

THEORETICAL RESEARCHES ON KINEMATICS OF CAM - ROCKER MECHANISMS FROM SEED DRILLS TRANSMISSION

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Design of seed drills transmissions, which are composed of cam-rocker mechanisms, starts from their functional role, namely providing a range of transmission ratios to fully cover the range needed to make seeding norms of any type of agricultural crop. The transmission ratio is dependent on the size of the input (rotation angle of the camshaft) and the amplitude of oscillation of the rocker. The mathematical model developed in this paper allows obtaining the theoretical transmission ratios of the mechanism at which the contact between the leading element (cam) and the follower (rocker) is directly made and the roller of rocker moves on a plate for limiting the stroke.

Keywords: mechanism, kinematics, transmission ratio

1. Introduction

To ensure the correspondence of the forward speed of the seed drill and the amount of seeds to be distributed per unit area, in practice there are two main types of transmissions, namely:

a) gear drive (Northon gearboxes), that deliver 24 ... 128 transmission ratios, for each transmission ratio value being achieved a certain seeding norm [1,2]. Disadvantages of using the Northon type gear mechanism in the transmission system are: large constructive dimension; high weight; complicated adjustments; require lubrication after few hours of operation; they have components subject to erosion.

b) transmissions with cam - rocker mechanisms, which provide the driving of devices for seeds distribution, through an intermittent motion (the case presented in this paper).

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Worldwide, renowned manufacturers of agricultural machinery (ex. Pottinger, Gaspardo) used cam mechanisms in transmission of seed drills, due to the multiple benefits they offer, namely: simplified construction and operations of distribution control, reduced manufacturing costs and improved machine design. Also, it ensures the very precise distribution of seeds and the possibility of easy adjustment of the amount of seed per hectare.

In Romania, investigations undertaken until year 1999, to promote transmission with cam mechanism on the seed drills, were nonexistent. Since 1999, National Institute of Research-Development for Machines and Installations Designed to Agriculture and Food Industry - INMA Bucharest was researched and introduced into series manufacturing a mechanism having two cams mounted parallel and out of phase, that can adapt on all machines for sowing cereals in operation [3].

In the paper [4], the author has studied the kinematics of a cam mechanism, cam having four lobes and the contact between the leading element (cam) and follower (rocker) is done via a roller and the return stroke of the rocker is limited by a limiting plate. Using computer-aided simulation with ProEngineer software design, author of paper [5] performed virtual prototyping and studied kinematic of a cam - rocker mechanism, cams having several lobes.

Cam mechanisms are composed of an upper kinematic coupling, carried out by contact of a leading element, having a certain profile, called cam and a follower called rocker, moving according to a law determined by the shape of the cam profile. Maintain permanent contact between the cam and the rocker is achieved through a resilient force generated by coil springs. For kinematic analysis of the cam mechanisms may be used graphical, grapho - analytical or analytical methods.

The graphic method or analytical charts method can be applied directly or by reversing motion of mechanism elements. This method consists of tracking of cam successive positions, during a complete rotation, which is difficult, leading to complex figures, difficult to analyze [6, 7]. With graphical method, derivation and graphical integration of kinematic diagrams introduces method errors, which are much higher as these operations are repeated.

Another method for the analysis of the kinematics of the cam mechanism is a mechanisms replacement method, which consists in replacing the upper kinematic coupling of the cam mechanism through a chain of lower couplings to obtain in the end a mechanism equivalent in terms of kinematics with that is analyzed [7].

Graphic - analytical analysis methods, based on mathematical algorithms more or less sophisticated and processing the results as graphs, are commonly used today, they ensure maximum precision of the results. Analysis can be performed with specialized programs made on the platform programming

language (ex. Basic, Pascal, C ++), with programs aided design (ex. SolidWorks, Catia, ProEngineer) or performance software based on the theory of mechanical multi-body systems (ex. Adams, Plexus).

In the present paper, we used the classical approach, using the programming language Turbo Pascal, the theoretical model of the mechanical system is described by analytic relations which materialize functions of motion, we transposed the mathematical relations into programming language code and set numerical algorithm for solving.

2. Object of kinematic analysis

Sketch of operating principle of the cam - rocker mechanism, submitted of kinematic analysis by graphic - analytical method is presented in Figure 1. The mechanism transforms the continuous motion, with constant angular velocity ω_1 of the input shaft 1, in intermittent motion, with variable angular velocity ω_2 of the output shaft 3. Underlying mechanism functioning on the principle of adding the rotation impulses, generated by the cam - rocker - one-way clutch assembly. The varying rotation of the output shaft is carried out by changing the active stroke Ψ of the rocker, by limiting the return stroke thereof, through the limiting plate 8.

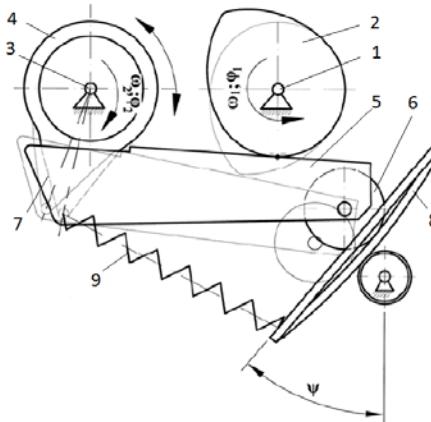


Fig. 1. Sketch of operating principle of the cam - rocker mechanism

During a complete rotation of the cam 2, the roller 6 of the rocker 5 moves on limiting plate, and the rocker prints to arm 7 which is articulated to, an oscillation with a certain angle of rotation. The arm 7, being mounted indirectly on the output shaft 3 via the one-way clutch 4, prints to the output shaft an intermittent rotation by an dependent angle on the stroke of the rocker. The rocker remain always in contact with the cam, this being achieved under the action of the coil springs 9.

3. Mathematical modelling of the cam - rocker mechanism

Sketch of operating principle of the cam - rocker mechanism was implemented in the kinematic sketch presented in Fig. 2. The significance of symbols and notations in Fig. 2 is presented in Table 1.

Table 1

The list of parameters in Figure 2

Name of parameter	Notation
Cam profile	<i>1</i>
Rocker profile	<i>2</i>
One-way clutch arm solidary with rocker	<i>3</i>
Roller of rocker	<i>4</i>
Angle of rotation of the cam	φ_1
Angle of rotation of the rocker	φ_2
Angle of rotation of the output shaft	φ_3
Angle of plate for stroke limiting to the vertical	ψ
Angle of plate for stroke limiting to the horizontal	Θ
Geometric parameters of the cam	a, r
Stroke of the roller of rocker	S_4

At the two profile which come into contact is attached a system of axis, namely: the system Fx_1y_1 for the profile *1* and the system Bx_2y_2 for the profile *2*.

The laws of motion of the two profiles are given by the movements of the origin of systems Fx_1y_1 and Bx_2y_2 , and by the rotation of these systems relative to fixed system of axis XFY , namely:

- the law of motion for profile *1*:

$$\begin{cases} X_F = 0 \\ Y_F = 0 \\ \varphi_1 = \varphi_1(t) \end{cases} \quad (1)$$

- the law of motion for profile *2*:

$r_B = r_A + R_3 u_1$, where $r_A = [X_A \ Y_A \ 0]^T$, $u_1 = [1 \ 0 \ 0]^T$
 $R_i = \text{rot}(z, \varphi_i) = [\sin \varphi_i \ \cos \varphi_i \ 0], (i=1, 2, 3); [0 \ 0 \ 1]$, we obtain:

$$\begin{cases} X_B = X_A + AB \cos \varphi_3 \\ Y_B = Y_A + AB \sin \varphi_3 \\ \varphi_2 = \varphi_2(t) \end{cases} \quad (2)$$

Parametric equations of the profile *1* are:

$$\begin{cases} x_1 = x_1(\lambda) \\ y_1 = y_1(\lambda) \end{cases} \quad (3)$$

$$\begin{cases} x_1 = a + r \cos \lambda \\ y_1 = r \sin \lambda \end{cases} \quad (4)$$

where λ is the curve generation parameter.

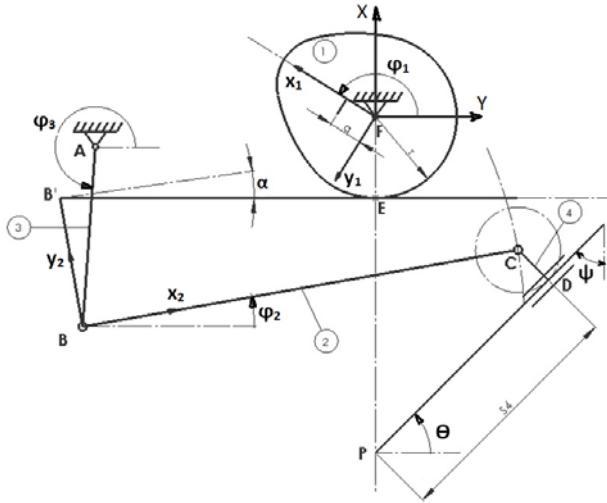


Fig. 2. The kinematic sketch of cam - rocker mechanism

Parametric equations of the profile 2 are:

$r_{2E} = BB' u_2 + [\vartheta \cos \alpha \ - \vartheta \sin \alpha \ 0]T$, where $u_2 = [0 \ 1 \ 0]^T$; we obtain:

$$\begin{cases} x_2 = x_2(\nu) \\ y_2 = y_2(\nu) \end{cases} \quad (5)$$

$$\begin{cases} x_2 = \nu \cos \alpha \\ y_2 = BB' - \nu \sin \alpha \end{cases} \quad (6)$$

where ν is the line generation parameter.

Having these data, on put contact and tangency conditions in the fixed system XFY . To write equations of contact between the two profiles is necessary to transform the coordinates of the two profiles from mobile systems in the fixed system. Contact conditions between the two profiles, are given in the fixed frame:

$$r_{10E} = r_{20E} \quad (7)$$

where: $r_{1E} = [x_1 \ y_1 \ 0]^T$, $r_{10E} = R_1 r_{1E}$, $r_{20E} = r_B + R_2 r_{2E}$;
we obtain:

$$\begin{cases} X_{10} = x_1 \cos \varphi_1 - y_1 \sin \varphi_1 = (a + r \cos \lambda) \cos \varphi_1 - r \sin \lambda \sin \varphi_1 \\ Y_{10} = x_1 \sin \varphi_1 + y_1 \cos \varphi_1 = (a + r \cos \lambda) \sin \varphi_1 + r \sin \lambda \cos \varphi_1 \end{cases} \quad (8)$$

$$\begin{cases} X_{10} = a \cos \varphi_1 + r \cos(\lambda + \varphi_1) \\ Y_{10} = a \sin \varphi_1 + r \sin(\lambda + \varphi_1) \end{cases} \quad (9)$$

$$\begin{cases} X_{20} = X_B + x_2 \cos \varphi_2 - y_2 \sin \varphi_2 = \\ = X_A + AB \cos \varphi_3 + v \cos \alpha \cos \varphi_2 - (BB' - v \sin \alpha) \sin \varphi_2 = \\ \quad X_A + AB \cos \varphi_3 - BB' \sin \varphi_2 + v \cos(\alpha - \varphi_2) \\ Y_{20} = Y_B + x_2 \sin \varphi_2 + y_2 \cos \varphi_2 = \\ = Y_A + AB \sin \varphi_3 + v \cos \alpha \sin \varphi_2 + (BB' - v \sin \alpha) \cos \varphi_2 = \\ \quad = Y_A + AB \sin \varphi_3 + BB' \cos \varphi_2 - v \sin(\alpha - \varphi_2) \end{cases} \quad (10)$$

Tangency condition in the fixed system, is given by:

$$\frac{\frac{\partial Y_{10}}{\partial \lambda}}{\frac{\partial X_{10}}{\partial \lambda}} = \frac{\frac{\partial Y_{20}}{\partial v}}{\frac{\partial X_{20}}{\partial v}} \quad (11)$$

or:

$$\left(\frac{dx_1}{d\lambda} \frac{dx_2}{dv} + \frac{dy_1}{d\lambda} \frac{dy_2}{dv} \right) \sin(\varphi_2 - \varphi_1) + \left(\frac{dx_1}{d\lambda} \frac{dy_2}{dv} - \frac{dy_1}{d\lambda} \frac{dx_2}{dv} \right) \cos(\varphi_2 - \varphi_1) = 0 \quad (12)$$

On the profile FABCDPF can be written the vector equation:

$$\overline{FA} + \overline{AB} + \overline{BC} + \overline{CD} = \overline{FP} + \overline{PD} \quad (13)$$

which produces two equations, namely:

$$\begin{cases} X_A + AB \cos \varphi_3 + BC \cos \varphi_2 + CD \cos(\theta - \pi/2) = X_P + S_4 \cos \theta \\ Y_A + AB \sin \varphi_3 + BC \sin \varphi_2 + CD \sin(\theta - \pi/2) = Y_P + S_4 \sin \theta \end{cases} \quad (14)$$

or:

$$\begin{cases} X_A + AB \cos \varphi_3 + BC \cos \varphi_2 + CD \sin \theta = X_P + S_4 \cos \theta \\ Y_A + AB \sin \varphi_3 + BC \sin \varphi_2 - CD \cos \theta = Y_P + S_4 \sin \theta \end{cases} \quad (15)$$

On the mechanism assembly, is written the complete system of nonlinear equations, as follows:

$$\left\{ \begin{array}{l} X_A + AB \cos \varphi_3 + BC \cos \varphi_2 + CD \sin \theta - X_P - S_4 \cos \theta = 0 \\ Y_A + AB \sin \varphi_3 + BC \sin \varphi_2 - CD \cos \theta - Y_P - S_4 \sin \theta = 0 \\ a \cos \varphi_1 + r \cos(\lambda + \varphi_1) = X_A + AB \cos \varphi_3 - BB' \sin \varphi_2 + v \cos(\alpha - \varphi_2) \quad (16) \\ a \sin \varphi_1 + r \sin(\lambda + \varphi_1) = Y_A + AB \sin \varphi_3 + BB' \cos \varphi_2 - v \sin(\alpha - \varphi_2) \\ \left(\frac{dx_1}{d\lambda} \frac{dx_2}{dv} + \frac{dy_1}{d\lambda} \frac{dy_2}{dv} \right) \sin(\varphi_2 - \varphi_1) + \left(\frac{dx_1}{d\lambda} \frac{dy_2}{dv} - \frac{dy_1}{d\lambda} \frac{dx_2}{dv} \right) \cos(\varphi_2 - \varphi_1) = 0 \end{array} \right.$$

The system (16) can be written as:

$$\left\{ \begin{array}{l} f_1(\varphi_2, \varphi_3, S_4, \lambda, v) = X_A - X_P + AB \cos \varphi_3 + BC \cos \varphi_2 - S_4 \cos \theta + CD \sin \theta = 0 \\ f_2(\varphi_2, \varphi_3, S_4, \lambda, v) = Y_A - Y_P + AB \sin \varphi_3 + BC \sin \varphi_2 - S_4 \sin \theta - CD \cos \theta = 0 \\ f_3(\varphi_2, \varphi_3, S_4, \lambda, v) = X_A + AB \cos \varphi_3 - BB' \sin \varphi_2 - a \cos \varphi_1 + v \cos(\alpha - \varphi_2) - r \cos(\lambda + \varphi_1) \\ f_4(\varphi_2, \varphi_3, S_4, \lambda, v) = Y_A + AB \sin \varphi_3 + BB' \cos \varphi_2 - a \sin \varphi_1 - v \sin(\alpha - \varphi_2) - r \sin(\lambda + \varphi_1) \\ f_5(\varphi_2, \varphi_3, S_4, \lambda, v) = \lambda + \alpha + \varphi_1 - \varphi_2 - 3\pi/2 = 0 \end{array} \right. \quad (17)$$

The unknowns of the system (17) are: $\varphi_2, \varphi_3, S_4, \lambda, v$.

To solve the system we used gradient method, a method described in detail in [8] and [9].

4. Results

The drawn up calculation program considered the successive positions of the cam during a complete rotation from 1° to 361° and various positions of the stroke limiting plate, transposed into the program through values of angle ψ .

After running the calculation program, we obtained values of parameter $\varphi_2, \varphi_3, S_4, \lambda, v$ as shown in Figure 3, these values were then imported and processed with *Excel*, an example being presented in Table 2.

C:\DOCUME-1\Dragos\Desktop\MECANISMATURBO.EXE					
1	-0.1500	1.3423	0.0751	3.0600	0.0537
2	-0.1467	1.3355	0.0756	3.0459	0.0532
3	-0.1434	1.3287	0.0760	3.0317	0.0526
4	-0.1425	1.3228	0.0753	3.0152	0.0518
5	-0.1408	1.3175	0.0766	2.9994	0.0514
6	-0.1393	1.3122	0.0768	2.9834	0.0509
7	-0.1381	1.3084	0.0771	2.9671	0.0503
8	-0.1372	1.3048	0.0772	2.9506	0.0498
9	-0.1366	1.3018	0.0774	2.9336	0.0493
10	-0.1363	1.2996	0.0775	2.9166	0.0488
11	-0.1366	1.2985	0.0776	2.8989	0.0483
12	-0.136	1.2973	0.0773	2.8814	0.0478
13	-0.1403	1.3048	0.0770	2.8603	0.0475
14	-0.1443	1.3135	0.0764	2.8388	0.0472
15	-0.1484	1.3222	0.0757	2.8172	0.0470
16	-0.1527	1.3321	0.0751	2.7955	0.0467
17	-0.1571	1.3417	0.0744	2.7737	0.0465
18	-0.1615	1.3516	0.0737	2.7518	0.0463
19	-0.1661	1.3616	0.0730	2.7298	0.0460
20	-0.1707	1.3718	0.0723	2.7077	0.0458
21	-0.1753	1.3820	0.0716	2.6856	0.0456
22	-0.1799	1.3922	0.0708	2.6636	0.0454
23	-0.1846	1.4024	0.0701	2.6415	0.0452
24	-0.1892	1.4128	0.0693	2.6193	0.0450

Fig. 3. A sequence in the results display window

Table 2

Results obtained after running the calculation program (sequence)

Successive positions of the cam	Rotation angle of the rocker, φ_2 [rad]	Rotation angle of the output shaft, φ_3 [rad]	Stroke of the roller of rocker, S_4 [m]	Curve generation parameter, λ	Line generation parameter, v
1	-0.15	1.3423	0.0751	3.06	0.0537
2	-0.1467	1.3355	0.0756	3.0459	0.0532
3	-0.1444	1.3288	0.076	3.0307	0.0526
4	-0.1425	1.3228	0.0763	3.0152	0.052
5	-0.1408	1.3175	0.0766	2.9994	0.0514
...
361	-0.1223	1.2699	0.0795	3.1955	0.0522

Fig. 4 shows the variation diagram of the rotation angle of the output shaft according to the rotation angle of the camshaft, the cam having one lobe. The motion of the output shaft is in the form of angular pulses, which highlights the fact, that underlying mechanism functioning on the principle of transformation continuous motion of rotation of the camshaft, in rotating intermittent motion of output shaft, its angle of rotation being dependent of the amplitude of rocker rotation motion.

For the maximum stroke of the rocker, to a full rotation of the cam, the output shaft rotates at an angle $\varphi_2=155^\circ$. Considering that at the start of the stroke for the withdrawal of the rocker, instantly unlocks one-way couplings, the theoretical transmission ratio achieved by the cam - rocker mechanism is:

$$i = \varphi_1 / \varphi_2 = 3.31 \quad (18)$$

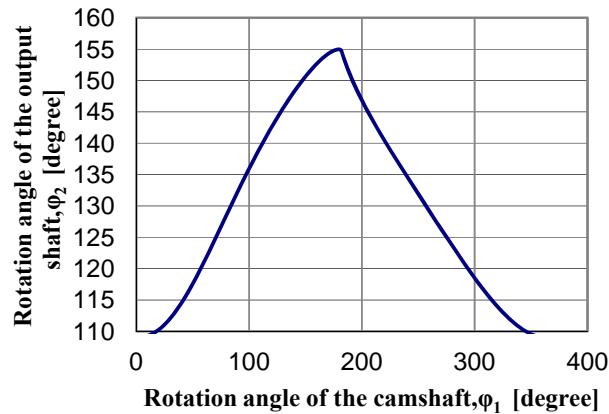


Fig. 4. The variation diagram of the rotation angle of the output shaft according to the rotation angle of the camshaft

The maximum stroke of the rocker is obtained for a positioning angle of the limitation plate, $\psi = 30^\circ$.

By reducing the stroke of the rocker, pulse amplitude of angular rotation of the output shaft decreased. Changing the active stroke of the rocker is accomplished by limiting its return stroke by limitation plate, which provides adjustment over a wide range of transmission ratio.

Figure 5 presents the variation diagram of the rotation angle of the output shaft for a complete rotation of the cam, taking into account the ten positions of limitation plate, which affect the oscillation of the rocker stroke.

The minimum stroke of the rocker is obtained for an angle of positioning of the limitation plate, $\psi = 50^\circ$.

For the minimum stroke of the rocker, to a full rotation of the cam, the output shaft rotates at an angle of $\varphi_2 = 146.7^\circ$. In this case, the theoretical transmission ratio achieved by the cam - rocker mechanism is:

$$i = \varphi_1 / \varphi_2 = 5.328 \quad (19)$$

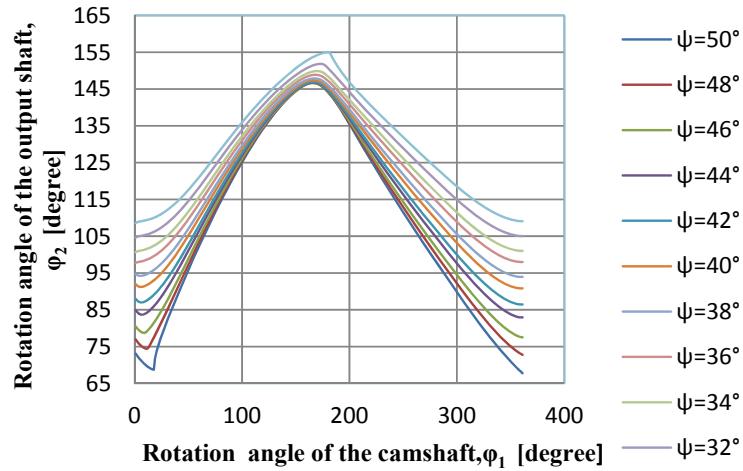


Fig. 5. The variation diagram of the rotation angle of the output shaft according to the rotation angle of the camshaft, for ten positions of the stroke limitation plate

In comparison, Fig. 6 presents the values of the output shaft stroke for three position of limitation plate. The chart highlights the dependence of the active stroke of output shaft according to the return stroke of rocker. The maximum angular stroke of the output shaft is obtained for a maximum oscillation of the rocker.

By graphical representation of amplitude variation of rocker oscillation according to the angle ψ of limitation plate inclination, the obtained graph is shown in Fig. 7.

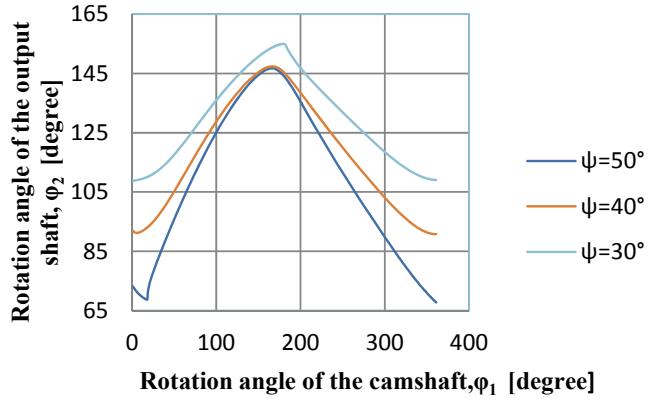


Fig. 6. The variation diagram of the rotation angle of the output shaft, for three specific positions (minimum, medium and maximum) of the stroke limitation plate

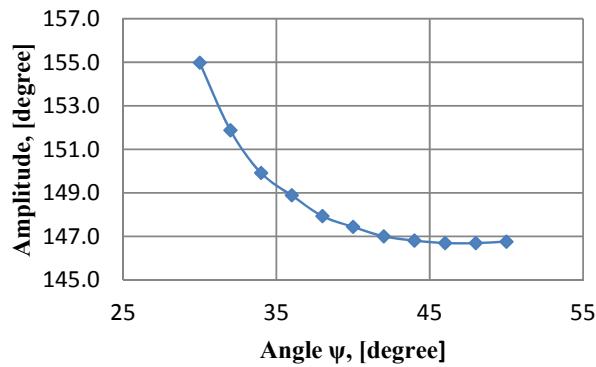


Fig. 7. The variation diagram of amplitude of rocker oscillation according to the angle of limitation plate inclination

Analyzing the diagram in Figure 7, it is found that by increasing the angle ψ of limitation plate inclination, rocker oscillation amplitude is reduced accordingly. Table 3 presents the values of rocker oscillation amplitude and transmission ratios that are made depending on the angular position of the stroke limitation plate.

By graphical representation of the transmission ratios values according to the lever position on graded sector, on obtained the adjustment curve of the theoretical transmission ratio of cam - rocker mechanism (Fig. 8).

Analyzing the diagram in Fig. 8, it appears that for the first positions of the lever, which correspond to high levels of positioning angles of the limitation plate, $\psi = 50^\circ \dots 40^\circ$, the value of the transmission ratios sharp drop, followed by a

decrease easier. The adjustment curve highlights the theoretical transmission ratio can take any value in the range 3 ... 5.328. From a theoretical standpoint, the minimum and maximum range values are less important that depend directly of constructive and geometric parameters of the mechanism. What is really important is the aspect of the adjustment curve of transmission ratios, aspect similar to that obtained by other authors (ex. [4]), but using other methods of analysis. Here deducting the usefulness of mathematical model and of the used analysis method, developed by the authors of paper.

Table 3

Values of the transmission ratios realized by the cam - rocker mechanism

Lever position on graded sector	The angle positioning of the limitation plate, ψ [degree]	Rocker oscillation amplitude, [degree]	Transmission ratio, i
0	50	146,8	5,328
10	48	146,7	4,959
20	46	146,7	4,656
30	44	146,8	4,354
40	42	147,0	4,176
50	40	147,4	3,975
60	38	147,9	3,842
70	36	148,9	3,683
80	34	149,9	3,573
90	32	151,9	3,437
100	30	155,0	3,310

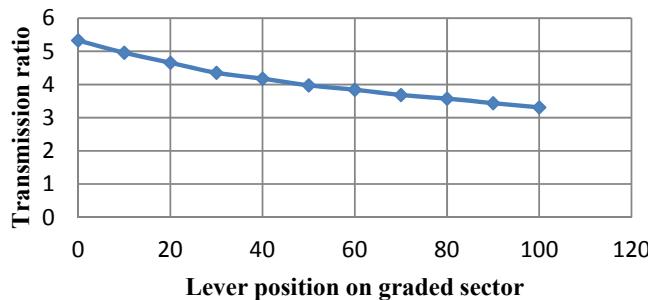


Fig. 8. The adjustment curve of the theoretical transmission ratio according to the lever position on graded sector

5. Conclusions

Given the direct link between the law of motion transmission or variation of rotation angle of the output shaft depending on the rotation angle of the camshaft and definition of transmission ratio, it is inferred that, in principle, ranging either kinematic parameters of mechanism (for example, the stroke of the rocker, in the case of cam-rocker mechanism) can influence the size of the average angular velocity output.

As a result, the processes of adjusting the transmission ratio of the cam mechanisms are, in fact, methods of adjusting the angle of oscillation ("pulse") of the follower (rocker).

Using graphic - analytical analysis in this paper, in determining the theoretical transmission ratios of cam - rocker mechanisms, represents an original approach, by reference to the theoretical research of the literature studied. The results obtained by the paper's authors allow extending the investigations on the devices made up of two or more cam - balance mechanisms - running parallel.

Kinematic analysis of cam - rocker mechanisms, reflected in this paper by graphical representation of the motion laws and determining transmission ratios, depending on the adjustment parameter (stroke of rocker) is a useful aid in both the design kinematics of new solutions construction, and in comparing the performance of existing solutions in order to optimize them.

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