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Overrunning Clutches in Designs of Inertial Continuously Variable Transmissions

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Abstract— **The inertial continuously variable transmissions are transmissions of mechanical type. They have a number of advantages in comparison with other types of transmissions. For example, they have a high coefficient of efficiency, since the principle of their action does not imply the need to convert energy from one type to another one. These transmissions have a compact design, a wide range of torque transformation. They can operate in direct mode, smoothing the torsional vibrations in the system. At the moment when operating element is stopped, the input transmission shaft continues to rotate, that prevents the engine from overloading. There are other advantages. But despite these advantages, the inertial transmissions are not widely used in the automotive industry. The main reason for this is the inadequate durability of the freewheel mechanisms involved in the designs of the inertial transmissions. The paper considers ways to improve the efficiency and durability of the inertial transmissions. Dynamics of the inertial transmissions is considered. For this purpose, physical and mathematical models of the inertial transmissions have been developed. The transmissions have a variable structure, nevertheless, new mathematical methods have been developed in the paper, which made it possible to describe the dynamics of the transmissions in the form of only one system of differential equations. On the basis of the developed mathematical model, periodic solutions of the system are constructed, and the external characteristic of the inertial transmissions is obtained. It is shown that the obtained external characteristic is close to ideal one. To increase the reliability and durability of the inertial transmissions, a new design of the free-wheeling mechanisms is proposed, which is distinguished by increased capacity of operating. The proposed designs have a new principle of activity. It is shown that the load on the details of the developed freewheel mechanisms can be reduced in several times in comparison with the existing designs of freewheel mechanisms. A mathematical model of the proposed designs of the freewheel mechanisms is constructed in this paper. Periodic solutions of systems of differential equations that describe the dynamics of the freewheel mechanisms with the new principle of action are constructed. It is shown that the application of the proposed designs allows increasing significantly the durability of the inertial transmissions. In this paper computer simulation was carried out, which confirmed the correctness of the results of the theoretical studies.**

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Index Terms— **Continuously variable transmissions, overrunning clutches, dynamics, modeling.**

I. Introduction

THE inertial continuously variable transmission is a mechanical transmission which is based on the principle of inertia [1]. This transmission has a lot of advantages: compactness, minimum friction losses, high efficiency, and wide range of the torque transformation. It does not need any conventional friction clutches. This transmission protects the engine from overload when output shaft is stopped. This drive guarantees optimum conditions of work for the engine regardless of the changing of load, and smoothly changes output speed with change the load. However, despite of these advantages, the inertial continuously variable transmissions are not widely used in the automotive industry. The main reason for this is that in the design of the transmissions there are two overrunning clutches which are characterized by low reliability under severe operating conditions: high dynamic loads and high on and off power frequency.

The purpose of the paper is the increase the load ability of the continuously variable transmission by reducing the number of the overrunning clutches.

II. The Physical Model of the Continuously Variable Transmission

In general, the inertial continuously variable transmission contains the pulsed mechanism with unbalanced inertial elements, for example, planetary gear with unbalanced satellites, and two overrunning clutches. The main purpose of the pulsed mechanism is to create alternative-sign impulses of the torque. One of the overrunning clutches (the output overrunning clutch) transmits direct impulses of the torque to the output shaft, the other one (the body overrunning clutch) transmits the reverse impulse of the torque on the body of the transmission. The presence of two overrunning clutches determines the possible way for increasing of the reliability of the transmission by reducing the number of the overrunning clutches. For example, the well-known Hobbs' inertial transmission [1] has only the body overrunning clutch.

In this paper, the design of the inertial continuously variable transmission without the body overrunning clutch is suggested. In the design solution, the output shaft of the pulsed mechanism is connected with the body of transmission, not through the overrunning clutch, but through an elastic element in the circumferential direction. A spring or a torsion shaft can be used as elastic element. The scheme of such a transmission is shown in Fig.1.

Here, 1 is the drive shaft of the transmission, 2 are unbalanced loading elements, 3 is the output shaft (the reactor) of the pulsed mechanism, 4 is the elastic element, 5 is the output overrunning clutch, and 6 is the output shaft of the transmission. The elastic link is shown schematically and this link determines the connection of the output shaft of the pulsed mechanism with the body of the transmission, not in the radial direction, but in circumferential one.

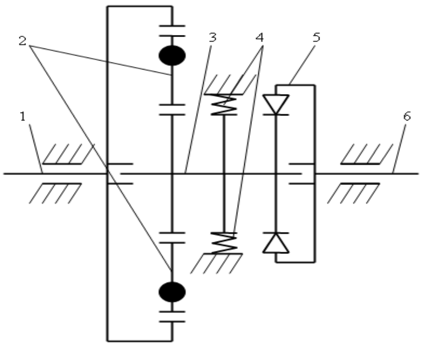

Fig. 1. Kinematic scheme of the inertial continuously variable transmission with high load ability

The principle of operation of the proposed transmission is as follows.

While the driving shaft 1 is rotating, the unbalanced loading elements 2 provide alternative-sign impulses of the torque acting on the output shaft 3 of the pulsed mechanism. Because the output shaft 3 is connected with the body of the transmission by means of the elastic link 4, the shaft performs alternating vibration. The elastic link accumulates the potential energy during the reverse impulse of the torque and transfers the energy during direct rotation of the shaft 3 on the output shaft 6. Such kind of the design of the inertial transmission allows using only one overrunning clutch, such as in Hobbs' transmission. In contrast to Hobbs' transmission, the suggested design provides high equability of the rotation of the driven shaft 6. In this case, the output shaft 3 of the pulsed mechanism is not rigid connected with the driven shaft 6 of the transmission. Transfer of the motion is occurred through the output overrunning clutch. The driven shaft 6 of the transmission has the ability for independent rotation.

Fig.2 illustrates the pulsed mechanism.

Average values of the torque acting on the driven shaft in the usual rigid design of the inertial transmission and the suggested transmission with the elastic link are defined by the expressions, respectively

$$M_{cp} = A\int_0^{\pi} \sin x dx = 2A; \quad M_{cp} = B\int_0^{2\pi} (1+\sin x)dx = 2\pi B,$$

where, A and B are coefficients depending on the parameters of the pulsed mechanism. Matching the average values, we obtain $B = A/\pi$. It is easy to determine that the maximum value of the torque acting on the driven shaft of the transmission in the second case decreases in $\pi/2$ times.

The above example is illustrated by Figure 3. Here, the thick line shows the graph of the torque acting on the driven shaft in the usual rigid design of the inertial transmission; the thin line is for the suggested transmission.

Fig. 2. Pulsed mechanism

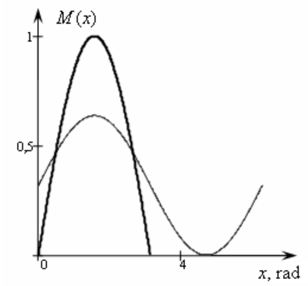

Fig. 3. Graphs of torques acting on the driven shaft

Moreover, if there are two pulsed mechanisms in the suggested design of the inertial transmission, we can have the reduction of the maximum value of the torque in π times. In this case, the value of the torque acting on the driven shaft of the transmission is constant. A similar ability to stabilize the torque in the pulsed continuously variable drive is described in [2].

III. The Mathematical Model of the Continuously Variable Transmission

The mathematical model of the inertial continuously variable transmission is based on Lagrange equations of the second kind.

For the transmission we consider the following sections: 1) the section of the separated motion of the output shaft of the pulsed mechanism and the driven shaft of the transmission; 2) the section of their joint motion. In each section the motion of the transmission is described by means of its own systems of differential equations. Besides, while we construct the differential equations, the dynamic characteristic of the drive engine [3] is taken into account.

As the generalized coordinates, we take the angle α of rotation of the driving shaft 1, the angle β of rotation of the output shaft 2 of the pulsed mechanism, and the angle γ of the driven shaft 6.

In the section of the separated motion of the output shaft of the pulsed mechanism and the driven shaft of the transmission, the system of differential equations takes the form

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$$\begin{cases} A_1\ddot{\alpha} + A_2\ddot{\beta} + A_4(\dot{\alpha} - \dot{\beta})^2 + A_6\dot{\beta}^2 = M_D, \\ A_2\ddot{\alpha} + A_3\ddot{\beta} + A_5(\dot{\alpha} - \dot{\beta})^2 - A_6\dot{\alpha}^2 + c\beta = 0, \\ J_4\ddot{\gamma} = -M_C, \\ M_D = M_H - T\cdot\dot{M}_Д - \dfrac{1}{\nu\cdot\omega_X}\cdot(\dot{\alpha} - \omega_H), \end{cases} \quad (1)$$

where,

$$A_1 = J_1 + nma^2 + \left(\frac{a}{k}+q\right)^2 nJ_3 + 2nmah\left(\frac{a}{k}+q\right)\cos\psi,$$
$$A_2 = nmab + \left(\frac{a}{k}+q\right)\left(\frac{b}{k}-q\right)nJ_3 + nmh\left(\frac{2ab}{k}+(b-a)q\right)\cos\psi,$$
$$A_3 = J_2 + nmb^2 + \left(\frac{b}{k}+q\right)^2 nJ_3 + 2nmbh\left(\frac{b}{k}-q\right)\cos\psi,$$
$$A_4 = -nmah\left(\frac{a}{k}+q\right)q\sin\psi,$$
$$A_5 = nmbh\left(\frac{b}{k}-q\right)q\sin\psi,$$
$$A_6 = nmkhq\sin\psi,$$
$$\psi = q(\alpha-\beta),$$
$$k = a+b,$$

J_1, J_2, J_4 are the moments of inertia of the driving shaft, the output shaft of the pulsed mechanism, and the driven shaft of the transmission, respectively,
nJ_3 is the total moment of inertia of the unbalanced elements relative to the geometric center,
nm is the total mass of the unbalanced elements,
h is the distance between the geometric center and the center of mass of the unbalanced elements,
a, b, q are parameters of the pulsed mechanism,
c is the angular stiffness of the elastic link,
M_C is the drag torque acting on the driven shaft of the transmission,
M_D is the engine torque acting on the drive shaft,
M_H is the rated torque of the engine,
ω_X, ω_H are the angular ideal idling and nominal velocities, respectively,
T is the electromagnetic constant of time of the engine,
ν is the slope coefficient of the static characteristic.

In the section of the joint motion of the output shaft of the pulsed mechanism and the driven shaft of the transmission, the system of differential equations takes the form

$$\begin{cases} A_1\ddot{\alpha} + A_2\ddot{\beta} + A_4(\dot{\alpha} - \dot{\beta})^2 + A_6\dot{\beta}^2 = M_D, \\ A_2\ddot{\alpha} + (A_3+J_4)\ddot{\beta} + A_5(\dot{\alpha} - \dot{\beta})^2 - A_6\dot{\alpha}^2 + c\beta = -M_C, \\ \ddot{\gamma} = \ddot{\beta}, \\ M_D = M_H - T\cdot\dot{M}_Д - \dfrac{1}{\nu\cdot\omega_X}\cdot(\dot{\alpha} - \omega_H). \end{cases} \quad (2)$$

We assume that the speed of the driving shaft is constant $\dot{\alpha} = \omega \equiv const$ [4]. This is acceptable because the moment of inertia of the driving component is usually much greater than the moments of inertia of the inertial transmission's other components. The driving component acts as a flywheel stabilizing its speed.

Assuming constant speed of the driving shaft, we may simplify the mathematical model of the transmission and reduce the order of the systems of differential equations. The equations will take the forms (3) and (4), respectively.

$$\begin{cases} A_3\ddot{\beta} + A_5(\omega - \dot{\beta})^2 - A_6\omega^2 + c\beta = 0, \\ J_4\ddot{\gamma} = -M_C. \end{cases} \quad (3)$$

$$\begin{cases} (A_3+J_4)\ddot{\beta} + A_5(\omega - \dot{\beta})^2 - A_6\omega^2 + c\beta = -M_C, \\ \ddot{\gamma} = \ddot{\beta}. \end{cases} \quad (4)$$

Using Heaviside function, we can write the systems (3) and (4) in the form of one system [5], as follows

$$\begin{cases} (A_3 + J_4\cdot(1-\Phi(\dot{\gamma}-\dot{\beta}))\ddot{\beta} + A_5(\omega-\dot{\beta})^2 - A_6\omega^2 + c\beta = \\ = -M_C\cdot(1-\Phi(\dot{\gamma}-\dot{\beta})), \\ J_4\ddot{\gamma} = -M_C\cdot\Phi(\dot{\gamma}-\dot{\beta}) + J_4\ddot{\beta}\cdot(1-\Phi(\dot{\gamma}-\dot{\beta})). \end{cases} \quad (5)$$

IV. Solution of the Mathematical Model of the Continuously Variable Transmission

As it was mentioned above, the inertial transmission is a system of variable structure [6-9]. A study of the dynamics of such systems is usually difficult [10-12]. The study assumes the consideration of the sections separately with further "matching" solutions for these sections. The representation of the mathematical model in the form of only one system (5) allows not caring about tracking movement from one section to another one. It is need only specification of the initial conditions. In this case, the procedure of the study of the dynamics of the transmission is greatly simplified. In addition, the Heaviside function can be approximated by analytic functions [6]. The mathematical model of the transmission in the form of the single system (5) can be used in a study of periodic solutions and their stability on the basis of analytical methods [13-16].

The numerical solution of the system (5) was received by using the Runge-Kutta method in MathCAD software. The transmission parameters were taken as follows:

$J_1 = 2$ кг$\cdot$м^2, $J_2 = 0{,}5$ кг$\cdot$м^2, $nJ_3 = 0{,}25$ кг$\cdot$м^2, $J_4 = 4$ кг$\cdot$м^2,
$nm = 5$ кг, $a = 0{,}02$ м, b $= 0{,}08$ м, k $= 0{,}1$ м, h $= 0{,}083$ м,
q $= 4/3$, $\mathrm{M_C} = 60$ Н$\cdot$м, $\omega = 150$ рад/с, $c = 500$ Н$\cdot$м.

The plots of the velocity of the output shaft of the pulsed mechanism (thin line) and the driven shaft (thickened line) depending on time are shown in Fig.4. As

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we can see, the driven shaft of the transmission quickly goes to the steady state of the motion from the initial conditions with a little ripple of the rotation. It confirms the advantages of the suggested transmission in compare with Hobbes' transmission.

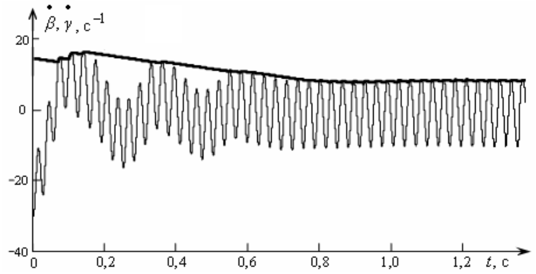

Fig. 4. Graphs of the velocities of the output shaft of the pulsed mechanism and the driven shaft of the transmission depending on the time

Fig.5 depicts a trajectory of the motion of the output shaft of the pulsed mechanism to the steady state in the periodic solution.

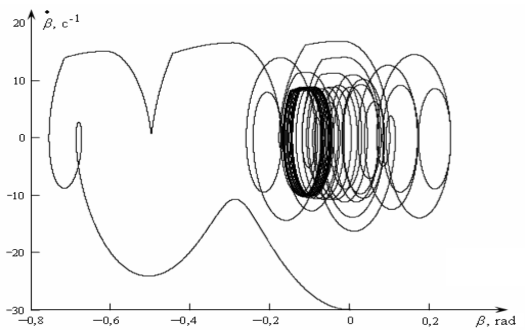

Fig. 5. The trajectory of the output shaft of the pulsed mechanism in phase space

One of the advantages of the traditional inertial continuously variable transmission is that it can work in the mode of a dynamic clutch. The suggested transmission also can work in this mode. For this purpose, it is sufficient to provide a constructive controlled connection of the elastic links to the body of the transmission.

Similarly, we can get the periodical solutions for other modes (Fig.6).

In the Fig.7 there is the external characteristic of the inertial continuously variable transmission.

As we can see from the graph shown in the Fig.7, the external characteristic of the inertial transmission almost meets the ideal characteristic, which once again underlines the prospects of using of the inertial transmission in machinery for various purposes.

V. Principle of Action of the Overrunning Clutch

Overrunning clutches transmit rotary motion in only one direction [17–19]. They are widely used, for example, in hydraulic transformers, pulsed continuous transmissions, inertial automatic torque transformers, electrical starters for motors, metal- and wood-working drives, and so on.

Unfortunately, existing overrunning clutches are insufficiently reliable and durable and in many cases limit the reliability of the drive as a whole. Thus, the insufficient life of overrunning clutches delays the use of inertial automatic continuous transmissions, which have many benefits over existing transmissions [20].

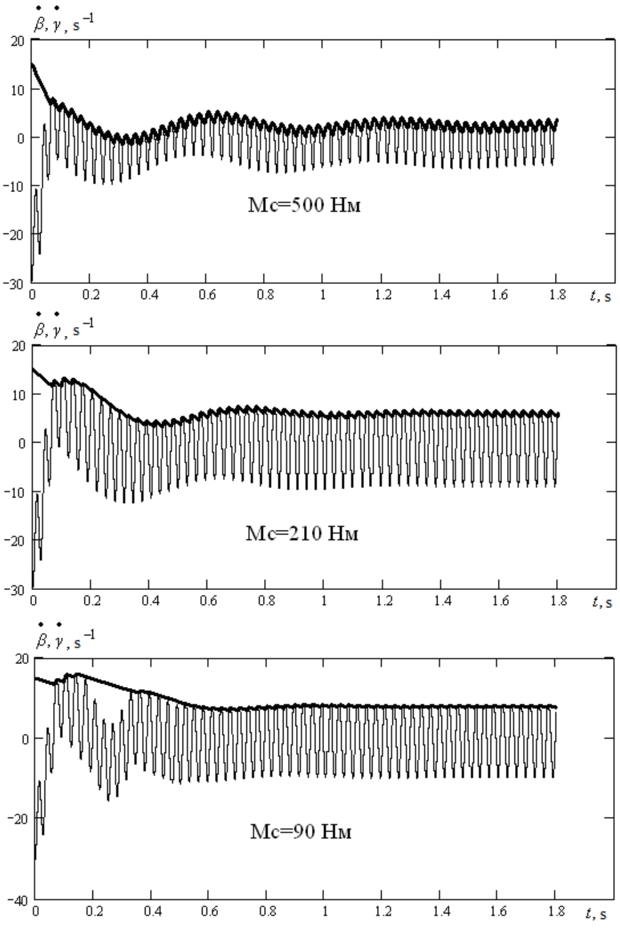

Fig. 6. Graphs of the angular velocities of the driven and the output shafts depending on time in the area of high values of the drag torque

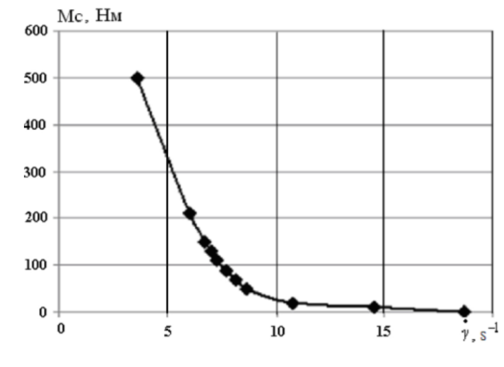

Fig. 7. Graph of the external characteristic of the inertial transmission

In most known overrunning clutches, the whole torque is transmitted through wedging (or, in other words, locking) elements such as balls, rollers, eccentric wheels, pawls, slide blocks, and wedges, whose operation at large loads may limit the life of the mechanism. Therefore, in the present work, we propose an overrunning clutch of relay type, in which only a small part of the torque is transmitted through the locking element [21]; the remainder is transmitted through long-lived elements.

The overrunning clutch of relay type reduces the load on the weak locking elements by one or two orders of magnitude in comparison with existing designs [22-24].

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Its use in inertial transmissions and other machines considerably improves drive performance [8,25,26]. Therefore, the development of a well-founded design method for such overrunning clutches will permit their broader introduction in industrial drives.

The operation of the overrunning clutch of relay type is analogous to that of an electrical relay, in which a small current flows through a weak electrical circuit and triggers the main electrical circuit, which transmits the main energy flux. In Fig.8 we present one design of overrunning clutch of relay type.

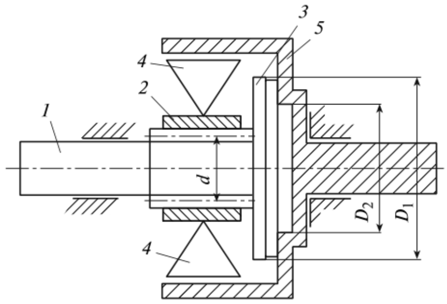

Fig. 8. Overrunning clutch of relay type: D_1, D_2 are external and internal diameters of the frictional surfaces of elements 3 and 5; d is mean diameter of the helical surface

Frictional disk 3 is attached to drive shaft 1, which is positioned in internal ring 2 by means of a screw–nut transmission. Locking elements 4 interact with external ring 5, which is connected to a driven shaft. The drive shaft 1 and driven shaft turn independently if the velocity of the drive shaft is less than that of the driven shaft or if the shafts turn in opposite directions. In that case, the torque in the drive shaft is not transmitted to the driven shaft. When the velocity of the drive shaft is equal to that of the driven shaft, elements 4 lock, and the torque is transmitted from drive shaft 1 through ring 2, elements 4, and ring 5 to the driven shaft. The drag torque on ring 2 results in rotation of the drive shaft relative to the internal ring.

VI. Distribution of the Torque between Parts of the Overrunning Clutch of Relay Type

The axial force on the drive shaft only appears when the overrunning clutch is locked. This benefit of the frictional disk clutch as a means of transmitting large torques at surface contact (and hence with small distributed loads) is fully apparent in overrunning clutches of relay type. In Fig.9 we show the forces in the screw pair of the overrunning clutch.

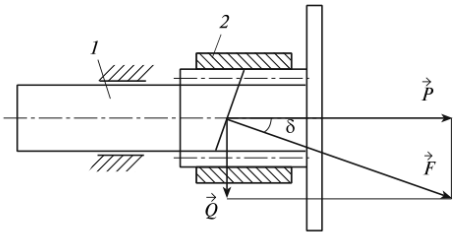

Fig. 9. Forces in the screw pair: (1) drive shaft; (2) internal ring of freewheel mechanism; $\vec{F}, \vec{P}, \vec{Q}$ are resultant, axial, and azimuthal forces on the drive shaft from the screw pair; δ is inclination of helical line

Torque M_1, transmitted through the wedging elements of the overrunning clutch, is determined by the formula

$$M_1 = Q \cdot r \ , \tag{6}$$

here $r = d/2$.

Torque M_2, transmitted through friction surface of the units 3 and 5, is found by the following expression

$$M_2 = \frac{2f \cdot P \cdot (R_1^2 + R_1 \cdot R_2 + R_2^2)}{3 \cdot (R_1 + R_2)}, \tag{7}$$

here f is coefficient of friction in the frictional contact;

$R_1 = D_1/2, R_2 = D_2/2.$

Full torque, which is transmitted from the drive shaft to the driven one, is found as the sum $M = M_1 + M_2$.

It is not so difficult to obtain the relationship between the scalars of forces P and Q (Fig.9)

$$P = Q \cdot \cot\delta^* . \tag{8}$$

We are interested in relationship between the components M_1 and M_2 of the full torque M, therefore, using (6), (7), and (8), we can write the ratio

$$\frac{M_2}{M_1} = \frac{2f \cdot (R_1^2 + R_1 \cdot R_2 + R_2^2) \cdot \cot\delta^*}{3r \cdot (R_1 + R_2)} . \tag{9}$$

Dividing numerator and denominator of the right-hand side of (9) on $R_1 R_2 \neq 0$ and denoting $k = R_1 / R_2$, after some transformations we find

$$\frac{M_2}{M_1} = \frac{2f \cdot R_1 \cdot (k^2 + k + 1) \cdot \cot\delta^*}{3r \cdot (k^2 + k)} . \tag{10}$$

Let $k^2 + k = p$. Then (10) can be rewritten in the form

$$\frac{M_2}{M_1} = \frac{2f \cdot R_1 \cdot \cot\delta^*}{3r} \cdot (1 + \frac{1}{p}). \tag{11}$$

It is clearly, that $R_1 > R_2$, therefore, $k > 1$, and $p > 2$. Taking these relations under consideration, let us do the following estimation $1 < 1 + \frac{1}{p} < \frac{3}{2}$. Then, using the expression (11), we can estimate the ratio of the moments $\frac{M_2}{M_1}$:

$$\frac{2}{3} A < \frac{M_2}{M_1} < A , \tag{12}$$

here $A = \frac{f \cdot R_1 \cdot \cot\delta^*}{r}$.

As it follows from (12), the distribution of the torque on the components, transmitted by the wedging elements and the friction pair, is determined by the value of A. This value depends linearly on coefficient of friction f and radius of the external circumference R_1 of the frictional contact. The dependence of A on the angle δ^* of the helix and the average radius r of screw thread is not linear (although it is monotonically decreasing in the real domain of the arguments) and represents a higher interest for further study.

Fig.10 shows the dependence of the ratio $\frac{M_2}{M_1}$ on the angle δ^* of the helix. Curve 1 corresponds to the lower (dashed line) and the upper (solid line) boundaries of the estimation (7) when values of the parameters are $f = 0,3, R_1 = 0,2\,\text{m}, r = 0,02\,\text{m}$. For the curves 2 radius $R_1 = 0,1\,\text{m}$. All other parameters were taken the same as for the curves 1.

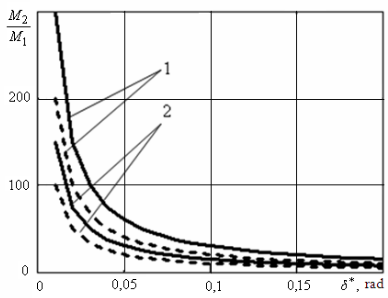

Fig. 10. Dependence of the ratio of the components M_1 and M_2 on the angle helix

From (11) it follows that the ratio of the components of the torque increases without limit as one of the following conditions (or a combination of these conditions) exists: $R_1 \to \infty, r \to 0, \delta^* \to 0$. Consequently, under these conditions it is possible to unload the wedging elements of the overrunning clutch on an arbitrarily large value. We do not consider the case $f \to \infty$ because the coefficient of friction is limited (usually $f \in [0,1 \ldots 0,4]$). Although it is clear that the more f the more the ratio of the moments (11), which also leads to the discharge of the wedging elements.

VII. Conclusions

The suggested transmission is more reliable and has high load ability in compare with the known designs of inertial continuously variable transmissions. It reduces number of overrunning clutches, which are limiting the reliability of the transmission. The elastic links which are setting into the design of the transmission instead of the body overrunning clutch does not limit the reliability of the transmission. The numerous examples of the successful use of elastic connections prove it.

In this paper, the physical and mathematical models of the suggested transmission are developed, and the mathematical model is reduced to only one system of nonlinear differential equations. The received solutions of the mathematical model confirmed the validity of the theoretical proposals. It is shown that the resulting external characteristic is close to an ideal one. This result indicates the potential for widespread usage of the inertial transmission in drives of different engineering systems for various applications.

It was proved that using the suggested design of the overrunning clutches of relay type it is possible to reduce load on wedging elements in tens or even hundreds times. It allows increasing durability and reliability of the clutches sharply.

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