THEORETICAL ASPECTS CONCERNING THE DYNAMIC MODELLING OF THE SAFETY CLUTCHES WORKING

Elena EFTIMIE
“Transilvania” University of Braşov, Department of Product Design and Robotics
e-mail: eftimiem@unitbv.ro

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Abstract: The clutches are used largely in machine buildings, and by the correct selection of these depends – to a great extent – the safe and long working both of these and of the kinematic chain equipped with them. The guarantee of these demands for the mechanical power transmission between shafts represents a ticklish problem for all areas and engineering applications that require compact, simple and reliable systems. By their advantages, the safety clutches are preferred in different top techniques areas: cars, naval industry and so on.

1. OBJECTIVES

For a correct, safe and economical machine working, it is necessary that the component elements of this to be designed and accomplished properly.

One of the directions through, still from the conceiving stage, it can be worked both on the machines and equipments gauge and on their reliability (so implicitly on the materials and energy consumptions) is the mechanical transmission protection against overloads, that produce their deteriorations, by the use of some safety clutches. In this way, the designers can decrease the value of the safety coefficient for the dimensioning of the mechanical transmissions of equipments.

The safety clutches fulfil – besides the main function of the torque transmission and rotational motion transmission between two consecutive elements of a kinematic chain - the function of transmitted torque limitation, in the case of some overloads occurrence, during the performance. In this way it is avoided the kinematic chain elements overstressing and their deterioration.

The overloads – that occur in transmission thanks to some causes like machine starting or stopping, the passing through resonance zone, too high overloads of the driven mechanism – can be dynamic (with shocks), with very short duration or quasi-static with long duration.

Indifferently of the overloads type, these can lead to the machine deterioration and its retirement. Taking into consideration all overloads, for the transmission calculus, can lead to an excessive over-measure of this, situation that cannot be accepted. If a safety clutch is assembled in the kinematic chain of the mechanical transmission, then the mechanical properties of the materials, for the transmission component elements can be used to maximum.

This paper has as a main objective the proposing of some models and analytical methods of dynamic study for the safety clutches as part of the mechanical transmissions. In keeping with the machine definition [1], for this study it is considered a machine whose scheme (Figure 1,a) includes: one motor, one transmission that contains the analysed clutch and one end-user (power consumer) that is the working machine.

For the dynamic modelling of the safety clutches, in Figure 1,b, an equivalent calculation scheme is proposed, where the component parts of the machine, denoted in Figure 1,a with I and II, are reduced to the semi-clutch shafts 1 and 3.

On the basis of the calculation scheme (Figure 1,b) the dynamical analysis algorithm proposes the following steps:

- the wording of the dynamic modelling problem;
- the static and kinematic modelling of the clutch;
- the modelling of the correlation induced by the mechanical characteristics of the motors and end-users;
- the modelling of the semi-clutches motions by means of the Lagrange equations of the second species.

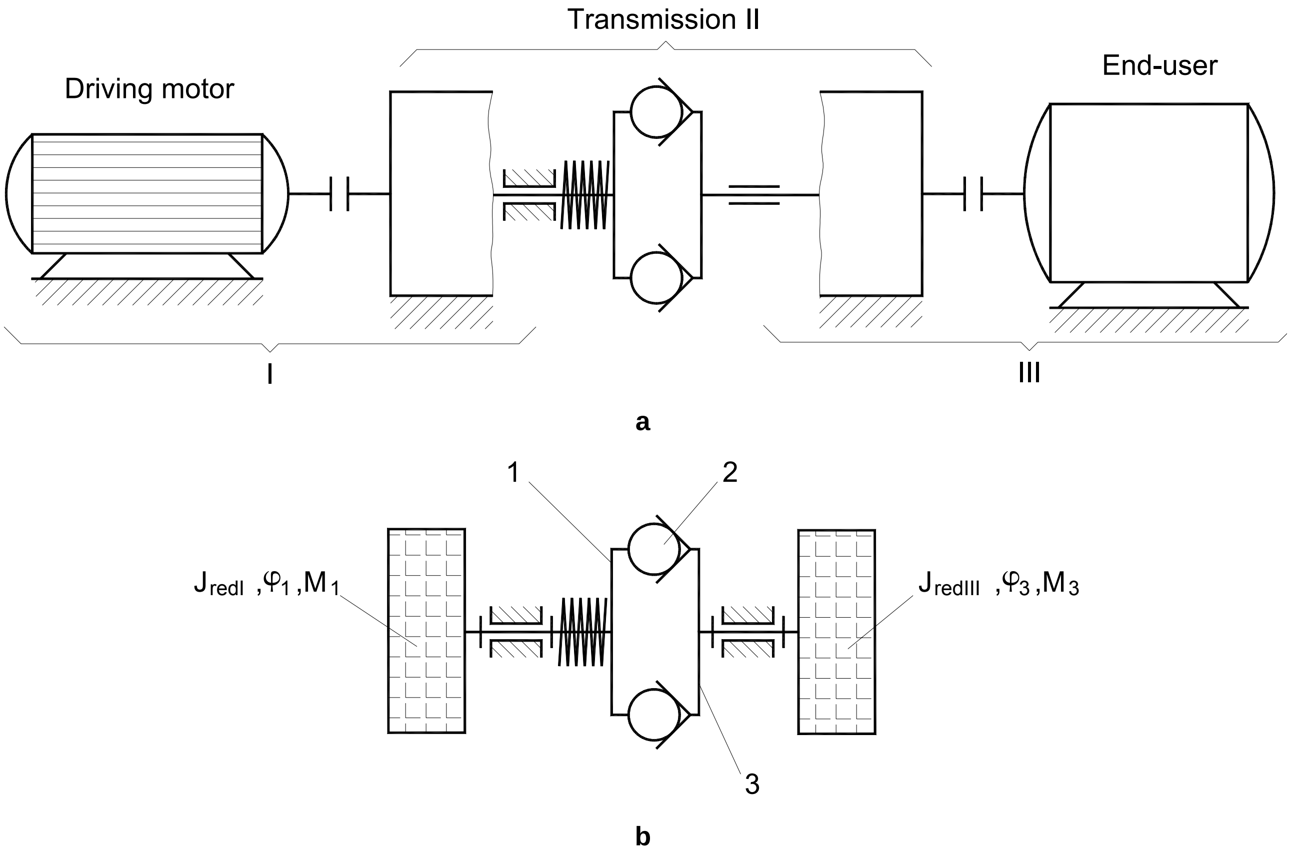


Figure 1. Equivalent calculation scheme of the machine

2. WORDING OF THE DYNAMIC MODELLING PROBLEM

From a structural point of view, the transmission II (Figure 2) is characterised by L=3 external connections and the degree of freedom M=2; from their interpretation it is obtained:

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

$$(\varphi_1, M_1), (\varphi_3, M_3), (s_a, F_a), \tag{1}$$

- M=2 $\Rightarrow$ 2 independent external motions:

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

$\Rightarrow$ 2 transmission functions of the forces:

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

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

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

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

$\Rightarrow$ 1 independent external force:

$$F_a . \tag{6}$$

Taking into consideration that the working of the mechanism as part of the machine, is characterised by motions and forces determined in time, it results that for the determination of the 6 external parameters (1) of the mechanism there are necessary 6 independent equations; L=3 from these equations characterise the mechanism connections (the transmission functions of this – relations (3), (4), (5)); the other L=3 necessary equations are described by the mechanical characteristics of the motor and resistant energy systems:

$$M_1 = M_1(\omega_1), M_3 = M_3(\varphi_3), F_a = F_a(s_a). \tag{7}$$

As a result, the wording of the dynamics problem consists of the determination of the 6 external parameters (1), as expressions depending on time, under the circumstances of dependencies definition due to the mechanism connections and of some given characteristics, proper to the external connections.

3. KINEMATIC AND STATIC MODELLING OF THE CLUTCHES

This stage has as a purpose the determination of the dependencies introduced by the mechanism among the motions and the external forces (under the assumption the transmission is isolated from machine), dependences expressed in the appearance of the motion transmission functions (5) and, respectively, of the loads (3), (4).

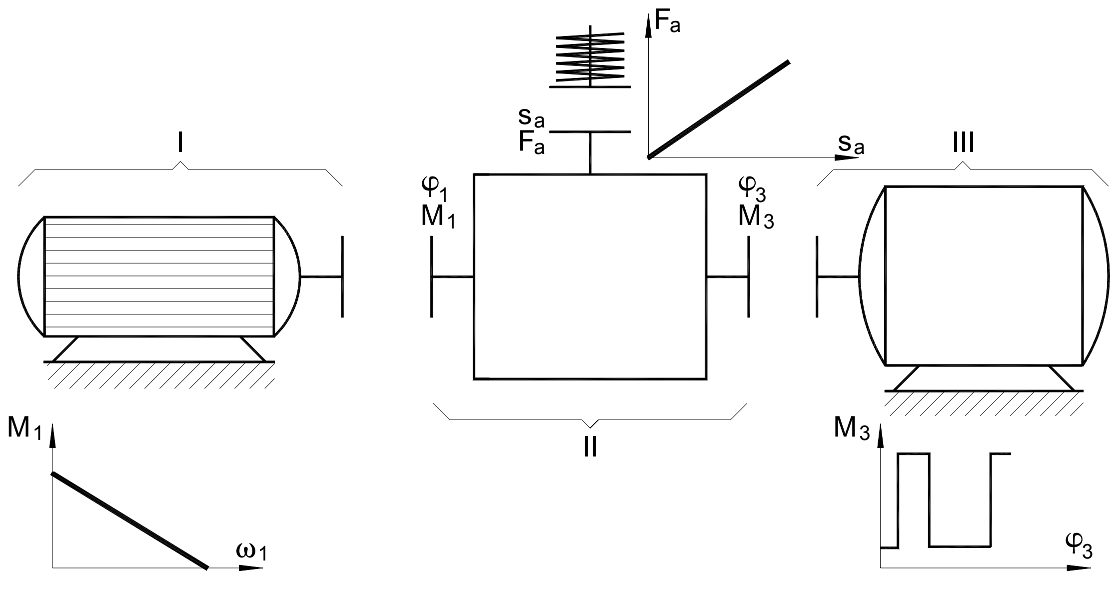


Figure 2. Structural scheme of the mechanical transmission

There are recommended the clutches modelling as bimobile mechanisms (M=2), with one input and one output, where the semi-clutches motions, φ_1 and φ_3, are considered as independent parameters. For the observance of the structural condition of mechanism existence [1]: L=3>M=2>0, where L represents the external connections number (power inputs and outputs), and M the degree of freedom, the third external connection will be considered the connection among the active intermediate elements (that carries the uncoupling out) and one of the semi-clutches (connections achieved by means of the used elastic elements).

4. MODELLING OF THE CORRELATIONS INTRODUCED BY THE MECHANICAL CHARACTERISTICS OF THE MOTORS AND END-USERS

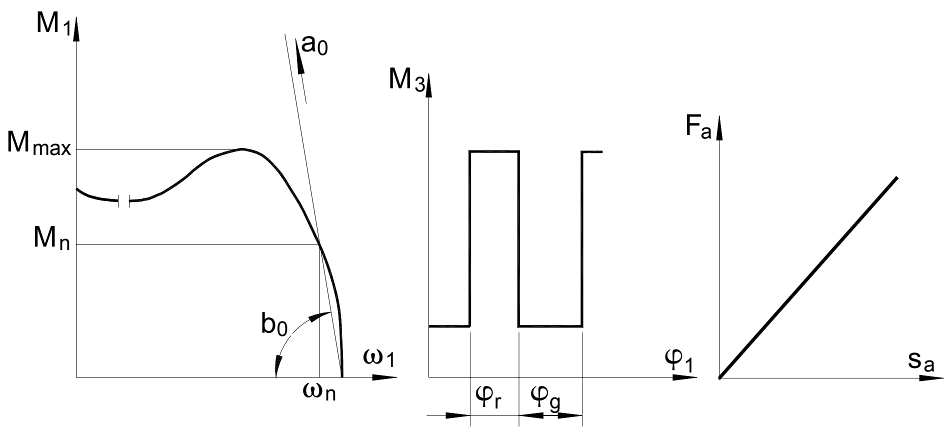
It is known, in the general case, the mechanism is a part of a machine (Figure 2); as a result, every input of the mechanism is connected to a motor energy system, and every output is connected to a consumer energy system. Each energy system (motor or consumer) is characterised by one equation of dependence among the external parameters, named mechanical characteristic of the energy system. Therefore, in the case of the considered clutch, besides the external parameters of the bimobile equivalent mechanism with L=3 inputs and outputs, the energy systems introduce more L=3 dependence equations:

$$M_1 = M_1(\varphi_1, \dot{\varphi}_1, t), \tag{8}$$

$$M_3 = M_3(\varphi_3, \dot{\varphi}_3, t), \tag{9}$$

$$F_a = F_a(s_a). \tag{10}$$

These equations, together with the equations described by the transmission functions of the motions and forces, will lead to a system of 2L=6 equations, that allow the determination of the 2L external parameters. Therefore, this stage has as an end in view the establishment of the L=3 external parameters remained undetermined, using the L=3 mechanical characteristics of the motor and consumer energy systems.



a b c

Figure 3. Mechanical characteristics of the motor and consumer energy systems

Motor moment modelling

Because the description and the representation, of the typical functions, of the driving moments are complicated from a mathematical point of view, as a rule, the simplified equivalent characteristics are used, accessible for calculus. For this case as a simplified solution, the linear working portion of the characteristic of an asynchronous motor $M_1=M_1(\omega_1)$ is proposed (Figure 3). The simple proposed case covers, in practice, a great number of load transitory processes, the motor moment equation having the following expression:

$$M_1 = a_0 - b_0\omega_1 = M_n\left(1+\frac{M_{max}}{M_n}\right) - \frac{M_n}{\omega_n}\frac{M_{max}}{M_n}\omega_1, \tag{11}$$

where: a_0 represents the conventional moment size; this is obtained if the given right line is extended to the intersection with the ordinate axis; b_0 is the angular coefficient of this right line; these coefficients can be written depending on the nominal technical characteristics of the chosen driving motor (nominal moment and angular velocity, determined at their turn depending on the nominal power and revolution) and depending on the ratio M_{max}/M_n, given by the motors catalogues [3].

Resistant moment modelling

For the resistant moment determination, it is considered the case of a diagram specified to striking machines, with big resistant moment in regime working and with small resistant moment for no-load operation. A characteristic that describes such a kind of working machine is represented in Figure 3,b and it can be described by the following equations:

$$\begin{aligned} M_3 &= M_{t0} && \text{for } \Delta\varphi \in \varphi_g, \text{ for empty trip,} \\ M_3 &= M_{t0} + M_{shock} && \text{for } \Delta\varphi \in \varphi_r, \text{ for working operation,} \end{aligned} \tag{12}$$

where: M_{t0} represents the moment transmitted by the clutch in the complete connected working situation, and the M_{shock} the value of the shock moment.

Spring force modelling

In the uncoupling process, the active intermediate elements displacement leads to the compression of the springs. As a result, at the spring compression, the resistant force F_a, operates on these elements. The resistant force expression depends on the type of the used elastic elements; for a linear elastic characteristic it can be written:

$$F_{ax} = F_{a1} + s_a K_a, \tag{13}$$

where: F_{a1} represents the pretension springs force; its expression is determined from condition that in the complete connected working situation, the transmitted torque to be the moment M_{t0} (the moment for the clutch design); k_a represents the spring rigidity.

The resistant force expression will be determined for all working stages, replacing for the displacement s_a the proper their expressions.

5. THE MODELLING OF THE SEMI-CLUTCHES MOTIONS BY MEANS OF THE LAGRANGE EQUATIONS OF THE SECOND SPECIES

For the bimobile equivalent mechanism of the clutch, the determination of the transmission functions for the forces (relations (3) and (4)) will be achieved using the Lagrange equations of the second species; this method is preferred to d'Alembert principle, due to some advantages, as the following:

- the Lagrange equations method is simpler to apply due to a smaller calculation volume;
- this method is suited, very well, to the use of the performable software of the dynamic analysis on the computer;
- replacing, in the transmission functions of the moments, the external moments on the basis of the mechanical characteristics of the motor and resistant energy systems, there are obtained two differential equations; these represent, in fact, the motion equations of the system.

6. CONCLUSIONS

About the tackled themes by the present paper, it must be specified that in the technical references exists – and still there are elaborated – studies and models concerning the dynamic behaviour description of the safety clutches, on purpose to identification of new ways of functional performances increasing. Although numerous, the existent studies and models present, as a rule, a high level of complexity, corroborated frequently with calculus errors and / or interpretation; all of these make actually their use in direct design difficult or even impossible.

The use of these studies in the design practice would suppose their process by a designer with a very rich experience and a serious training in the area, that to allow him the solving of the difficult aspects and to correct the errors sources concerning both calculus and interpretation.

Starting from this reality, the paper proposes to remodel the main dynamic aspects of the safety clutches in view of the designer.

In this way, it is taken into consideration the fact that the modern design of the mechanical transmissions requires the finding of the optimum solutions from a constructive and functional point of view, and the constructive design must be correlated with that technological. In this way, it is possible the obtaining of mechanical components with reduced gauge and **small weight**, with **high reliability** and **low cost**.

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