Volume-4, Issue-10, Oct-2015 • ISSN No 2277 - 8160

Research Paper | Engineering

Study of The Elastic and Safety Clutch with Equiangular Cam and Camfollower

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ABSTRACT

This paper presents the theoretical and experimental research of the elastic and safety clutch. The type of the studied clutch fulfils the functions of both an elastic and a safety clutch. The constructive solution of the clutch is being presented. The study of the clutch is realized by determining the forces and the torque momenta. The clutch is cinematically modeled and its elastic capacity is presented. The theoretical research is illustrated by the dynamic characteristic corresponding to an average value of a shock. The experimental diagram emphasizes the importance of such a clutch within a mechanical transmission.

KEYWORDS : mechanical transmission, clutches, elastic, safety, functions, simple, multiple

INTRODUCTION

The load torque of the driven mechanism, also known as static couple, is the couple that appears at the shaft of the mechanism. The couple is the results of all the static forces: friction forces, effective forces etc. The design of mechanical transmission is necessary to know the mechanical characteristics of the engine as well as working machines If the engine-mechanism system rotates with variable velocity, the driving load differs from the load torque. In these cases the velocity variation is accompanied by variation of kinetic energy stored by the moving masses of the group formed by driving motor, transmission and working machines

Continuous evaluation of energetic phenomena, conditioned by the driving of the moving masses, is obtained by the general equation of motion. In this case the kinetic equilibrium equation includes a term with depends on the variation of kinetic energy, i.e. on the dynamic couple. The potential static couples which act on the elastic bodies can store potential energy. The potential static couples, unlike the reaction, can be positive or negative, depending on whether they promote the motion or oppose to it.

In the first moments of the start-up, when the angular velocity is zero, the driving mechanism absorbs a large quantity of energy. On the other hand, in the dynamic regime of the start-up the couple's equilibrium is done by the kinetic energy variation expressed by the dynamic couple. The difference between the driving torque and load torque produces the acceleration or deceleration torque.

During the operation of a mechanical transmission disturbance factors appear, that act upon the system. The most important are the shocks and torsional vibration. Under these conditions, for storing the kinetic energy it is necessary to use an elastic clutch. In order to eliminate the destruction of the drive machine, the transmission and the machine is necessary to use a safety clutch.

In the case of diverse applications, when the mechanical transmission imposes it, there can be combined the simple functions of one clutch type with the simple functions of another clutch type, obtaining a combined coupling. In this case, the combined coupling is obtained by the connection, of two or more simple clutches, in a certain manner, on purpose to accomplish accordingly the imposed complex functional role of mechanical transmission [5], [7].

Taking the data above into consideration, this paper will present a new type of clutch, whose component elements can accomplish the functions of a combined clutch. It will be presented by the name "Elastic and Safety Clutch" [1], [2].

CONSTRUCTION OF THE CLUTCHE

Figure 1 presents the structural schemes of the elastic and safety clutch and figure 3 presents the transversal section through the clutch with flat translation followers [2]. The elastic and safety clutch presented in figure 2 belongs to the category of clutches generated from mechanisms with rotation cam and flat translation follower; the cam as well the follower is non-degenerated [8], [9].

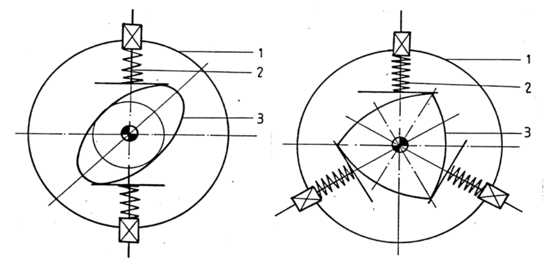


Figure 1: Structural scheme of the elastic and safety clutch with flat translation followers

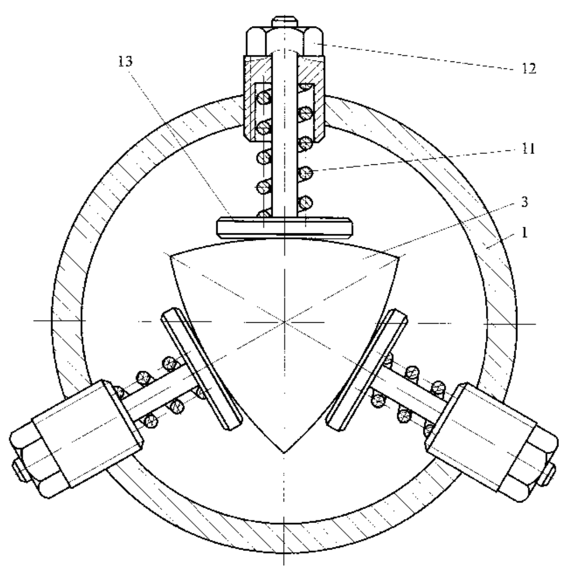


Figure 2: Constructive scheme of the safety clutch

The constructive solution allows the pretension adjustment without to take to pieces the clutch. The clutch contains the equiangular cam 3, with three prominences, which represents one of the semi-clutches and the second semi-clutch is represented by the flange 1. On this flange there are mounted the three flat translation followers 13 – that are equiangular disposed – as well as the fastening and adjusting elements 12, that are necessary for the pretension of the compression spiral springs 11 [8].

The driving semi-clutch 1, transmits the torsion moment by means of the equiangular cam 3, that are in contact with the three flat translation follower 13, due to the pressing force of the compression spiral springs 11. Knowing the elastic characteristic of the compression springs, the pressing force necessary for the torque transmission can be adjusted by means of the special screws 12.

The new type of clutch accomplishes the functions of the elastic and safety clutches and it assures a good damping of torsion shock and vibrations; it also provides the compensation of axial, radial and angular deviations in relative large limits

ANALYTICAL DETERMINATION OF THE TORQUE FOR THE CLUTCH AND THE ELASTC CHARACTERISTIC

In this chapter the torque expression is analytical modelled and in the second part of the chapter there are determined the theoretical and experimental elastic characteristic for the first representative variant considered [2], [7], [8].

Starting from the structural scheme from the Figure 1 and from the constructive variant from Figure 2, in Figure 3 and Figure 4 are presented the geometrical models for the determination of torsion moment that can be transmitted by the clutch on the curved radius M_0M_1 and M_1M_2:

$$\mathbf{T}^* = [\mathbf{k_a}(\mathbf{s_2} + \delta)] \cdot \mathbf{OB}, \qquad (1)$$

$$\mathbf{T} = \mathbf{n}[\mathbf{k_a}(\mathbf{s_2} + \delta)] \cdot \mathbf{OB}, \qquad (2)$$

$$T = \mathrm{n}T^* = \mathrm{n}\,\mathrm{k}_a[(r_1 - r_0)(1 - \cos\varphi_1) + \delta]\{(r_1 - r_0) \cdot$$
$$\cdot \sin\varphi + \mu[(r_1 - r_0)(1 - \cos\varphi_1) + r_0]\}, \qquad (3)$$

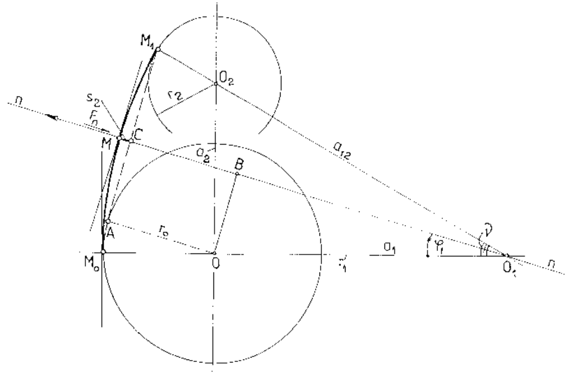


Figure 3: Geometrical model for the torque determination on the curved radius M_0M_1

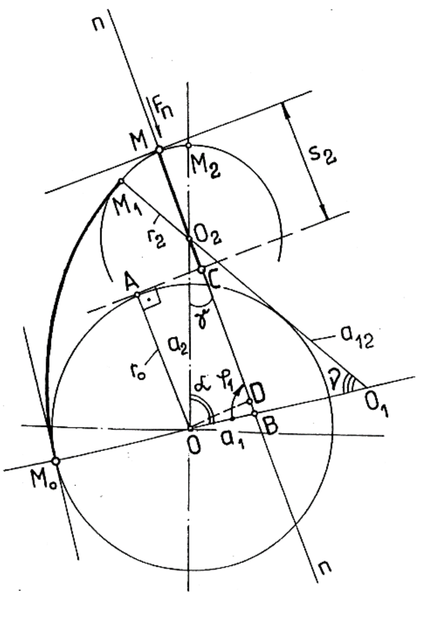


Figure 4: Geometrical model for the torque determination on the curved radius M_1M_2

$$\mathbf{T} = \mathbf{n}\mathbf{T}^* = \mathbf{n}\mathbf{k_a}(\mathbf{s_2} + \delta)[\mathbf{a_2}\sin\gamma + \mu(\mathbf{s_2} + \mathbf{r_0})] =$$
$$= \mathbf{n}\mathbf{k_a}\left[\mathbf{a_2}\frac{1}{\sin\varphi_1}(\sin\alpha - \sin\gamma\cos\varphi_1) + \mathbf{r_2} - \mathbf{r_0} + \delta\right] \cdot$$
$$\cdot\left[\mathbf{a_2}\sin\gamma + \mu\left(\mathbf{a_2}\frac{1}{\sin\varphi_1}(\sin\alpha - \sin\gamma\cos\varphi_1) + \mathbf{r_2}\right)\right], \qquad (4)$$

where by T* it had been written down the moment that corresponds to a single proeminence

($T^* = T/n$, n – the number of the prominences of the cam and the number of the flat translation followers).The determination of the s_2 - change of place will be perform distinctly, for each circle arc of the cam profile.

$$\mathbf{K}(\varphi) = \frac{\mathbf{dT}(\varphi)}{\mathbf{d}\varphi}, \qquad (5)$$

$K(\varphi)$express the tangent to the curve moment which is traced in function oh the φ relative rotation angle between the semi-clutches. For the elastic characteristics determination of the clutches with flat translation followers utilising circle arc profiles (v. Figure. 3 and Figure 4), it is necessary the determination of $K(\varphi)$ for the two characteristic sections M_0M_1, M_1M_2, Figure 5 [9].

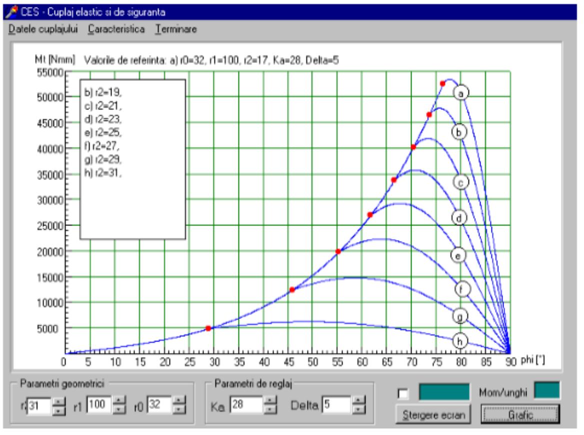


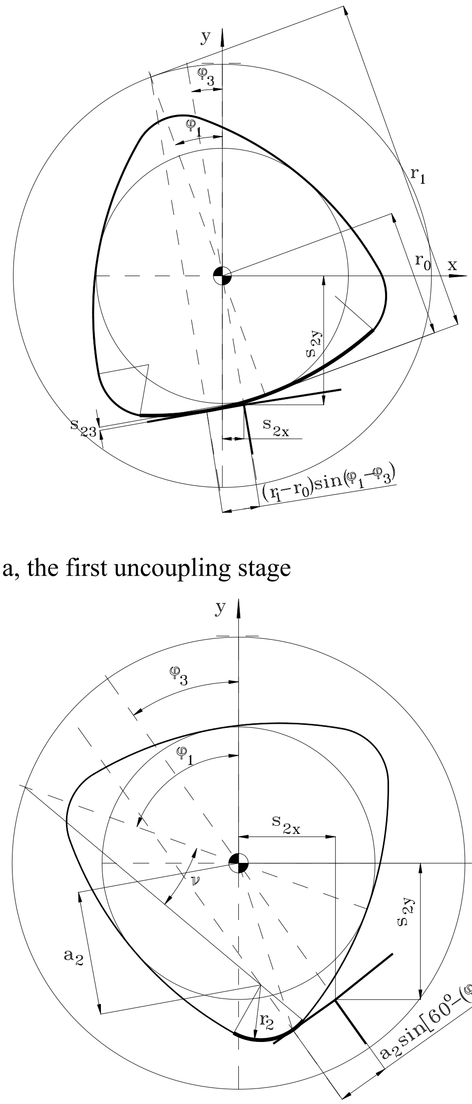
Figure 5: The elastic characteristic of the clutch

KINEMATIC MODELLING OF THE ELASTIC AND SAFETY CLUCHE

The dynamic study of the clutch proposed, will be achieved on the basis of an equivalent plane mechanism with two degrees of freedom and with three external connections, namely the input (φ_1, M_1), the output (φ_3, M_3) and the contact cam - flat translation follower (s_a, F_a), (Figure 2); the semi-clutches motions φ_1 and φ_3 are considered the independent parameters

The adopted mechanism model is defined considering the following assumptions: the kinematic elements are rigid; the kinematic pairs are permanent, geometrical, and stationary.

Figure 6 represents the cam positions for the first and the second uncoupling stages, considering both semi-clutches moving; thus the angular relative displacement between the two semi-clutches is $\varphi = \varphi_1 - \varphi_3$. Knowing the geometry of the cam (Figure 7) there can be determined the transmission function $s_a = s_a(\varphi_1, \varphi_3) = s_{23}$, as well as the relative and absolute position functions of the flat translation followers 2 (13): s_{21} and s_2. Differentiating these position functions, there can be also determined the velocity and acceleration functions of the flat translation follower 2 (13), depending on the relative rotation angle between the semi-clutches ϕ, [8].



a, the first uncoupling stage

b, the second uncoupling stage

Figure 6: Definition scheme for the position functions

- The relative rotation angle
  - The first uncoupling stage

$$\varphi_1-\varphi_3 \in [0,\varphi_{maxI}],$$

$$\varphi_{maxI}=60^o-\arcsin\left(\frac{r_1-r_0}{r_1-r_2}\sin 60^o\right). \qquad (6)$$

  - The second uncoupling stage

$$\varphi_1-\varphi_3 \in [\varphi_{maxI},\varphi_{maxII}],\ \varphi_{maxII}=60^o\ . \qquad (7)$$

- Position function of the flat translation follower 2 (absolute co-ordinates)

$$[s_2]=\begin{bmatrix} s_{2x}\\ s_{2y}\end{bmatrix}=\begin{bmatrix} k_2\sin\varphi_3\\ -k_2\cos\varphi_3\end{bmatrix}. \qquad (8)$$

- Velocity function of the flat follower 2

$$[v_2]=\begin{bmatrix} v_{2x}\\ v_{2y}\end{bmatrix}=\begin{bmatrix} k_1\sin\varphi_3(\dot\varphi_1-\dot\varphi_3)+k_2\cos\varphi_3\dot\varphi_3\\ -k_1\cos\varphi_3(\dot\varphi_1-\dot\varphi_3)+k_2\sin\varphi_3\dot\varphi_3\end{bmatrix}$$

(9)

- Acceleration function of the flat translation follower 2
  - The first uncoupling stage

$$[a_2]=\begin{bmatrix} (r_1-r_0)\cos(\varphi_1-\varphi_3)\sin\varphi_3(\dot\varphi_1-\dot\varphi_3)^2+2k_1\cdot\\ \cdot\cos\varphi_3(\dot\varphi_1-\dot\varphi_3)\dot\varphi_3-k_2\sin\varphi_3\dot\varphi_3^{\,2}+k_1\sin\varphi_3\cdot\\ \cdot(\ddot\varphi_1-\ddot\varphi_3)+k_2\cos\varphi_3\ddot\varphi_3\\ -(r_1-r_0)\cos(\varphi_1-\varphi_3)\sin\varphi_3(\dot\varphi_1-\dot\varphi_3)^2+2k_1\cdot\\ \cdot\sin\varphi_3(\dot\varphi_1-\dot\varphi_3)\dot\varphi_3+k_2\cos\varphi_3\dot\varphi_3^{\,2}-k_1\cos\varphi_3\cdot\\ \cdot(\ddot\varphi_1-\ddot\varphi_3)+k_2\sin\varphi_3\ddot\varphi_3\end{bmatrix}$$

(10)

  - The second uncoupling stage

$$[a_2]=\begin{bmatrix} -a_2\cos[60-(\varphi_1-\varphi_3)]\sin\varphi_3(\dot\varphi_1-\dot\varphi_3)^2+\\ 2k_1\cos\varphi_3(\dot\varphi_1-\dot\varphi_3)\dot\varphi_3-k_2\sin\varphi_3\dot\varphi_3^{\,2}+k_1\cdot\\ \cdot\sin\varphi_3(\ddot\varphi_1-\ddot\varphi_3)+k_2\cos\varphi_3\ddot\varphi_3\\ a_2\cos[60-(\varphi_1-\varphi_3)]\cos\varphi_3(\dot\varphi_1-\dot\varphi_3)^2+\\ 2k_1\sin\varphi_3(\dot\varphi_1-\dot\varphi_3)\dot\varphi_3+k_2\cos\varphi_3\dot\varphi_3^{\,2}-k_1\cdot\\ \cdot\cos\varphi_3(\ddot\varphi_1-\ddot\varphi_3)+k_2\sin\varphi_3\ddot\varphi_3\end{bmatrix}$$

(11)

- Position function of the flat translation follower 2 depending on the driven semi-clutch 3

$$s_{23}=k_2-r_0\ . \qquad (12)$$

- Velocity function of the flat translation follower 2 depending on the driven semi-clutch 3

$$v_{23}=k_1(\dot\varphi_1-\dot\varphi_3). \qquad (13)$$

- Position function of the flat translation follower 2 depending on the driving semi-clutch 1
  - The first uncoupling stage

$$s_{21}=r_1(\varphi_1-\varphi_3)-(r_1-r_0)\sin(\varphi_1-\varphi_3). \qquad (14)$$

  - The second uncoupling stage

$$s_{21}=r_1\varphi_{maxI}+r_2(\varphi_1-\varphi_3-\varphi_{maxI})-$$
$$-a_2\sin[60-(\varphi_1-\varphi_3)]. \qquad (15)$$

- Velocity function of the flat translation follower 2 depending on the driving semi-clutch 1

$$v_{21}=k_2(\dot\varphi_1-\dot\varphi_3). \qquad (16)$$

Notations

- The first uncoupling stage

$$k_1=(r\ -r_0)\sin(\varphi_1-\varphi_3),$$
$$k_2=r_1-(r_1-r_0)\cos(\varphi_1-\varphi_3), \qquad (17)$$
$$k_3=(r_1-r_0)^2\sin[2(\varphi_1-\varphi_3)].$$

- The first uncoupling stage

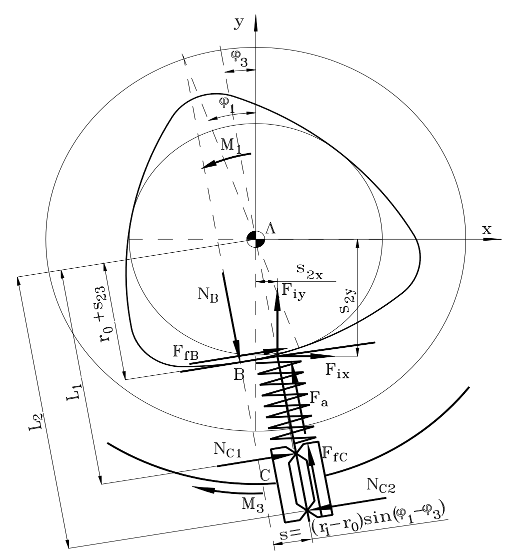
$$\begin{aligned}
&\mathbf{k_1 = a_2 \sin\left[60 - (\varphi_1 - \varphi_3)\right]},\\
&\mathbf{k_2 = r_2 + a_2 \cos\left[60 - (\varphi_1 - \varphi_3)\right]},\\
&\mathbf{k_3 = -a_2^2 \sin\left[2\left(60 - (\varphi_1 - \varphi_3)\right)\right]},\\
&\mathbf{a_2 = \frac{(r_1 - r_0)\sin\varphi_{maxI}}{\sin(60 - \varphi_{maxI})}}.
\end{aligned} \tag{18}$$

THEORETICAL AND EXPERIMENTAL RESEARCHES CONCERNING THE DYNSMIC TESTING

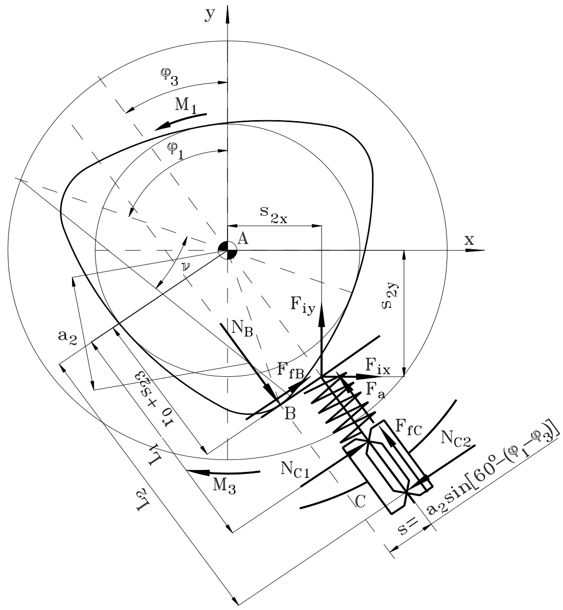
This stage intents to determine the expressions of the friction forces F_{fB} and F_{fC}, their expressions being necessary for the determination of the generalised forces Q_1, and Q_3 [2]. Using the force scheme presented (figure 7), it can be written the equilibrium equation system of the element.
Solving the system (1) there are resulted the following expressions for friction forces [5], [8].

The mechanism presented is bimobile (figure 8) and it has two unknown motions $(\varphi_1, \dot{\varphi}_1, \ddot{\varphi}_1)$ and $(\varphi_3, \dot{\varphi}_3, \ddot{\varphi}_3)$. For the determination of the two motions, there will be use the two transmission functions of the external forces; these functions are described by two Lagrange equations of the second species:

$$\begin{cases}
\mathbf{\dfrac{d}{dt}\left(\dfrac{\partial E_c}{\partial \dot{\varphi}_1}\right) - \dfrac{\partial E_c}{\partial \varphi_1} = Q_1},\\
\mathbf{\dfrac{d}{dt}\left(\dfrac{\partial E_c}{\partial \dot{\varphi}_3}\right) - \dfrac{\partial E_c}{\partial \varphi_3} = Q_3},
\end{cases} \tag{19}$$



a, the first uncoupling stage



b, the second uncoupling stage

Figure 7: The scheme force for the first and the second stage

where E_c represents the kinetic energy of the mechanism and Q_1, Q_3 are the external generalised forces (external forces reduced at the element of whose motion is unknown - element 1, respectively 3). The determination of the generalised forces are obtained from the expression of the total mechanical power, that can be written as follows

$$\begin{aligned}
&\mathbf{Q_1\dot{\varphi}_1 + Q_3\dot{\varphi}_3 = M_1\dot{\varphi}_1 - M_3\dot{\varphi}_3 - 3F_a v_{23} -}\\
&\mathbf{- 3F_{f21} v_{21} - 3F_{f23} v_{23}},
\end{aligned} \tag{20}$$

where: M_1 represents the motor moment (the input moment at se driving semi-clutch); M_3 - the output moment (the moment at the driven semi-clutch); F_a - the resistant force of the arc on its pressing direction; $F_{f\,21}$, $F_{f\,23}$ - the friction forces between follower and semi-clutches; v_{21} - the relative velocity between follower and driving semi-clutch; v_{23} - the relative velocity between the follower and driven semi-clutch.
Solving the Lagrange equations, for the first and the second stage, there will result the characteristic motion equations of the mechanism. In this way making the intermediate calculus for the left members of equation (1), it results

$$\begin{cases}
\mathbf{\left(J_1 + 3m_t k_1^2\right)\ddot{\varphi}_1 - 3m_t k_1^2 \ddot{\varphi}_3 + 0.5 \cdot 3m_t k_3 \cdot}\\
\mathbf{\cdot (\dot{\varphi}_1 - \dot{\varphi}_3)^2 - 3m_t k_1 k_2 \dot{\varphi}_3^2 = Q_1},\\
\mathbf{\left(J_3 + 3m_t k_1^2 + 3m_t k_2^2\right)\ddot{\varphi}_3 - 3m_t k_1^2 \ddot{\varphi}_1 - 0.5 \cdot 3m_t k_3 \cdot}\\
\mathbf{\cdot (\dot{\varphi}_1 - \dot{\varphi}_3)^2 + 3m_t k_1 k_2 \dot{\varphi}_3^2 + 6m_t k_1 k_2 (\dot{\varphi}_1 - \dot{\varphi}_3)\dot{\varphi}_3 = Q_3}
\end{cases} \tag{21}$$

where: J_1 and J_3 represent the moments of inertia for the driving and driven semi-clutch, $\dot{\varphi}$ - the angular velocity of the reduction element; the generalised forces Q_1, and Q_3 are given by the relations

$$\mathbf{Q_1 = M_1 - 3(F_a - F_{fC})k_1 - 3F_{fB}k_2}\,, \tag{22}$$

$$\mathbf{Q_3 = -M_3 + 3(F_a - F_{fC})k_1 + 3F_{fB}k_2}\,. \tag{23}$$

Simulation of the uncoupling process

By means of the motion equations (6), it can be simulate the clutch working. In this way Figure 8 presents the dynamic diagrams for a situation when the resistant moment has a variation corresponding to a medium shock for a short period of time, followed by a decrease of this under the initial value (the moment for what the clutch was designed) [2].

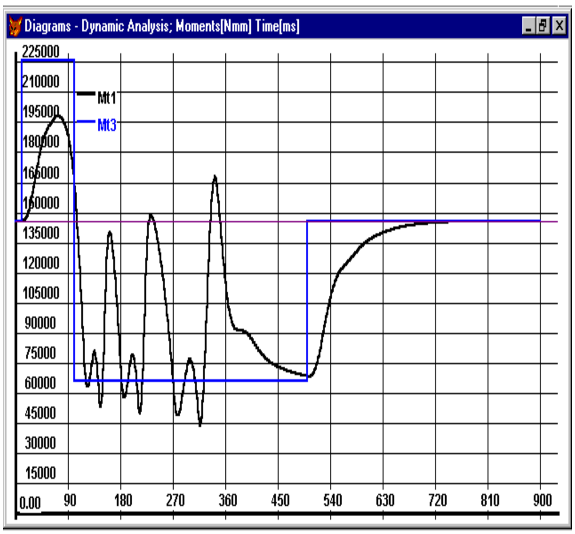


Figure 8: Dynamic diagrams for a medium shock value

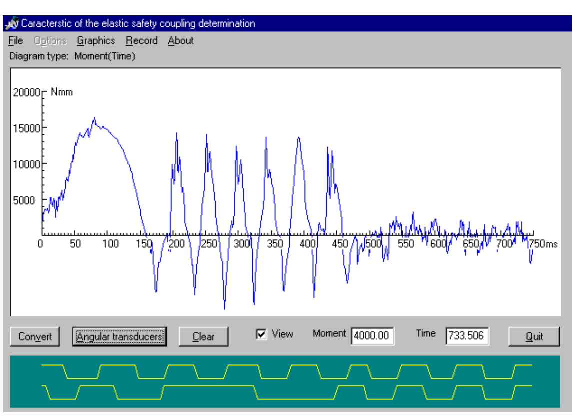


Figure 9: Dynamic diagrams for a big shock value

The resistant moment increase leads at its beginning to the decrease of the angular velocity, the motor moment increases; at the resistant moment decrease takes also place the motor moment decrease, that leads to repeated uncoupling, until its stabilisation. After the return of the resistant moment to the initial value, the clutch returns to the nominal working point (the initial angular velocity for the two semi-clutches and the initial value for the motor moment).

Experimental dynamic diagrams for the elastic and safety clutches

Figure 9 presents the testing dynamic diagram of the elastic and safety clutch on a testing stand.The testing regime corresponds to the shock moment increase up to a value corresponding to the repeated uncoupling situation, followed be the return to the normal working after the elimination of the shock moment.

CONCLUSIONS

The analytic modeling of the torque moment in the case of the clutch with degenerated followers and lamellar bows disposed equiangular is based on the next conditions [8], [10]:

- the modeling is made on a simplified geometrical model;
- the cam profile is defined by circular arcs;
- the lamellas package is replaced by a single lamella, represented in the geometrical model by the median line;
- the lamella is considered free propped on a bolt andjointed on the other;
- the came actuates over the lamella with a normal force orientated after the normal line, in the point of contact;
- the friction effects between the lamella and bearings are neglected;
- the angular deformations of the lamellas are small, because the cams are small in comparison with the opening of the lamellas bearings;
- in the calculus of the torque moment there are considered: the normal force effect as well as the friction effect;
- in the case of the degenerated came, with constant raises, the torque moment is transmitted only by the cam-lamella friction; the elastic and safety clutch becomes a safety clutch by friction.

The elastic and safety clutches of the [9], [10] mechanisms with cam and flat translation followers' type present the next advantages:

- the clutches have a simple construction, cabaret reduced dimension, chip price;
- the clutches ensure the compensation of axial,

radial and angular deviation in relative large limits;

- the clutches ensure the a relative movement between the semi – clutches, in function of the nature and of the disposing mode of the component elements; above the accepted limits, the elastic clothe becomes a safety one;
- the clutches ensure the limitation and the adjustment of the transmitted moment.

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