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Original Research Paper

Engineering

CONSTRUCTIVE AND FUNCŢIONAL GENERATION OF ELASTIC AND SAFETY COUPLINGS WITHIN MECHANICAL TRANSMISION

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ABSTRACT The paper presents the constructive and functional generation of elastic and safety couplings. The method of structural and constructive generation of these couplings is presented. Starting from the mechanic coupling scheme, the following dynamic analysis algorithm is proposed: formulation of problems from dynamic modelling; kinematic and static modelling of elastic and safety couplings; modelling the induced correlations of the mechanical characteristics of the engines and effector; modelling the semi clutches movement using Lagrange educations - species II. The proposed algorithm is verified by the exemplification on a coupling realized and subjected to variable requests and to the constructive and functional deviation. The conclusions obtained highlight the role and importance of elastic and safety couplings in the field of mechanical transmissions

KEYWORDS : Mechanical Transmission, Clutches, Elastic, Safety Structure, Kinematic, Dynamic

INTRODUCTION

The main parameter that derives from the main characteristic of the elastic and safety couplings is the torque. Another parameter is the rigidity or elasticity, which represents the dependence of the relative rotation angle φ of the semi-couplings, depending on the value of the torque of Mt. This feature, which represents the static rigidity of the elastic and safety coupling, is dependent on the construction of the coupling. The verification of the proposed mathematical model for determining the elastic characteristic of the elastic and safety couplings, as well as the validation of the adopted constructive and technological solution, is performed by comparing the theoretic diagrams with the experimental ones, determined in static and dynamic.

The mechanical coupling has the role of transmitting the rotational motion and the torque within a mechanical transmission. Structure of an ether coupling consisting of elemental transmitting motion, intermediate elementary and elemental receiving motion figure 1

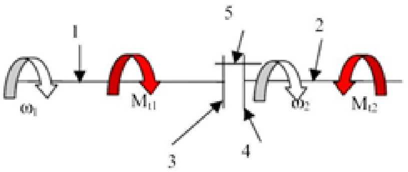

Figure 1: functional diagram of the coupling

The mechanical couplings in addition to the main function of momentum transmission and the transmission of the rotational motion through the constructive frame can perform a number of functions imposed on the transmission.

PROCEDURE FOR DERIVING STRUCTURAL SCHEMES

The main parameter that derives from the main characteristic of the elastic and safety couplings is the torque. Another parameter is the rigidity or elasticity, which represents the dependence of the relative rotation angle φ of the semi-couplings, depending on the value of the torque of Mt. This feature, which represents the static rigidity of the elastic and safety coupling, is dependent on the construction of the coupling. The verification of the proposed mathematical model for determining the elastic characteristic of the elastic and safety couplings, as well as the validation of the adopted constructive and technological solution, is performed by comparing the theoretical diagrams with the experimental ones, determined in static and dynamic regime [1]. The elastic and safety couplings are characterized by the following functions (functional technical criteria):

make the connection between two shafts (with fixed or variable relative position) and ensure the transmission of the moment and the rotation motion between the shafts (according to the general definition);

the transmission of power is interrupted when the resistance moment exceeds an imposed limit value; the interruption of the energy flow is realized on the basis of the deformation of an elastic element (when the deformation reaches the value corresponding to the limit moment, the connection between semi-couplings is interrupted).

From the analysis of the properties corresponding to the elastic and safety coupling, a special importance rests with the modeling of the elastic element, so as to ensure the automatic interruption of the energy flow, at the limit value of the torque

From the critical analysis of the mechanisms used in the technique it turned out that the cam mechanism figure 2 is best suited to the previously formulated requirements, thus:

The component elements and their functions are:

the element 1 (connection between the cam 3 and the pad 2) materializes one semi coupling, and the other semi coupling is represented by the cam element 3;

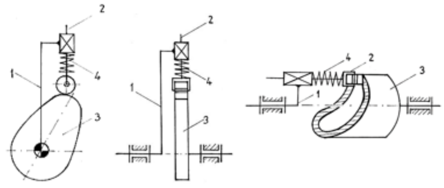

Figure 2: Structural diagrams mechanisms with cam

the relative displacement between the shafts (between the cam element 3 and the element 1) is translated by the linear and / or angular displacement of the base plate, through which the deformation of the elastic element 4 (spring) materializes;

during the power transmission, the relative position between the semi-couplings (elements 1 and 3) remains unchanged if the transmitted moment is constant and less than the limit torque; the value of the transmitted moment is directly measurable (transposed), by the deformation of the elastic element; if the transmitted moment is variable but lower than the limit, the relative position between the semi-couplings and implicitly the deformations of the elastic element are also

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variable;

reaching the limit moment corresponds to the maximum displacement between the elements 1 and 2 and implicitly the maximum deformation of the elastic element; In this case, the semi-couplings perform a relative movement corresponding to a step (cycle).

The actual generation considers the following objectives:
 the obtained coupling must have a symmetrical construction, in order to achieve simple balancing;

a semi-coupling must contain a flat multiple rotating cam (with at least two equilateral elevations), or a spatial one;

the second semi coupling must use translational fastening, oscillating or flat, radially, axially and equilocularly arranged;

from the point of view of the adjustment of the torque to be transmitted by the coupling, two variants can be made:
- with the initial adjustment of the restressing of the elastic element;
- with the possibility of adjusting the pretensioning without dismantling the coupling, depending on the torque transmitted;

from a constructive point of view, elastic and safety couplings can be generated in four main variants:

couplings derived from mechanisms with the cam itself and the stud itself;

couplings derived from mechanisms with proper cam and degenerate stud in elastic stud;

couplings derived from mechanisms with degenerate cam and the actual stud;

couplings derived from mechanisms with degenerate cam and degenerate stud.

CONSTRUCTIVE GENERATION OF ELASTI AND SAFETY COUPLING

- For the design of the elastic and safety couplings, the transition from the structural to the constructive schemes is presented where the following constructive generation criteria can be formulated:
- the coupling must take axial, radial and angular deviations;
- the relative movement between the semi-couplings, as well as the load decoupling to be made without shocks;
- the coupling must have a reduced rigidity, it is recommended a characteristic of the torque as a function of angle Mt () with an increasing slope and a high damping capacity;
- the elasticity of the coupling can be modified by changing or adding some elastic construction elements;
- when rotating the coupling, no large axial forces appear;
- it will be taken into account the release of the heat generated by the damping of the oscillations or the relative rotation between the semi-couplings;
- the coupling must not go out of operation immediately when an elastic element is destroyed;
- elastic construction elements, which can be destroyed quickly, should be easily replaced, if possible without dismantling the coupling or moving the shaft ends axially;
- changing the direction of rotation is allowed without play;
- In order to increase the safety in operation, the components of the coupling should not be protruding.
- Based on these criteria, constructive variants and design calculation of elastic and safety couplings can be generated.

DYNAMIC MODELING OF ELASTIC AND SAFETY COUPLINGS

Dynamic modeling of elastic and safety couplings

The work has as a priority the dynamic modeling of couplings, considered included in the transmission of a machine. According to the definition of the machine [1], for the dynamic study a mechanical transmission is considered whose scheme figure 2 comprises: a drive motor, a transmission, which includes the analyzed coupling, and an effector (power consumer).

For the dynamic modeling of the coupling, in fig. 2 an equivalent calculation scheme has been proposed in which the component parts of the machine, noted in fig. 1. with I and III, they are reduced to the semi-coupling shafts 1 and 3.

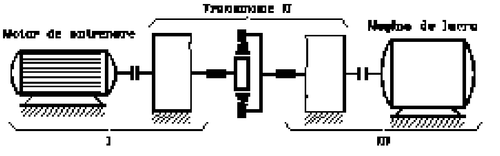


Figure 3: mechanical transmission

In order to dynamically model the coupling, an equivalent calculation scheme was proposed in figure 4 in which the component parts of the machine, noted in figure 3 with I and III, are reduced to the semi-coupling shafts 1 and 3.

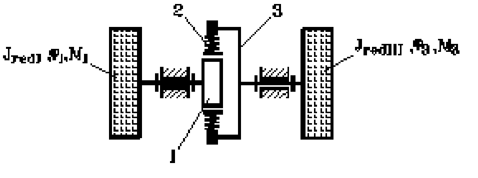
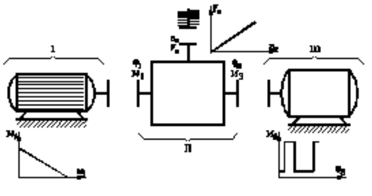


Figure 4: equivalent calculation scheme

Based on the equivalent calculation scheme, Figure 4, the following dynamic analysis algorithm is proposed:
- formulation of the dynamic modeling problem;
- kinematic and static modeling of the coupling;
- modeling the correlations induced by the mechanical characteristics of the motors and effector;
- modeling the motion of the semi-couplings using the Lagrange equations of the second species.

FORMULATION OF THE DYNAMIC MODELING PROBLEM

From a structural point of view, the transmission II figures 5 is characterized by L = 3 external links and the degree of mobility M = 2. From their interpretation, we obtain:



Figures 5: scheme of motor and resistant energy systems

• $L = 3 \Rightarrow 2L$ external parameters:

$$(\varphi_1, M_1),\ (\varphi_3, M_3),\ (s_a, F_a) \qquad (1)$$

• $M = 2 \Rightarrow 2$ independent external movements:

$$(\varphi_1, \dot{\varphi}_1, \ddot{\varphi}_1),\ (\varphi_3, \dot{\varphi}_3, \ddot{\varphi}_3) \qquad (2)$$

$\Rightarrow$ 2 forces transmission functions

$$M_1 = M_1(F_a, \varphi_1, \varphi_3) \qquad (3)$$

$$M_3 = M_3(F_a, \varphi_1, \varphi_3) \qquad (4)$$

L-M=1 $\Rightarrow$ o motion transmission function:

$$s_a = s_a(\varphi_1, \varphi_3) \qquad (5)$$

$\Rightarrow$ an independent external force: F_a

Considering that the operation of the mechanism in the machine is characterized by movements and forces determined in time, it turns out that for the 6 external parameters (1) of the mechanism, 6 independent equations are required; L = 3 of these equations characterize the links of the mechanism (its transmission functions - relations (3), (4), (5)), and the other L = 3 equations required are described by the mechanical characteristics of the motor and resistant energy systems:

$$M_1 = M_1(\omega_1), M_3 = M_3(\varphi), F_a = F_a(s_a) \quad (7)$$

Therefore, the formulation of the dynamics problem consists in establishing the 6 external parameters (1), as expressions of time, under the conditions of defining the dependencies introduced by the links of the mechanism and of some given characteristics, corresponding to the external links.

CINEMATIC AND STATIC MODELING OF THE CLUTCH

This subchapter aims to determine the dependencies introduced by the mechanics between the movements and the external forces (assuming that the transmission is isolated from the machine), dependencies expressed in the form of the transmission functions of the movements and the tasks, respectively.

$$s_a = s_a(\varphi_1, \varphi_3) \quad (8)$$

$$M_1 = M_1(F_a, \varphi_1, \varphi_3) \quad (9)$$

$$M_3 = M_3(F_a, \varphi_1, \varphi_3) \quad (10)$$

The proposed elastic and safety coupling can be modeled as a two-capacitive (M = 2), "capacitive" type (by analogy with AC circuits), with an input and an output, in which the movements of the semicouplers, and, are considered to be independent parameters. The mechanical capacity of the coupling, materialized by its elastic elements, transforms the coupling between the stud 2 and the semi-coupling 3 [2], [3] into a connection which is still considered to be external. In this way, the structural condition of existence of the mechanism [1] is ensured: L = 3> M = 2> 0, in which L - represents the number of external connections (power inputs and outputs), and M - the degree of mobility.

MODELAREA CORELAŢIILOR DINTRE MOTOARE ŞI EFECTOARE

Each energy system (engine or consumer) is characterized by an equation of dependence between the external parameters, called the mechanical characteristic of the energy system.

Consequently, in the case of the considered coupling, between the external parameters of the two-vehicle equivalent mechanism with L = 3 inputs and outputs, the energy systems still introduce L = 3 type dependence equations:

$$M_1 = M_1(\varphi_1, \dot{\varphi}_1, t) \quad (11)$$

$$M_3 = M_3(\varphi_3, \dot{\varphi}_3, t) \quad (12)$$

$$F_a = F_a(s_a) \quad (13)$$

These equations, together with the equations described by the transmission functions of the motions and forces, will form a system of 2L = 6 equations, which allow the 2L external parameters to be determined. The purpose of this paper is to establish the L = 3 external parameters that remain undetermined, by introducing the L = 3 mechanical characteristics of the motor and consuming energy systems. These equations, together with the equations described by the transmission functions of the motions and forces, will form a system of 2L = 6 equations, which allow the 2L external parameters to be determined. The purpose of this paper is to establish the L = 3 external parameters that remain undetermined, by introducing the L = 3 mechanical characteristics of the motor and consuming energy systems.

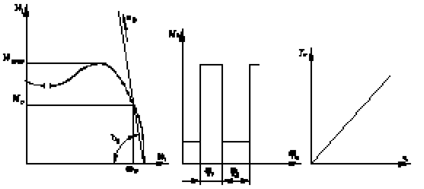


Figure 6: the mechanical characteristics of the motor and resistant energy systems

MODELING THE MOTOR TIME

As the description and representation of the typical functions of the training moments are mathematically complicated, for this reason, simplified equivalent features, accessible to calculations, are used. In the case studied, it is proposed as a simplified solution the linear operation part of the characteristic of an asynchronous motor.

The simple case proposed covers, in practice, a large number of transient loading processes, the equation of torque of the motor having the following expression

$$M_1 = a_0 - b_0\omega_1 = M_n\left(1 + \frac{M_{max}}{M_n}\right) - \frac{M_n}{\omega_n}\frac{M_{max}}{M_n}\omega_1 \quad (14)$$

where a_0 represents the size of the conventional moment, which is obtained if the given right is extended to the intersection with the ordinate axis, and b_0 - the angular coefficient of this straight line.

These coefficients can be written according to the nominal technical characteristics of the chosen drive motor (nominal moment and angular speed, determined in turn according to the rated power and speed) and according to the ratio Mmax / Mn, given in the engine catalogs.

MODELING THE RESISTANT MOMENT

For the determination of the resistant moment, it will be considered the case of a diagram specific to the impacting machines, with a high resistance moment at the operating mode (Moc) and with a small monom resistant to idling (Mtc). Such a feature is represented in figures 6 and is expressed by the equation,

$$M_{t3} = M_{tc} \text{ for } \varphi \in \varphi_g$$

$$M_{t3} = M_{tc} + M_{soc} \text{ for } \varphi \in \varphi_r \quad (15)$$

ARCH FORCE MODELING

In the decoupling process, the displacement of the patches leads to the compression of the coil springs. Therefore, when compressing the springs, the resistant force acts on the tachosFa. Knowing the linear elastic characteristic of the arc, one can write the expression of the force in the arc

$$F_a = F_{a1} + s_{23}K_a \quad (16)$$

in which Fa1it represents the pretensioning force of the springs, k_a este rigiditatea arcului.

SPECIFICATIONS ON THE APPLICATION OF LAGRANGE EQUATIONS

In the study of the motion of a system of rigid bodies, analytical mechanics starts from a series of hypotheses. Of these, the most important is the hypothesis of ideal connections, which precludes the application of the principle of virtual mechanical work (virtual mechanical power). In this premise, the application of Lagrange's equations usually excludes the consideration of frictional forces. To circumvent this disadvantage, the frictional forces will be considered as external forces, their determination being done separately, by the method of Alembert.

The equivalent mechanism proposed being two-wheeled, has two independent movements:

$(\varphi_1, \dot{\varphi}_1, \ddot{\varphi}_1)$ and $(\varphi_3, \dot{\varphi}_3, \ddot{\varphi}_3)$.

To determine them, the M = 2 transmission functions of the external forces, described by the Lagrange equations of the second species, will be used. As a result, including in them the explanations offered by the mechanical characteristics of the motor and resistant systems, the equations of motion of the mechanism are obtained [1].

The general expressions of Lagrange equations of type II have the form:

$$\begin{cases} \frac{d}{dt}\left(\frac{\partial E_c}{\partial \dot{\varphi}_1}\right) - \frac{\partial E_c}{\partial \varphi_1} = Q_1, \\ \frac{d}{dt}\left(\frac{\partial E_c}{\partial \dot{\varphi}_3}\right) - \frac{\partial E_c}{\partial \varphi_3} = Q_3, \end{cases} \tag{17}$$

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Where Ec is the kinetic energy of the mechanism, and Q1 and Q3 respectively are the generalized external forces (external forces reduced to the element whose motion is unknown - element 1, respectively 3).

CONCLUSIONS

Following the kinematic and dynamic modeling of the elastic and safety couplings and the numerical simulations performed, some more important conclusions can be formulated.

- Passing the cam-to-tach contact point from one operating phase to the other is accomplished with acceleration jump which leads to large variations of torque momentum - both at the driver and the driven semicoupling - due to the jumps of moments produced by the inertia the driving and driving parts of the coupling.
- The coupling operation on the cam connection area is unstable due to the sudden decrease of the normal force arm which at a relative rotation between the two semi-couplings becomes 0. For tracking the operating phases, some positions of the cam were shown in the relative movement between the semi-couplings. Although on the cam connection portion there is an increase in momentum resistance, it is not recommended to operate the coupling on this portion both, due to the high deformation force of the elastic elements, as well as due to the instability of the operation.
- The adjustment of the force of the pretensioning of the springs has a special influence on the torque transmitted by the coupling; thus, the increase of the pretensioning force leads to the increase of the capacity to take the shocks; the increase of the pretensioning force corresponds to the operation of the coupling - at the nominal operating point - at a relative rotation angle between the small value semi-couplings. If the load decoupling is desired when the smallest shocks occur, it is recommended to use smaller pretensioning forces (corresponding to a relative rotation angle between the semi-couplings - at the nominal operating point - values close to the maximum angle of the first phase)

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