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A NEW TYPE OF CAM MECHANISM

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Abstract: The paper presents a new cam-mechanism. Noticing that the mobility of a mechanism with a roller follower is 2, from which a passive one, the work propose replacing the rolling element with a special one, V-shaped, that has two contact points with the profile of the cam. Therefore, the mobility of the new mechanism decreases to 1. Differing from the mechanisms presented in the literature that have class 2 and order 2, the new replacing mechanism has the class and order equal to 3, attesting the structural complexity of the proposed mechanism.

1. INTRODUCTION

The courses on Machines and mechanisms theory present a chapter on cam mechanisms, [1], [2], [3], [4]. These mechanisms are initially depicted and afterwards they are arranged following diverse criteria. One of the criteria considers enclosing the mechanism into plane or spatial mechanisms category and another criterion states the constructive form of the follower that can be with tip or flat face, Figure1.

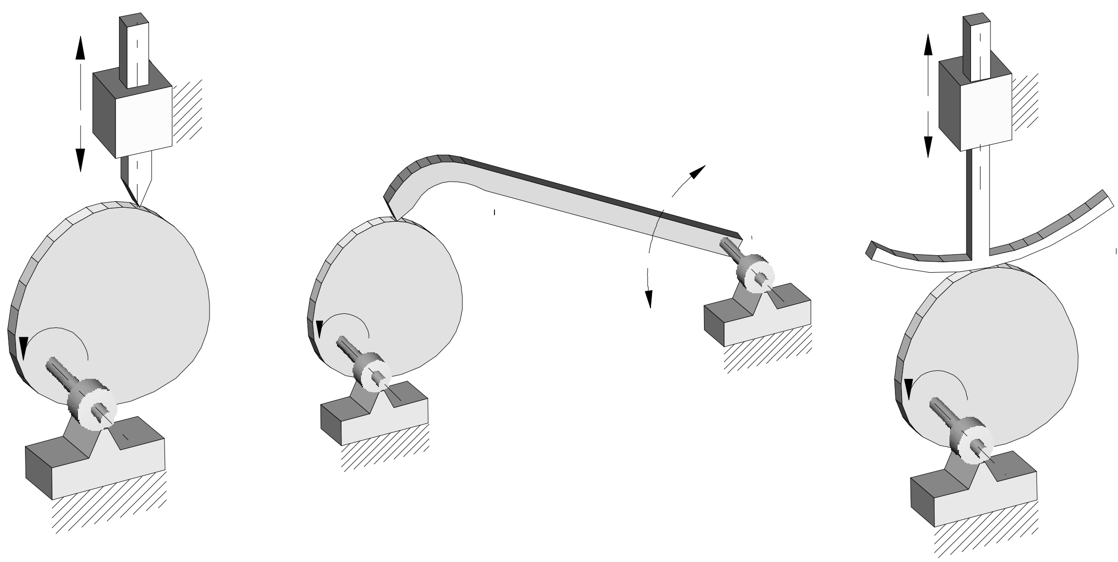

Figure 1. *Plane mechanisms with rotating cam and translation follower with tip (a), (b) and with flat face, (c)*

The major inconvenience of these mechanisms involves increased wear and decreased efficiency due to sliding friction from the cam-follower higher pair. For the tip follower mechanisms, this problem can be reduced by introducing a roller situated on the tip of the follower, as shown in Figure 2. The roller transforms the sliding friction between cam and follower into rolling friction which reduces the lost work and causes less wear. The literature emphasize the following feature: after introducing a roller into the structure of the mechanism, the degree of freedom increases one unit, [5], [6]. The reason consists in the passive rotation motion of the roller around its own axis, motion that doesn't influence the kinematics of the mechanism.

As an illustrative example, the mechanism with oscillating follower is presented, the other cases being similarly treated.

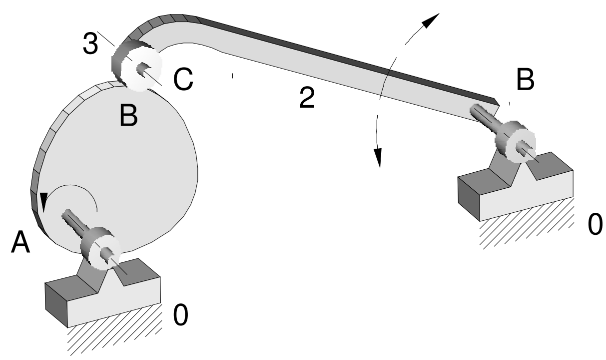

Figure 2. *Mechanisms with rotation cam and oscillating follower with roller*

The relation of Kutzbach, [1], is used for figuring the degree of mobility:

$$M_3 = 3(n-1) - c_4 - 2c_5 \quad (1)$$

where:

M_3 is the degree of mobility of the mechanism;

$n = 4$, $(0;,1;2;3)$, represents the number of elements from the structure of the mechanism (including the frame);

$c_4 = 1$, (B), number of higher pairs (the cam-roller pair in this case);

$c_5 = 3, (A,C,B)$, number of lower pairs.

Applying the relation (1) for the above parameters, it actually results $M = 2$.

2. A NEW MECHANISM

The idea that stands at the base of the new mechanism followed the use of the degree of freedom introduced by the roller to become *desmodrome.*

To this purpose, the roller is replaced by a "V' shaped element (snake tongue profile), jointed to the frame and that creates two pairs with the cam, Figure 3. Applying the relation of degree of freedom (1), and considering:

$n = 4$, $(0;,1;2;3)$, number of elements, including the frame, from the mechanism;

$c_4 = 2$, (B,C), number of higher pairs (in the case studied, the roller-cam pair)

$c_5 = 3, (A,D,E)$, number of lower pairs

$$M_3 = 3(4-1) - 2 - 2 \cdot 3 = 1.$$

The mechanism has the degree of freedom, $M = 1$, as shown in Figure 3.

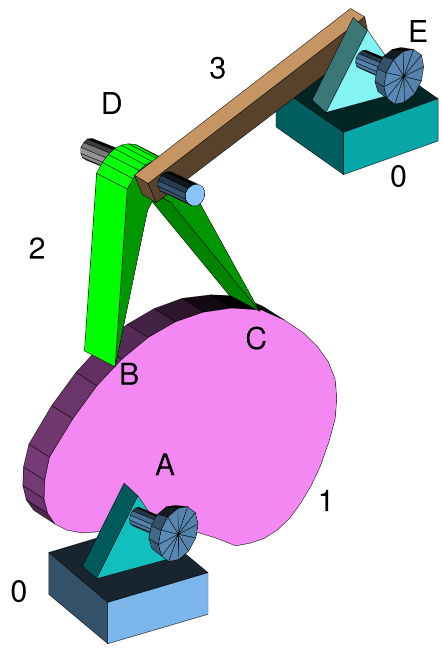

Figure 3. *New mechanism*

3. PARTICULARITIES OF THE STRUCTURE AND KINEMATICS STUDY OF THE MECHANISM

Special circumstances occur if kinematics study of the mechanism using the replacing mechanisms method is stipulated. The higher pair substitution is made under the known conditions:

- the degree of freedom of the mechanism should remain unmodified;
- the relative motion between the elements from the higher pair must be unchanged.

Figure 4 presents both the real mechanism and the replacing mechanism.

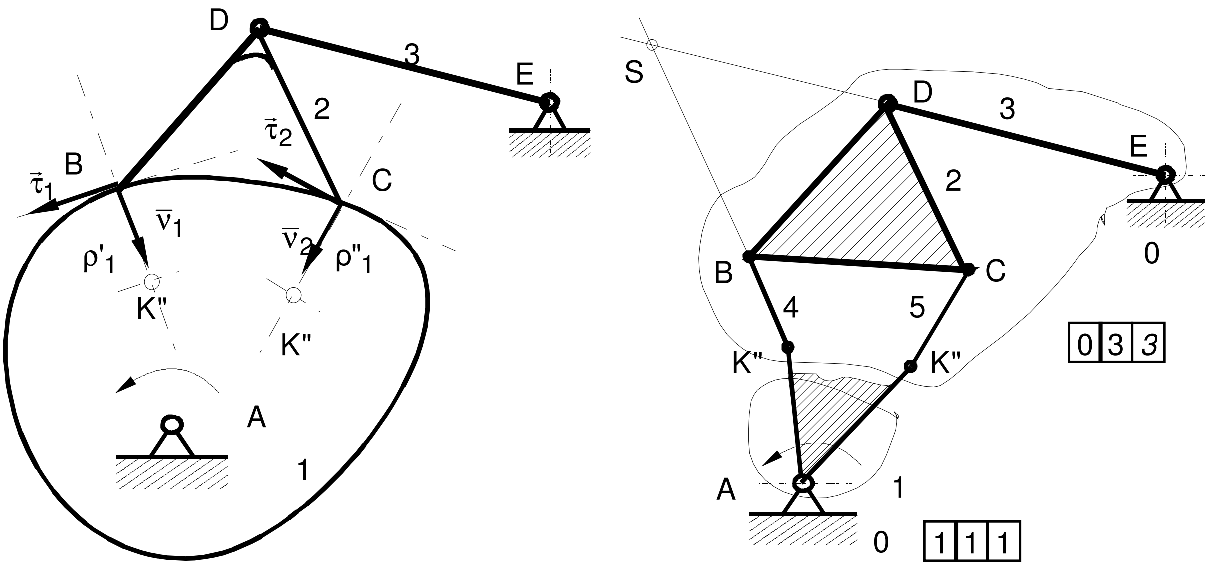

Figure 4. *Existent mechanism and the replacing mechanism*

The higher pairs in B and C were replaced by the elements 4 and 5 having the lengths the same as the curvature radius ρ'_1 and ρ''_1 of the cam profile from these points. At the ending of these elements the lower pairs K' and B, K" and C respectively, where K' and K" denote the curvature centers of the cam corresponding to points B and C. After decomposition in structural grouping, according to [4], one can observe that the mechanism consists of a leading group, namely the leading element 1 and the leading pair A and an Assur group, class 3 order 3, as seen from the figure, differing from all the cam mechanisms where from decomposition always an Assur group class 2 was obtained.

As known [5], the kinematics analysis of an Assur group class 3 order 3 using vector equations method requires using a singular point of the group, S, defined as the point fixed to the ternary element 3, and found at the intersection of the directions of two binary elements of the group, elements 3 and 4 in this case. The kinematics analysis of the real mechanism is straightforward.

The following geometrical parameters of first order must be determined:

$\bar{v}_{B_2B_1}$, relative velocity from pair B, parallel to the unit-vector of the tangent, $\bar{\tau}_1$;

$\bar{v}_{C_2C_1}$, relative velocity from pair C, parallel to the unit-vector of the tangent, $\bar{\tau}_2$;

ω_2, modulus of angular velocity of element 2, normal to the plane of motion;

ω_3, modulus of angular velocity of element 3, normal to the plane of motion.

Denoting by $\bar{\nu}_1$ and $\bar{\nu}_2$ the unit-vectors normal to the cam profile in points B and D respectively and by $\bar{k}$ the unit-vector normal to the motion plane, having the direction chosen to fulfill the following relation, [7]:

$$\bar{k} = \bar{\tau}_1 \times \bar{\nu}_1 = \bar{\tau}_2 \times \bar{\nu}_2 . \tag{2}$$

The subsequent relation can be written:

$$\bar{v}_{B_2B_1} = \bar{v}_{B_1} + v_{B_2B_1}\bar{\tau}_1; \tag{3}$$

$$\bar{v}_{C_2C_1} = \bar{v}_{C_1} + v_{C_2C_1}\bar{\tau}_2,$$

where:

$$\bar{v}_{B_1} = \bar{v}_A + \bar{v}_{BA} = \bar{v}_A + \bar{\omega}_1 \times \overline{AB} = \bar{v}_A + \omega_1 \bar{k} \times \overline{AB}; \tag{4}$$

$$\bar{v}_{C_1} = \bar{v}_A + \bar{v}_{CA} = \bar{v}_A + \bar{\omega}_1 \times \overline{AC} = \bar{v}_A + \omega_1 \bar{k} \times \overline{AC},$$

and:

$$\bar{v}_{B_2} = \bar{v}_D + \bar{v}_{B_2D} = \bar{v}_2 + \bar{\omega} \times \overline{DB} = \bar{v}_2 + \omega_2 \bar{k} \times \overline{DB}; \tag{5}$$

$$\bar{v}_{C_2} = \bar{v}_D + \bar{v}_{C_2D} = \bar{v}_2 + \bar{\omega} \times \overline{DC} = \bar{v}_2 + \omega_2 \bar{k} \times \overline{DC}.$$

The velocity of the point D is found using the expression:

$$\bar{v}_D = \bar{v}_E + \bar{v}_{DE} = \bar{v}_D + \bar{\omega}_3 \times \overline{ED} = \bar{v}_E + \omega_3 \bar{k} \times \overline{ED}. \qquad (6)$$

In the above equations, it must be considered that:

$$\begin{aligned} \bar{v}_A &= 0, \\ \bar{v}_E &= 0 \end{aligned} . \qquad (7)$$

By substitution in equation (3) from equations (4), (5), (6) and (7), it results the following system:

$$\omega_3 \bar{k} \times \overline{ED} + \omega_2 \bar{k} \times \overline{DB} = \omega_1 \bar{k} \times \overline{AB} + v_{B_2B_1} \bar{\tau}_1; \qquad (8)$$

$$\omega_3 \bar{k} \times \overline{ED} + \omega_2 \bar{k} \times \overline{DC} = \omega_1 \bar{k} \times \overline{AB} + v_{C_2C_1} \bar{\tau}_2.$$

The system of equations (8) consists of two plane vector equations, where $v_{B_2B_1}$, $v_{C_2C_1}$, ω_3 and ω_2 are unknown geometrical parameters.

Using the dot product between the equations of the system and the unit-vectors $\bar{v}_1$ and $\bar{v}_2$ respectively, the following two scalar equations are obtained:

$$\begin{cases} (\bar{v}_1 \cdot \overline{ED})\omega_3 + \omega_2 (\bar{v}_1 \cdot \overline{DB}) = (\bar{v}_1 \cdot \overline{AB})\omega_1 \\ (\bar{v}_2 \cdot \overline{ED})\omega_3 + \omega_2 (\bar{v}_2 \cdot \overline{DB}) = (\bar{v}_2 \cdot \overline{AB})\omega_1 \end{cases}; \qquad (9)$$

Where the following relations were taken into consideration:

$$\bar{v}_1 = \bar{k} \times \bar{\tau}_1, \; \bar{v}_2 = \bar{k} \times \bar{\tau}_2; \qquad (10)$$

$$(\bar{a} \times \bar{b})(\bar{c} \times \bar{d}) = (\bar{a} \cdot \bar{c})(\bar{b} \cdot \bar{d}) - (\bar{a} \cdot \bar{d})(\bar{b} \cdot \bar{c}). \qquad (11)$$

The solution of the system (8) can be expressed under the form:

$$\begin{aligned} \omega_3 &= \frac{(\bar{v}_1 \times \bar{v}_2)(\overline{AB} \times \overline{DB})}{(\bar{v}_1 \times \bar{v}_2)(\overline{ED} \times \overline{DB})} \omega_1 \\ \omega_2 &= \frac{(\bar{v}_1 \times \bar{v}_2)(\overline{AB} \times \overline{ED})}{(\bar{v}_1 \times \bar{v}_2)(\overline{ED} \times \overline{DB})} \omega_1 \end{aligned}; \qquad (12)$$

and imposing

$$\bar{v}_1 \times \bar{v}_2 \neq 0. \qquad (13)$$

The modulus of relative velocity $v_{B_2B_1}$ from B pair is found by dot product between the first equation of system (8) and unit-vector, $\bar{\tau}_1 = \bar{v}_1 \times \bar{k}$, considering relation (11).

After some calculus, it is obtained:

$$v_{B_2B_1} = (\overline{\nu}_1 \cdot \overline{AB})\omega_1 - (\overline{\nu}_1 \cdot \overline{DB})\omega_2 - (\overline{\nu}_1 \cdot \overline{ED})\omega_3 . \qquad (14)$$

In an analogous manner, for $v_{C_2C_1}$, it is obtained:

$$v_{C_2C_1} = (\overline{\nu}_2 \cdot \overline{AB})\omega_1 - (\overline{\nu}_2 \cdot \overline{DC})\omega_2 - (\overline{\nu}_2 \cdot \overline{ED})\omega_3 . \qquad (15)$$

In a similar approach, the unknown components from acceleration equations, which are relative derivatives of the parameters determined in velocity study, are found, as it follows:

$\overline{a}^r{}_{B_2B_1}$, relative acceleration from B pair, parallel to the unit-vector of the tangent $\overline{\tau}_1$;

$\overline{a}^r{}_{C_2C_1}$, relative acceleration from C pair, parallel to the unit-vector of the tangent $\overline{\tau}_2$;

ε_2, modulus of angular acceleration of element 2, a vector normal to the plane of motion:

ε_3, modulus of angular acceleration of element 3, a vector normal to the plane of motion.

The equations between accelerations of the points overlaying in higher pairs B and C have the expressions:

$$\overline{a}_{B_2} = \overline{a}_{B_1} + \overline{a}^c_{B_2B_1} + \overline{a}^{tn}_{B_2B_1} + \overline{a}^r_{B_2B_1} .;$$
$$\overline{a}_{C_2} = \overline{a}_{C_1} + \overline{a}^c_{C_2C_1} + \overline{a}^{tn}_{C_2C_1} + \overline{a}^r_{C_2C_1} .; \qquad (16)$$

where:

$$\overline{a}_{B_1} = \overline{a}_A + \overline{a}^n_{BA} + \overline{a}^t_{BA},$$
$$\overline{a}^n_{BA} = -\omega_1^2 \overline{AB},$$
$$\overline{a}^t_{BA} = \overline{\varepsilon}_1 \times A\overline{B} = \varepsilon_1 \overline{k} \times \overline{AB}$$

$$\overline{a}_{C_1} = \overline{a}_A + \overline{a}^n_{CA} + \overline{a}^t_{CA}, \qquad (17)$$
$$\overline{a}^n_{CA} = -\omega_1^2 \overline{AC},$$
$$\overline{a}^t_{CA} = \overline{\varepsilon}_1 \times \overline{AB} = \varepsilon_1 \overline{k} \times \overline{AC}$$

And

$$\begin{aligned}&\overline{a}^{c}_{B_2B_1} = 2\overline{\omega}_1 \times \overline{v}_{B_2B_1} = 2\omega_1 \cdot v_{B_2B_1}\overline{k} \times \overline{\tau}_1 = 2\omega_1 \cdot v_{B_2B_1}\,\overline{\nu}_1 \\ &\overline{a}^{tn}_{B_2B_2} = \frac{v^2_{B_2B_1}}{\rho'_1}\overline{\nu}_1,\ \overline{a}^{r}_{B_2B_1} = a^{r}_{B_2B_1}\overline{\tau}_1\end{aligned}; \tag{18}$$

$$\begin{aligned}&\overline{a}^{c}_{C_2C_1} = 2\overline{\omega}_1 \times \overline{v}_{C_2C_1} = 2\omega_1 \cdot v_{C_2C_1}\overline{k} \times \overline{\tau}_2 = 2\omega_1 \cdot v_{C_2C_1}\,\overline{\nu}_2 \\ &\overline{a}^{tn}_{C_2C_2} = \frac{v^2_{C_2C_1}}{\rho''_1}\overline{\nu}_2,\ \overline{a}^{r}_{C_2C_1} = a^{r}_{C_2C_1}\overline{\tau}_2\end{aligned};$$

$$\overline{a}_{B_2} = \overline{a}_D + \overline{a}^{n}_{B_2D} + \overline{a}^{t}_{B_2D}; \tag{19}$$

$$\overline{a}_{C_2} = \overline{a}_D + \overline{a}^{n}_{C_2D} + \overline{a}^{t}_{C_2D};$$

$$\overline{a}_D = \overline{a}_E + \overline{a}^{n}_{DE} + \overline{a}^{t}_{DE}. \tag{20}$$

In equations (17) and (18), it must be considered that:

$$\begin{aligned}&\overline{a}^{n}_{DE} = -\omega_3^2\overline{ED},\ \overline{a}^{t}_{DE} = \overline{\varepsilon}_3 \times \overline{ED} = \varepsilon_3\overline{k} \times \overline{ED}; \\ &\overline{a}^{n}_{B_2D} = -\omega_2^2\overline{DB},\ \overline{a}^{t}_{B_2D} = \overline{\varepsilon}_2 \times \overline{DB} = \varepsilon_2\overline{k} \times \overline{DB}; \\ &\overline{a}^{n}_{C_2D} = -\omega_2^2\overline{DC},\ \overline{a}^{t}_{C_2D} = \overline{\varepsilon}_2 \times \overline{DC} = \varepsilon_2\overline{k} \times \overline{DC}.\end{aligned} \tag{21}$$

The angular acceleration $\overline{\varepsilon}_1$ of the cam is either specified or it is considered nil for the regime phase. In addition, it is obvious that:

$$\begin{aligned}&\overline{a}_A = 0, \\ &\overline{a}_E = 0\end{aligned}. \tag{22}$$

By substituting relations (17) - (22) into equations (16), the following system is obtained:

$$\begin{cases}\varepsilon_3\,\overline{k} \times \overline{ED} - \omega_3^2\overline{ED} + \varepsilon_2\,\overline{k} \times \overline{DB} - \omega_3^2\overline{DB} = \\ = \varepsilon_1\,\overline{k} \times \overline{AB} - \omega_1^2\overline{AB} + 2\omega_1 \cdot v_{B_2B_1}\overline{\nu}_1 - \dfrac{v^2_{B_2B_1}}{\rho'_1}\overline{\nu}_1 + a_{B_2B_1}\overline{\tau}_1 \\ \varepsilon_3\,\overline{k} \times \overline{ED} - \omega_3^2\overline{ED} + \varepsilon_2\,\overline{k} \times \overline{DC} - \omega_3^2\overline{DC} = \\ = \varepsilon_1\,\overline{k} \times \overline{AB} - \omega_1^2\overline{AB} + 2\omega_1 \cdot v_{C_2C_1}\overline{\nu}_2 - \dfrac{v^2_{C_2C_1}}{\rho''_1}\overline{\nu}_2 + a_{B_2B_1}\overline{\tau}_2\end{cases}; \tag{23}$$

The system (23) is resolved pursuing the same conduct as used in solving system (8).

4. CONCLUSIONS:

The paper suggests a new cam mechanism. The newness of the mechanism consists in the fact that the roller is replaced by an element "snake tongue" shaped that structures two higher pairs with the cam profile. The result reflects upon the class and order of the replacing plane mechanism. Differing from the mechanisms presented in the literature that have class 2 and order 2, the new replacing mechanism has the class and order equal to 3, attesting the structural complexity of the proposed mechanism.

In the case when the profile of the cam has a region with constant curvature radius, the replacing mechanism has, for this region, the class and order equal to 2.

The relations for calculation of kinematics parameters of order one and two are specified. One must emphasize that the solutions of the kinematics analysis are presented in vector form, independent of the referential chosen.

REFERENCES

1. Shigley, J. E., *Kinematic Analysis of Mechanism*, Ed. 2, Mc Graw-Hill, Series in Mechanical Engineering, 1969.
2. Artobolevski, I., *Les mecanismes dans la tehchnique moderne*, Vol. 1-5, MIR, Moscou, 1978.
3. Uicker, J. J., Jr., Pennock, G., Shygley, J., Theory of Machines and Mechanisms, Oxford University Press, 2003.
4. Artobolevski, S. I., *Teoria mecanismelor şi maşinilor*, Editura Tehnică, Bucureşti, 1955.
5. Duca, C., *Mecanisme*, Institutul Politehnic Iaşi, 1983.
6. Handra-Luca, V., Stoica, I. A:; *Introducere în teoria mecanismelor*, Vol. 1., Editura Dacia, Cluj-Napoca, 1983.
7. Şabac, Gh., *Matematici speciale*, Vol. 1., Editura Didactică şi Pedagogică, Bucureşti, 1964.