Research of friction safety multidisk coupling

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Abstract. Safety friction couplings have been applied in mining, industry, construction, agriculture, transportation, and public utilities. They are used in free shafts uncoupling in critical situations when the value of torque exceeds the permissible value under short-term overloads. Compared to other safety couplings such as destructive element couplings (with shear pins), gear clutches (cam, ball), friction couplings transmit much larger torques being relatively small in dimensions. They have a small locking force, and better smooth running characteristics. The advantage of using multidisk couplings is the possibility of increasing the value for the limit torque by increasing the number of disks (due to the increased area of adjacent surfaces). Operation of safety friction multidisk couplings is influenced by various factors, an understudied relationship between the response speed of a friction coupling and the rate of the applied load rise being among them. The paper presents experimental data on a safety friction multidisk coupling operation on the test stand, and that allows one to fill out this gap.

1. Introduction
Safety friction multidisk couplings are used in trenchers for earthworks, facing stone production, cutting slots and trenches for cables and pipelines, in the rotation mechanisms of cranes for crane superstructure rotation, track-laying tractors, in drainage excavation as part of reclamation work, soil puddling, and for protection in high-precision testing devices and stands. The advantages of friction couplings are the transmission of large torques, relatively small dimensions, a small locking force, compactness, and smoothness of operation [1]. The purpose of couplings is to join shafts and transmit torque from a driving shaft to a driven shaft. An essential condition for the operation of couplings is a mechanical power transmission without changing its value. A distinctive feature of safety couplings is protection of a machine from breakage under overload. Prevention of a breakdown is achieved by setting a coupling at a specific torque value referred to as a limit torque. This torque being exceeded, the coupling is actuated and shafts are disconnected. Coupling actuates when coupling disks slip relative to one another. The design of the safety friction disk coupling and its diagram are shown in Figure 1 and Figure 2.

Figure 1. Safety friction disk coupling with components:
a friction disk and steel disk without a lining.
Safety friction multidisk coupling [2] includes a set of steel disks and so-called "clutch" disks, which are usually steel ferodo-lined disks or other frictional materials. The disks are arranged alternatively one after another. A frictional material should meet stringent requirements. First of all, it is a high friction coefficient which determines smooth braking, minimum slipping, and wear resistance of disks. The material should provide a good running-in ability, minimum jamming, and have a high thermal conductivity. Composite materials meet all these requirements being a combination of different materials with different properties. One of these materials is ferodo, which is a friction heat-resistant phenol-formaldehyde resin composite material. It is composed of the heat-resistant filler, reinforcing component (steel, organic or mineral fibers), abrasive additives and friction additives. Sunk rivet fixing of friction disks is widespread, but today glueing of lining has been used due to the increased development of heat-resistant and durable adhesives. Figure 3 shows components of a safety friction disk coupling.
2. Task Description
Operation of safety friction multidisk coupling takes place under frequent short-term overloads and high angular velocities. Work of a coupling is influenced by a variety of factors. Possible adhesion of disks can result in a later slipping of disks (coupling actuation takes place at a torque higher than the calculated one). Lubricant properties of the exposed wear surfaces of disks are changed over time (this changes the value of the friction torque). In operation of a coupling the tension of a pressing spring weakens (instead of the control mechanism, springs are embedded into overload couplings), and friction surfaces of disks wear out (correct interaction of the components against each other is broken, which brings to inaccurate operation). All these factors cause not only quantitative, but also qualitative changes in the operation of couplings [3]. Modeling the process of coupling operation is extremely difficult due to the uncertainty in time and actuating quantity of this or that factor [4]. The use of a design program complex involves excessive computing time, as it requires enumeration of a considerable amount of versions, and this substantially complicates the solution of the problem. In some cases it is less complex to determine experimentally the dependence of the input and output shaft turnover frequency on time, and the dependence of the turnover frequency on the torque (load) allows one to obtain the required data on coupling operation with less complexity. Due to the reasons above the determination of the dependency between the response delay of a friction coupling and the rate of the applied load rise does not have a computational solution. That is why an experimental determination of this dependency is relevant in terms of practice. In future this experimental study can be used for assessment of the computational solution.

3. Theory
A safety friction multidisk coupling is used in devices subjected to frequent short-term overloads. It proves itself be good under shock loads. The design of this coupling is similar to a multidisk coupling without a drive control, with the friction disks being permanently compressed by springs [5]. The springs are adjusted for the transmission of a certain specified torque. Its value being exceeded, the coupling actuates, and the disks slip relative to one another. It should be noted that the contact between disks brings to surface wearing, thus their thickness reduces, and the spring force weakens. This fact defines the necessity for spring condition monitoring and regulation. The safety friction multidisk coupling (a is the design; b is the cutaway design) is shown in Figure 4.

![Figure 4. Safety friction multidisk coupling: a – design; b – cutaway design.](image)

The frictional torque of a coupling

\[ K \cdot T = T_F = F_a \cdot f \cdot r_{mr} \]  

(1)

where \( K \) is the factor of safety; \( K = 1.5...2.0 \); \( f \) is the friction coefficient; \( r_{mr} \) is the mid-radius of working surfaces; \( r_{mr} = (D_1 + D_2)/4 \); \( T \) is the transmitted torque; \( F_a \) is the required half-coupling force.

The required half-coupling force

\[ F_a = K \cdot T / (f \cdot r_{mr}) \]  

(2)

The specific pressure

\[ \rho = 4F_a / (\pi (D_{12} + D_{23})) \leq \rho \]  

(3)
where \([\rho]\) is the permissible specific pressure;

The precision factor of coupling response
\[ K \cdot T = \frac{T_{\text{lim}}}{T_{\text{limmin}}} \]  

(4)

where \(T_{\text{limmax}}\) and \(T_{\text{limmin}}\) are the maximum and minimum values of the limit torque.

The coefficient of the residual torque
\[ k_{\text{sl}} = \frac{T_{rd}}{T_{lm}} \]  

(5)

Where \(T_{rd}\) and \(T_{lm}\) are the mean values of the slip torque and limit torque.

The mean value of the static friction coefficient
\[ f_s = \frac{2T_{lm}}{(F_{lm} \cdot D_{mn} \cdot z)} \]  

(6)

The mean value of the coefficient of sliding friction
\[ f_d = \frac{2T_{rd}}{(F_{lm} \cdot D_{mn} \cdot z)} \]  

(7)

4. Experimental Results

In the experiments temporary dependence of the input and output shaft turnover frequency on time and the dependence of the turnover frequency on the torque (load) were defined. The study of the dependence between the time of delay of a friction coupling operation and the speed of load increase is established. It was found that when the critical moment was reached in 17 S, the delay of the clutch operation was 40 ms, and when the critical moment was reached in 7.2 S, the delay of the clutch operation was 25 ms. To study the operation of safety friction multidisk coupling and its effect on the rotation of an output shaft, a test stand was used, its kinematic configuration being shown in Figure 5. The test stand consists of an enclosed ventilated electric motor of the unified series AIR 1, a worm reducer 2, a powder breaker 3, a brake inner shaft 4, an output outboard shaft 5, and a safety friction multidisk coupling under study 6. The following measurement units were chosen for constructing graphs: torque is in N·m; shafts speed is in rpm; time is in second. The friction coefficient \(f = 0.25\) (ferodo on steel). As far as the pressure between the friction surfaces in the coupling is created by a spring, it was taken into account that this spring should have been adjusted to the transmission of the limit torque.

![Figure 5. Kinematic configuration of test stand: 1 – electric motor; 2 – reducer; 3 – powder breaker; 4 – inner brake shaft; 5 – output outboard shaft; 6 – safety friction multidisk coupling.](image-url)
the adjusted limit torque of this coupling. The limit torque was set by changing the position of the adjustable component on the thread section of the driven shaft. The torque was recorded by inductive sensors. The results of the experiments are presented in Figure 6, Figure 7 (1st series of experiments), and Figure 8, Figure 9 (2nd series of experiments).

**Figure 6.** Dependence of the rotation speed of the driving and driven shafts, and the torque on time and dependence of the response delay of a friction coupling on the applied load rise (the 1st series of experiments).

**Figure 7.** Dependence of the rotation speed of the driving and driven shafts on load (the 1st series of experiments).
Figure 8. Dependence of the rotation speed of the driving and driven shafts, and the torque on time and dependence of the response delay of a friction coupling on the applied load rise (the 2d series of experiments).

Figure 9. Dependence of the rotation speed of the driving and driven shafts on load (the 2d series of experiments).

5. Results and Discussion
1. Under the load rise there is a coincidence of the input shaft rotation speed and the output shaft rotation speed (rotation is transmitted without changes which agrees with the theory of coupling operation). When the limit torque is reached (its value is fixed in the graphs), the safety coupling is actuated. The rotation speed of the input shaft remains the same, the rotation of the output shaft stops (it is defined by the time of coupling response) and resumes automatically under the load reduction.
2. The experiments showed the dependence between the response delay of a safety friction multidisk coupling and the rate of the applied load rise (a response delay of a friction coupling with a decrease of the rate of the applied load rise is observed).

6. Conclusion
The obtained experimental data are used to determine parameters of the safety friction multidisk coupling.
1. The dependence between the time of delay of a friction coupling operation and the speed of load increase is established.
2. It was found for the test friction coupling that when the critical moment was reached in 17 S, the delay of the clutch operation was 40 ms, and when the critical moment was reached in 7.2 S, the delay of the clutch operation was 25 ms.

7. References
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