the accuracy of the “ideal” safety clutch.

A diagram illustrating the process of the appearance of overloads with different values of the friction coefficient is shown in Figure 1. In the coordinate axes, straight line 1 shows the load characteristic of
the “ideal” safety clutch.

Figure 1. Scheme of action overloads.

In accordance with the definition of the “ideal” safety coupling, straight line 1 is parallel to the abscissa axis.

The overload scheme is shown by curve 2, according to which [1]:
- at the initial value of the friction coefficient, the beginning of curve 2 is at the point belonging to straight line 1 and corresponding to the specified value of the friction coefficient;
- the magnitude of the overload, with an increase in the friction coefficient of the value, depends on the current value of the friction coefficient, and increases with the growth of the latter;
- after moving the rolling body relative to the forming wall of the socket to the position corresponding to the friction coefficient reached, the safety clutch will transmit the nominal torque, as the pressure angle and the gain factor increase, this process is reflected in Figure 1 line segments 3;
- at the initial value of the friction coefficient, different from, the process of the action of overloads in the safety coupling is shown in Figure 1 by curves 4 and straight lines 5.

Consider the scheme shown in Figure 2. The diagram shows the process of moving the rolling element during automatic regulation.

Figure 2. The movement of the rolling body in the nest.

Position I of the rolling element corresponds to the operation of the safety clutch with the minimum value of the friction coefficient. The abscissa axis passes through the center of the rolling body, the ordinate axis through the point of contact of the side wall of the nest and the rolling body.

In the specified position of the rolling element, the initial ordinate of the contact point is equal to.
It is logical to assume that the maximum overload of the safety clutch will occur at the maximum value of the friction coefficient. In this regard, as the second position of the rolling body relative to the profile of the side wall of the socket, we consider position II, corresponding to the maximum value of the friction coefficient. This position corresponds to the ordinate of the point of contact.

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