ORIGINAL RESEARCH PAPER

Engineering

STUDY OF COUPLING COMPENSATION IN MECHANICAL TRANSMISSIONS

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ABSTRACT

The paper presents the constructive and functional importance of couplings in a mechanical transmission under given operating conditions. On a mechanical transmission, the equivalent calculation scheme is presented and an algorithm is proposed that defines the coupling behaviour in different operating modes. The structure of the transmission is characterized by external links and mobility of the mechanism. The equivalent scheme of the motor and resistive power system defines independent external parameters and movements. There is a dependence between the movements and the external forces expressed in the form of the movements of the movements. Depending on the geometry of the component elements of the coupling, the function of the movement of the movements and of the positioning functions is defined.

INTRODUCTION

In the case of mechanical couplings, the moment of torsion and the rotation movement are the main coupling parameters that characteristically define the mechanical couplings. Another parameter is rigidity or elasticity, which is the dependence of the relative rotation angle of the semicuplations, depending on the value of torque Mt. This feature, which represents the static stiffness of the elastic and safety coupling, is dependent on the coupling construction. The validation of constructive solutions is given by the mathematical model defined in the theoretical determination of the elastic characteristic. The theoretical characteristic is characterized by comparison of the experimental one. The validation of the constructive solution is given by the mathematical model, defined in the theoretical determination of the elastic characteristic by comparing the experimental one with the theoretical characteristic determined in the static and dynamic regime. If the two theoretical and experimental characteristics are close in size and shape, it can be considered that the theoretical model adopted is the right one and the desired solution. For coupling modeling it is considered to be implemented in a mechanical transmission.

EQUIVALENT SCHEME OF MECHANICAL TRANSMISSION

For the study of the couplings, it is considered a mechanical transmission comprising an electric drive motor, a mechanical transmission containing the coupling and a working machine, figure 1.

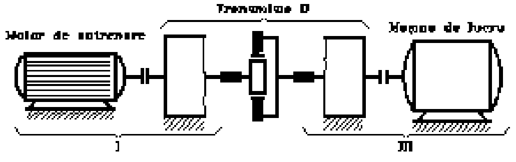

Figure 1: mechanical transmission

An equivalent scheme is proposed in figure 2 for dynamic coupling modeling. Mechanical transmission components are reduced to semi-couplings.

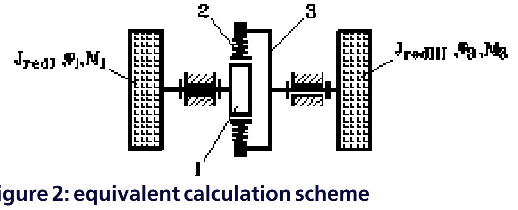

Figure 2: equivalent calculation scheme

The calculation scheme allows for the formulation of dynamic analysis by kinematic, static and dynamic modeling of the coupling. Mechanical characteristics of engines and work machines are correlated. For the modeling of the movement between semi coupling Lagrange equations of II a are used. Figure 3 shows the transmission II characterized by external links and mobility. Parameter interpretation determines static, cinematic and dynamic behavior.

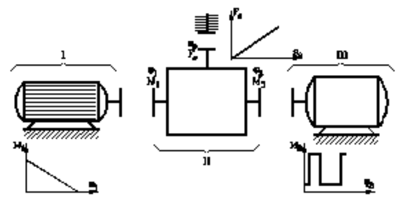

Figure 3: Scheme of power and power systems

The outer parameters of the mechanism for L = 3 are:

$$(\varphi_1, M_1)(\varphi_3, M_3)(s_a, F_a) \tag{2}$$

It results in 2 forces transmitting functions:

$$M_1 = M_1(F_a, \varphi_1, \varphi_3) \tag{3}$$

$$M_3 = M_3(F_a, \varphi_1, \varphi_3) \tag{4}$$

The difference between L-M = 1 determines the movement of the movement:

$$s_a = s_a(\varphi_1, \varphi_3) \tag{5}$$

respectively an independent external photograph Fa.

The operation of the entire mechanism is a function of time, forces and movements. 6 independent equations are required to define the 6 parameters (1).

The forwarding functions of the relations (3), (4), (5) characterize the links of the mechanism. The mechanical characteristics of the motor and the working machine are described by the other equations:

$$M_1 = M_1(\omega_1) M_3 = M_3(\varphi) F_a = F_a(s_a) \tag{6}$$

The external parameters, which define the mechanism, are a function of time, its characteristics and the external links that allow the dynamic formulation of the mechanical system.

CINEMATIC AND STATIC MODELING OF THE COUPLING

The modeling is made on a resilient and flexible coupling that performs the functions of an elastic coupling and a safety coupling. This coupling can be modeled as a bimobile mechanism in which the independent parameters define the semi couplings φ1and φ3, figure 4, (4].

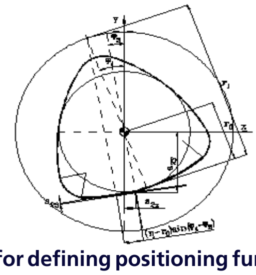

Figure 4: Scheme for defining positioning functions

The outer link between the semi coupling is realized by the kinematic coupler made between the two-piece stalk, belonging to the semi-coupling 1, and the semi-coupling 3, [1], [4]. Thus, the structural link of the mechanism L=3>M=2>0, L defines the external link while M represents the degree of mobility [1].

For the modeling of the mechanism, the following calculation assumptions are applied: the kinematic elements are rigid, the kinematic bonds are permanent. The analytical correlation is performed between the half-coupling movements and the torque between the two coupling components. Here are the positions of the cam that corresponds to the function of the cam in the radius circle r_0. The relative movement between semi couplings is determined by the difference between the angles $\varphi1$ and $\varphi3$.

The semi couplings led causes relative angular displacement. Depending on the geometry of the cam are expressed the expressions of the motion transmission functions, the relative and absolute positions of the stick [4].

Defining positioning functions determines the expressions of speed and acceleration functions taking into account the independent movements of the mechanism.

The following are the expressions of these position, speed and acceleration functions:

- Angle of relative rotation between semi couplings

$$\varphi_1 - \varphi_3 \in (0, \varphi_{\max I})$$

$$\varphi_{\max I} = 60^o - \arcsin\left(\frac{r_1 - r_0}{r_1 - r_2}\sin 60^o\right)$$

- Button position function 2 (absolute coordinates)

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

- Button speed function 2

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

- Throttle acceleration function 2

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

- Relative position function of the rod 2 against the driven semi coupling 3

$$s_{23} = k_2 - r_0$$

- Relative speed of the stem 2 relative to the steering half-coupling 3

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

- Relative position of the rod 2 relative to the steering half-coupling 1

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

- Relative speed of tiller 2 relative to the leading half-coupling 1

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

Remarks:

$$k_1 = (r - r_0)\sin(\varphi_1 - \varphi_3)$$

$$k_2 = r_1 - (r_1 - r_0)\cos(\varphi_1 - \varphi_3)$$

$$k_3 = (r_1 - r_0)^2 \sin[2(\varphi_1 - \varphi_3)]$$

By neglecting friction and moments of inertia, the expressions of transmitted torsional moments are expressed in moments capable of generating angular displacement. If we take into account the friction forces and inertia moments, the force transmission functions can be determined by the Lagrange equations.

CONCLUSIONS

The kinematic model of the elastic and safety couplings defines the following conclusions: the contact cam cam point out the coupling operation taking into account the geometry of the cam, defining the phases of operation and the passage from one phase to the other; the transition from one phase to another is realized and highlights the accelerations that induce torsion moments with high leaps; these high jump torques are caused by the inertia moments of the semi-coupled and driven clutch.

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