Vibration damping in torsionally flexible metal clutch for applications in mining machines

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Vibration damping in torsionally flexible metal clutch for applications in mining machines

K Filipowicz

Silesian University of Technology, Faculty of Mining and Geology, 2 Akademicka Street, 44-100 Gliwice, Poland

E-mail: krzysztof.filipowicz@polsl.pl

Abstract. This article presents the theoretical and experienced determination of damping coefficient, which carried out for a new construction torsionally flexible metal clutch, intended for use in mining machines drives. The theoretical methodology for determining the damping factor, includes primarily affected the structural parameters of the clutch. In addition, the concept of construction of hydraulic vibration damper used in the clutch is presented. This allows for the modernization, which helps to make possible shaping one of the basic characteristics of the clutch-additional damping.

1. Introduction

The torsionally flexible metal clutch, the idea of which has been developed in the Department of Mechanization and Robotics of Mining of the Silesian University of Technology in Gliwice, is a new solution that can be applied, inter alia in drives of scraper conveyors (figure 1) [1, 2, 3, 4, 5]. As a result of using this clutch, the dynamic loads of the conveyor drive system will be significantly reduced. This can be a protection against occurring overloads in conveyor working cycles.

The clutch is characterized by specific features of elastic and deadening having primary impact on the work of the drive system by changing course and stabilize the torsional vibrations and loading torque. The result is decreased dynamic interactions between the elements of the drive system, which directly affects increasing of their durability and reliability.

The principle of the torsionally flexible metal clutch is the working torque, which acts on the active side of the clutch directly via the shaft (1) and then is transmitted to the sliding sleeve (2) by using a multi-turn threaded mechanism. The increasing torque value rotate the shaft (1) relative to the sleeve (2) at the same time with the clutch housing (4) and the resulting longitudinal force in the thread mechanism initiates the displacement of sliding sleeve along the shaft axis (clutch axis). The limitation of the displacement of the sleeve only to the sliding one is accomplished by a splined connection (5) made between the sleeve (2) and the clutch housing (4). The sliding displacement of the sliding sleeve causes simultaneous compression of a set of disc springs (3), appropriately matched to the assumed characteristics of the clutch. Compression of the springs induces the inner strength of elastic deformation of this set of springs. This force, in each temporary fixed position of the sliding sleeve, balances the longitudinal force arising in the thread mechanism, which is the result of the external operating torque.

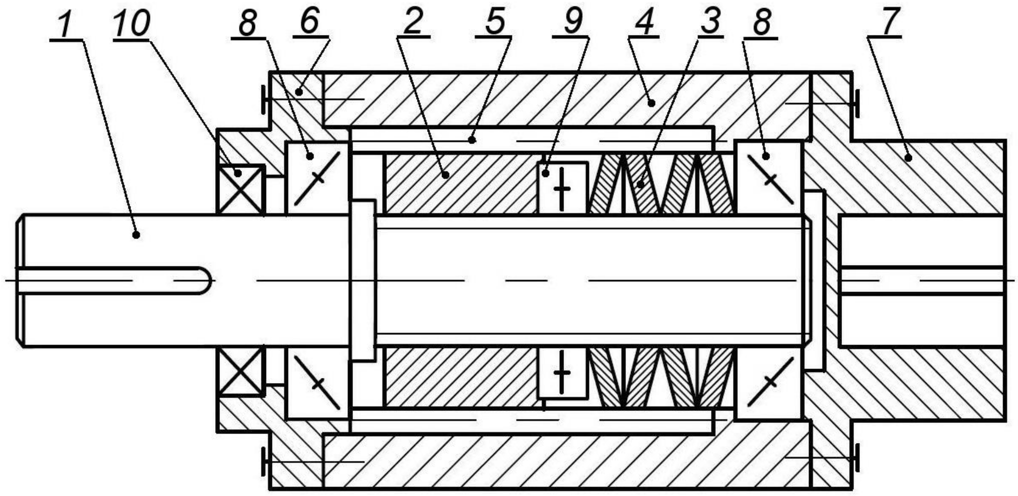

Figure 1. Diagram of one-way torsional flexible clutch [6]: 1 - clutch shaft, 2 - sliding sleeve, 3 - plate spring assembly, 4 - clutch housing, 5 - splined connection, 6 - closure cover, 7 - clutch hub, 8 - tapered bearings, 9 - thrust bearing, 10 - sealing ring.

The equilibrium of forces in the clutch thread mechanism, defined by the temporary fixed position of the sliding sleeve (2) relative to the shaft (1) and clutch housing (4) defines the relative rotation angle of the active and passive clutch elements as well, at which angle the current value of the operating torque is transferred from the active to the passive side of the clutch.

Each momentary overload of the drive with the operating torque results causes additional compression of the elastic elements of the clutch and reduce the burden on their stress relieving.

After the complete relief of the drive system, the sliding sleeve (2) pressed by the relieving set of spring, returns to its initial position, structurally determined in relation to the axis of the clutch shaft.

2. Damping in the clutch mechanism

As mentioned before, the operation of a torsionally flexible metal clutch susceptible is characterized by two basic parameters - stiffness and damping. They are strictly dependent on the structural elements and the construction of the clutch itself. The amount of damping, i.e. the mechanical dissipation energy in the clutch is called damping factor.

The damping factor ψ_{cal} of the clutch is primarily depends on the following design parameters:

- the helix angle γ;
- coefficients of friction in clutch mechanisms;
- the layout and number of springs in the package.

The damping factor can be determined experimentally based on the static characteristics of the clutch (figure 2) using the equation:

$$\psi_{cal} = \frac{A_r}{A_s} \tag{1}$$

where:
A_r is the damping energy, and A_s is the work of elastic deformation.

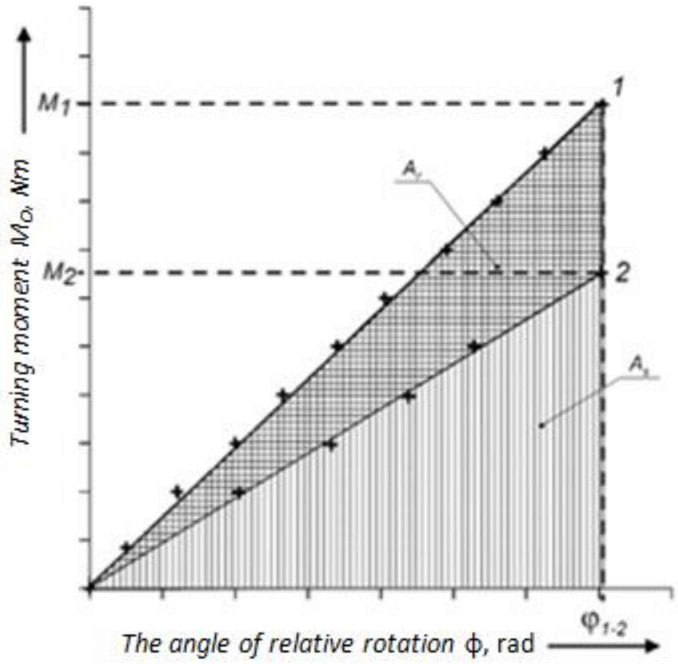

Figure 2. Characteristic of the torsionally flexible metal clutch with the hysteresis loop [6].

The theoretical determination of the damping coefficient considers the occurrence of the hysteresis loop, which is recorded in the form:

$$\psi_{cal} = \frac{\frac{M_1 \cdot \varphi_{1-2}}{2} - \frac{M_2 \cdot \varphi_{1-2}}{2}}{\frac{M_1 \cdot \varphi_{1-2}}{2}} = \frac{M_1 - M_2}{M_1} \quad (2)$$

where: M_1, M_2 are the torques at points 1 and 2 characteristic of the elastic clutch (figure 2), φ_{1-2} is the relative rotation angle which defines the "displacement" of the clutch operation characteristics points from point 1 to point 2. It results from the operation of thread mechanism. Under the influence of the elastic force accumulated in the set of spring, there is a change in the motion of the sliding sleeve, connected with the splined clutch housing, to the rotational movement of the clutch shaft.

M_1 torque depends on a number of construction parameters of the clutch and is determined from equation [3]:

$$M_1 = \frac{\varphi_{1-2} \cdot c_{set} \cdot d_s^2 \cdot tg\gamma \cdot tg(\gamma + \rho')}{3{,}88 \cdot \left[1 - \mu_M \cdot (n \ \ -1) - \mu_R\right]} \quad (3)$$

where: c_{set} is the stiffness of the springs in the set, d_s is the working diameter of the thread, γ is the helix angle, ρ' is the apparent friction angle in the thread mechanism, μ_M is the coefficient of friction in the splined connection, n_{spr} is the number of springs in the package, μ_R is the coefficient of friction in a set of disc springs.

Figure 3 shows the arrangement of forces operating in the clutch mechanism, starting from point 0 to point 1.

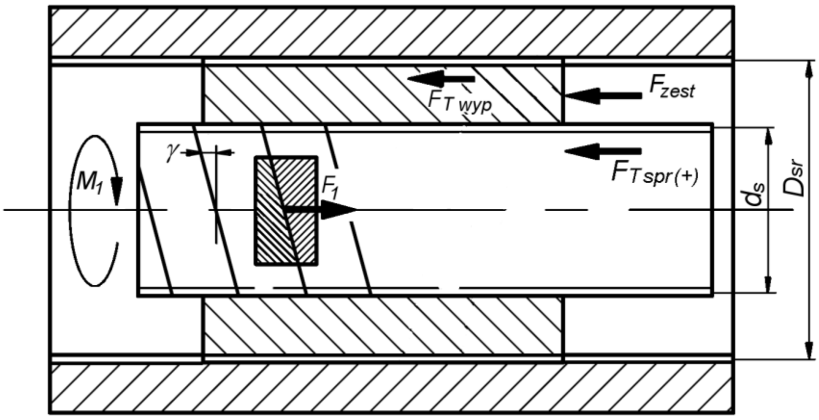

Figure 3. The forces appearing in time of work the clutch from point 0 to point 1, where D_{sr} the average spline diameter, F_1 axial force in the clutch thread mechanism (resulting from impact M_1), $F_{T\,wyp}$ friction force in splined connection, F_{zest} elastic force in a set of springs, $F_{T\,spr}$ generalized friction force in a set of springs [6].

The M_2 torque corresponds to point 2 of the characteristic (figure 2) can be determined from the equation:

$$M_2 = \frac{\varphi_{1-2} \cdot c_{set} \cdot d_s^2 \cdot tg\gamma \cdot tg(\gamma - \rho') \cdot \left[0{,}03 + \frac{1}{1 + \mu_M \cdot (n_{spr} - 1) + \mu_R} \right]}{4} \tag{4}$$

Figure 4 shows the forces arrangement acting in the clutch mechanism, starting from point 2 to point 0.

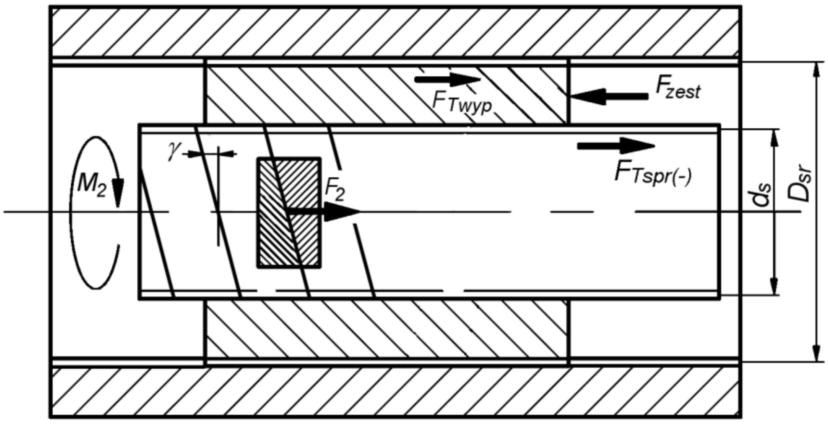

Figure 4. The forces appearing in time of work the clutch from point *2* to point *0,* where F_2 axial force in the clutch thread mechanism (resulting from impact M_2) [6].

Substituting equations (3) and (4) to equation (2), a calculated clutch damping factor is finally obtained ψ_{cal}:

$$\psi_{cal} = \frac{tg(\gamma+\rho') - 0{,}97 \cdot tg(\gamma-\rho') \cdot \left[0{,}03 + \frac{1}{1+\mu_M \cdot (n_{spr}-1) + \mu_R} \right] \cdot \left[1 - \mu_M \cdot (n_{spr}-1) - \mu_R \right]}{tg(\gamma+\rho')} \tag{4}$$

The apparent coefficient of friction ρ' is determined by the equation:

$$\rho' = arc\ tg \frac{\mu}{\cos\alpha_{rn}} \tag{5}$$

where: μ is the sliding friction coefficient between the bolt and the nut, and α_{rn} is the angle of the side inclination of the profile in the normal plane.

3. Hydraulic vibration damper in torsionally flexible metal clutch

The construction of the torsionally flexible metal clutch allows via a simple modification of its construction to the use of an additional hydraulic vibration damper [7, 8]. This solution allows to insert additional damping into the system, expressed as a drag damper coefficient of the hydraulic clutch ψ_{dd}, which allows adjusting the clutch characteristics to the dynamic nature of the load on the machine's drive system. Figure 5 shows the construction of a hydraulic vibration damper, integrated with a torsionally flexible metal clutch (in the two-way operation).

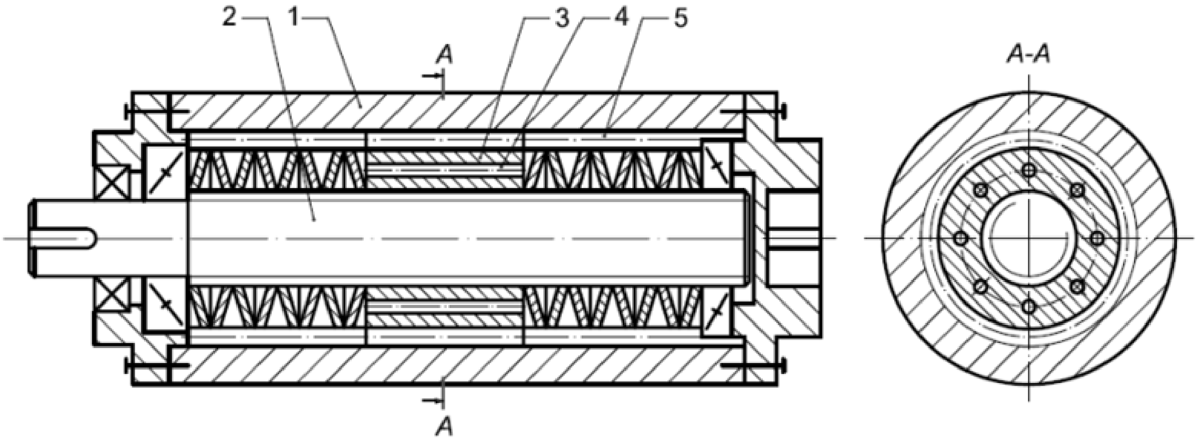

Figure 5. Construction of two-way torsionally flexible metal clutch with a hydraulic vibration damper, where: 1- housing, 2 - screw, 3 - movable element (nut), 4 - holes, 5 - sliding shape connection [7].

Figure 6 shows the operation principle of the hydraulic vibration damper. The displacement of the moving element along the axis of the clutch shaft (one way), is caused by the acting longitudinal force created in the thread mechanism, that derives from the torque M_{ot} (output torque), and in the other side of the force accumulated in the compressed spring set. Each of these displacements is counteracted by friction forces which comes from the thread mechanism, splined connection and friction in the spring set, which is indicated in the second point.

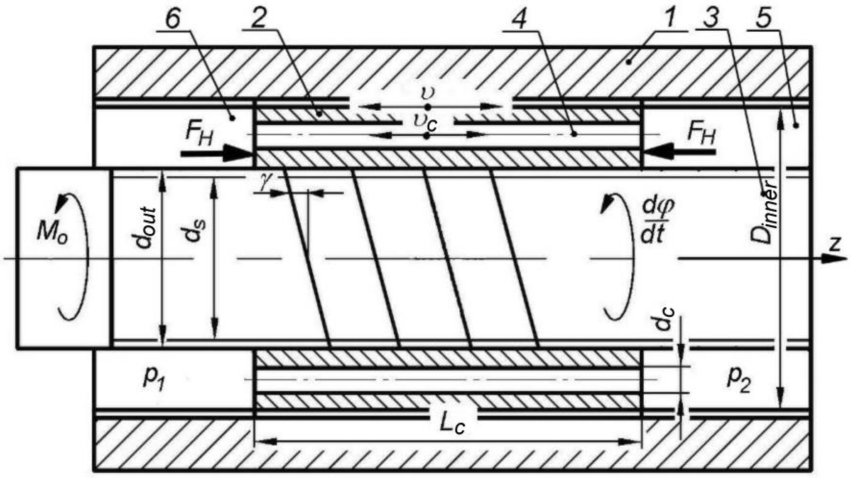

Figure 6. Operation principle of the hydraulic vibration damper in the torsionally flexible metal clutch, where: 1 - housing, 2 - movable element (nut), 3 - screw, 4 - holes, 5, 6 - clutch internal chambers [9].

The introduction of additional hydraulic damping results in an action on the movable element (2) of the additional hydraulic resistance force F_H. This outcome from the pressure difference p_1 - p_2 in the clutch chambers on both sides of the movable element. The value of this force is influenced by the resistance of the working fluid flow through the canals (4) and the moving velocity of the movable element.

The idea of the construction and operation of the damping system in the clutch consists of:

- filling the internal chambers of the clutch (5) and (6), on both sides of the movable element (2) with the operation fluid, e.g. hydraulic oil of known kinematic viscosity ν and specific gravity γ_H;
- making canals (4) in a movable element (2) of specified length L_c, calibrated diameter d_c and N_c number.

During the operation of the clutch, when the operation displacement of the movable element - nuts (2) occurs, exists damping caused by the displacement resistance of the movable element in the environment of the operation fluid flowing through the canals (4).

The force of the hydraulic resistance F_H in a typical hydraulic damper counteracts the displacements of the movable element-nut, in proportion to the velocity υ of these displacements, according the equation [9]:

$$F_H = \psi_{dd} \cdot \upsilon = \psi_{dd} \cdot \frac{dz}{dt} \tag{6}$$

where: ψ_{dd} is the drag coefficient of the hydraulic damper N·s/m, υ and dz/dt is the velocity of displacement of the movable clutch element m/s.

The linear velocity υ of the displacement of the movable element depends on the construction parameters of the clutch, i.e. from the pitch P_h of the thread mechanism, the pitch diameter d_s and the helix angle γ, and is determined according the equation:

$$\upsilon = \frac{dz}{dt} = \frac{P_h}{2\pi} \cdot \frac{d\varphi}{dt} = \frac{d_s \cdot tg\gamma}{2} \cdot \frac{d\varphi}{dt} \tag{7}$$

where:
φ is the angle of relative rotation of the clutch members in rad.

After substituting the values of the angle φ in radians within the equation (7), we get a dependency, indicating the speed of displacement of the moving element υ inside the clutch:

$$\upsilon = \frac{dz}{dt} = 0{,}5 \cdot d_s \cdot tg\gamma \cdot \frac{d\varphi}{dt} \tag{8}$$

After substituting the equation (8) with the equation (6), we receives an equation determining the value of the hydraulic resistance force F_H, which takes into account the construction parameters of the clutch and the angular velocity $d\varphi/dt$ occurring during the relative rotation of the clutch members:

$$F_H = \psi_{dd} \cdot \upsilon = \psi_{dd} \cdot 0{,}5 \cdot d_s \cdot tg\gamma \cdot \frac{d\varphi}{dt} \tag{9}$$

The displacement of the movable element inside the clutch filled with the fluid is accompanied by the flow of operation fluid between the existing spaces on both sides of the movable element through the canals.

From the condition of the continuity of the flow movement follows:

$$Q = N_c \cdot \upsilon_c \cdot \frac{\pi \cdot d_c^2}{4} = \upsilon \cdot \frac{\pi}{4} \cdot \left(D_{inner}^2 - d_{out}^2\right) \tag{10}$$

where: Q is the volume strain of the fluid in m^3/s, υ_c is the flow velocity of the operation fluid in the connecting canals in m/s, D_{inner} is the inner diameter of the clutch housing in m, d_{out} is the external diameter of the screw mechanism thread in m.
After considering the equation (7), the equation (10) will be:

$$Q = N_c \cdot \upsilon_c \cdot \frac{\pi \cdot d_c^2}{4} = \frac{\pi}{4} \cdot \left(D_{inner}^2 - d_{out}^2\right) \cdot 0{,}5 \cdot d_s \cdot tg\gamma \cdot \frac{d\varphi}{dt} \tag{11}$$

Assuming that in the canals the fluid moves in the stratified motion, using the Hagen law [9], we obtain:

$$Q = \frac{\pi \cdot (p_1 - p_2) \cdot d_c^4 \cdot N_c}{128 \cdot \rho_H \cdot L_c \cdot \nu} = \frac{\pi}{8} \cdot \left(D_{inner}^2 - d_{out}^2\right) \cdot d_s \cdot tg\gamma \cdot \frac{d\varphi}{dt} \tag{12}$$

where: p_1, p_2 are the pressure in the spaces on both sides of the movable element in Pa, ρ_H is the density of the operation fluid in kg/m^3, ν is the kinematic viscosity of the spray fluid in m^2/s.

The force of hydraulic resistance F_H, acting during displacement of the movable element, can be determined from the equation considering the pressure difference p_1 and p_2, prevailing in operation spaces on both sides of the movable element of the clutch, namely:

$$F_H = \frac{\pi}{4} \cdot (p_1 - p_2) \cdot \left(D_{inner}^2 - d_{out}^2\right) \tag{13}$$

The pressure difference is determined from the equation:

$$(p_1 - p_2) = \frac{16 \cdot \rho_H \cdot L_c \cdot \nu \cdot \left(D_{inner}^2 - d_{out}^2\right)}{N_c \cdot d_c^4} \cdot d_s \cdot tg\gamma \cdot \frac{d\varphi}{dt} \tag{14}$$

By substituting equation (14) to equation (13), we obtain the dependency specifying the value of the hydraulic resistance force F_H, acting during the displacement of the movable element in a torsionally flexible metal clutch:

$$F_H = \frac{4 \cdot \pi \cdot \rho_H \cdot L_c \cdot \nu \cdot \left(D_{inner}^2 - d_{out}^2\right)^2}{N_c \cdot d_c^4} \cdot d_s \cdot tg\gamma \cdot \frac{d\varphi}{dt} \tag{15}$$

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Using the equations (9) and (15), the equation specifying the value of the drag coefficient of the hydraulic damper ψ_{dd} in a metal torsionally flexible metal clutch will be:

$$\psi_{dd} = \frac{8 \cdot \pi \cdot \rho_H \cdot L_c \cdot \nu \cdot \left(D_{inner}^2 - d_{out}^2\right)^2}{N_c \cdot d_c^4} \tag{16}$$

Based on the equation (16), the drag coefficient of the hydraulic damper ψ_{dd} depends directly on the properties of the operation fluid expressed in density ρ_H and kinematic viscosity ν, and the geometric parameters of the clutch, i.e the length of the connecting canals L_c and the internal dimensions of the D_{inner} housing and the external diameter of the d_{out} thread on the input shaft on the clutch. On the other hand, its value is inversely proportional to the number of the connecting canals N_c and the diameter d_c of the connecting canals of the operation spaces of the clutch.

Modification of the above constructional parameters allows for a sufficiently wide selection of the properties of the hydraulic vibration damper in a metal torsionally flexible metal clutch.

4. Summary

In the process of mitigating dynamic changes occurring in the propulsion system of mining machines, the dispersion of mechanical energy plays a significant role. The task is mainly filled by torsionally flexible metal clutches.

In a torsionally flexible metal clutch, energy dissipation takes place through friction in moving joints and through structural friction in a set of springs. Therefore, it was necessary to determine the damping factor for this new construction.

The value of the damping factor can be determined experimentally. For this purpose, a test should be carried out to determine the elastic characteristics of the clutch. The damping factor can also be determined theoretically. Its value depends on such design parameters of the clutch as: the value of the helix angle of the thread mechanism, the system and the number of springs in the package, and coefficients of friction in the thread association and in the set of disc springs.

In the construction of the clutch, adding an additional damper allows to fit the characteristics of the clutch to the dynamic nature of the mining machine`s drive system loads. The modification of the construction parameters presented in this study allows for a sufficiently wide selection of the properties of the hydraulic vibration damper in a torsionally flexible metal clutch, intended for use in mining machines drives.

5. References

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