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ELASTIC AND SAFETY CLUTCH WITH RADIAL TAPERED ROLLER AND METALLIC ELASTIC ELEMENTS AXIALLY ARRANGED

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***Abstract*—**The paper presents a new type of clutch named "Elastic and Safety Clutch" that can accomplish the functions of the elastic and those of the safety clutches, but it is not a combined clutch. The proposed clutch is an elastic and safety clutch with metallic intermediate elements. The paper presents the elastic and safety clutch with radial tapered roller and metallic elastic elements axially arranged. The design and verification computing relations of the elastic and safety clutch are established.

***Keywords*—**clutch, elastic, safety, mechanism with cam, multiple functions

I. Introduction

A condition imposed to the elastic clutches is that at the breakage of an element, the clutch does not fail immediately. If there is only an elastic element, the total breakage of the clutch has to be inferred, in case of partial fractures or fissures. Another condition imposed to elastic clutches is that the elastic elements that can rapidly be destroyed, to be easily replaced – if it is possible without the clutch disassembling or without the axial displacement of the axle stubs [1].

The elastic clutch could take over, overloads up to a certain torque value. Over this value, when the transmission is not provided with a load disengagement system, the elastic elements of the clutch are destroyed, this corresponding with the placing of transmission out of the operation [2].

The safety clutch fulfils, besides the main function of the torque transmission, the function of torque limitation or automatic interruption of the connection between the coupled shafts, in the case of some overloads' occurrence during the performance.

The safety clutches assure the transmission performance up to the limit torque value for which the safety elements are designed. Over this value, the relative slipping between the semi-clutches appears which consequently involves the mechanical transmission protection. Based on the aforementioned facts, it results the necessity of some clutches, that by the associated functions to allow the load disengagement before that the elastic elements will be destroyed.

In the case of diverse applications, the simple functions of the two clutches can be combined, obtaining a combined clutch. In this case, the combined clutch 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.

II. Elastic and Safety Clutch

The elastic and safety clutches are characterized by the following functions (functional and technical criteria):

1) they make the bundle between two shafts (with relatively fix variable position) and ensure the torque transmission and the rotation motion between the shafts (according to the general definition);

2) the power transmission is braked off when the resistive moment outruns an imposed limit value;

3) the braking off of the energy flux is realized on the basis of an elastic element deformation.

By analyzing the properties of the elastic and safety clutches, a distinctive importance goes to the modeling of the elastic element, in order to ensure the automated braking off every flux, at the torque limit value.

From their use within technique and from the criteria analysis of the mechanisms, one can conclude that the cam mechanism (Fig. 1) lends itself (the best) to the demands previously formulated [3].

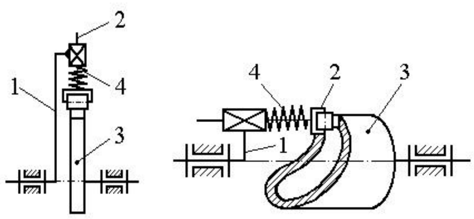

Fig. 1. Structural scheme

Fig. 1 presents the structural scheme of an elastic and safety clutch with metallic intermediate elements. The clutch contains the following elements [4]:

1) *the semi-clutches, 1 and 2;*
2) *the equiangular cam 3, assembled on the semi-clutch.*

The elastic intermediate elements are executed from spring steel, with a segments' shape. The spring segments are assembled in packets and the number of packets is imposed by the cam type. The segments can have different thickness and widths, depending on: the transmitted torque, the cam width, and the maximum value of the axial displacement that can be take over [5].

III. Structural Scheme the Elastic and Safety Clutch with Radial Tapered Roller and Elastic Elements Axially Arranged.

Fig. 2 presents the structural scheme of an elastic and safety clutch with metallic intermediate elements. The clutch contains the following elements:

1) *the semi-clutches, 1 and 3;*
2) *the equiangular cam 2, assembled on the semi-coupling 3, the tapered roller 2 .*

The elastic intermediate elements are executed from spring steel, in the form of plate packs 4. The spring segments are assembled in packets, the number of packets being imposed by the cam type, number of tapered roller 2. The segments can have different thickness and widths, depending on: the transmitted torque, the length of the blades is a function of the allowed axial displacement.

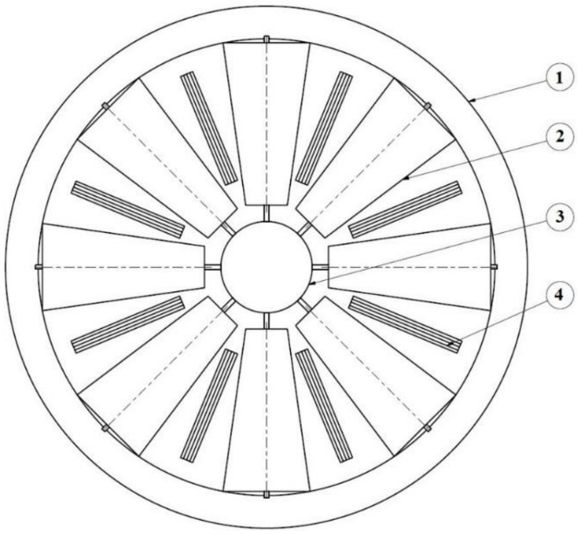

Fig. 2. Structural scheme of the elastic and safety clutch with cam and degenerate followers

The equiangular cam represents the semi-clutch 2 Fig. 1 that in Fig. 2 is represented by the combined tapered element 2 and 3, [6].

Relationships computing design and verification of elastic and safety clutch tapered slats arranged radially and axially will perform and follow the steps as in the linear clutch feature, but applying the peculiarities related to the operation characteristic nonlinear clutch.

IV. The Torque and the Elastic Characteristic

Starting from the structural schemes and from the representative functions and proprieties – of the elastic and safety clutch, the next anterior of constructive generation can be formulated:

1) *the clutch must absorb radial and angular tilts;*
2) *the relative movement between the semi-clutches, as well as the releasing must be made without shocks;*
3) *the clutch must have a reduce rigidity; it is suggested a characteristic $M_t(\varphi)$ with a rising inclination and a big damping capacity ;*
4) *the clutch elasticity could be modify – by changing or adding of a constructive elastic elements;*

Fig. 3 presents the geometrical model for the torque determination.

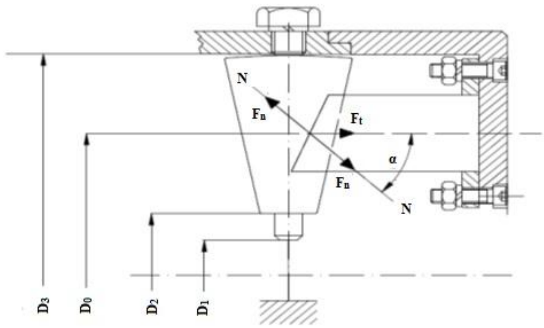

Fig. 3. Geometrical model for the torque determination

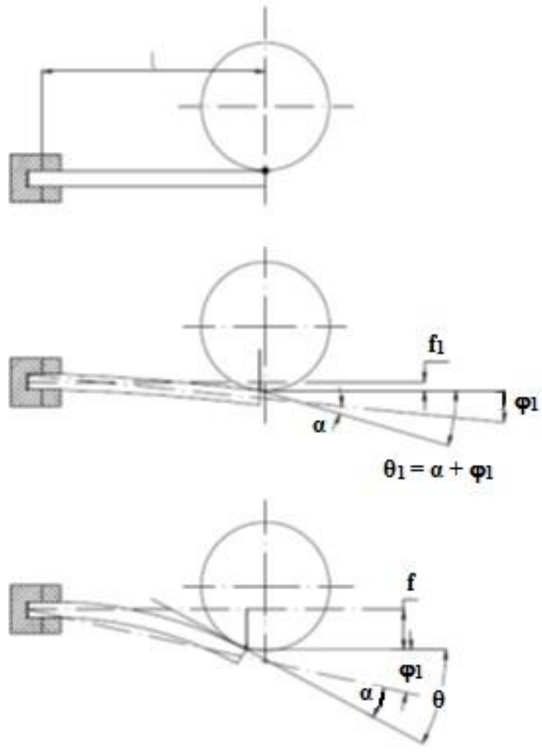

Fig. 4. The simplified geometrical model

Fig. 4 presents the simplified geometrical model.

For establishing the design and verification relations the following assumptions are accepted:

1) *the elastic clutch is considered symmetrically loaded;*
2) *the resilient is embedded in the driver semi-clutch, which allows relative movements between the semi-clutches;*
3) *due to the relative motion between the spring elements of the packs, friction forces appear between the blade packs and the tapered roller which are considered uniformly distributed along the contact surfaces;*
4) *relative rotation of the clutch half inserted between the resilient bending loads and twisting;*
5) *as a result of the clearance of insertion of the elastic pocks and the tapered rollers, it is considered that not all lode packs are under uniform stress, which imposes the correction of the calculations using a coefficient of unevenness of k=0,8…0,85;*
6) *installation games affect the relative angle of rotation of the clutch half in the sense of increasing their operation because of their processing;*
7) *on the occurrence of radial, angular , or combined misalignment the elastic elements are under additional bending stress;*
8) *the elastic elements deform accumulating potential energy, which allows the taking over of moment beats, which can be taken over by the clutch.*

The clutch belongs to the category of derivate clutch from a mechanism with degenerate cam and degenerate follower. The connection between the two semi-clutches 1 and 3 is realized with a package of lamellar bows axially disposed 4.

At the outrunning of the limit torque, which can be transmitted by the clutch, between the semi-clutches appears a relative rotation movement, which permits the load me clutch of the mechanic transmission.

The elastic and safety clutches are characterized by a variable rigidity (nonlinear characteristic) as presented in Fig. 5 and refer to (1); the security condition of the mechanic transmission is presented in (1), [7].

$$\mathbf{k}(\varphi)=\frac{\mathbf{dM_t}(\varphi)}{\mathbf{d}(\varphi)} \tag{1}$$

$$\mathbf{M_{t\,lim}}(1+\Delta)\le \mathbf{M_{t\,maxa}} \tag{2}$$

where: $k(\varphi)$ - represents the tangent to the curve of the torque, which is written depending on the relative displacement; φ - the relative displacement angle between the semi-clutches; $M_t(\varphi)$ - the torque corresponding to the clutch deformation with the angle φ; $M_{t\lim}$ - the torque when the clutch disengage; $M_{t\max a}$ - the maximum torque admitted by the strength of the most weak clutch element; Δ - the relative error reset inputs in function of the clutch.

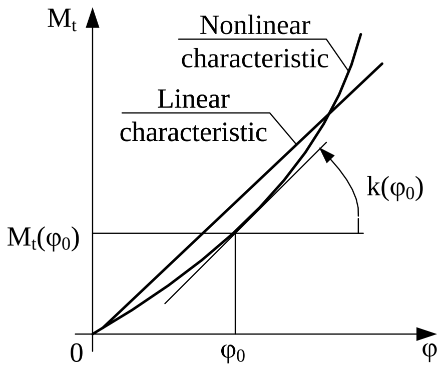

Fig. 5. The characteristics of the elastic clutches

During torque occurs roller contact pressure and elastic lamellae. Contact moves from the initial point made by slide and roller blades taper towards the top. The point of application of the resultant contact pressure F_n, with direction normal to generator running on tapered roller element of a cone envelope and slide, so the direction remains parallel, overlapping the normal continuous joint NN, [8].

Due to the slippage between the rollers and blades friction forces arise that force F_n summed give a resultant which closes with the direction of the normal joint angle equal to the angle of friction. The magnitude of this angle is small and therefore the friction forces are generally neglected, assuming that the resultant force of the contact roll - slide joint acts in the direction normal to the two surfaces (cam, solid) the point of contact.

Normal force F_n is determined by torque roll M_t working relationship:

$$\mathbf{F_n}=\frac{\mathbf{M_t}}{\mathbf{D_0}}=\frac{\mathbf{2M_t}}{\mathbf{D_0\cos\alpha}} \tag{3}$$

Normal force is decomposed into tangential and radial component

$$\mathbf{F_t}=\mathbf{F_n\cos\alpha}=\frac{\mathbf{2M_t}}{\mathbf{D_0}} \tag{4}$$

$$\mathbf{F_r}=\mathbf{F_n\sin\alpha}=\mathbf{F_t\cdot tg\alpha} \tag{5}$$

The clutch calculation follows:

1) *Calculation of maximum bending unit effort for overload springs;*
2) *Determination of the bending angle α;*
3) *Determination of the relative rotation angle clutch half;*
4) *Calculating the number of springs in a pack.*

Feature of this clutch is a straight (stiffness constant) as long as no contact spring package with tapered roller element of a cone envelope (no relative motion occurs between clutches), Fig. 6 points 0, 1. At increasing time passed, the contact takes place element of a cone circle, the point of application moves continuously and becomes nonlinear characteristic points 1, 2 (see Fig. 6).

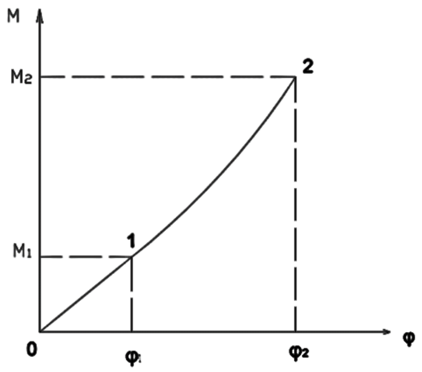

Fig. 6. The characteristic of the elastic clutch

$$\theta_1 = \alpha + \varphi_1 \tag{6}$$

where:

1) *the modeling is made on a simplified geometrical model;*
2) θ_1 *is the angle average fiber;*
3) α *semi-angle composed of tangency between the roller and blade;*
4) φ_1 *relative rotation angle of the clutch half.*

$$\theta_1 = \frac{F \cdot l}{2EI_z} \tag{7}$$

$$F = \frac{M_t}{z \cdot l} \tag{8}$$

$$I_z = n\frac{b \cdot h^2}{12} \tag{9}$$

where:

5) *F is the tangential force that returns a set of blades;*
6) I_z *sectional moment of inertia pack of blades;*
7) *z number of packages;*
8) *n is the number of springs (lamellae) of a package;*
9) *b, h the thickness of a lamella width respectively;*
10) *l blade length.*

The displacement at the end of the package:

$$f = (\varphi_1 + \alpha) \cdot l \tag{10}$$

$$M_{t\,lim} = \frac{3}{8} \cdot \frac{D_0^2 znEI_z \varphi_{max}}{\left(1 - n\frac{\mu h}{l}\right) l^3} \geq M_{tc} \tag{11}$$

The relative rotation angle between the clutch parts is:

$$\varphi = \frac{8}{3} \cdot \frac{M_t l^3}{D_0^2 nzEI_z}\left(1 \pm n\frac{\mu h}{l}\right) \tag{12}$$

The clutch stiffness is given by:

$$K = \frac{3}{8} \cdot \frac{D_0^3 nzEI_z}{l^3\left(1 \pm n\frac{\mu h}{l}\right)} \tag{13}$$

The ability to accumulate the clutch of the mechanical work strain is given by:

$$W_{mec} = \frac{2M_{tc}}{nzD_0^2 \sigma_{ai}}\left(1 - n\frac{\mu h}{l}\right) \tag{14}$$

Calculation and verification of safety elastic clutch tapered slats arranged radially and axially elastic element is:

$$\sigma_i = 12\frac{M_{tc}}{D_0^2 nzbh^2}\left(1 - n\frac{\mu h}{l}\right) \leq \sigma_{ai} \tag{15}$$

V. CONCLUSION

The analytic modeling of the torque in the case of the clutch with degenerated followers and lamellar bows disposed equiangular is based on the next conditions:

1) the modeling is made on a simplified geometrical model;
2) the lamellas package is replaced by a single lamella, represented in the geometrical model by the median line;
3) the came actuates over the lamella with a normal force orientated after the normal line in the point of contact;
4) the friction effects between the lamella and bearings are neglected;
5) the clutches have a simple construction, cabaret reduced dimension, low price;
6) the clutch ensure the compensation of axial, radial and angular tilt;
7) 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 clutch becomes a safety one.

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