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SIMPLE MECHANICAL CLUTCH WITH MULTIPLE FUNCTIONS – THE CAM OF THE CLUTCH WITH FOUR PROFILES

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***Abstract*—**The modular design imposes finding the optimal solutions from constructive and functional point of view. The constructive design must be correlated with the technological one. Thus, it is possible to obtain mechanical components with reduce building limit and weight, with high durability and small price.This 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 analytic modeling of the torsion moment in case of the elastic and safety clutches with degenerated followers into lamellar bows that are equiangular disposed.

***Keywords*—**clutch, 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.

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.

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 [1], [5], [7].

II. ELASTIC AND SAFETY CLUTCH

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

- they make the bundle between two shafts (with relatively fix variable position) and ensure the moment transmission and the rotation motion between the shafts (according to the general definition);
- the power transmission is braked off when the resistive moment outruns an imposed limit value;
- 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 [2], [3] 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.

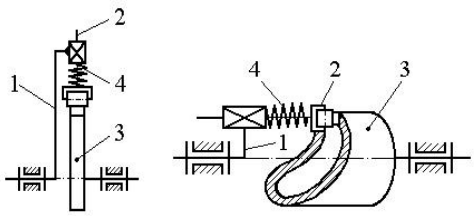

Fig. 1. Structural scheme

Fig. 1. presents the structural scheme of an elastic and safety clutch with metallic intermediate elements. The clutch contains the following elements [2], [3]:

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

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 torsion moment, the cam width, and the maximum value of the axial displacement that can be take over.

III. Structural Scheme and the Geometrical Model

Fig. 2. presents the structural scheme of an elastic and safety clutch with metallic intermediate elements. The clutch contains the following elements [7], [8]:

- the semi-couplings, 1 and 2;
- the equiangular cam 3, assembled on the semi-coupling.

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

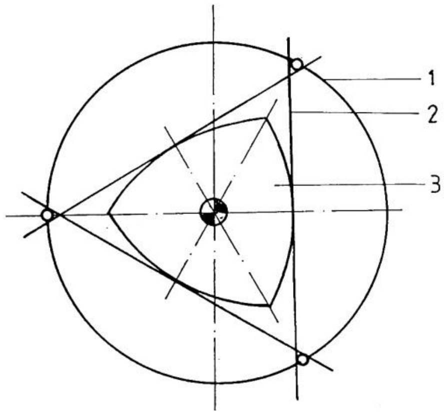

Fig. 2. Structural scheme of the elastic and safety clutch with cam and degenerate followers

The equiangular cam represents the semi-clutch 2 and it can be worked directly on the output shaft or it can be assembled on the shaft. There can be used cams with more profiled faces, as well as different shapes of contact surface.

Fig. 3. presents the structural scheme of an elastic and safety clutch with metallic intermediate elements and the equiangular cam 3. The equiangular cam 3 is provided with four profiles which have the shape of arcs. The element denoted 4 in Fig. 3. represents a compressive spiral spring that has the role to achieve the necessary strength for transmitting the torsion moment [4].

The relative displacement between the semi-clutches is graphically illustrated in Fig. 4. In the initial phase, Fig. 4.a. the clutch has the two semi-clutches in the position when the cam r_0 has the minimum radius (see Fig. 5.).

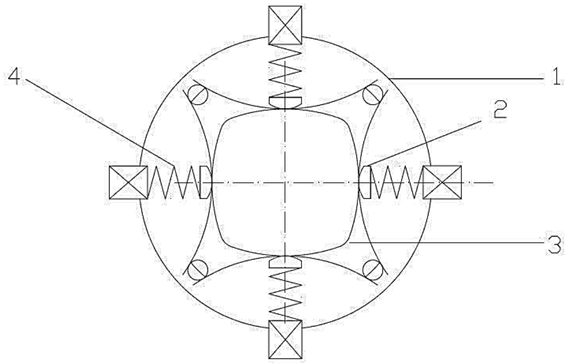

Fig. 3. Structural scheme of the elastic and safety clutch with metallic intermediate elements and equiangular cam

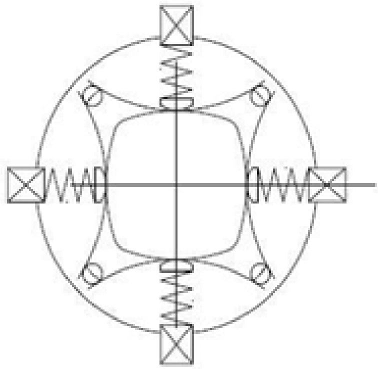

Fig. 4.a. Structural scheme – the cam is in initial position

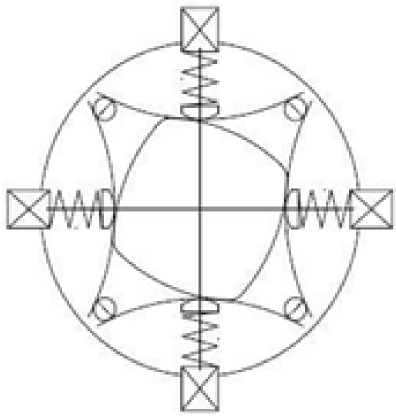

Fig. 4.b. Structural scheme – the cam is rotated with 22.5°

During the transmission of the torsion moment it takes place the relative displacement between the semi-clutches, defined by the angle φ, up to a value of 22.5° (see Fig. 4.b.). If the resistive moment has a value higher than the value of the moment, which is able to be transmitted by the working machine, the decoupling from load takes place. The relative displacement between the two semi-clutches takes place until the angle φ reaches a value of 45° (see Fig. 4.c.). As it can be observed from Fig. 5., the moment has the value of the radius r_1.

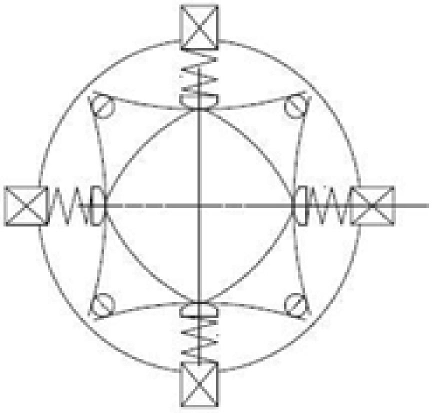

Fig. 4.c. Structural scheme – the cam is rotated with 45°

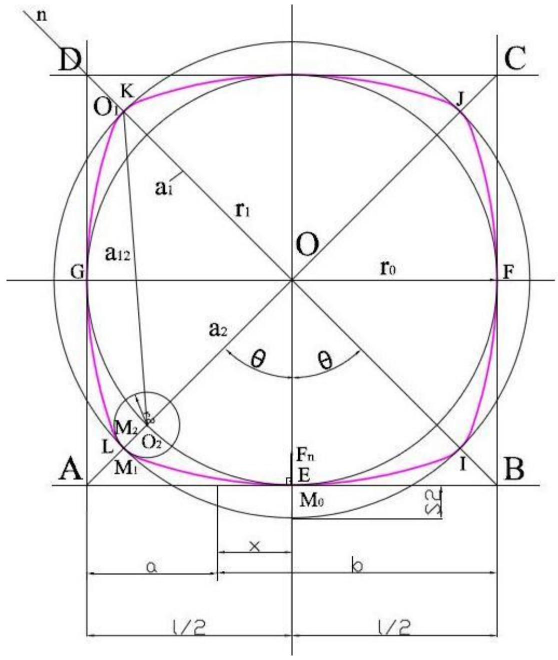

Fig. 5. The simplified geometrical model

Fig. 5. presents the geometrical model of an elastic and safety clutch with metallic intermediate elements. The cam profile can be realized in the shape of an arc and the calculus of the cam radius of curvature can be eliminated. The cam geometry is described simply, by the circle radius, which define it, and by the distance between the centers of these circles [2], [3], [6].

The simplified model considers the following assumptions:

- in calculus, 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 and jointed on the other;
- the bearings that support the lamellas are considered point-like; the distance between the bearings is L;
- the cam actuates over the lamella with a force F_n orientated after the normal line, in the point of contact;
- the angular deformations of the lamella are small (8°….10°);
- it considers that over the lamella actuates only the F_n force;
- the friction effects between the lamella and the bearing are neglected;
- in the calculus of the torsion moment there are considered: the normal force F_n effect as well as the friction force μF effect;
- the normal force (F_n) is considered vertical, and the friction force (μF_n) will be considered horizontal ($\mu F_o \approx F_o$).

IV. THE TORSION MOMENT

According to the adopted model, the determination of the torsion moment is realized in two stages, as it is presented in Fig. 6 [8], [9]:

a) The cam is in contact with the lamellas on the circle of radius r_1, so on M_0M_1 section;
b) The cam is in contact with the lamellas on the circle of radius r_1, so on M_1M_2 section.

$$\mathbf{T = T_1 + T_2} \tag{1}$$

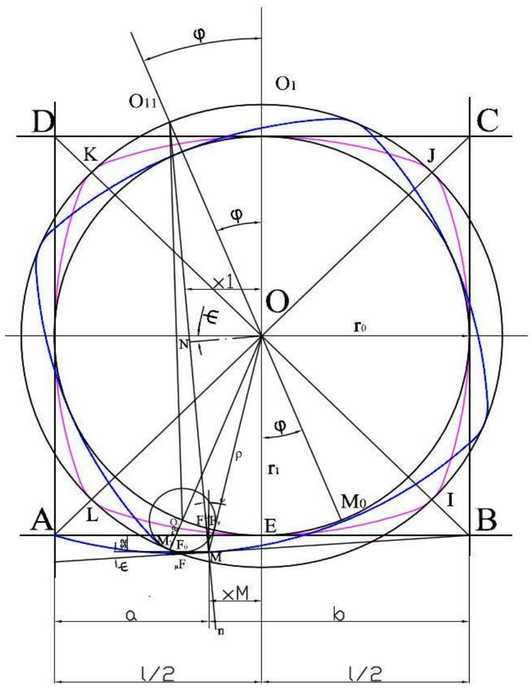

Fig. 6. The geometrical model

$$\mathbf{T_1 = nF_nOB = n(F_v x_M + F_o y_M)} \tag{2}$$

$$\mathbf{T_2 = nF_n r_1} \tag{3}$$

where n represents the number of profiles of the cam.

$$\begin{aligned}&\mathbf{T=T_1+T_2=nF_nOB+nF_nr_1=nF_n(OB+r_1)=}\\&\mathbf{n[(F_vxm+F_oy_M)+(F_vx_m+y_m)]=}\\&\mathbf{2n(F_vx_m+F_oy_m)}\end{aligned} \tag{4}$$

$$\mathbf{x=x(\varphi_1)}\ ,\ \mathbf{a=\frac{l}{2}-x(\varphi_1)} \tag{5}$$

$$\mathbf{S_2=S(\varphi_1)}\ ,\ \mathbf{b=\frac{l}{2}+x(\varphi_1)} \tag{6}$$

where φ_1 represents the relative displacement angle between the semi-clutches;

$$\begin{aligned}&\mathbf{T=2n[F_vx(\varphi_1)+\mu F_v(r_0+S_2(\varphi_1))]=}\\&\mathbf{=2nF_v[x(\varphi_1)+\mu(r_0+S_2(\varphi_1))]}\end{aligned} \tag{7}$$

To compute the aforementioned equations assumes that one may know the parameters $F_v, x(\varphi_1)$ and $S_2(\varphi_1)$

The equation of the displacement $S_2(\varphi_1)$ is expressed as:

$$S_2(\varphi_1)=\frac{F_v\left(\frac{l}{2}-x(\varphi_1)\right)^2\left(\frac{l}{2}+x(\varphi_1)\right)^2}{3EI_zl} \tag{8}$$

$$\mathbf{F_v=\frac{3EI_zlS_2(\varphi_1)}{\left(\frac{l}{2}-x(\varphi 1)^2\right)^2\left(\frac{l}{2}+x(\varphi_1)\right)^2}} \tag{9}$$

$$\mathbf{S_2(\varphi_1)=S_2\approx O_1A=r_1-(r_0+a_1\cos\varphi_1)} \tag{10}$$

$$\mathbf{F_v=3EI_2l\frac{r_1-r_2-a_1\cos\varphi 1}{\left(\frac{l}{2}-a_1\sin\varphi_1\right)^2\left(\frac{l}{2}+a_1\sin\varphi_1\right)^2}} \tag{11}$$

By introducing (10) and (11) into (7), the torsion moment can be obtained as:

$$\begin{aligned}&\mathbf{T=2nF_v[x(\varphi_1)+\mu(r_0+S_2(\varphi_1))]=}\\&\mathbf{=6nEI_zl\frac{r_1-r_0-a_1\cos\varphi_1}{\left(\frac{l}{2}-a_1\sin\varphi_1\right)^2\left(\frac{l}{2}+a_1\sin\varphi_1\right)^2}}\cdot\\&\mathbf{\cdot[a_1\sin\varphi_1+\mu(r_1-a_1\cos\varphi_1)]}\end{aligned} \tag{12}$$

V. CONCLUSION

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

- 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 and jointed 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.

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